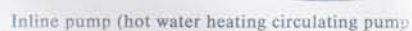


Centrifugal Pump

method of i.b.p.a. in centrifugal means of an axially sliding adjustment inside the hollow → pump in the case of large → cooling water absorb adjustment forces up to 600 higher in certain cases. The axial of the adjustment rod is shifted mechanical screw thread drive a hydraulic piston. Manual a reduction gear is after → propeller the

ded types of for a disturb- the pumps. The ensures that the receive a disturbance- when if the flow into the from one side (emer- of failure of the



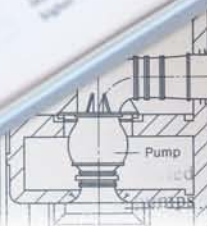
The arbitrary selection of an axial, or peripheral i , is however justified by the fact that the \rightarrow rotation of the i , diameter D must not turn the i into a set of optically active in

Positive Displ.
Verdrängerpumpe
Pompe volu-
P.d.p., also
collective
cording
pe



These
Kommunally
Gruppe in...

Page 10 of 10



Missão

Nossa missão é suprir o mercado de bombas, válvulas e sistemas correlatos, propondo soluções e fornecendo equipamentos e serviços, sempre embasados em nossa tradição de tecnologia, qualidade e confiabilidade.

A operação deve proporcionar a satisfação de nossos clientes, colaboradores, acionistas e da comunidade, garantindo o contínuo desenvolvimento da organização.

Visão

Desejamos consolidar a posição da KSB como líder no mercado brasileiro de bombas.

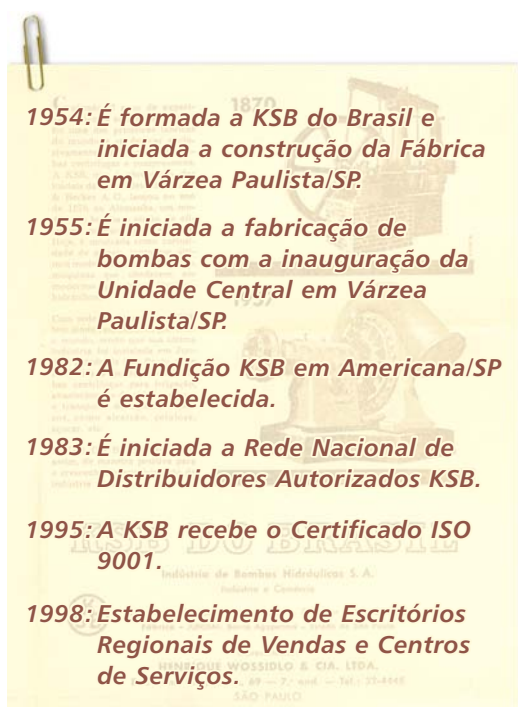
Pretendemos introduzir nossas linhas de válvulas, posicionando a KSB entre as principais empresas desse mercado.

Visamos ampliar a atuação da KSB como supridora de soluções integradas e prestadora de serviços em sistemas de bombeamento.

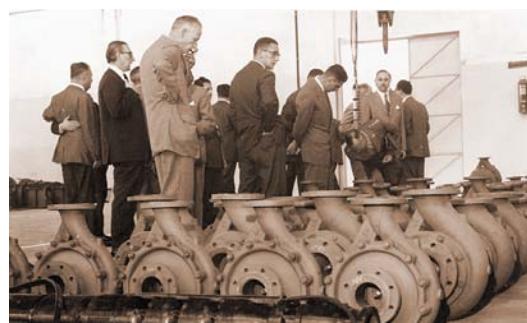
Queremos também, aumentar nossas exportações e participação em projetos internacionais, principalmente no continente americano.

KSB no Brasil

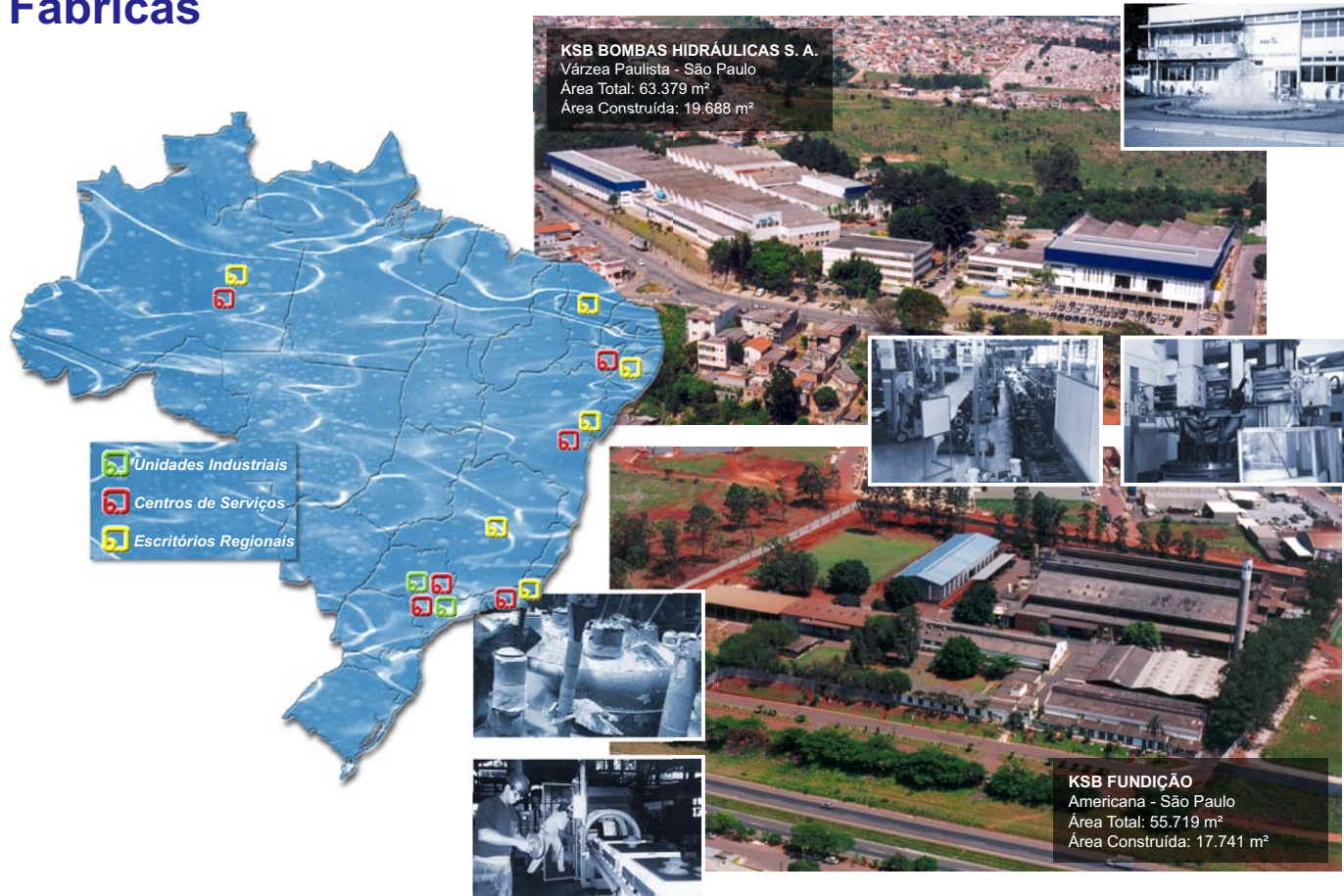
KSB e você... fluindo juntos, crescendo juntos...



Desde o início de suas atividades no Brasil, a KSB vem investindo na contínua melhoria dos meios de produção, formação profissional, qualidade, pesquisa e desenvolvimento, serviços a clientes e cobertura territorial.



Fábricas



Operações

Fundição

Capacidade instalada: 600 t/mês

Principais Ligas:

- **Ferro Fundido**
Cinzento - ASTM A48 CL 25/30/35/40
Nodular - ASTM A536 Gr 60-40-18
- **Aço**
Carbono - ASTM A216 WCB
Inox austenítico - ASTM A743 Cf3 / CF8 / CF8M
Inox martensítico - ASTM A743 CA6NM / CA15 / CA40
Inox duplex - ASTM A743 CD4MCu
Inox super duplex - ASTM 890 5A CE3NM
- **Bronze** - GCu Sn10 e SAE 40
- **Ligas especiais sob consulta**



Usinagem e Montagem

Todo o fluxo de produção foi recentemente estudado, simulado e instalado com apoio de recursos da Unicamp-SP utilizando os mais sofisticados softwares.

Como resultado a KSB dispõe de instalações industriais de Classe Mundial onde os Sentidos de Organização, Utilização, Saúde, Limpeza e Disciplina (5S) são praticados pelos colaboradores.

A KSB vem constantemente efetuando importantes investimentos e intensos programas visando o aperfeiçoamento de suas unidades industriais e de seu padrão de qualidade.



Tecnologia e Qualidade

Soluções exatas em bombeamento...

A motivação das atividades da KSB é a certeza que a tecnologia deve construir pontes, não barreiras. Tecnologia é a nossa força... Servir é o nosso compromisso.

Os produtos da KSB transportam líquidos com eficiência, qualidade e confiabilidade a qualquer lugar desejado. A KSB não é apenas líder na tecnologia de manuseio de fluidos mas uma companhia que apresenta soluções com foco no cliente.

Satisfazer clientes requer que estejamos em contato direto e frequente com os mesmos. Isto significa oferecer-lhes produtos e soluções de primeira classe, valor para os recursos dispendidos, atendimento pronto e eficaz, centros de serviços regionais e todo o suporte que é esperado.

Qualidade significa cliente satisfeito...

A Qualidade KSB começa com nossos colaboradores, os quais são continuamente treinados e aperfeiçoados em métodos de auto-controle, planejamento, direção, verificação e auditoria.

O comprometimento da Alta Direção da KSB é expresso em suas ações diárias, orientando e acompanhando os indicadores que compõem nossa Política da Qualidade.

O sistema da qualidade é continuamente controlado por auditores independentes de acordo com os requisitos da Norma NBR ISO 9000:2000.



BANCO DE PROVAS
Potência: até 2.200 HP
Vazão: até 24.000 m³/h
Pressão: até 250 Kgf/cm²



Estação de TRATAMENTO DE ÁGUA - Bombas RDL

Água e Meio Ambiente

Água é vida. E nós a tratamos com cuidado...

Transportar água é nosso negócio e o fazemos em todas as aplicações e formas. As linhas de bombas e válvulas KSB são utilizadas em irrigação, captação, tratamento e distribuição.

A KSB também fornece sistemas completos incluindo estudo e instalação de proteção contra transientes hidráulicos, assim como a reabilitação e expansão de estações de bombeamento existentes.

Cuidar e proteger o meio ambiente também é missão da KSB.

Para tanto a linha de produtos KSB conta com bombas e válvulas aplicáveis em captação e tratamento de esgotos domésticos, efluentes industriais e águas residuais em geral.



Válvulas Borboleta
Diâmetro até 3200 mm



IRRIGAÇÃO com Pivot Central - Bombas WKL



Estação de TRATAMENTO DE ESGOTO
ETE CEDAE ALEGRIA / RJ - Bombas SPY V

Energia, Óleo e Mineração

Produtos e serviços para toda forma de geração de energia e extração / processamento de óleo e minérios...

Energia mantém o mundo em movimento. Para hidroelétricas, termoeletricas, cogeração ou usinas nucleares a KSB dispõe dos mais variados tipos de bombas para aplicações como drenagem de barragens / turbinas, circulação, alimentação de caldeira, extração de condensado, refrigeração e demais serviços auxiliares.



REFINARIAS - Bomba RPH API 610

A KSB Brasil também é um dos centros de produção e desenvolvimento de bombas para todas as etapas da indústria de óleo, upstreams e downstreams, ou seja, desde a extração em plataformas off shore e em todos os processos de uma refinaria, com bombas API 610.



ALIMENTAÇÃO DE CALDEIRA - Bomba HDB de Alta Pressão



MINERAÇÃO - Bomba de Polpa LCC

Também na extração e processamento de minérios aplicamos nossas bombas projetadas e construídas para as mais severas condições de trabalho e atendendo requisitos quanto à abrasão, corrosão e resistência mecânica.

Indústria

Nossos produtos atuam na fabricação de seus produtos...

As bombas e válvulas KSB estão presentes em todos os principais campos industriais seja diretamente envolvidas no processo produtivo ou em atividades auxiliares da produção.

Para sistemas de pintura de automóveis, torres de refrigeração, circulação de óleo térmico, movimentação de produtos químicos abrasivos ou corrosivos, sucos de frutas, pasteurização em cervejarias, fabricação de açúcar e álcool e outras incontáveis aplicações industriais a KSB oferece

bombas e válvulas de alta confiabilidade, comprovada eficiência e que agregam valor aos produtos fabricados por nossos clientes proporcionando segurança, economia, rentabilidade e elevada vida útil.



LINHA MEGA para as mais diversas aplicações industriais



COMBATE À INCÊNDIO, bombas e sistemas com certificado NFPA 20 e FM - Factory Mutual



VÁLVULAS BORBOLETA com atradores pneumáticos e sistema de monitoração eletrônica

Rede de Distribuição e Building

Distribuidor KSB - Nossa presença nacional...

A Rede Nacional de Distribuidores Autorizados KSB conta com 15 membros cobrindo o território nacional oferecendo produtos e serviços autorizados e dispondo de estoque de bombas e peças sobressalentes para atendimento imediato das necessidades de nossos clientes.

Com profissionais categorizados e continuamente treinados, o Distribuidor KSB atua no assessoramento a clientes para dimensionamento das instalações de bombeamento, seleção, aplicação, montagem e manutenção de bombas.

KSB trabalhando para o conforto da população...

Nas cidades ou no campo as bombas KSB contribuem para a contínua melhoria das condições de vida. Para centrais de ar condicionado, abastecimento de água potável, sistemas de pressurização, proteção contra incêndio, drenagem de águas servidas, irrigação de jardins ou pequenas propriedades a KSB oferece a mais ampla linha de produtos com soluções simples, eficazes e econômicas.



Service

KSB Service - Pessoas a seu serviço...



A KSB Service tem evoluído continuamente colocando à disposição de nossos clientes, profissionais altamente qualificados na prestação de serviços e treinados para pronto e eficaz atendimento. A KSB dispõe de estoque de peças sobressalentes para grande parte da linha de produtos ou esquema emergencial para produção das peças de bombas especiais que eventualmente sejam necessárias em situações de urgência.

A atuação da KSB Service compreende:

- Contratos de manutenção e assistência técnica - "In site" ou convencional
- Atendimento 24 horas em caso de emergência
- Serviços de engenharia de campo
- Montagem e partida de equipamentos rotativos
- Gestão do estoque de peças sobressalentes
- Planejamento e execução de manutenção preditiva, preventiva e corretiva.



Preface

As is the usual custom in general reference works, no mention is made in this book of existing patents, registered designs or trade marks. Despite the absence of any special identification, the fact that no such reference has been made should not lead the reader to assume that any such proprietary names are not protected within the scope of the trade mark and brand name protection legislation and that they can therefore be freely used by anyone.

Technical editors:

Dr. Ing. Kurt Holzenberger

Dipl. Ing. (FH) Klaus Jung All rights reserved Copyright 1990 by KSB Aktiengesellschaft, D-6710 Frankenthal (Pfalz), Federal Republic of Germany

Introduction to the 3. Edition

14 years after the first and 9 years after the second slightly revised edition of *KSB's Centrifugal Pump Lexicon* you now have the third, thoroughly revised edition of this technical reference book in your hands. Where updating was required, the contents were brought up to the state of the art; new technical standards were taken into account, and the range of technical terms was extended, however without changing the well-proven layout of this lexicon.

Owing to our closer business ties in Europe we supplemented all technical terms with their German and French equivalents, which means that *KSB's Centrifugal Pump Lexicon* may also be used as a translation aid for more than 700 terms. Last but not least an alphabetic list of technical terms is provided in the annex as quick reference for the user.

We hope that this third edition, too, will meet with lively interest and that the supplements and revisions will further increase the benefit of using *KSB's Centrifugal Pump Lexicon*. On this occasion we would also like to thank more than 25 experts of our company who made valuable contributions to this edition by providing the revisions or translations as well as Mr. Fischer of the printing agency for their commitment to this cause.

Frankenthal, September 1990

The Editors

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A

Abrasion

Abrasion

Abrasion

A. is a form of wear and is caused by the mechanical attack of the fluid on the surface of the material (in particular when the fluid contains entrained solid particles). In the case of centrifugal pumps specially designed for the hydraulic hydrotransport, special provisions as regards design and selection of materials must be made to ensure that this wear (erosion) is kept within economically tolerable bounds.

A. wear will tend to increase as a result of the following influences:

- with the square or cube of the relative velocity between the pumped medium and the pump material (materials),
- in cases where the pump materials are simultaneously exposed to attack by corrosion, cavitation or mechanical a.,
- proportionally to the concentration of solid particles,
- with the ratio of solids hardness (e.g. in the case of quartz) to pump material hardness,
- with the ratio of the density of solids (e.g. in the case of ore) to the density of the carrier liquid,
- when sharp-edged solids are conveyed as opposed to round shaped particles (e.g. in the case of sinter),
- with the grain size (diameter) of the solids (e.g. in the case of gravel),
- with larger deviations of the operating point from the optimum capacity (capacity).

In the case of conventional centrifugal pumps (i.e. pumps not incorporating any special wear- inhibiting features) the permissible percentage of solids (i.e. the solids concentration) and, if necessary, the grain diameter and the flow velocity must therefore be limited in accordance with practical experience.

In the case of centrifugal pumps designed for the conveyance of solids (hydrotransport), the following design features are adopted to counteract a. wear: extremely thick walls are provided at those places where wear by a. is anticipated. High flow velocities are avoided as far as possible, and consequently such centrifugal pumps generate relatively small pressure differentials only (pressure). Components which are likely to be subjected to a high rate of wear after a given period of operation are designed in such a way that they can be replaced by new ones with a minimum amount of manipulation and with as short as possible an interruption of operation. So-called armoured pumps are designed along these lines; their wear parts (or armour) consist of materials of extreme hardness and are designed in such a way that they only require mechanical machining in a very few places, and this machining is limited to grinding only. All these pumps for the hydraulic conveying of solids are fitted with sturdy outboard bearings and amply sized shafts, and in many cases they are also provided with speed (control), in order to prolong the operational running time of the pump as far as possible, despite increasing out-of-balance (unbalance of centrifugal pumps) and deteriorating head after onset of wear, and because valves and fittings for throttling control (control) would wear out very quickly.

The following correlation exists between the influences of material hardness and angle of impact or impingement: if the solid particles impinge on a hard, brittle surface at a large angle of impact (90° approx.), they will finally succeed in dislodging particles out of the wall, after initial compaction and subsequent fatiguing of the wall material; on the other hand, ductile materials are affected to a lesser extent by this *impact wear*. If the solid particles impinge at a small angle of impact (15° approx.) on a soft, ductile surface, they will abrade particles of material from the wall in a similar way to abrading with emery paper, hard materials are affected to a lesser extent by such *erosion* (also referred to as hydroabrasive wear) (Fig. 1). In narrow slit seals, solid material leads to so-called *grain-friction wear* (all three terms according to DIN 50320), which can be countered by flushing with clear water (very complicated and expensive) or by using extremely hard materials (plain bearing).

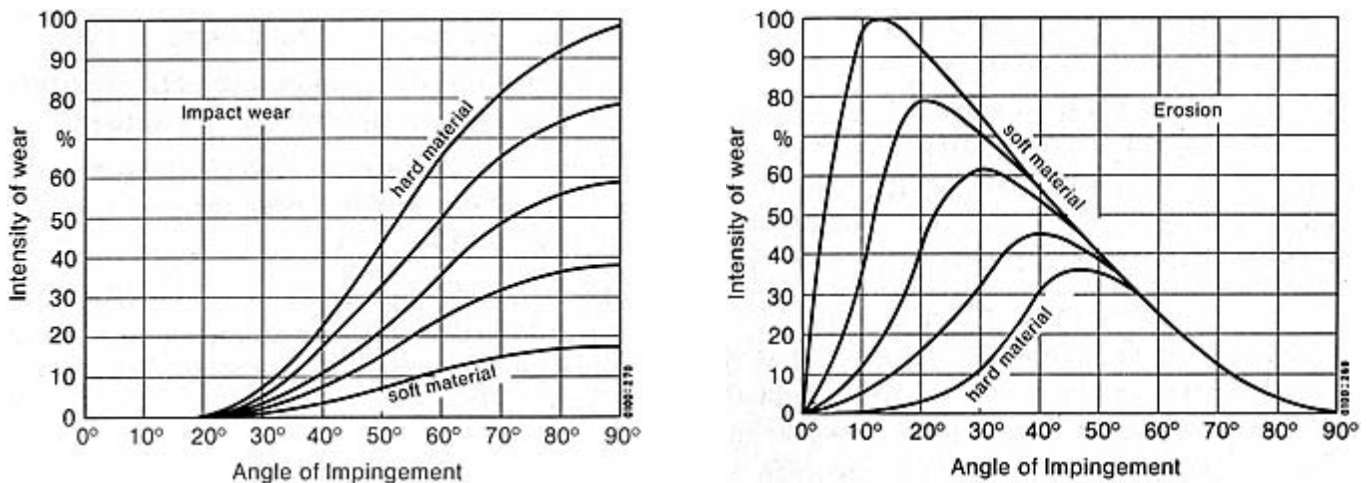


Fig. 1: Intensity of wear in function of the material and angle of impingement

Most a. damage in a centrifugal pump is caused by erosion and grain-friction wear, and consequently very hard materials (NORIHARD[®], chilled silicon cast iron, NIHARD[®] hard facings with tungsten carbide or cobalt alloys, ceramics) present advantages. On the other hand, only a few spots in the pump are exposed to impact wear, e.g. the front edges of volute spurs, impeller and guide vanes (blade), and the front faces of deflector baffles which are arranged in opposition to the direction of flow. To the extent possible, ductile materials should be used in such places (Fig. 2).

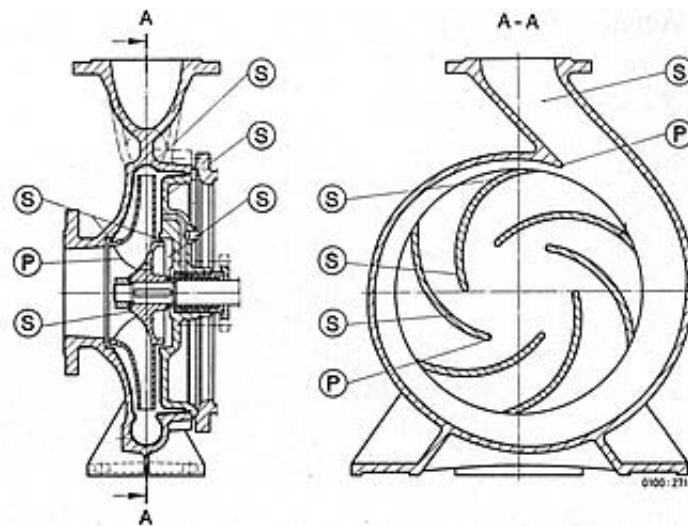


Fig. 2: Points subject to wear in a volute casing pump (P) = impact wear; (S) = erosion

Absolute Velocity

Absolutgeschwindigkeit
 Vitesse absolue

The concept of a.v. is used predominantly in the design of fluid flow machines. The a.v. is the velocity v of a particle of fluid in relation to its stationary surroundings (e.g. pump casing). The a.v., together with the relative velocity w and the circumferential velocity u forms the velocity triangle.

Acceptance Test

*Abnahmeversuch
Essai de réception*

In the a.t. of a centrifugal pump, the technical guarantees specified in the supply contract between the purchaser and the vendor are proved by measurement of the guaranteed data.

The following data are generally measured in the a.t.: rotational speed, head, capacity and shaft power of the pump. If necessary, the NPSH required (net positive suction head) of the centrifugal pump is also determined. In addition, the a.t. can also be used to verify other technical data and properties of the pump, in so far as they form part of the technical guarantee, e.g. stable throttling curve, power absorbed at pump shut-off-point (shaft power), maximum (or upper limit) shaft power, (shaft power), reverse rotational speed, leakage loss, sound pressure level (noise in pumps and pumping installations), quietness of centrifugal pumps, etc. The acceptance test codes for centrifugal pumps are the determining guidelines for the execution and evaluation of an a.t.

It should be borne in mind that every additional proven datum or property will increase the costs of the a.t.; therefore a request should only be made for the proven data and properties which are essential for the planned use of the centrifugal pump. The costs of the a.t. should represent a reasonable proportion of the costs of the centrifugal pump. Requirements specifying high degrees of accuracy also entail additional cost; details on the degrees of accuracy I, II and III for the a.t. can be found under guarantee.

Suitable locations for the execution of the a.t. include the pump test bed of the pump manufacturer, of the operator or of a neutral institute; in the case of pumps having a very high shaft power (in excess of 2 MW) and of pumps with a casing which cannot be dismantled (e.g. pumps with a concrete casing, see Fig. 6 under cooling water pump), the at. can be carried out in situ. The location, date, extent and cost of the at. shall be mutually agreed in the supply contract.

As regards the execution of the a.t., the following points should be agreed between purchaser and vendor in good time before commencement of the tests: permissible uncertainties of measurements, operating points which can be tested on the pump test bed using cold water or the actual liquid pumped, whether the tests could or should be carried out at full or at reduced rotational speed, which formula of efficiency reevaluation shall be applied, if necessary, and, in the case of an a.t. in situ, what operating conditions prevail or still remain to be arranged.

Acceptance Test Codes for Centrifugal Pumps

*Abnahmeregeln für Kreiselpumpen
Règles de réception de pompes centrifuges*

This concept encompasses the guidelines for carrying out acceptance tests. The a.t.c.f.c.p. are usually in the form of a standard, and generally incorporate:

- a) definitions of all entities required to demonstrate the guarantees, i.e. mainly: rotational speed, pressure, flow velocity, head, pump output, capacity, NPSH (net positive suction head), shaft power, pump cross-sections, head loss, efficiency;
- b) stipulations concerning technical guarantees and the fulfilment of said guarantees;
- c) recommendations for the preparation and execution of acceptance tests destined to verify the guarantees;
- d) stipulations concerning the comparability of the test results with corresponding guaranteed values, and the conclusions to be drawn from this comparison;
- e) recommendations concerning the compilation of the test report (acceptance report);
- f) descriptions of the results and testing methods (measuring technique) authorized for the proving of the guarantees, including the execution and evaluation of said methods, taking the overall tolerance into account.

The concept of "acceptances" shall be interpreted in its technical, and not in its legal sense in this context. It follows that a successfully performed acceptance test does not in itself represent an "acceptances" in the legal sense in accordance with par. 640 of the German Federal Legal Code (BGB).

Examples of the most widely used a.t.c.f.c.p. include the following:

- ISO 2548 or 3555 Centrifugal, mixed flow and axial pumps - Code for acceptance tests - Class C or B;
 - EUROPUMP rules for Acceptance of Centrifugal Pumps;
 - British Standard 599 - Methods of Testing Pumps,
 - Hydraulic Institute Standards Section: Centrifugal pump, test standards (Cleveland, USA);
 - DIN 1944 Acceptance Tests on Centrifugal Pumps (VDI rules for Centrifugal Pumps);
 - DIN 4325 Acceptance Tests on Storage Pumps;
 - IEC-Recommendation, Publication 198, International Code for the Field Acceptance Tests of Storage Pumps;
 - API Standard 610: Centrifugal Pumps for General Refinery Services, Section 4: Inspection + Tests, Section 5: Guarantee + Warranty;
 - ASME PTC 8.2 Centrifugal Pumps Power Test Code.
-

Acid Pump

Säurepumpe
Pompe à acide

see [Chemical Pump](#)

Acoustics

Akustik
Acoustique

see [Noise in Pumps and Pumping Installations](#)

Adjustable Vane

Einstellbare Schanfel
Pale à pas réglable

see [Blade](#)

Adjustment Gear

Verstellgetriebe
Mécanisme réglable

see [Impeller Blade Pitch Adjustment](#)

Aerofoil Theory

Tragflügeltheorie
Théorie de l'aile portante

The a.t. has gained much importance in the calculation of axial beading (impeller, blade). Whereas radial and mixed flow beading calculations are usually based on the representation of a flow channel (decreased output), the a.t. considers the blade as an aerofoil wing in conjunction with a cascade. Lift coefficients, drag coefficients and other characteristic magnitudes of an aerofoil wing in function of its angle of incidence (flow profile) are mainly obtained experimentally (e.g. in a wind tunnel with threecomponent force measurements) but can also be calculated theoretically.

According to the a.t., one can calculate the frictionless incompressible flow (potential flow) of an aerofoil, of unlimited width, using either a conformal transformation (which leads the problem back to flow around a cylinder) or the method of singularities (where the aerofoil will be simulated by distribution of vortices, sources and sinks along a center line or simply by vortices along the contour of the aerofoil). The method of singularities, in an expanded form, is also useful for cascades.

These procedures had a large use in the calculation of axial hydraulic blading since sufficient computer power was available (flow profile).

The basis of the a.t. for aerofoils of finite width goes back to PRANDTL. According to this theory, the aerofoil is modelled with horse shoe vortices, through which the pressure equalization on the aerofoil tip can be considered.

Affinity Law

Affinitätsgesetz
Loi de similitude

The a.1. is a special case of the model laws for centrifugal pumps, and it states that the throttling curves (characteristic curve) of a centrifugal pump at various rotational speeds, in the conventional representation, can be derived from the following relationship (affinitive representation):

$$Q_1 = \frac{n_1}{n_2} \cdot Q_2 ,$$

$$H_1 = \frac{n_1^2}{n_2^2} \cdot H_2 \text{ or } Y_1 = \frac{n_1^2}{n_2^2} \cdot Y_2$$

where
Q = capacity,
H = head,
Y = specific energy,
n = rotational speed.

Because $P_Q = \rho \cdot g \cdot Q \cdot H$ (pump output) and $P = P_Q / \eta$ (shaft power), and assuming a constant density ρ of the medium pumped and a constant pump efficiency η we have:

$$P_{Q,1} = \frac{n_1^3}{n_2^3} \cdot P_{Q,2} \text{ or } P_1 = \frac{n_1^3}{n_2^3} \cdot P_2$$

Equivalent points with the subscripts 1 and 2 all lie on parabolas having their apex at the origin of the QH or QY coordinate system. A prerequisite for points of equivalence is the similarity of the respective velocity triangles.

As different REYNOLDS numbers (similarity conditions, model laws) are associated with different rotational speeds, a physical similarity of the frictional effects cannot be achieved.

The a.l. is therefore strictly valid only in the case of an in viscid, incompressible, non-cavitating pumped medium.

Aggressive Fluid

Angreifende Flüssigkeit
Liquide agressif

see [Table of Corrosion Resistance](#)

Air Lift

Druckluftheber
Pompe à émulsion d'air

see [Pump Types](#)

Air Pump

Luftpumpe
Pompe à air

see [Self-priming Pump](#)
see [Flow Profile](#)

Air Vessel

Windkessel
Réservoir à air

see [Pressure Vessel](#)

Alternating Current

Wechselstrom
Courant alternatif

A.c., more accurately single-phase a.c. is usually tapped from three-phase current networks; it is tapped in the form of line-to-line (interlinked) voltages L1-L2 or L2-L3 or L3-L1 in the case of networks of two main conductors, in the case of networks with a center conductor N, it is tapped in the form of a phase voltage (voltage to neutral) (terminal designation) from the center conductor and one of the three conductors (L1-N or L2-N or L3-N).

The phase voltage is used for the connection of small single-phase consumers such as electric light bulbs, heating appliances, single-phase motors. It is tapped between the center conductor (N, star point, earth, neutral conductor) and one main conductor L1, L2 or L3 of the three-phase network.

For the operating voltage	U	220	380	500V
is the phase voltage	U _{Str}	127	220	290V

Altitude

Höhenlage
Cote

see [Geodetic Altitude](#)

Amortization

Abschreibung, Amortization
Amortissement

see [Economics](#)

Angle of Incidence

Anstellwinkel
Angle d'incidence

see [Flow Profile](#)

Angle Valve

Eckventil
Soupape d'équerre

see [Valves and Fittings](#)

Angular Velocity

Winkelgeschwindigkeit
Vitesse angulaire

see [Rotational Speed](#)

Annular Casing

Ringgehäuse
Colimaçon torique

see [Pump Casing](#)

Anti-Friction Bearing

Wälzlager
Roulement

The a.f.b., very frequently used to support shafts, consists of the bearing races or bearing discs, the rolling elements, which can be spherical cylindrical, tapered or barrel-shaped (Fig.1) and frequently also of a cage (Fig.2) which prevents contact between adjoining rolling elements.

Depending on the direction of the force, distinction is made (Fig. 3) between transverse (radial) bearings, longitudinal (thrust) bearings and transition designs such as radial deep-groove ball bearings for both radial and axial forces.

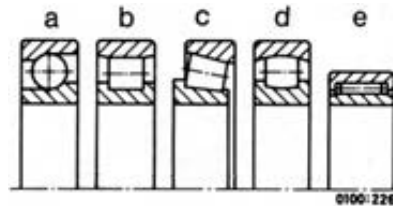


Fig. 1: Rolling element shapes
a) ball; b) cylinder; c) tapered; d) barrel; e) needle

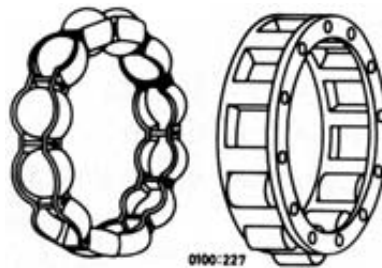


Fig. 2: Anti-friction bearing cages
a) sheet steel cage for balls; b) solid cage for rollers

Fig. 4 illustrates a typical shaft bearing arrangement for a low pressure pump with radial deep-groove ball bearings.

In radial bearings, the rolling elements run between races or rings (ring bearings), and in axial bearings they run between discs (disc bearings).

The friction coefficient of a.f.b.'s is some 25 to 50% lower than for plain bearings. Other advantages of a.f.b.'s include more accurate running because of the closer clearances that can be achieved, smaller space requirements, easier maintenance and lack of problems in lubrication. Because of extensive standardization, interchangeability is assured. Disadvantages include sensitivity to shock loads, noisier running. As a general rule, a.f.b.'s are more expensive than comparable plain bearings.

The load capacity and service life of a.f.b.'s are standardized (DIN 622). A distinction is made between:

- dynamic load capacity* (service life), which is the number of revolutions or operating hours which the bearing will sustain without symptoms of metal fatigue on all the bearing components, and
- static load capacity*, which is the static load force which causes a permanent deformation on the rolling body at the point of contact without impairing the functioning of the bearing. The calculation of the dynamic and static bearing capacity is preferably carried out in accordance with DIN 622 or in accordance with the a.f.b. manufacturer's recommendations.

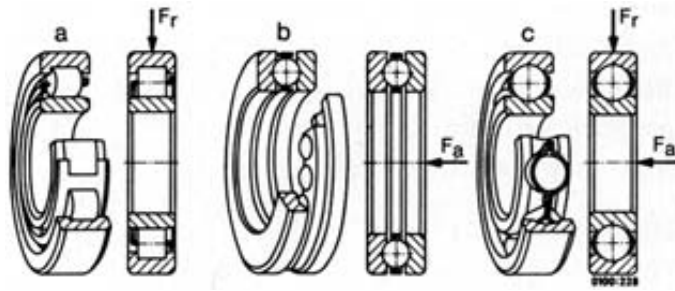


Fig. 3: Anti-friction bearings for various directions of load

- a) cylindrical roller bearing for radial loading;
- b) axial deep-groove ball bearing for axial loading;
- c) (radial) deep-groove ball bearing for both radial and axial loadings

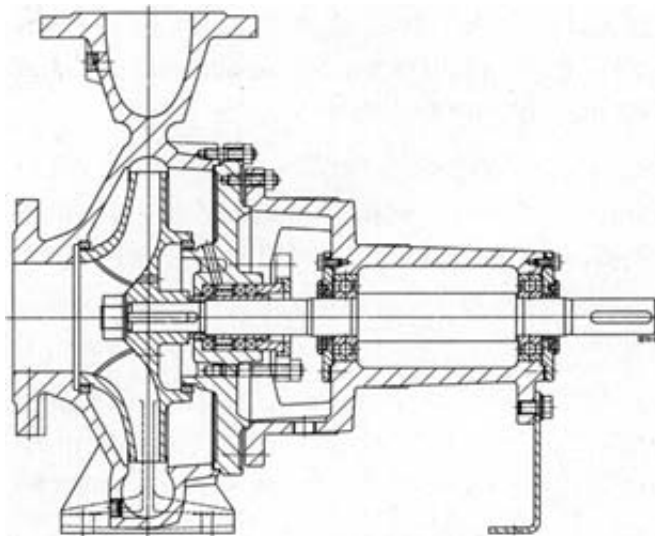


Fig. 4: Anti-friction bearings in a centrifugal pump

As the rolling velocity increases, so do the centrifugal outward forces acting on the rolling elements. As a result of the higher friction losses, the operating temperature of the bearing increases; thus each standard bearing has an upper limit of rotational speed. This limit speed can easily be calculated from formulas supplied by the a.f.b. manufacturers in their catalogues, in function of the a.f.b. type, a.f.b. size, type of lubrication and loading.

Because of the rolling contact motion of the rolling elements there is no complete separation of the rubbing surfaces by a load-bearing film of lubricant in a.f.b.'s, as is the case for plain bearings. Conventional lubrication by oil or grease of the desired consistency is adequate. The bearing manufacturer usually provides lubrication recommendations which include the influence of the operating temperature of the bearing, the ageing stability of the recommended greases and oils under the prevailing operating conditions and generally the necessity of topping up the lubricant fill. Present-day trends are towards "lubrication for life", i.e. a once only initial lubricant fill lasting the life of the bearing.

The following grease types are used for the lubrication of a.f.b.'s:

- a) lime (calcium) soap greases: they are water-repellent. Topping up of the grease fill is required at frequent intervals. Operating temperatures from -20 °C to 50 °C;
- b) soda soap greases: they have good lubricating properties, but absorb water (they are washed away if water penetrates inside the bearing). Operating temperatures from -30 °C to 110 °C;
- c) lithium soap greases: they are water-repellent and can sustain high loadings. Operating temperatures from -30 °C to 125 °C;
- d) complex (mixed base) soap greases (barium calcium or lithium-magnesium-strontium): they are water-repellent, not subject to any restrictions in their application, but more expensive. Operating temperatures from -25 °C to 150 °C.

The following solid lubricants are available for special applications:

- a) graphite: often in conjunction with other carriers or lubricants. Operating temperature below 400 °C, otherwise danger of oxidation;
- b) molybdenum disulphide: available in the trade in powder form or mixed with pastes, greases or oils. Very low coefficient of friction, which sinks even lower with increasing loadings. Can be used in powder form for temperatures from -180 °C to 450 °C.

Aperture Ratio

Öffnungsverhältnis
Rapport des sections

see [Orifice Plate](#), [Standard Orifice](#)

Apex

Scheitelpunkt
Apex

see [Siphoning Installation](#)

Apparent Power

Scheinleistung
Puissance apparente

see [Power](#)

Application Fields for Pumps

Pumpenverwendungsgebiete
Camps d'application des pompes

1. **Process technology**

1.1 *Chemical industry and general process technology*

Chlor-alkali electrolysis

Standard chemical pump, [rubber-lined pump](#), [plastic pump](#), [chemical pump](#), [canned motor pump](#), chemical sump pump.

Acid plants

Standard chemical pump, [rubber-lined pump](#), [plastic pump](#), [chemical pump](#), chemical sump pump, non-clogging impeller pump.

Copper and zinc electrolysis

Standard chemical pump, chemical sump pump, non-clogging impeller pump.

Fertilizers

Standard chemical pump, chemical sump pump, elbow casing pump with axial impeller, non-clogging impeller pump.

Cellulose and paper

Standard chemical pump, rubber-lined pump, low pressure pump, pulp pump, non-clogging impeller pump, circulating pump.

Sugar

Standard chemical pump, rubber-lined pump, low pressure pump, elbow casing pump with axial impeller, non-clogging impeller pump.

Foodstuffs, semi-luxury and beverages

Standard chemical pump, chemical sump pump, low pressure pump, elbow casing pump with axial impeller, non-clogging impeller pump.

High temperature and heating plants, hot water, organic heat transfer media and molten salts

Standard chemical pump, rubber-lined pump, plastic pump, chemical pump, canned motor pump, chemical sump pump, heat transfer pump, low pressure pump.

1.2 Pumps for refineries, petrochemical industry, fibre technology and cryogenics

Pipelines

Vertical can-type pump, high pressure pump, ring section pump, pipeline pump, volute casing pump with radial impeller, tubular casing pump with mixed flow impeller.

Refineries

Refinery pump, process pump, high pressure pump, ring section pump, vertical can-type pump.

Gas wash

Standard chemical pump, chemical sump pump, refinery pump.

Chemical-fibres

Standard chemical pump, chemical sump pump, non-clogging impeller pump, process pump, high pressure pump, ring section pump.

Cryogenics, liquefied gases

Standard chemical pump, canned motor pump, refinery pump, process pump, high pressure pump, ring section pump, vertical can-type pump.

2. Domestic appliances

Central heating plants

Circulating pump, twin pumping set, inline pump, low pressure pump, heat transfer pump.

Water supply

Circulating pump, side channel pump, underwater motor pump, self-priming pump, low pressure pump, peripheral pump, high pressure pump, ring section pump.

Dewatering and sewage disposal

Sewage pump, submersible motor pump, drainage pump, self-priming pump, non-clogging impeller pump, single vane impeller pump.

Gardens and swimming pools

Self-priming pump, sewage pump, submersible motor pump, circulating pump, close-coupled pumping set.

3. Power stations

3.1 Conventional power stations

Boiler feed water

High pressure pump, barrel pump, boiler feed pump, ring section pump, volute casing pump.

Condensate

Vertical can-type pump, ring section pump, condensate pump.

Boiler circulation

Circulating pump.

Cooling water

Tubular casing pump, volute casing pump, concrete tubular casing pump, concrete volute casing pump, cooling water pump.

3.2 Nuclear power stations

Coolant

Reactor circulating pump, volute casing pump, reactor pump.

Feedwater

Ring section pump, reactor feed pump, volute casing pump, reactor pump, boiler feed pump.

Condensate

Vertical can-type pump, ring section pump, condensate pump.

Cooling water

Tubular casing pump, volute casing pump, concrete tubular casing pump, concrete volute casing pump, cooling water pump.

4. Shipbuilding

Cooling water (sea water, fresh water)

Volute casing pump with radial impeller or helical impeller, annular casing pump with mixed flow impeller, elbow casing pump with axial propeller.

Fire-fighting

Volute casing pump with radial impeller, self-priming pump.

Bilge and ballast duty

Volute casing pump with radial impeller or helical impeller, annular casing pump with mixed flow impeller, reciprocating piston pump.

Condensate

Volute casing pump with radial impeller, condensate pump.

Hot water

Volute casing pump with radial impeller, hot water pump, circulating pump, low pressure pump.

Cargo oil

Volute casing pump with radial impeller, cargo oil pump.

Manoeuvring aid

Bow-thruster.

Boiler feed

High pressure pump, ring section pump, boiler feed pump, self-priming pump.

Sanitation

Volute casing pump with radial impeller, self-priming pump.

Brines, lyes

Low pressure pump, chemical pump.

Heat transfer media

Low pressure pump, heat transfer pump.

5. *Hydroeconomy*

Untreated and pure water supply

Underwater motor pump, borehole shaft driven pump, tubular casing pump with axial propeller or mixed flow impeller, low pressure pump, volute casing pump with mixed flow impeller or radial impeller, high pressure pump, ring section pump.

Sewage disposal and treatment

Single vane impeller pump, non-clogging impeller pump, submersible motor pump, sewage pump, volute casing pump with mixed flow impeller, tubular casing pump with semi-axial propeller or axial propeller or helical impeller, high pressure pump.

Irrigation and drainage

Underwater pump, borehole shaft driven pump, tubular casing pump, irrigation pump, low pressure pump, volute casing pump with mixed flow impeller or radial impeller, high pressure pump, ring section pump.

Sea water desalination

Tubular casing pump with mixed flow impeller or axial propeller, vertical can-type pump, circulating pump, low pressure pump, volute casing pump with radial impeller or mixed flow impeller, standard chemical pump.

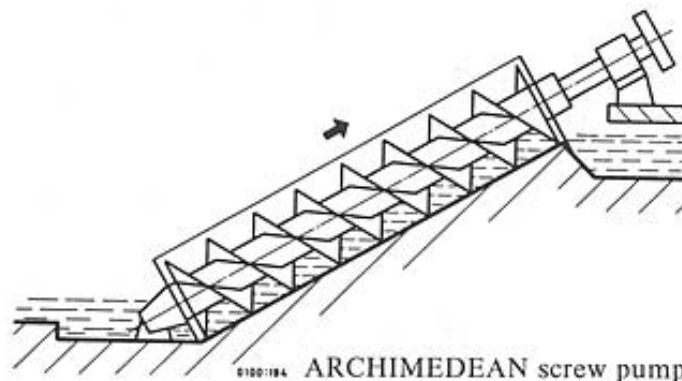
Waterworks

Volute casing pump with radial impeller, high pressure pump, ring section pump, tubular casing pump with axial propeller or mixed flow impeller, underwater pump, single vane impeller pump, non-clogging impeller pump, waterworks pump.

ARCHIMEDEAN Screw Pump

Schneckenentropumpe
Vis d'ARCHIMEDE

The A.s.p. is the pump of a constant pressure elevator which lifts liquids to a given geodetic altitude with the aid overstatic screw (true helicoid in accordance with the irrigation device invented by ARCHIMEDES) (see illustration). The screw is open on all sides, can be several metres long, is usually fabricated from steel plate, and can have up to three leads (thread starts); it rotates in an open semi-circular trough inclined at an angle of 30° approx. The screw scoops up a limited volume of water from the lower water reservoir at each revolution (volume determined by the inclination angle, diameter and lead (or pitch) of the screw) and lifts it at a relatively low rotational speed (less than 100 min⁻¹) to the top end of the trough. Depending on the submerged depth of the bottom end of the screw, the volume flow (capacity) of the A.s.p. adjusts itself within certain limits, by differing depths of liquids in the individual screw passages. This method of operation presupposes that the screw is constantly submerged in the liquid to at least half the diameter of its leading screw passages and does not have too large a clearance gap width in relation to the steel plate or concrete trough, so as to minimize the leakage losses. The lower bearing is an underwater bearing (plain bearing) which must be lubricated by fresh water or grease; the upper bearing also absorbs axial forces and is often an anti-friction bearing.



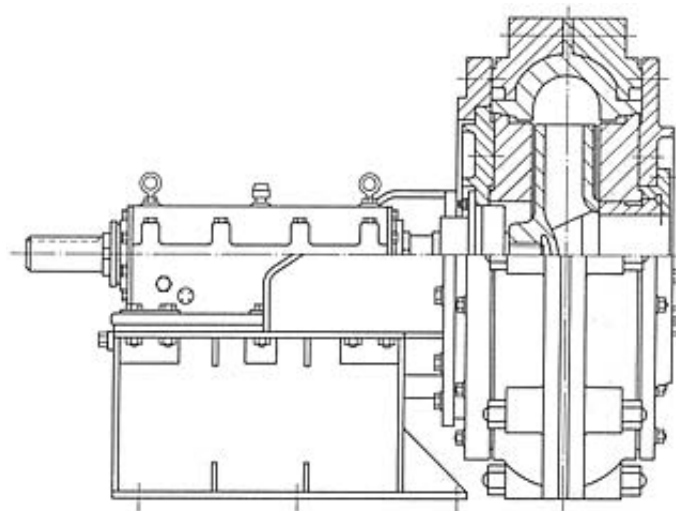
The gearbox is coupled to the upper end of the screw A.s.p.'s are capable of pumping heavily contaminated surface water or sewage, and even sand, if they are specially designed for this purpose. The head is purely geodetic and may reach 6 m at efficiencies between 60 and 80 % Depending on the capacity, the screw diameter can attain up to several metres. A.s.p.'s are simple and adaptable in service and operate economically, but are also often replaced by submersible motor pumps.

Armoured Pump

Panzerpumpe

Pompe à revêtement anti-abrasif

A.p.'s are centrifugal pumps which are exposed to abrasive wear (erosion) to an exceptional degree. A.p.'s are therefore constructed in such a way that the components exposed to wear (rings, bushes, discs, casing inserts, impellers, all types of linings) can easily be replaced by new ones. This method of construction presupposes that the wear parts are reasonably cheap, and consist of materials which are particularly resistant to erosion or can easily be provided with a hard facing (e.g. by metal deposition by welding). The operating requirements also include ease and rapidity of dismantling of the pump (see illustration), and a reliable spare parts service for the renewable wear parts. A.p.'s are used in cases where the medium pumped contains highly abrasive solid particles (abrasion), e.g. suspensions of slag, coal or ore in mining service; flotations, sinter sludges; also used in gravel dredging operations and hydraulic hydrotransport.



Armoured pump

Asynchronous Motor

Asynchronmotor

Moteur asynchrone

The a.m., also known as three-phase motor, has two armature windings, viz. the primary winding, generally executed in the form of a stator winding, and the secondary winding, generally executed as a rotor winding. The energy is transmitted via the rotating field generated in the stator winding (three-phase current), which generates a torque in the rotor winding, when the rotor rotates "asynchronously" in relation to the rotating field. The difference between the synchronous speed and the speed of the a.m. is known as the slip speed. The slip constitutes the quotient of slip speed and synchronous speed; in most cases this slip speed is very small, because of the low rotor resistances, but it increases with increasing load and attains values between 2 and 6 % of the synchronous speed at full load.

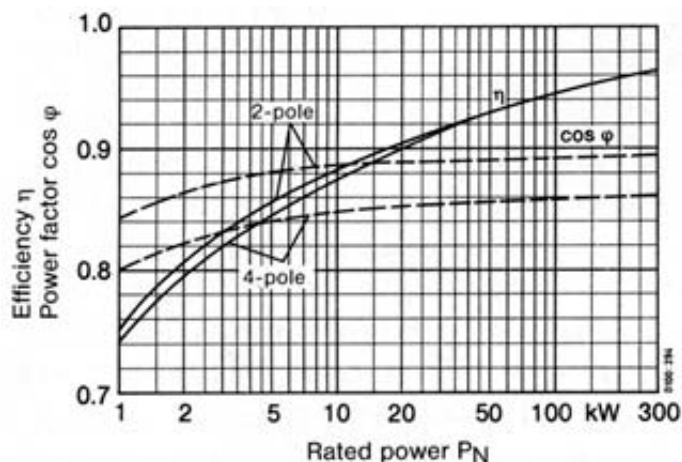


Fig. 1: Typical efficiencies η and power factor $\cos \varphi$ of standard-type IP 54 motors at 50 Hz, as functions of rated motor power P_N

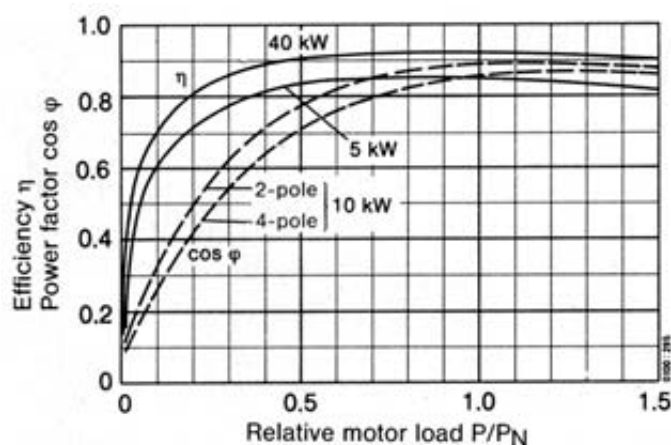


Fig. 2: Efficiency η and power factor $\cos \varphi$ as functions of the relative motor load P/P_N

Figs. 1 and 2 show how the efficiency and the power factor $\cos \varphi$ of an a.m. are influenced by the power output and changes in load.

In accordance with the standards at present in force, certain deviations from the nominal values are permissible in respect of the voltage and frequency of the main supply. When this happens, certain characteristics of a.m.'s vary as indicated in Table 1:

Table 1: Influence of voltage and frequency deviations on asynchronous motors

Characteristics	Voltage		Frequency	
	110% of nominal value	90% of nominal value	105% of nominal value	95% of nominal value
starting and pull out torque	approx. 20% increase	approx. 20% decrease	approx. 10% decrease	approx. 10% increase
synchronous speed	no change	no change	5% increase	5% decrease
full load speed	small increase	small decrease	5% increase	5% decrease
starting current	approx. 10% increase	approx. 10% decrease	approx. 5% decrease	approx. 5% increase

A.m.'s can also be fed from mains supplies with voltages and frequencies which vary from those indicated on the motor rating plate, provided that the voltage and frequency vary proportionally (constant magnetic flux). In such cases, the nominal rating of the motor will also vary proportionally. For example, an a.m. designed for 380 V and 50 Hz can also be run on 440 V (more precisely on 456 V) and 60 Hz, as long as the critical speed and the centrifugal forces (which increase with the square of the speed) allow a speed increase that is proportional to the frequency. The motor's output increases by 20% under such conditions. This is not entirely true in the case of two-pole a.m.'s however, for which the possible increase in output is somewhat lower due to the attendant rise in temperature.

An a.m. cannot be run regardless of the consequences at a constant voltage but at a different frequency. Thus a 50 Hz motor operating on a 60 Hz mains supply at the same voltage only has a starting torque and pull-out torque of some 70% of the original. Its rated output should therefore be reduced to 70% approx. On the other hand, a 60 Hz motor operating on a 50 Hz mains supply at the same voltage would be operating under a magnetic flux of 120% of the nominal value.

This cannot be tolerated in modern motors which are magnetically saturated to the limit, because of the excessive saturation. The magnetizing current and the iron losses would rise excessively and resulted the destruction of the winding insulation by overheating.

Depending on the type of rotor winding, a.m.'s are grouped into squirrel-cage (or short-circuit) rotor a.m.'s and into slip-ring rotor a.m.'s.

In the case of *slip-ring rotor motors*, the three-phase rotor winding is connected via slip rings to adjustable rheostats (resistances) usually executed in the form of liquid starters. Slip-ring rotor motors provide a smooth run up to full operating speed (starting process) without any shock loading to the mains, and also allow a limited speed variation, which is however linked with a higher power loss.

Slip-ring rotor motors are used if the following requirements have to be met: speed adjustment (control), frequent starting of large gyrating masses (moment of gyration), special requirements relating to starting currents (starting process). The starting torque can be increased up to the value of the pull-out torque, depending on the position of the starter and therefore on the magnitude of the resistance connected in series in the rotor circuit. Depending on the motor rating and the >> number of poles, the pull-out torque amounts to 1.8 to 3.5 times the rated torque. The starting current is proportionally the starting torque. The speed can generally be reduced down to 70% of the rated speed by rheostats in the rotor circuit. Of course the losses increase accordingly in this case (order of magnitude: slip X power absorbed). This technique is economically justifiable in the case of centrifugal pump drives (drive) up to medium range outputs.

The rotor winding of *squirrel-cage motors* (sometimes also called short-circuit motor) usually consists of single or double conductor bars, short-circuited at the front and back ends by an annular conductor. The squirrel-cage rotor motor is very simple, reliable in operation and maintenance-free.

The principal construction types of squirrel-cage rotors comprise the single cage rotor, the eddy current rotor and the double rotor.

Single cage rotors are used up to 35 kW in two pole form, and up to 25 kW approx. in four pole form. They are suitable, amongst other things, for direct-on-line starting (starting process); under these conditions,

- the switching-on current amounts to 4.5 to 5.5 times the rated currents
- the starting torque amounts to 1.7 to 2.5 times the rated torque.

If star-delta starter circuit is used, the above values are reduced by 70% approx.

Eddy current rotors (high bar rotors or key bar rotors) are built from approx. 25 kW up to the highest ratings. They can be adapted to most requirements and are particularly suitable for centrifugal pump drives by virtue of their torque characteristics (starting process). They can be started direct-on-line or in star-delta starter circuit. In the case of direct-on-line starting,

- the starting current amounts to 4 to 6 times the rated current
- the starting torque amounts to 0.9 to 1.8 times the rated torque.

In the case of star-delta starter circuit, the above values are reduced by 70% approx.

Double cage rotors are used in the 20 to 200 kW range, particularly in cases where a high starting torque is required (starting process). Their torque characteristic is however not suitable for centrifugal pump drives, particularly in the case of star-delta starter circuit, because the pullout torque is too low. This means that if the pump is started up against an open discharge valve in the star circuit, the >> rotational speed it can attain is insufficient, and an unduly high current surge often occurs on switching over to delta. However, double cage rotors for both modes of starting are eminently suitable for many other types of drive. In the case of direction-line starting,

- the starting current amounts to 4.5 to 5.6, times the rated current,
- the starting torque amounts to 1.7 to 2.6 times the rated torque.

In the case of star-delta starting, the above values are reduced by 70% approx.

Quite independently of the rotor construction type, the following characteristics apply to all *squirrel-cage rotor motors*. the rotational speed varies only slightly when the load varies (similar to the shunt characteristics of the direct current motor). They can be made to run at different speeds by pole changing (number of poles) with a special winding. By means of frequency change and therefore proportional voltage change, a proportional speed change (control) at constant torque can be achieved. The direction of rotation can be reversed by crossing over two of the power supply lead connections.

With regard to contact with water, one can differentiate between methods of construction of a.m. with a short-circuited rotor (see also Table 2):

- **Dry motor** with several types of protection against water.
- A **submersible motor** is either entirely or partially submerged and is preferably mounted vertically. It is cooled by the surrounding medium. Its trademark is the wet outside housing (submersible motor pump). Depending on the moisture on the inside and the depth of submerging, one can differentiate:
Oil or air-filled submersible motors for slight to medium submersion, for example in sewage submersible motor pumps (Figs. 3 and 4 under sewage pump).
Wet submersible motors (submersible motors, abbreviated U-motors) that are wet on both the inside and the outside, and can be built to operate at any depth. Due to uses in cases of underwater motor pumps in drilling shafts, these pumps must have the smallest possible diameter, however, they are built longer. Submersible motors are wet rotor motors and can be constructed with wet stator windings (waterproof isolated by synthetic materials) or with the aid of a can (canned motor pump) with dry winding.
- The **wet rotor motors** of a glandless pump is filled with liquid, its housing outside is, however, not wetted (as opposed to submersible pumps). It has a fluid-lubricated bearing (plain bearing) and forms, with the pump, a hermetically sealed pump unit (a so-called glandless pump). Even this type of pump can be built with wet stator windings or with a canned rotor (canned motor pump), and is often applied for circulating pumps.

Table 2: Environment-dependent nomenclature of asynchronous motors

Interior environment of the		Exterior environment of the casing					
rotor	winding	dry	wet = submersible motor				
dry	dry	<table><tr><td>dry motor (with or without protection against water)</td><td>dry submersible motor</td></tr><tr><td>wet rotor motor of a glandless pump</td><td>wet submersible motor</td></tr></table>		dry motor (with or without protection against water)	dry submersible motor	wet rotor motor of a glandless pump	wet submersible motor
dry motor (with or without protection against water)	dry submersible motor						
wet rotor motor of a glandless pump	wet submersible motor						
wet = wet rotor motor	dry = canned motor						
	wet = wet motor						

Atmospheric Pressure

Atmosphärendruck
Pression atmosphérique

The a.p. is the absolute barometric pressure at the installation site of the pump, averaged over a long period of time. In most cases, the a.p. can be taken as equal for the elevations of both the inlet and outlet cross-sections of the pumping plant. As the altitude increases above mean sea level (NSL), the a.p. decreases (see Table). The a.p. should only be substituted for the absolute barometric pressure (which varies with location and with time) in rough

calculations of the head, NPSH (net positive suction head). The SI unit of a.p. is 1 Pascal (Pa); in centrifugal pump technology however, the unit 1 bar or 1 mbar = 1 hPa is generally used.

Table: Atmospheric pressure in function of altitude starting at 1013 mbar as datum ("physical atmosphere") at 0 meters above mean sea level (NSL)

Altitude above NSL	Atmospheric pressure	
	Pa	mbar = hPa
0	101300	1013
500	95500	955
1000	89900	899
2000	79500	795

Automatic Recirculation Valve

Freilauf-Rückschlagventil

Soupape de retenue et de décharge

see Valves and Fittings

Automatic Switch

Automatischer Schalter

Conjoncteur

A.s. effect the switching operation automatically by means of electromagnetic contactors (electrical switchgear), the coil current of which is controlled via quick-action or permanent contactors.

Availability

Verfügbarkeit

Disponibilité

The a. is a characteristic of the reliability of a system or unit, or even of a machine. It is defined as the percentage of operating hours plus stand-by within a given period of time. The a. determines the probability that a machine is able to operate properly at a given point in time.

Axial Force

Axialkraft

Force axiale

see Axial Thrust

Axial Impeller

Axialrad

Roue axiale

see [Impeller](#)

Axially Split Casing

Längsteilung

Corps à joint longitudinal

A a.s.c. of a [centrifugal pump](#) is a [pump casing](#) split along the shaft centralize (Figs. 8 and 11 under pump casing), in contrast to a [radially split casing](#).

Axial Pump

Axialpumpe

Pompe axiale

see [Propeller Pump](#)

Axial Thrust

Achsschub, Axialschub

Poussée axiale

The a.t. in [centrifugal pumps](#) is the resultant of all the axial forces F acting on the pump rotor. In the case of a single stage centrifugal pump, the following axial forces act on the rotor (Fig. 1):

1. the axial pressure forces acting on the impeller shroud at the suction side (subscript s) and on the discharge side (subscript d) produce the axial force on the impeller $F_1 = F_d - F_s$;
2. the momentum force F_J (theorem of momentum, [fluid dynamics](#)) $F_J = \rho \cdot Q \cdot \Delta v_{ax}$

where:

Q = [capacity](#) (rate of flow),

ρ = [density of pumped medium](#),

Δv_{ax} = difference between axial components of [absolute velocity](#) at impeller inlet and outlet;

3. the resultant pressure forces arising from the static [pressures](#) in front of and behind the [shaft seal](#) (subscript Wd) on the relevant shaft cross-section A_{Wd}
 $F_{Wd} = A_{Wd} \cdot \Delta p_{Wd}$;
4. special axial forces, e.g. during the [starting process](#), sheathe vortex conditions in the impeller shroud clearance spaces ([impeller side friction](#)) vary;
5. other axial forces, such as the force of the rotor weight F_G on non-horizontal centrifugal pumps, F_{mech} caused by magnetic pull in the electric motor, e.g. in [close-coupled pumping sets](#), etc.

The a.t. component $F_1 + F_J$ of closed [impellers](#) without balance holes (that is, impellers with suction side shrouds) is:

$$F_1 + F_J = \alpha \cdot \rho \cdot g \cdot H \cdot D_{2m}^2 \cdot \frac{\pi}{4}$$

with:

- α = a.t. coefficient,
 ρ = density of pumped medium,
 g = gravitational constant,
 H = head
 D_{2m} = medium impeller diameter = $(D_{2i} + D_{2a})/2$ (Fig. 2).

The a.t. coefficient α is substantially dependent on the specific speed n_q .

For radial and mixed flow impellers one can use the following equation in the area of $6 < n_q < 130 \text{ min}^{-1}$:

$$\alpha = 0.5 (D_{sp}/D_{2m})^3 + 0.09$$

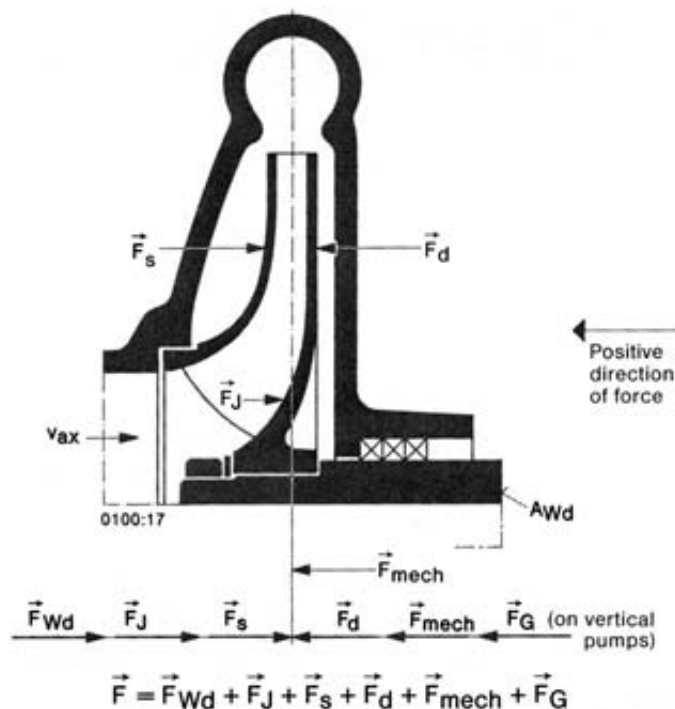


Fig. 1: Axial forces in a single stage centrifugal pump

with:

- D_{sp} = diameter of the slit seal on the suction side of the impeller inlet shroud (Fig. 2).

This equation for α is valid for capacities Q ranging from $0.8 \cdot Q_{opt}$ to $1.0 \cdot Q_{opt}$ and for clearance gap widths $s = 0.1 \text{ mm}$; with double the clearance gap width the α increases by 8.0 %.

In the case of multistage pumps equipped with diffusers (e.g. boiler feed pump), F_1 will vary very markedly depending on the axial location of the impeller in relation to the diffuser.

In the case of open type radial impellers, which have no impeller shroud on the suction side, F_s is much lower than on closed impellers, and therefore F_1 is higher. Open impellers with cutouts in the impeller shroud between adjoining impeller vanes (Fig. 13 under impeller) develop a lower pressure force F_d and consequently a lower axial force F_1 than impellers with a full shroud on the discharge side.

With axial propellers (Fig. 5 under impeller), the a.t. coefficient α is nearly equal to the degree of reaction r_{th} . The a.t. can then be roughly calculated by:

$$F_1 + F_J \approx 0.8 \cdot \rho \cdot g \cdot H \cdot D_A^2 \cdot \frac{\pi}{4}$$

with:

D_A = outside diameter of propeller

The following proportionality applies to the constituent F_1 of a.t. of geometrically similar pumps:

$$F_1 \sim n^2 \cdot \rho \cdot D_2^4$$

where:

n = rotational speed,

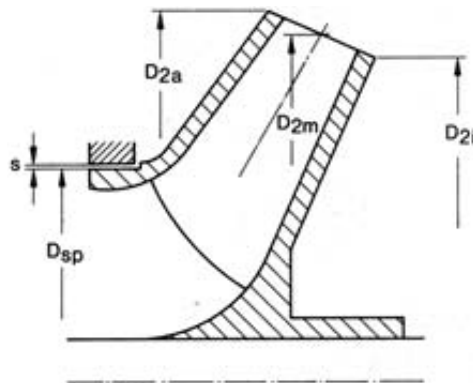


Fig. 2: Non-balanced impeller with conical impeller outlet area

ρ = density of pumped medium,

D_2 = max. impeller diameter

The proportion of the total thrust listed under 1. above, and consisting of the pressure forces F_d and F_s , is higher during the starting process than in the steady state operating condition. The reason for this is as follows: the medium in the impeller shroud clearance spaces only starts to rotate gradually during the starting process as a result of the impeller side friction, and it is entrained on the one hand by the action of the impeller shrouds and retarded on the other hand by the braking effect of the stationary casing surfaces. The mean angular velocity (rotational speed) of the rotating medium attains on average approx. half the angular velocity of the impeller. In addition, the inward-directed gap flow in the clearance space at the suction (outer) end further excites the impeller side turbulence, as a result of CORIOLIS accelerations. In the discharge (inner) end shroud clearance space of multistage pumps, this process is reversed as a result of the outward-directed gap flow, i.e. the vortex motion is damped, and F_d increases, therefore F_1 increases also.

Axial thrust balancing (or compensation) can be achieved in several ways:

1. mechanically by the complete absorption of the at. by a thrust bearing (plain bearing, anti-friction bearing),

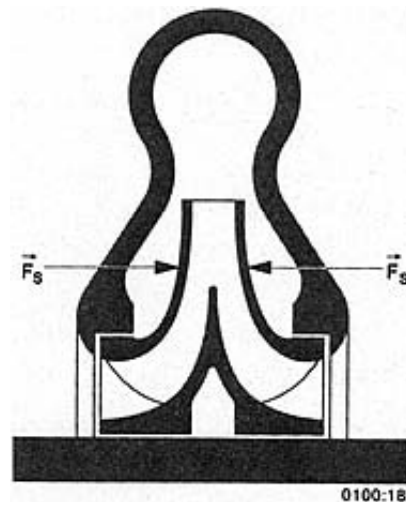


Fig. 3: Axial force, double suction impeller arrangement

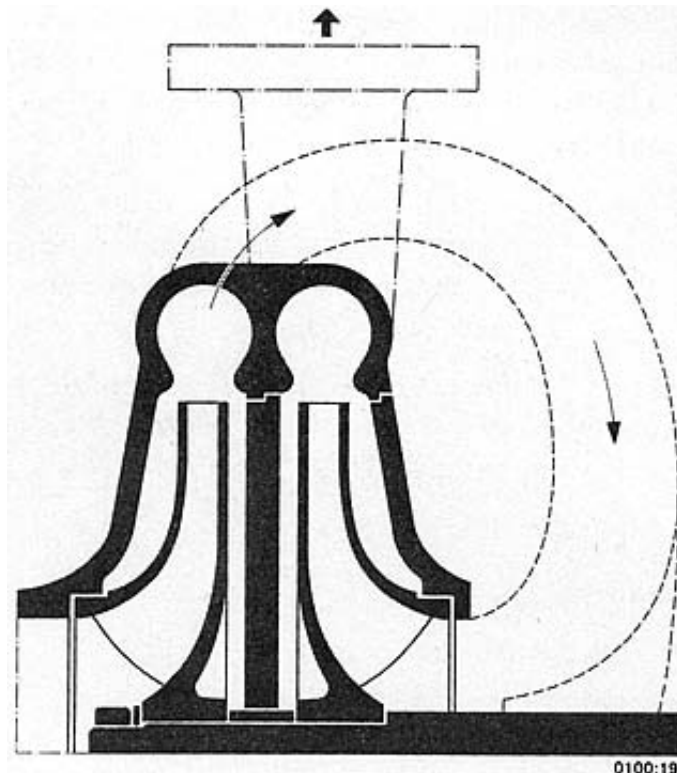


Fig. 4: Axial force, two stage back-to-back impeller arrangement

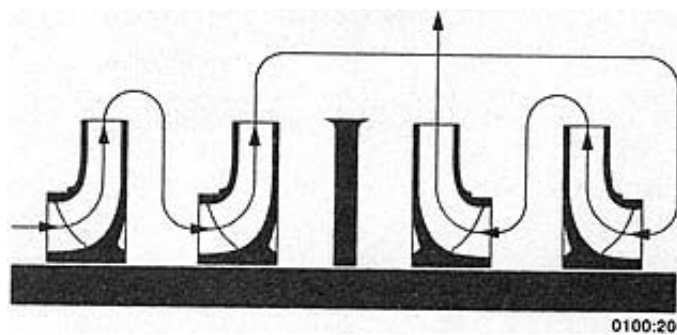


Fig. 5: Axial force, four stage pipeline pump with two sets of two stage series-coupled opposed impellers

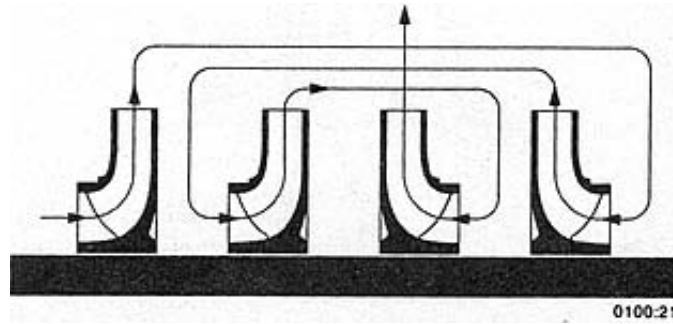


Fig. 6: Axial force four stage pipeline pump, with sets of parallel-coupled opposed impellers

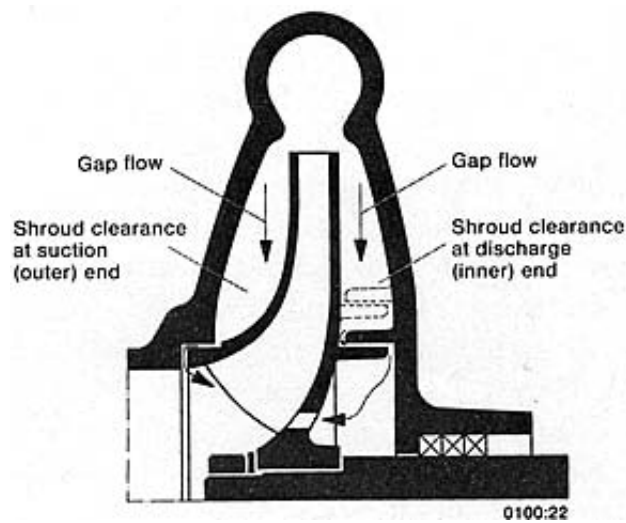


Fig. 7: Axial force, single stage centrifugal pump with sealing gap at discharge end and balance holes

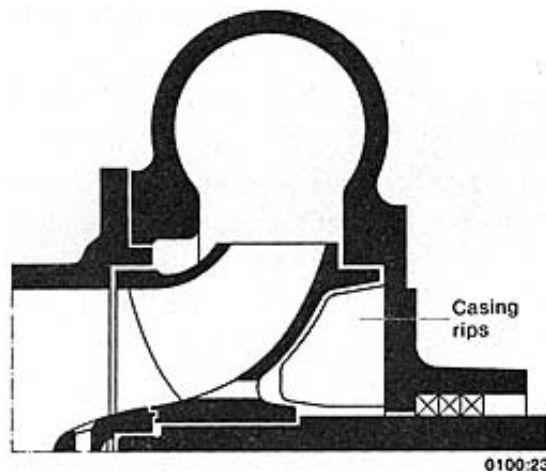


Fig. 8: Axial force, single stage centrifugal pump with sealing gap at discharge end balance holes and casing ribs within the sealing annulus

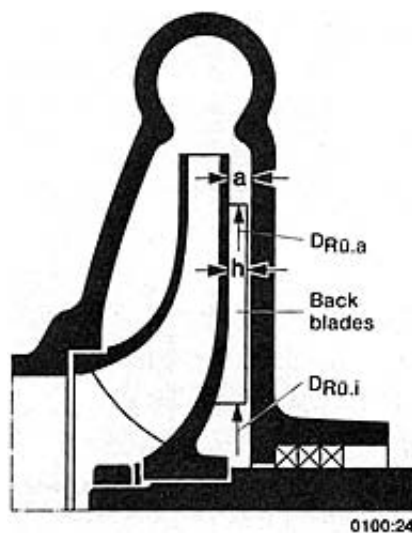


Fig. 9: Axial force single stage centrifugal pump with back shroud blades

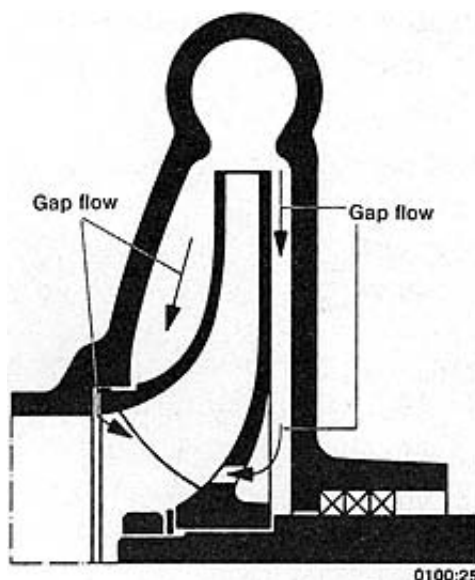


Fig. 10: Balancing the axial thrust in a single stage centrifugal pump with balance holes only

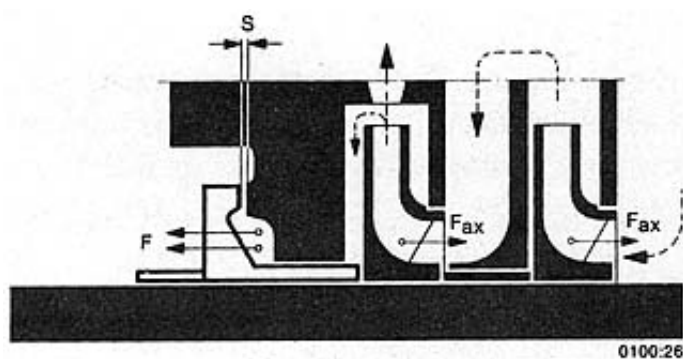


Fig. 11: Balancing device with balance disc

2. by design features (see Figs. 3 to 6) by the back-to-back arrangement of impellers or stages (back-to-back impeller pump). This arrangement does not fully balance the a.t., and the residual a.t. must therefore be absorbed by a thrust bearing (see Figs. under pipeline pump, booster pump),
3. balancing or reduction of the a.t. on the individual impeller by:
 - throttling action (sealing ring at discharge end, combined with balance holes, Figs. 7 and 8) or
 - dynamic action (back shroud blade, Fig. 9, balance holes only, Fig. 10),
4. balancing of complete rotating assembly by balancing device with:
 - automatic balancing by balance disc and balance disc seat (Fig. 11),
 - partial balancing by balance piston (Fig. 12, the residual thrust being absorbed by a thrust bearing),
 - balancing by a double piston (Fig. 13, additional thrust bearing).

The following comments apply to these four methods of balancing the a.t.:

Method 1: The absorption of the a.t. by an anti-friction bearing is the most efficient solution, but the efficiency gain may be nullified by complicated thrust bearings and the price advantage nullified by the absence of special devices.

Method 2: The simplest means of balancing the a.t. almost entirely are

- a double flow impeller arrangement (multisuction pump, impeller) as illustrated in Fig. 3;
- a two stage back-to-back impeller arrangement (multistage pump, back-to-back impeller pump, impeller) as illustrated in Fig. 4;
- a multistage series-coupled opposed impeller arrangement as illustrated in Fig. 5, or

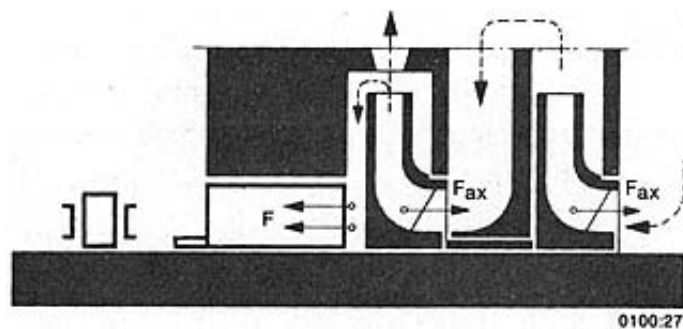


Fig. 12: Balancing device with balance piston and thrust bearing

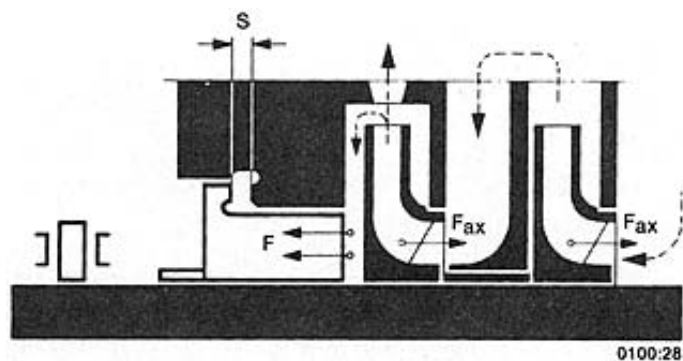


Fig. 13: Balancing device with double piston and thrust bearing

- a parallel-coupled opposed impeller arrangement as illustrated in Fig. 6, an arrangement sometimes adopted for pipeline pumps.

In the impeller arrangement according to Fig. 5, the max. a.t. which can arise amounts to twice the stage a.t., if two stages happen to cavitate (net positive suction head, the cavitation criteria under 6.) due to operating conditions in the installation. In the arrangement according to Fig. 6, the max. a.t. arising under such conditions is the thrust of a single stage. Such pumps therefore require the additional incorporation of a thrust bearing.

Method 3: Fig. 7 illustrates the oldest method adopted for balancing the a.t. The pressure is reduced in a chamber equipped with a throttling gap, usually down to the pressure reigning at the impeller inlet- the pressure equalization in today's pumps is effected simply via so-called balance holes (Fig. 7) in the impeller. This type of balancing can lead to variations in the a.t. when the inlet conditions vary, caused by instability in the flow at the mouth of the balance holes on the blade side of the impeller. The dotted lines in Fig. 7 indicate how the force exerted by the weight of the rotor of vertical pumps, acting in the direction of the suction end, can be compensated by displacing the location of the discharge side sealing gap radially. Fig. 8 illustrates a special construction with casing ribs within the sealing gap on the discharge side. These ribs exert a braking effect on the rotating fluid (vortex in the impeller shroud clearance space) and thus help to reduce F_1 ; this feature is mainly used on mixed flow impellers, where for design reasons the location of the sealing gap is fixed. The dynamic action on the magnitude of the a.t. consists in influencing the angular velocity (rotational speed) of the impeller side space eddies. The increase in the rotational speed of the impeller side space eddies is effected usually by means of back shroud blades, arranged radially on the back plate of the impeller (Fig. 9). The higher mean angular velocity of the impeller side space eddies results in a lower static pressure on the impeller shroud at the discharge (inner) end. This results in a lower F_d and thus a lower F_1 .

Most radial back shroud blades (Fig. 9) are designed with diameters $D_{Rü,a}$ and $D_{Rü,i}$, side space depths a and blade height h and number of blades z which vary according to requirement.

The power absorbed by this method of a.t. balancing depends on the sizing of the back shroud blades; in practice, the pump efficiency is reduced by up to three points by back shroud blades. The angular velocity of the impeller side space eddy can also be increased by a helical inward-guided path of the gap flow, as illustrated in Fig. 7. The excitation effect caused by the CORIOLIS acceleration previously mentioned is exploited for impeller balancing purposes in the arrangement illustrated in Fig. 10 which only has balance holes but no second sealing ring on the discharge side.

All hydraulic balancing devices illustrated in Figs. 7 to 10 are fully effective at a certain capacity (Q as close to Q_{opt} as possible) only; under all other duty conditions, in particular under off design conditions (operating behaviour), considerable residual forces develop, which have to be balanced by the thrust bearings.

Method 4: There are three main types of balancing device:

1. balance disc with balance disc seat and recirculation of the balance water stream (Fig. 11),
2. balance piston with recirculation of the balance water stream and thrust bearing (Fig. 12),
3. double piston with recirculation of the balance water flow and thrust bearing (Fig. 13).

In all three types, the balance water flow (which acts as a by-pass) is returned to the pump suction branch (after being cooled if necessary) or to the suction vessel of the centrifugal pump. In the case of a balance disc (Fig. 11), the gap flow (clearance gap loss in centrifugal pumps) is low because the axial gap s which adjusts itself of its own accord remains very narrow, and consequently the pump efficiency decreases only slightly; however in the case of a balance piston (Fig. 12), the radial clearance gap widths are greater, and the gap flow much higher, causing a greater drop in efficiency, which is further decreased by the necessity of fitting a thrust bearing. Labyrinth type slit seals are fitted to minimize the high gap flow. The so-called double piston balancing devices (Fig. 13) enable an additional thrust bearing to be fitted (thanks to their higher axial play) which is mainly designed to prevent mechanical seizure (scuffing) of the balancing device. Scuffing can arise in certain cases during start-up (starting process), during operation at extreme overloads (caused by plant operation) (operating behaviour) or when cavitation occurs. The position of the pump rotor and consequently the degree of wear of the balancing device or of the thrust bearings can be ascertained by means of a simple seizure check indicating device which can be observed whilst the pump is running.

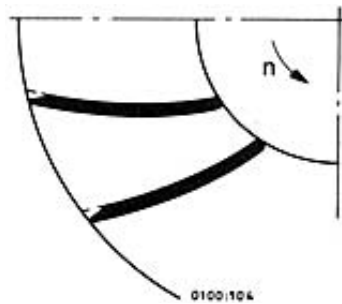
B

Back Filing

Hinterfeilen

Dégagement à la lime

B.f. is the subsequent machining (see illustration) of impeller vanes (blade) on centrifugal pumps of low specific speed for the purpose of slightly increasing the head at the design duty point. When b.f. the concave side of double-curvature impeller vanes (see illustration, blade), a somewhat steeper blade outlet angle is formed. As a general rule, the characteristic curve is displaced upwards only in the region of maximum efficiency by this procedure; this results in an increase in head in this region of the order of 3% (this depends to a large extent on the nature of the b.f. and on the type of impeller). In some cases, b.f. also improves the efficiency slightly.



Back filed blades of a. radial impeller

Back Pull Out Pump

Herausziehbare Pumpe

Pompe à rotor démontable par le haut

The ability to pull out a pump depends almost entirely on the installation and removal of the rotating parts (impeller, pump shaft) of a vertically installed tubular casing pump. The entire rotating parts can be removed from the pump casing, sometimes even the diffuser and wear ring, without removing the pump casing from the piping or the foundation base (see illustration under tubular casing pump).

Back Shroud Blade

Rückenschaufel

Aube dorsale

The b.s.b. is a radial narrow blade on the back plate (viewed from the direction of flow) of a radial impeller or of a mixed flow (impeller), designed to balance the axial thrust.

Back-to-Back Impeller Pump

Gegenläufige Pumpe

Pompe à flux opposé

The axial thrust arising during operation of a centrifugal pump can be very effectively balanced out by arranging several impeller pairs in back-to-back on the common pump shaft, apart from the various other devices (balancing device) adopted to compensate the axial thrust. The axial flow towards the impeller therefore takes place in opposite direction, either in pairs or groups of impellers (opposed flow).

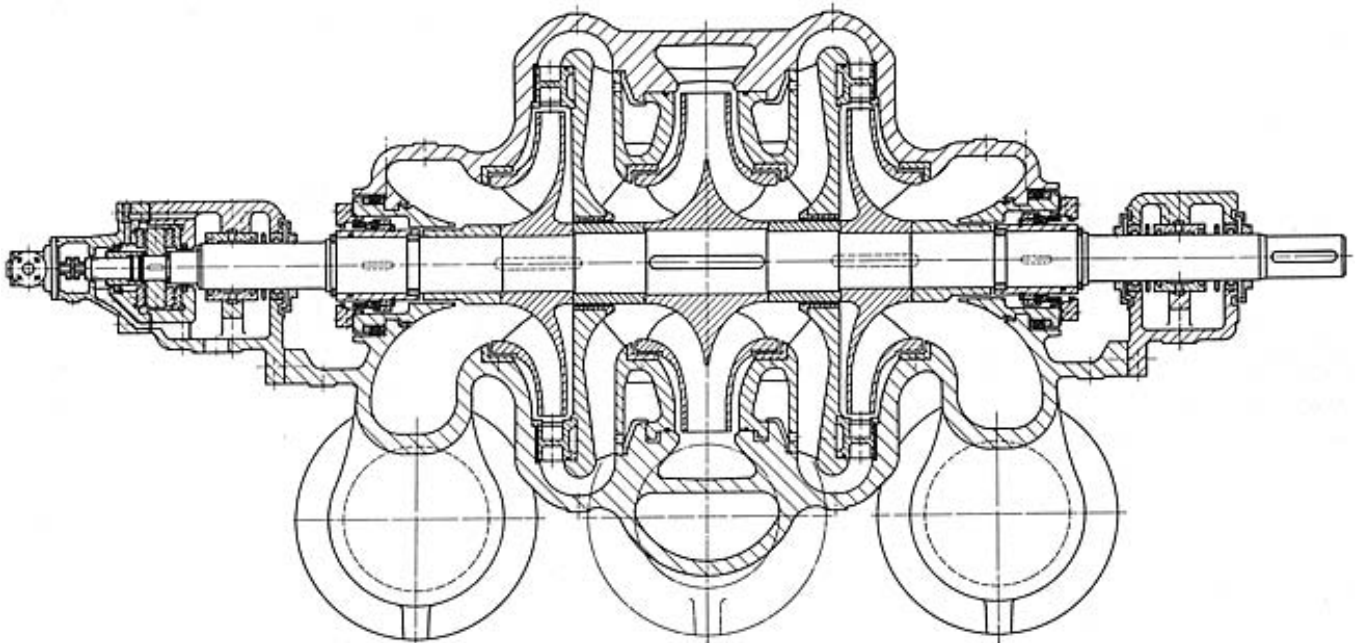


Fig. 1: Double suction two stage volute casing pump (waterworks pump)

This type of construction is particularly attractive in the case of multisuction pumps, especially centrifugal pumps fitted with a double flow impeller (Fig. 1). But it can also be adopted on multistage pumps, e.g. pipeline pumps or boiler feed pumps; this requires the provision of crossover pipes (Fig. 2), and hence a pump casing axially split at shaft centreline. The casing of such multistage centrifugal pumps tends to become rather long and relatively expensive, so that b.t.b.i.p.'s with more than one pair of impellers are only adopted in cases where special operating conditions (as a consequence of failure during operation) or when the ensuing damage (e.g. if a balancing device rails) justify the increased cost.

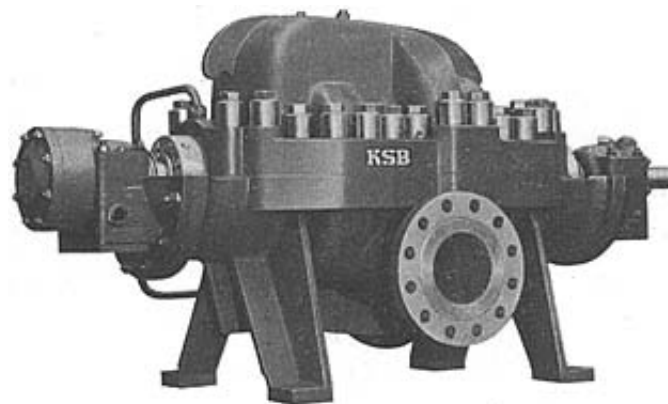


Fig. 2: Two stage single suction pipeline pump with cross-over pipes, see also Fig. 2 under refinery pump

Balance Hole

Entlastungsbohrung
Trou d'équilibrage

see Axial Thrust

Balance Water Flow

Entlastungsstrom
Débit d'équilibrage

B.w.f. is the necessary flow volume QE for the operation of the balancing device of a centrifugal pump, through the clearance gap width between the rotating parts of the balancing device and the stationary parts. The b.w. f. increases clearance gap losses in centrifugal pumps, but allows for an economical method of balancing the axial thrust.

Balancing

Auswuchten
Équilibrage

see Unbalance of Centrifugal Pumps

Balancing Device

Entlastungseinrichtung
Dispositif d'équilibrage

The b.d. of a centrifugal pump is designed to absorb the axial thrust of the pump rotor via a balance disc and balance disc seat. Constructions incorporating a balance piston require a thrust bearing to absorb the residual thrust. Another type of b.d. incorporates a double piston with an additional thrust bearing (axial thrust).

With the pump in operation, the b.d. requires a certain balance water flow, the clearance gap width of which separates the rotating from the nonrotating parts of the b.d. B.d.'s are adopted especially in the case of very high axial thrusts, e.g. on very high pressure pumps (boiler feed pump).

Ball Passage

Kugeldurchgang
Passage entièrement dégagé

see Impeller

Barometric Pressure

Barometerstand, Luftdruck
Pression atmosphérique, pression barométrique

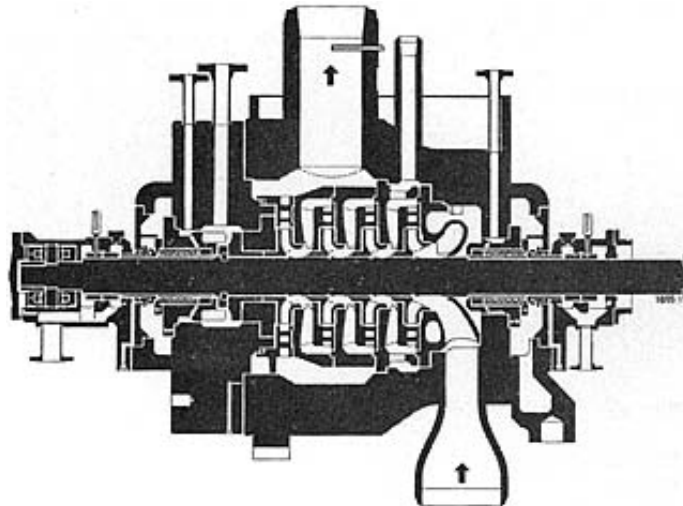
The b.p. Pb (suffix b = barometric) at the site location of a centrifugal pump is subject to large fluctuations caused by meteorological conditions. The legal unit of b.p. is 1 bar, and the unit adopted in meteorology is 1 mbar = 1 hPa (hectopascal). When carrying out an accurate investigation of the suction behaviour of a centrifugal pump, Pb should be measured at hourly intervals. For the preliminary calculation of the suction behaviour of a fluid flow machine the atmospheric pressure relating to the altitude of the pump site is often adopted.

Barrel Pump

Topfgehäusepumpe
Pompe barrel

B.p.'s (also known as double casing pumps or jacket casing pumps) are centrifugal pumps surrounded by a barrel-shaped casing (see illustration). The barrel is equipped with a suction branch and sometimes also with an outlet branch, and is tightly bolted together with a radially split cover. The drive shaft passes through the cover and is sealed by a shaft seal. When the pump is dismantled, the barrel casing can remain in situ, attached to the pipings and to the pump foundation.

B.p.'s are usually multistage horizontal or vertical pumps; they are used as high pressure and super pressure pumps particularly as boiler feed pumps (see illustration). The b.p. should not be change by mistake with the so-called vertical can-type pump (condensate pump, see Fig. 2 under refinery pump).



Barrel pump (full load boiler feed pump)

Baseplate

Grundplatte
Socle commun

see Installation of Centrifugal Pumps

Bearing

Lager
Palier

see Anti-Friction Bearing, Plain Bearing

Belt Drive

Riementrieb
Transmission par courroie

General explanations

B.d.'s are in general friction drives, in which forces are transmitted from belt to pulley or vice versa by friction. Both V-belts and flat belts (Figs. 1 and 2) are used, and in the case of positive-locking b.d.'s, toothed belts are used. B.d.'s play an important part in centrifugal pump technology, e.g. because of the simple adaptation of a required pump speed to fixed step drive speeds, and also because of the flexibility of b.d.'s. The main geometric dimensions relating to b.d.'s are shown in Figs. 1 and 2. Table 6 lists the main formula symbols, designations and units used in conjunction with b.d.'s.

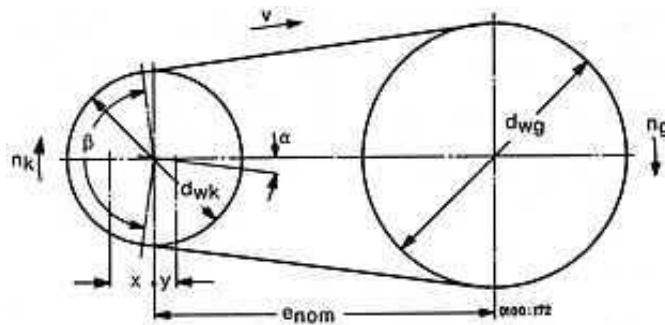


Fig. 1: Diagram of belt drive

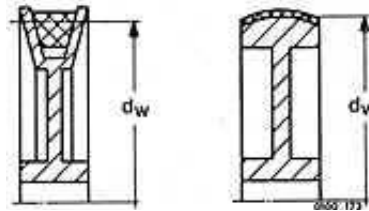


Fig. 2: Effective pulley diameter for V-belts and flat belts

In the engineering industry, the narrow V-belts (wedge belts) with profiles designated by the ISO symbols SPZ, SPA, SPB and SPC, in accordance with DIN 7753, are gaining increasing popularity. The belt manufacturers can provide details of these profiles. The values quoted here apply to belts in accordance with DIN 7753.

1. **V-belt drive** (symbols, designations and units [see table 6](#))

In the case of flat belts, synthetic fibre belts with chrome leather contact faces have largely superseded the classic leather belt. These flat belts have the advantage over narrow V-belts that they can still be used for $v > 40$ m/s (in special cases up to $v = 100$ m/s).

These belts are neither standardized nor are the data provided by the manufacturers uniform. The data given below for transmittable power, bending frequency etc. are attainable values. For practical applications, the drive should be sized in accordance with the data supplied by the manufacturer, or one should checkup what extent the proposed belt complies with the characteristics listed here.

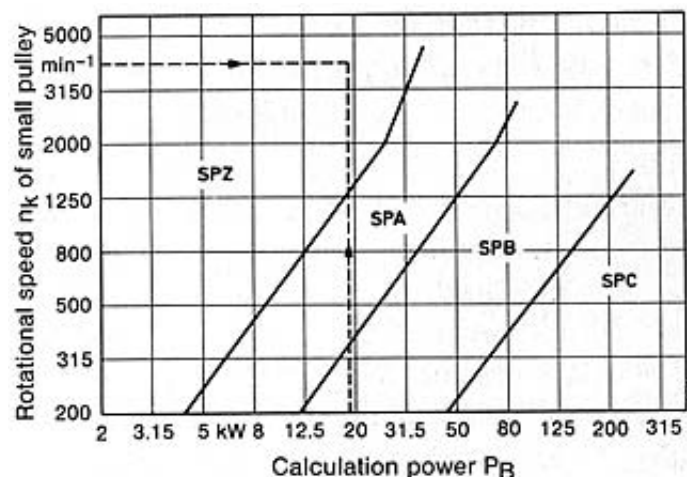


Fig. 3: Selection of profile in accordance with power and speed

Available:

driven engine	V-belt drive for increase in speed $n_g < n_k$ three-phase motor with starting torque up to 1.8 times rated torque (<u>starting process</u>)
<u>drive rating</u> P_M	$P_M = 18.5 \text{ kW}$
rated speed n_g	$n_g = 2860 \text{ min}^{-1}$
effective diameter d_{wg} of drive (large) pulley (V-belt) (manufacturer's recommendation on account of bearing loading)	$d_{wg} = 224 \text{ mm}$
driven equipment	<u>centrifugal pump</u>
<u>shaft power</u> P	$P = 16 \text{ kW}$
pump rotational speed n_k	$n_k = 4000 \text{ min}^{-1}$
daily operating time	approx. 14 hours
operating conditions	normal

Determine:

operating factor c_2 from Table 1	$c_2 = 1.2$
calculation power $P_B = P \cdot c_2$	$P_B = 16 \cdot 1.2 = 19.2 \text{ kW}$
selection of belt profile in acc. with Fig. 3	SPZ
transmission ratio $i = n_g/n_k = d_{wk}/d_{wg}$	$i = 2860/4000 = 0.715$; $1/i = 1.40$
effective diameter d_{wk} of the driven (small) pulley (V-belt) $d_{wk} = d_{wg} \cdot i$.	$d_{wk} = 224 \cdot 0.714 = 160 \text{ mm}$.
The pulley diameters in acc. with DIN 2211 (Figs. 4 and 5) should preferably be adopted; $d_w < d_{w \text{ min}}$ should be avoided.	
Distance between shaft centres e' (preliminary) recommended: $e'_{\text{max}} = 2 (d_{wg} + d_{wk})$, $e'_{\text{min}} = 0.7 (d_{wg} + d_{wk})$, selected (taking the pump and motor dimensions into account).	
Effective length L_w of V-belt (approximate) $L_w \approx 2 e' + 1.57 (d_{wg} + d_{wk}) + (d_{wg} - d_{wk})^2 / (4 e')$	$e'_{\text{max}} = 2 (224 + 160) = 768 \text{ mm}$ $e'_{\text{min}} = 0.7 (224 + 160) = 268.8 \text{ mm}$ $e' = 550 \text{ mm}$. $L_w \approx 2 \cdot 550 + 1.57 (224 + 160) + \frac{(224 - 160)^2}{4 \cdot 550} =$ 1705 mm
closest standard lengths L_r in accordance with <u>Table 3</u> or DIN 7753, Part 1 (1988)	$L_r = 1800 \text{ mm}$

distance between pulley centres e (finally)
 for $L_r > L_w$: $e \approx e' + (L_r - L_w)/2$
 for $L_r < L_w$: $e \approx e' - (L_w - L_r)/2$.

Adjustability of distance between shaft centres (see Fig. 1)

for laying of V-belts: $y \geq 0.015 \cdot L_r$
 for tensioning of V-belts: $x \geq 0.02 \cdot L_r$.

$$\text{V-belt speed } v = \frac{d_{wk}}{1000} \cdot \pi \cdot \frac{n_k}{60}$$

bending frequency of V-belt $f_B = 2 \cdot 1000 \cdot v/L_r$

strand inclination $(d_{wg} - d_{wk})/e$

V-belt angle factor c_1 (function of strand inclination) from [Table 2](#)

V-belt length factor c_3 (function of V-belt profile and standard length L_r) from [Table 3](#)

$$e \approx 550 + (1800 - 1705)/2 = 597.5 \text{ mm.}$$

$$y \geq 0.015 \cdot 1800 = 27 \text{ mm}$$

$$x \geq 0.02 \cdot 1800 = 36 \text{ mm.}$$

$$v = \frac{160}{1000} \cdot \pi \cdot \frac{4000}{60} = 33.5 \text{ m/s}$$

$$f_B = 2 \cdot 1000 \cdot 33.5/1800 = 37.2 \text{ s}^{-1}$$

$$(224 - 160)/597.5 = 0.107$$

$$c_1 = 0.985$$

$$c_3 = 1.01 \text{ for SPZ and } L_r = 1800 \text{ mm}$$

Power transmission P_R per V-belt (function of V-belt profile, d_{wk} , n_k and v from Fig. 4 or Fig. 5)

$$P_R = 10 \text{ kW}$$

number of V-belts $z = \frac{P \cdot c_2}{P_R \cdot c_1 \cdot c_3}$, rounded to next

$$z = \frac{16 \cdot 1.2}{10 \cdot 0.985 \cdot 1.01} = 1.93, \text{ selected } z = 2.$$

higher value.

For multi-groove drives use matched V-belts, for tolerances see [Table 4](#).

Centrifugal force portion ϵ from Fig. 6
 minimum strand force F_T per V-belt in stationary condition

$$F_T \approx \frac{500.3 \cdot (2.5 - c_1) \cdot P_B}{c_1 \cdot z \cdot v} + \epsilon$$

$$\epsilon = 80 \text{ N}$$

$$F_T \approx \frac{500.3 \cdot (2.5 - 0.985) \cdot 19.2}{0.985 \cdot 2 \cdot 33.5} + 80 = 300.5 \text{ N}$$

$$\cos \frac{\beta}{2} = (224 - 160)/(2 \cdot 597.5) = 0.0536; \beta = 173.9^\circ.$$

angle of belt contact (belt wrap) β on small pulley

approximate from $\cos \beta = (d_{wg} - d_{wk})/e$

exact from $\cos \frac{\beta}{2} = (d_{wg} - d_{wk})/(2e)$.

Minimum force on pulley shaft $F_A = 2 F_T \cdot z \cdot \sin \frac{\beta}{2}$

$$F_A = 2 \cdot 300.5 \cdot 2 \cdot \sin 86.9^\circ = 1200 \text{ N}$$

$$E = 1.8 \text{ mm}$$

$$L = 597.5 \cdot \sin 86.9^\circ = 596.6 \text{ mm}$$

$$E_a = 1.8 \cdot 596.6/100 = 10.7 \text{ mm.}$$

indentation depth E of strand per 100 mm strand length (function of V-belt profile and strand force F_T) from Fig. 7

$$\text{drive strand length } L = e \cdot \sin \frac{\beta}{2}$$

total indentation depth of strand $E_a = E \cdot L/100$.

Table 1: Operating factor c_2

Drive	Driven machine			Daily operating time
alternating current and three-phase motors with starting torque	Centrifugal pumps and turbo compressors		piston pumps and compressors	
	< 7.5 kW	> 7.5 kW		in h
up to 1.8 times operating torque, internal combustion engines and turbines with $n > 600 \text{ min}^{-1}$	1.1	1.1	1.2	< 10
	1.1	1.2	1.3	> 10 < 16
	1.2	1.3	1.4	> 16
exceeding 1.8 times operating torque, internal combustion engines and turbines with $n < 600 \text{ min}^{-1}$	1.1	1.2	1.4	< 10
	1.2	1.3	1.5	> 10 < 16
	1.3	1.4	1.6	> 16

In the above example the belts have the correct pre-tension if they can be pressed down by a test force F_P which is dependent on the profile (SPZ: $F_P = 25 \text{ N}$, Fig. 7). Devices for testing the pre-tension are available commercially. After commissioning, the belt drive should be tautened again after a short period of running (approx. 1 to 2 hours), in order to compensate the initial slackening (expansion). Thereafter the belt drive should only be checked for tautness at fairly lengthy intervals.

Designation of belts, e.g.:

1 matched set consisting of 2 narrow V-belts, SPZ X 1800 L_r , DIN 7753.

Designation of belt pulleys, e.g.:

1 belt pulley in acc. with DIN 2211, SPZ profile, $d_w = 224 \text{ mm}$,
2 grooves, dynamically balanced.

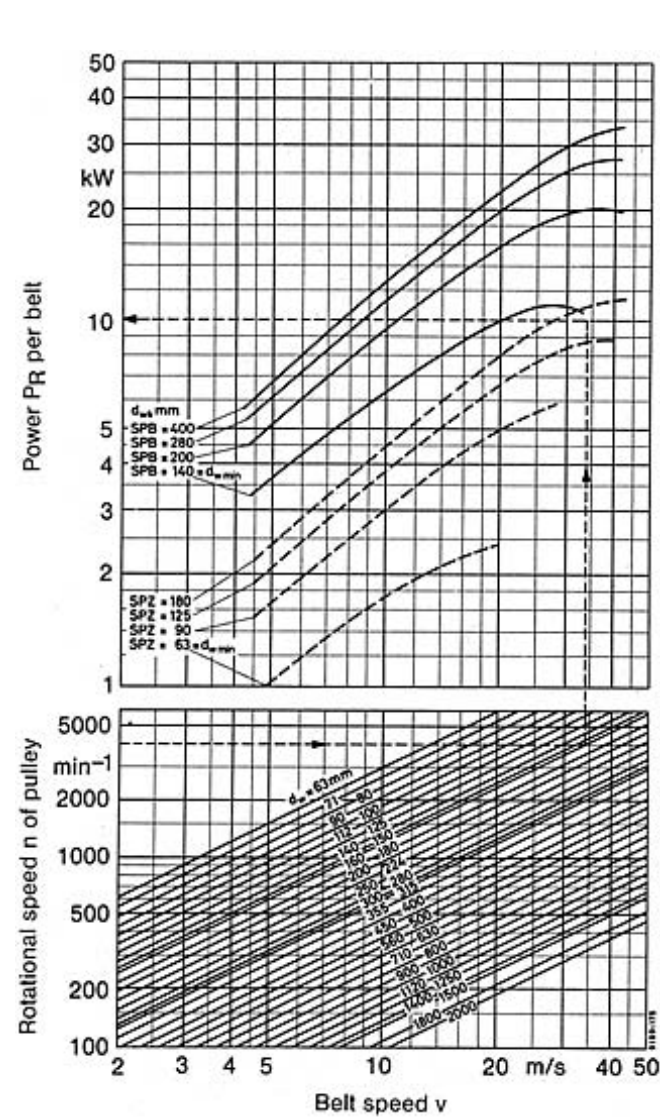


Fig. 4: Power per belt for SPZ and SPB profiles for $l/i = 1.20$; service life 24 000 h approx.
For $l/i \neq 1.20$, the above values should be upgraded or downgraded as follows (average values):

	$d_w \text{ min} \longleftrightarrow d_w \text{ max}$			
$i = 1.0$	-15 %	-6%	-3.5%	-2%
$i \geq 3.0$	+15%	+6%	+3.5%	+2%

(see enlarged diagram)

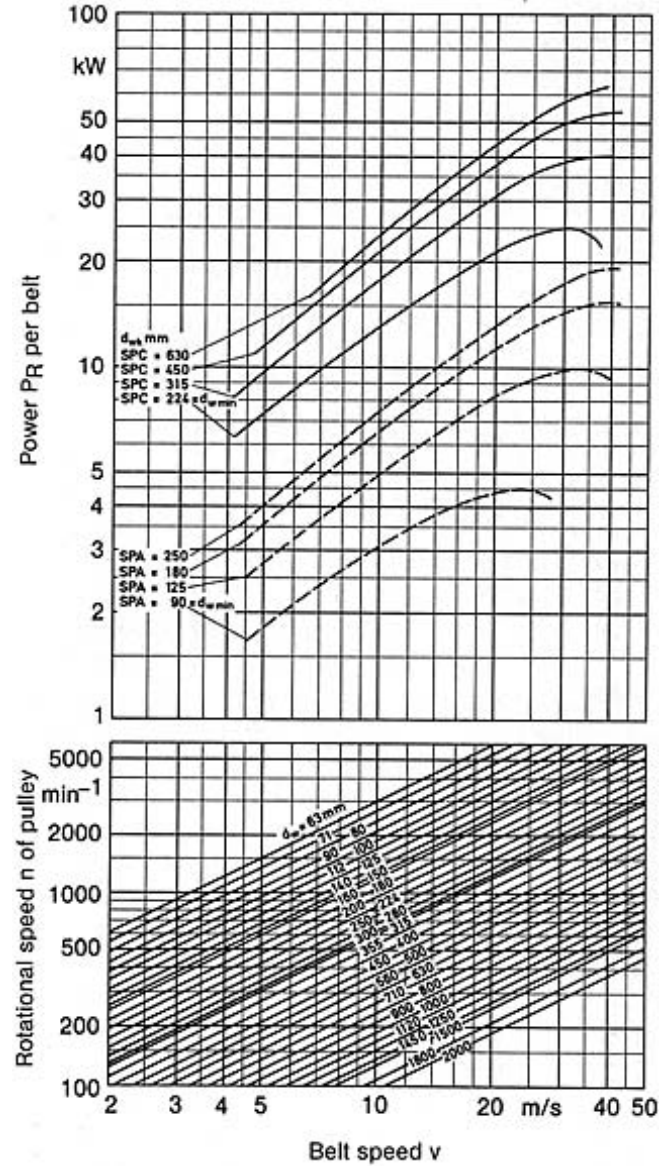


Fig. 5: Power per belt for SPA and SPC profiles for $l/i = 1.20$; service life 24 000 h approx.
For $l/i \neq 1.20$, the above values should be upgraded or downgraded, see Table under Fig. 4 (average values) (see enlarged diagram)

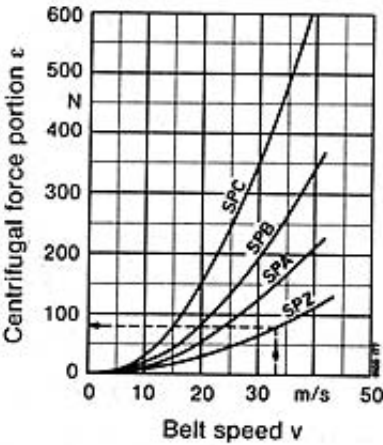


Fig. 6: Centrifugal force portion of the strand tension force F_p

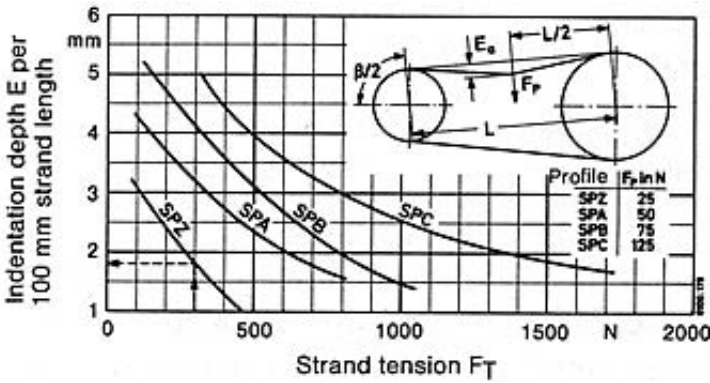


Fig. 7: Indentation depth of strand under testing

Table 2: Angle factor c_1

$\frac{d_{wg} - d_{wk}}{e}$	c_1	β	$\frac{d_{wg} - d_{wk}}{e}$	c_1	β
0	1.00	180°	0.75	0.88	136°
0.15	0.98	170°	0.85	0.86	130°
0.29	0.96	163°	1.00	0.82	120°
0.40	0.94	156°	1.15	0.78	110°
0.50	0.92	150°	1.30	0.73	100°
0.64	0.90	143°	1.45	0.68	90°

Table 3: Standard lengths L_r and length factor c_3

L_r in mm				
SPZ	SPA	SPB	SPC	c_3
630	800	1250	-	0.82
710	900	1400	2240	0.84
800	100	1600	2500	0.86
900	1120	1800	2800	0.88
1000	1250	2000	3150	0.90
-	1400	2240	3550	0.92
1120	1600	-	-	0.93
1250	1800	2500	4000	0.94
1400	2000	2800	4500	0.96
-	2240	3150	5000	0.98
1600	2500	3550	5600	1.00
1800	2800	-	-	1.01
2000	-	4000	6300	1.02
-	3150	4500	7100	1.04
2240	3550	5000	8000	1.06
2500	4000	5600	9000	1.08
2800	4500	6300	10000	1.10
3150	-	7100	11200	1.12
3550	-	8000	12500	1.14

Table 4: Permissible differences ΔL_r between the standard length L_r of one and the same matched set on multi-groove drives

L_r in mm	ΔL_r in mm
630 to 2000	2
2240 to 3150	4
3550 to 5000	6
5600 to 8000	10
9000 to 12500	16

The permissible deviation in standard lengths L_r amounts $\pm 1\%$

If necessary, the hub length and hub bore, keyway, set-screw, balancing requirements etc. should also be specified. For $v < 25$ m/s, the pulleys are statically balanced, for $v > 25$ m/s and $dw/B < 4$ they are dynamically balanced (B = width of belt pulley, unbalance of centrifugal pumps).

The material for standard construction is cast iron (GG-20).

2. Rat belt drive

Sizing of flat belt drives (calculation method and example).

The definition of the problem and the calculation method correspond to those applying to V-belt drives (see under 1. V-belt drive) unless anything to the contrary is stated.

Example: given are the power consumption $P = 16$ kW, the diameter of the small pulley $d_k = 160$ mm and the bending frequency $f_B = 40$ s⁻¹.

Selection of belt type from Fig. 8 for d_k , f_B .

Selected: belt type B. Belt length $L_r = 1800$ mm and belt speed $v = 33.5$ m/s are known from the V-belt example.

Specific belt power P_R^* per cm belt width from Fig. 9 for the appropriate belt type: $P_R^* = 3.1$ kW/cm.

Belt width $b = \frac{P \cdot c_2}{P_R^* \cdot c_1} = \frac{16 \cdot 1.2}{3.1 \cdot 0.985} = 6.29$ cm = 62.9 mm with c_2 taken from Table 1 and c_1 taken from Table 2.

Selected from Table 5 (standard width in acc. with DIN 111) : $b = 71$ mm.

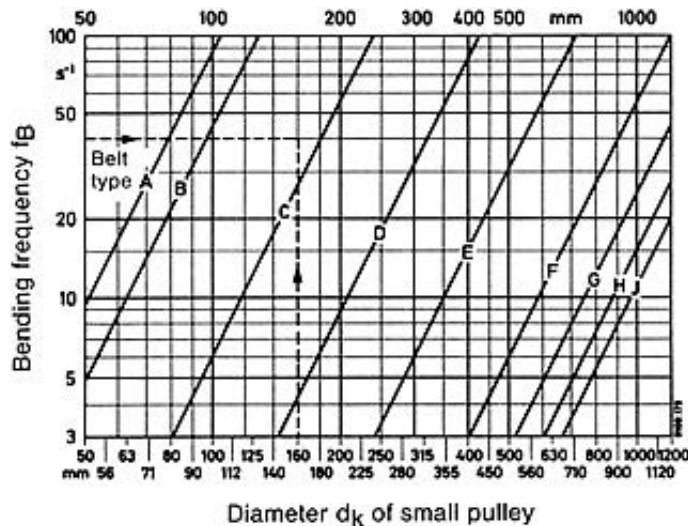


Fig. 8: Permissible bending frequency f_B in function of pulley diameter d_k for various flat belt types A to J

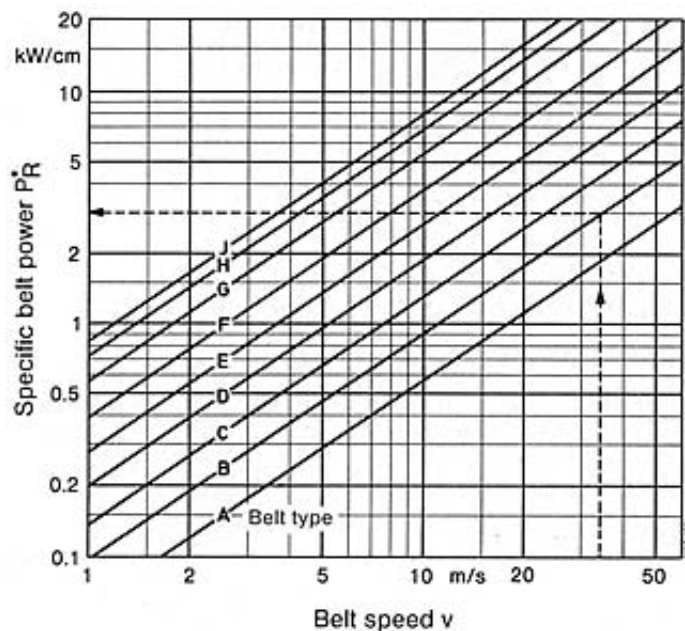


Fig. 9: Specific belt power P_R^* in function of belt speed v for various flat belt types A to J

Table 5: Pulley width B in mm and belt width b in mm, in accordance with DIN 111

B	b	B	b	B	b
20	16	80	71	200	180
25	20	100	90	224	200
32	25	125	112	250	224
40	32	140	125	280	250
50	40	160	140	315	250
63	50	180	160		

The corresponding standard pulley width in accordance with Table 5 (DIN 111) is: $B = 80$ mm.

$$\text{Peripheral force } F_U = \frac{P \cdot 1000}{v} = \frac{16 \cdot 1000}{33.5} = 477.6 \text{ N.}$$

$$\text{Force on pulley shaft during operation: } F_A \approx (2.0 \text{ to } 2.2) \cdot C_1 \cdot F_U \approx 2.1 \cdot 0.985 \cdot 477.6 = 988 \text{ N.}$$

Belt pre-tensioning:

The belt elongation is in the region of 1 to 3 % approx., and the lower figure should be selected in cases of uniform loading, whereas the higher figure applies to heavy shock-type loading. These figures only apply to new belts.

The basic consideration is that the belt should only be tautened to the extent that it is capable of transmitting the peak power without slippage.

3. **Jockey pulley** (tensioning pulley)

As modern belts only expand very slightly it is only necessary to provide a jockey pulley in cases where e.g. the drive cannot be shut down even very occasionally for the purpose of re-tensioning. Fig. 10 illustrates the two most suitable arrangements. In principle, the jockey pulley should be arranged on the slack strand. The jockey pulley diameter should not be selected too small ($\approx d_{w \min}$).

For f_B close to $f_{B_{\max}}$: fit jockey pulley on the inside, close to the large pulley,
 for $f_B < f_{B_{\max}}$: lit jockey pulley on the outside, close to the small pulley.

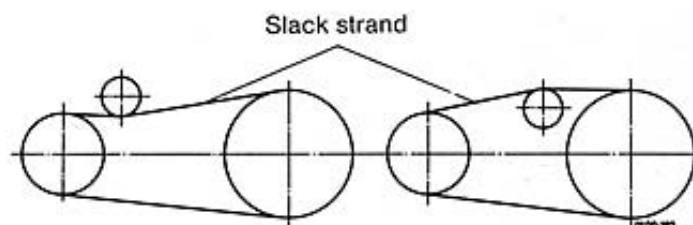


Fig. 10: Arrangement of jockey pulley

Table 6: Explanation of symbols (belt drive)

Symbol	Designation	Unit
B	width of belt pulley	mm
b	width of belt	cm, mm

c_1	V-belt angle factor	
c_2	operating factor	
c_3	length factor	
d_{wg}	effective ϕ of large pulley (in accordance with DIN 2211, V-belts)	mm
d_{wk}	effective ϕ of small pulley (in accordance with DIN 2211, V-belts)	mm
$d_{w \min}$	minimum effective ϕ (V-belt)	mm
$d_g = d_{wg}$	diameter of large pulley (flat belts)	mm
$d_k = d_{wk}$	diameter of small pulley (flat belts)	mm
E	indentation depth per 100 mm strand length	mm
E_a	total indentation depth of the strand	mm
e'	distance between shaft centres (preliminary)	mm
e	distance between shaft centres (finally)	mm
F_A	minimum force on pulley shaft	N
F_T	minimum force on strand in static condition, per belt	N
F^P	testing force	N
F_U	peripheral force	
f_B	bending frequency (number of bending changes per second)	s^{-1}
i	transmission ratio	
L	drive strand length	mm
L_w	effective belt length	mm
L_r	standard length	mm
ΔL_r	permissible difference between the standard lengths or belts in one and the same matched set of belts	mm
n_g	rotational speed of large pulley	\min^{-1}
n_k	rotational speed of small pulley	\min^{-1}
P	power to be transmitted by belt drive	kW
P_B	power for calculation purposes	kW
P_R	power per belt	kW
P_R^*	specific power per cm belt width	kW/cm
v	belt speed	m/s
x	adjustability of distance between shaft centres e for tensioning and subsequent re-tensioning of belt	mm

y	adjustability of distance between shaft centres e for effortless laying of the belt	mm
z	number of belts	
α	strand inclination angle $\alpha = 90 - \frac{\beta}{2}$	°
β	angle of contact (belt wrap) on small pulley	°
ϵ	centrifugal force portion	N

BERNOULLI Equation

BERNOULLI-Gleichung
Équation de BERNOULLI

see [Fluid Dynamics](#)

Blade

Schaufel
Aube

The b.'s of a centrifugal pump are either firmly fixed to or inserted (impeller blade pitch adjustment) into the impeller or diffuser (guide wheel); they represent the most important structural element of a pump for the conversion of mechanical power (shaft power) into pump output (see also internal and hydraulic efficiency).

The b.'s are bounded in the direction of flow by the inlet edge (also called suction edge) and the outlet edge (discharge edge), and, at right angles to the direction of flow, by the hub on the inside (in the case of axial and mixed flow impellers and diffusers) or by the inner (discharge side, rear) coverplate or cheek (in the case of radial and mixed flow impellers), and by the pump casing on the outside, or by the outer (suction side, front) coverplate or cheek (in the case of closed impellers).

A b. is called **adjustable** if it is an inserted b. and if its pitch angle can be adjusted as desired and to a fixed position when the pump is dismantled.

It is called a **variable pitch** b. if its pitch angle can be altered while the pump is running (impeller blade pitch adjustment).

The external shape of the b. is usually given as a so-called circular projection in the meridian section (longitudinal section along the rotation axis of the pump, flux line). The illustrations depicted under impeller give a synopsis of the possible impeller b. shapes used in centrifugal pump technology; in the case of diffuser b. shapes, we also have the conventional axial, mixed flow and radial b.'s, but without a clear differentiation of the direction of flow from the inside to the outside or vice versa. Thus on mixed flow tubular casing pumps (mixed flow pump) we have onion-shaped diffusers in many instances, with mixed flow diffuser b.'s traversed by the flow from the inside to the outside at the diffuser entry and from the outside to the inside at the diffuser exit.

Because no normal components of the relative velocity can arise perpendicularly to the b. on the impeller, or no normal components of the absolute velocity in the case of diffusers, the b. surfaces represent so-called flow surfaces consisting of flow lines infinitely close to each other. The effective shape of a b. can only be determined along a flow line, with regard to the hydrodynamic flow deflection (fluid flow machine); this is usually complicated, because the determination of the precise flow pattern in the impeller and diffuser requires a great deal of effort and is only possible under certain assumptions.

The velocity triangles on a flow line at the b. inlet (subscript 1) and on the same flow line at the b. outlet (subscript 2) already determine the outline of the b., taking the b. thickness and possible b. cascade reactions into account. The connection from b. inlet to b. outlet (median line, flow profile) is frequently created by means of an arc of circle (circular arc b.), sometimes by means of an arc of parabola, an S-shaped curve or another analytical curve. As a general rule, the b. inlet is designed as a shock-free entry in an inviscid approach flow (vortex flow). A well-known exception to this rule is given by the inducer. The outlet angle of the b. is more or less steep, depending on the head to be achieved (fundamental equation, fluid dynamics). On radial b.'s, β_2 is usually less than 90° (from 17° to 40° approx.); the b. in this case is called a "backward curved" b., $\beta_2 = 90^\circ$ characterizes a "radial ending" b., and $\beta_2 > 90^\circ$ a "forward curved" b. (with extremely high pressure coefficients, characteristic number).

The b. thickness is mainly governed by the centrifugal force stresses and the manufacturing method. In the case of profiled propeller b.'s, the thickness distribution is mainly governed by hydrodynamic considerations (flow profile). The minimum b. thickness is about 3 mm for cast iron, 4 mm for cast steel, and in special cases (e.g. inserted steel sheet b.'s) thinner b.'s can be realized.

Blade Angle

Schaufelwinkel
Angle d'aube

see Flow Profile

Blade Pitch Adjustment Device

Schaufelverstelleinrichtung
Dispositif réglable d'aube

see Impeller Blade Pitch Adjustment

Bleed-Off Stage

Anzapfstufe
Étage de soutirage

see Boiler Feed Pump

Boiler Circulating Pump

Kesselumwälzpumpe
Pompe de circulation de chaudière

see Circulating Pump

Boiler Feed Pump

Kesselspeisepumpe
Pompe d'alimentation de chaudière

The duty of the b.f.p., also called feed pump, consists in feeding to a steam generator (e.g. boiler, nuclear reactor) a quantity of feedwater corresponding to the quantity of steam generated. Today, nearly all d.f.p.'s are centrifugal pumps, as opposed to positive displacement pumps previously used for this duty. The construction of b.f.p.'s in respect of shaft power, material, pump types and drive is largely governed by the developments which have taken place in power station technology.

The trend in fossil fuel power stations is persistently towards larger and larger power block units (at the present time up to 750 MW, apart from a very few exceptions up to 1200 MW), and this has led to b.f.p.'s with drive ratings of 30 MW, with some exceptions up to 50 MW approx.

Up to 1950, the average discharge pressure of b.f.p.'s (pressure at outlet cross-section) lay in the 200 bar region. In 1955, these discharge pressures had risen up to 400 bar.

The mass flows were in the region of 350 t/h in 1950, and they have risen today up to 2500 t/h (4000 t/h) in conventional plants.

B.f.p.'s operate at temperatures of the fluid pumped of 160 to 180 °C, and in exceptional cases higher temperatures still.

Feed pumps for nuclear power stations of 1300 MW output are presently constructed for mass flows up to 4000 t/h and discharge pressures of 70 to 100 bar.

Up to 1950 approx. b. f. p.'s were constructed of unalloyed steels, and since then of 13 to 14 % chrome steels. This change of materials was made necessary by the introduction of new feedwater treatment processes. The development of high strength, corrosion-resistant martensitic chrome steels with emergency running characteristics paved the way for the present-day b.f.p. with speeds of 5000 to 6000 min⁻¹. The mass flows of b.f.p.'s rose rapidly in conjunction with the rise of power block outputs, and today's fullload b.f.p.'s for conventional 750 MW power blocks are constructed with four or five stages, with stage pressures up to 80 bar. Feed pumps for 1300 MW nuclear power stations are of the single stage type.

Feed pump drives. In the case of conventional plants above 400 MW, the drive of the full load feed pump is usually via a steam turbine today. In most cases condensing turbines running at 5000 to 6000 min⁻¹ are used.

Electric motors (asynchronous motor) are mainly used to drive half load feed pumps, both in fossil fuel and in nuclear power stations. Speed adjustment of electrically driven b.f.p.'s is effected via a fluid coupling (variable speed turbo gear) or by electrical regulation (control), e.g. by thyristors (electrical switchgear, used today up to 18 MW drive input approx.).

The slow speed booster pump is usually driven off the extension stub shaft of the b.f.p. via a speed reducing gear, in the case of turbine drive, or direct off the extension stub shaft of the electric motor. Fig. 1 gives three diagrammatic examples of feed pump layouts. The single or double suction booster pump (Fig. 2, compare illustration under booster pump) has the duty of generating the necessary NPSH value (net positive suction head) of the installation (short: NPSHav) for the high speed b.f.p. connected downstream.

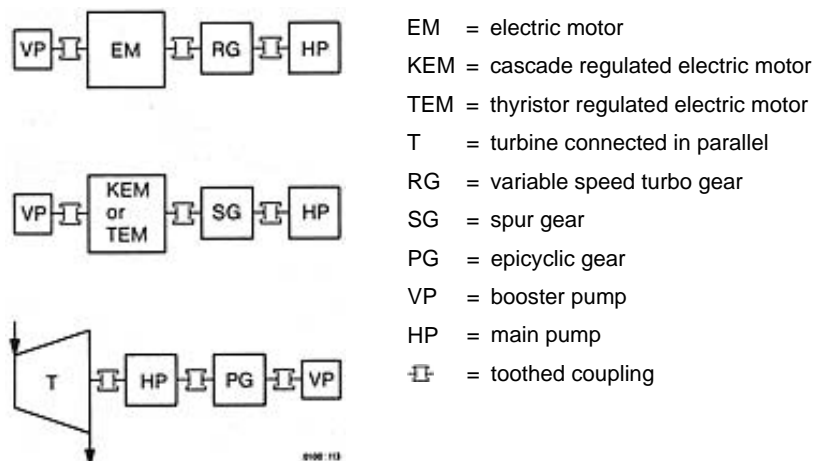


Fig. 1: Examples of different layouts of feed pumps

Types of feed pumps. For conventional power stations, b.f.p.'s are constructed in the form of multistage barrel pumps (Fig. 3) or ring section pumps (multistage pump) (Fig. 4).

These two types differ in the construction of the pump casing and in manufacturing costs. There are however no differences with regard to reliability in operation and ability to withstand maltreatment. The dimensions of the rotating parts and of the ducts through which the fluid flows can be made identical in both types. Two reasons for choosing either one or the other type are described below:

- a) The material and manufacturing costs of barrel pumps increase in inverse ratio to the rate of mass flow, when compared with ring section pumps.
- b) Barrel pumps have some advantages over ring section pumps when it comes to repairing a b.f.p. in a plant. If a rotor has to be replaced, the barrel (pump casing) can remain in situ in the piping. This is important with regard to the availability for service of the power block, if there is no 100% standby pump installed.

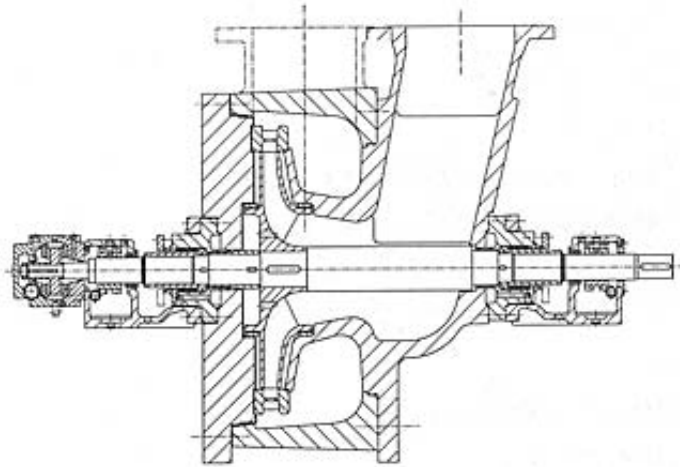


Fig. 2: Single suction booster pump for a boiler feed pump

In the case of nuclear power stations, feed pumps (reactor pump) of single stage type with a double suction impeller (multisuction pump) and a double volute casing (pump casing, see also Fig. 6 under volute casing pump) are usually adopted. The cast pressurized casing parts are increasingly replaced by forged parts. Such a pump is illustrated in Fig. 5, for a capacity of 4200 m³/h approx., a head of 700 m approx. and a speed of 5300 min⁻¹ approx. Present-day heads of reactor feed pumps are in the region of 800 m for boiling water reactors and 600 m for pressurized water reactors; the capacities are however about twice as high as those of a comparable b.f.p. in a fossil fuel power station.

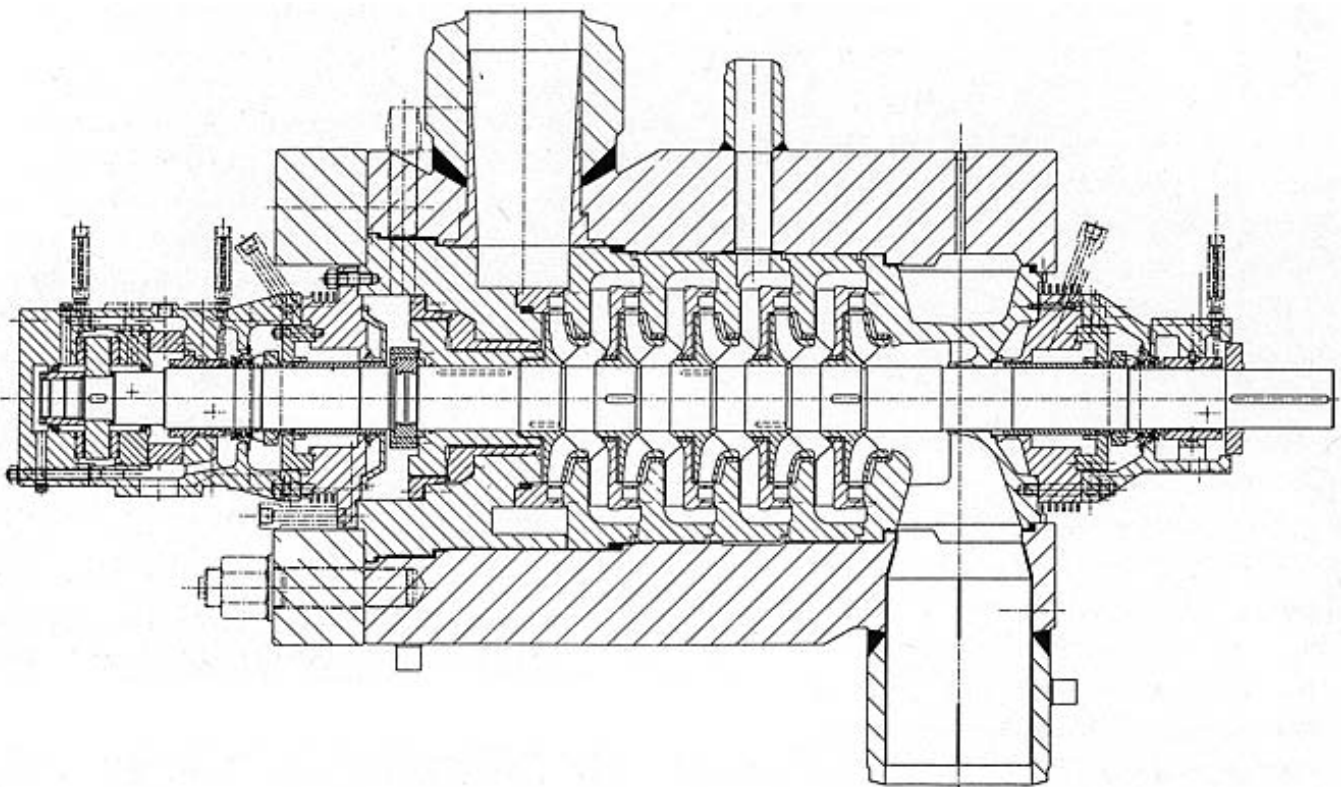


Fig. 3: Boiler feed pump in barrel insert design with tapping stage

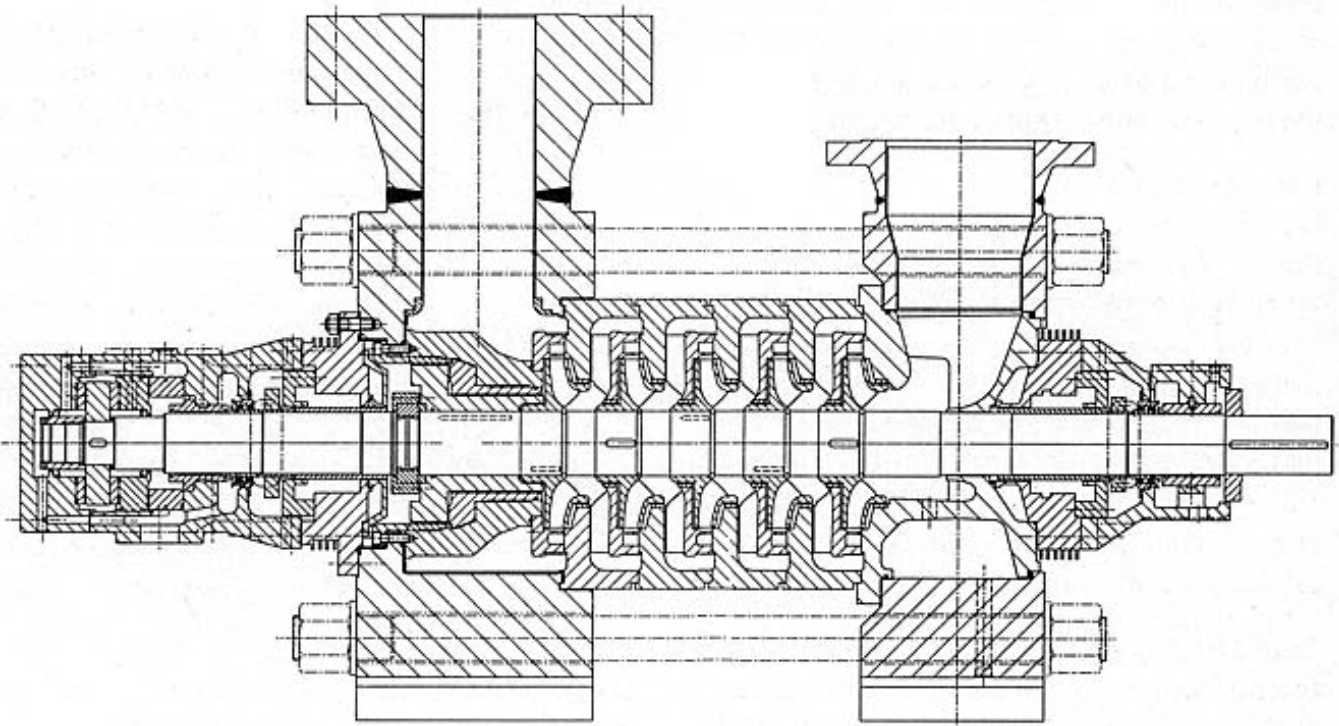


Fig. 4: Boiler feed pump of ring section type

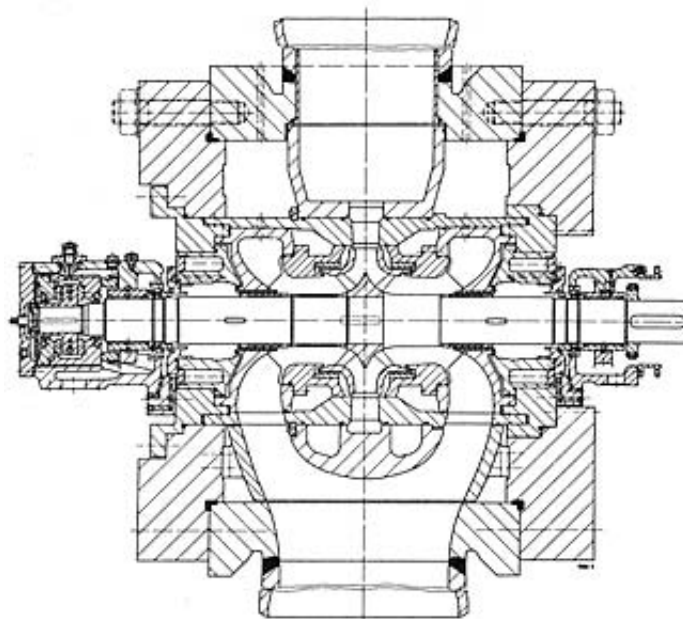


Fig. 5: Double suction reactor feed pump with forged casing

Casing of single and multistage pumps. The pump casings of b.f.p.'s must be considered from two points of view: the wall thickness must be appropriate on the one hand to satisfy the pressure loading requirements, and on the other hand to adapt itself to the temporary temperature fluctuations which arise. These two criteria are satisfied by adopting a high strength ferritic casing material which enables the wall thicknesses to be kept thin enough to avoid any overspreading as a result of temperature fluctuations, yet of adequate thickness to guarantee the requisite safety against the internal pressure.

Barrel casings are usually made of unalloyed ductile forged steel, and all surfaces in contact with the headwater are coated with an austenitic material by deposit welding. In order to weld the pump casing into the pipeline, an intermediate piece (adapter) made of a material compatible for welding into the pipeline and the pump casing is welded onto the pump suction and discharge branches. The cover on barrel pumps is sealed by flattening a cellular metal asbestos spiral-wound gasket (seals).

The casings of ring section pumps are manufactured from forged carbon steel with austenitic cladding (materials). The sealing (seals) of the individual stage casings (stage) against one another is by metal-to-metal contact, the individual casings being clamped together axially by tiebolts between the suction and discharge pump casings. Temperature shocks are absorbed mainly by additional stresses on the tiebolts and sealing faces of the stage casings. The thicker the wall thicknesses of the >> pump casing have to be, the more sensitively will the casing react to thermal shocks.

Water injection at a pressure situated between the suction and discharge pressures of the pump is a frequent service requirement, and this is taken care of by tapping water from one of the pump stages, both in the case of barrel pumps (Fig. 3) and of ring section pumps. In the case of barrel pumps, the inside of the barrel is subdivided into three pressure zones when an intermediate tapping is required, so that a partial flow at the required intermediate pressure can be led off directly to the outside.

These pressure zones are sealed off from one another by flexible spiral-wound gaskets, and the flexibility and thermal shock behaviour are suitably matched to one another. In the case of ring section pumps, a partial flow at an intermediate pressure can easily be led off through a stage casing fitted with an extraction nozzle.

Rotor construction. B.f.p.'s are usually fitted with pump shafts which have as small a distance between bearings as possible, combined with a relatively large shaft diameter; the impellers are usually shrunk on the shaft, and consequently the static shaft sag is very small; the shaft is largely insensitive to vibrations, and in normal running the conditions are smooth, without any undesirable radial contact with the casing (quietness of centrifugal pumps). The hub diameter is increased at the back of the impeller, and the impeller entry geometry is designed to keep the diameter as small as possible, so as to reduce the axial forces which have to be absorbed by the balancing device (axial thrust).

Single stage reactor feed pump rotors are even stiffer in comparison with multistage b.f.p. rotors, and their static shaft sag is smaller.

Balancing of axial thrust. If the impellers of a b.f.p. for conventional power stations are arranged as illustrated in Figs. 3 and 4, an axial thrust is generated. During operation of the pump, the magnitude of this axial thrust will depend on the position of the operating point on the throttling curve (operating behaviour of centrifugal pumps) and on the amount of wear on the internal clearances (slit seal). Additional disturbing forces can arise in the event of abnormal operating conditions, e.g. if the pump starts to cavitate (cavitation, net positive suction head). On the larger b.f.p.'s, the balancing of the axial thrust on the pump rotor is effected by means of a balancing device through which the pumped fluid flows, combined with an oil-lubricated thrust bearing (plain bearing). The hydraulic balancing device may comprise a balance disc with balance disc seat, or a balance piston or double piston with the associated throttling bushes (axial thrust). Pistons and double pistons can also be combined with a balance disc.

Axial thrusts arising in reactor feed pumps with double suction impeller (multisuction pump) are balanced hydraulically (Fig. 5); residual thrusts are absorbed by an oil-lubricated thrust bearing (plain bearing).

Balancing of radial forces on the pump rotor. Radial forces arise from the weight of the rotor, from mechanical out-of-balance, and from the radial thrust. The balancing of the radial forces is effected by two oil-lubricated radial bearings and by throttling gaps through which the fluid flows axially. Such throttling gaps through which the fluid flows axially are located at the impeller neck; or - in the case of multistage b.f.p.'s of conventional power stations - in the throttling bushes of the diffuser plates and on the balance piston. If the rotor is slightly eccentric, a centering restoring force will be generated in these gaps, and this force will be largely dependent on the pressure differential and on the gap geometry. This restoring action is usually termed LOMAKIN effect. It is much reduced when, due to abnormal operating conditions, the headwater in the gap flow is not in the purely liquid phase (cavitation). The hydrostatic action of the throttling gaps in respect of mechanical stiffness can greatly exceed the shaft stiffness. The system is tuned in such a way that the critical speed of rotation always remains well away from the operating speed, and hydraulic exciting forces, particularly under part load operation, can be absorbed in addition.

The radial thrust of reactor feed pumps (Fig. 5) is reduced by using double volutes (Fig. 6 under volute casing pump) and an additional diffuser, if required.

Shaft seals. Soft-packed stuffing boxes, mechanical seals, floating seals, and labyrinth seals (seals) are used on b.f.p.'s.

The application limit of soft-packed stuffing boxes is largely governed by the existing possibilities for the removal of the frictional heat. In the case of high duty soft-packed stuffing boxes, there is usually a pre-cooling of the leakage water and an ambient cooling of the stuffing box housing, shaft protecting sleeve and gland. The packing material

usually consists of braided asbestos-graphite or teflon twine. This type of shaft seal is used with success e.g. on full load feed pumps up to 150 MW power block size.

The very small leakage of mechanical seals (shaft seals) is emitted into atmosphere in the vapour phase at the exit. The frictional heat generated is less than in the case of soft-packed stuffing boxes. A closed circuit cooling system is generally adopted, which is driven by a circulator device on the rotating seal ring when the pump is running, and by natural circulation (thermosiphon action) when the pump is stopped.

Fig. 6 illustrates a floating seal. This type of seal can be used for high circumferential velocities and high sealing pressures. The floating seal consists of a series of short throttling rings which can be displaced radially. A stream of cold sealing water injected into the seal ensures that no hot water can leak out of the pump. This sealing water feed must be kept going while the pump is running or standing under pressure. The control of the sealing condensate injection (Fig. 7) into a floating seal can be effected by differential pressure regulation or by differential temperature regulation of the sealing condensate.

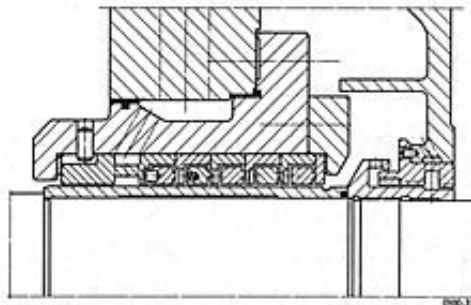


Fig. 6: Floating seal (differential temperature regulation)

In the case of differential pressure regulation, the difference between the injection pressure and the feed pressure (positive suction pressure) is pre-set, and a small flow of sealing condensate enters the pump. The pre-condition for this arrangement is that the sealing condensate must be absolutely clean and free of gas, so as not to contaminate the main flow of fluid pumped.

In the case of differential temperature regulation, the difference between the temperature at the condensate drain and at the condensate injection is pre-set, and a very small flow of feedwater penetrates from the inside of the pump into the floating seal. This precludes any ingress of cold water into the pump proper. A labyrinth seal (shaft seals) can be provided in lieu of a floating seal. This consists of a fixed throttling bush with an annular grooved section. There is no radial displacement facility in this case, and the clearance on the diameter of a labyrinth seal must therefore be made larger than on a floating seal. Consequently a greater flow of sealing water is required in this case.

Warming up procedure. If a b.f.p. is switched on and off frequently, it is desirable to avoid thermal stratification and warping of the casing after the pump has stopped, so as to prevent premature internal wear at the sealing gap. In principle, the construction materials are selected in such a way that the b.f.p. can be started up from any thermal condition (cold start, semi-warm start). However, a physical contact between rotor and casing at locations with a close clearance cannot be avoided under certain circumstances of abnormal operation, e.g. when cavitation occurs, or during a "semi-warm" start, when the b.f.p. is warped. The affected locations are the throttling gaps at the impeller inlet, at the throttling bush in the diffuser and at the balancing device. The matching of appropriate construction materials at these locations, consisting of corrosion-resistant chrome steels with special alloy additions ensures relatively good emergency running conditions even at high circumferential velocities. Any >> wear at close clearance gaps is always associated with a drop in efficiency.

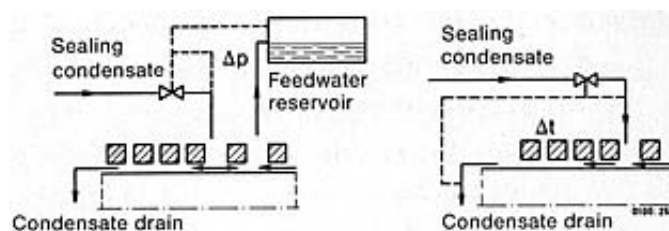


Fig. 7: Regulation systems of sealing condensate feed to floating seals

The reason for warping of the pump is a thermal stratification during the cooling down process. There are three different ways of avoiding this thermal stratification:

- a) circulation of the water content inside the pump by means of an auxiliary pump,
- b) injection of hot water on the discharge side of the pump,
- c) drainage of the water trapped inside the pump.

These three methods are used with success on ring section pumps. With method a) the pump cools down completely. In the case of barrel pumps, only method c) can be used, because of the construction of the casing; the water is drained from the lowest point on the pump. All these methods are controlled automatically.

The number of cold starts on a barrel pump with very thick casing walls has to be limited on account of thermal shock stresses. if no preliminary warming up procedure is adopted.

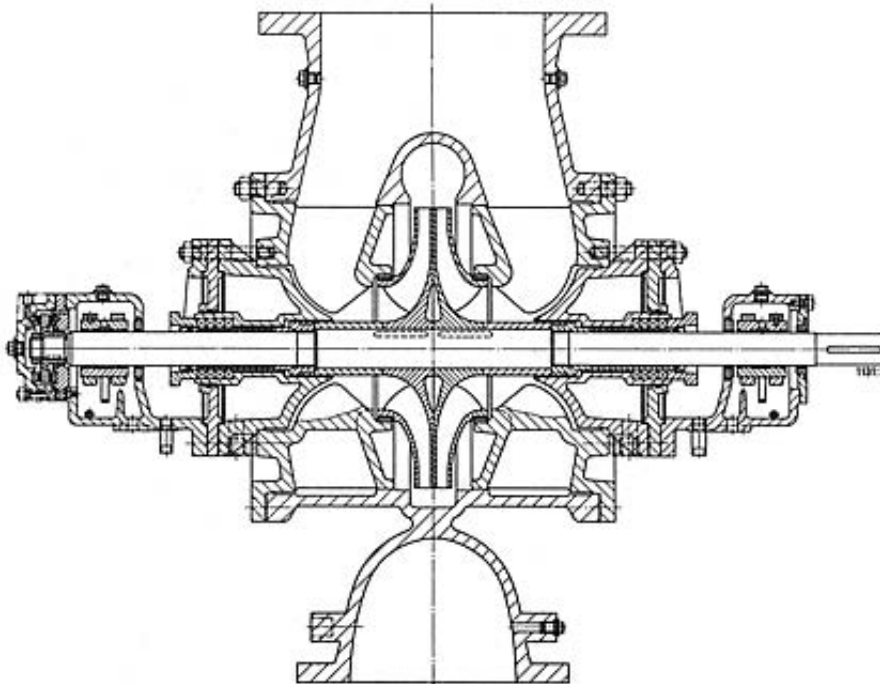
If for plant-specific reasons reactor feed pumps have to be operated under warm conditions, the shaft warping occurring at the mechanical seal must be counter-acted by shutting down its thermosiphon circuit.

Minimum flow valve. A so-called minimum flow valve (e.g. an automatic leak-off valve, valves and fittings), arranged downstream of the outlet branch of the b.f.p., ensures that a minimum flow (capacity) is always present and prevents any damage that might arise during very low load operation (part load operation, operational behaviour), as a result of excessive overheating and evaporation of the contents of the pump, or as a result of cavitation at part load operation.

Booster Pump

*Boosterpumpe, Vorpumpe, Zubringerpumpe
Pompe be mise en charge*

A b.p. is in most cases single or double suction backing-up pump (centrifugal pump) for a main pump. The b.p. has the duty of generating the necessary positive suction head (suction behaviour) for the trouble-free operation of the main pump, by boosting the pressure (illustration, see also Fig. 2 under boiler feed pump).



Double suction booster pump for a boiler feed pump

Borehole Pump

Bohrlochpumpe
Pompe de forage

see [Borehole Shaft Driven Pump](#), [Submersible Motor Pump](#), [Underwater Motor Pump](#)

Borehole Shaft Driven Pump

Bohrlochwellenpumpe
Pompe de forage à ligne d'arbre

A b.s.d.p. is a vertical [centrifugal pump](#) installed in a borehole or well, with a drive shaft ([pump shaft](#)) extended up to ground level and coupled at ground level to a motor or gear, also quill motor or quill gear, installed dry (in contrast to a [submersible motor pump](#) with a motor submerged in the fluid pumped).

The fact that the borehole is very narrow (only a few decimetres in diameter; usually ranging from DN 100 to DN 600) governs the design of the b.s.d.p. which has a small outer diameter but extends axially over a considerable length. B.s.d.p.'s are [multistage pumps](#) usually fitted with mixed flow impellers ([impeller](#)).

The suction strainer (suction strainer basket, [pressure loss](#), [valves](#) and fittings), [foot valve](#) (commonly used for installation depths of more than 10 m), pump and vertical discharge piping (rising main) are all suspended from the drive stool, which supports the driving motor and also incorporates the [discharge elbow](#) of the pump (see illustration).

The thrust bearing ([plain bearing](#), [anti-friction bearing](#)) is arranged in the motor lantern, in the quill motor (illustration) or the quill gear. The drive shaft, which if necessary consists of several lengths of shafting, transmits the [axial thrust](#) as well as the driving torque; it runs inside the rising main and is guided in water-lubricated [plain bearings](#) inside the rising main. If the borehole or well is very deep, the shafting and bearings tend to push up the total cost of the plant ([economics](#)), and it becomes more advantageous to install a [submersible motor pump](#).

Boundary Layer

Grenzschicht
Couche limits

B.I. in the flow of viscous media along stationary walls, is the flow layer in immediate proximity to the wall in which the velocity rises asymptotically from the value at the wall (no-slip condition) to the value of the main flow which is no longer influenced by the wall friction. The b.l. thickness is usually defined as the distance from the wall at which the flow velocity attains 99% of the value of the main flow velocity. A very steep rise in velocity occurs perpendicularly to the wall in the b.l., which is very thin in the case of high REYNOLDS numbers ([model laws](#)) of the main flow. In contrast to the main flow, which for practical purposes can be considered as a [potential flow](#) i.e. friction-free, the friction cannot be disregarded within the b.l., because the inertia forces and friction forces here are of the same order of magnitude. The friction forces also act on the wall and cause the frictional resistance along it. The main flow and the b.l. flow also react mutually on one another, in so far as the main flow on the one hand is deflected from the wall by the displacement thickness of the b.l., and on the other hand the outside flow imposes a pressure pattern on the b.l. which has a marked influence on its development.

The flow in the b.l. can be laminar or turbulent ([fluid dynamics](#)). For the same main flow velocity, a laminar b.l. is thinner than a turbulent b.l., but in the case of a turbulent b.l. flow the velocity profile has got more body. with a steep velocity gradient towards the wall, resulting in a much higher frictional resistance than in the case of a laminar b.l. In immediate proximity to the wall, even a turbulent b.l. possesses a laminar sublayer, because all transverse movements, including the turbulent fluctuations, must by definition disappear at the wall itself. In a flow around a body, there is at first a laminar b.l. which increases in the direction of flow, and which becomes unstable after having travelled a certain distance, and switches over to a turbulent flow form under the influence of disturbances e.g. wall roughness (Table 1 under pressure loss) or turbulent fluctuations in the outside flow.



Borehole shaft driven pump with quill motor

In curved ducts or rotating systems the equilibrium in the main flow between the pressure forces on the one hand and the centrifugal or CORIOLIS forces on the other hand is upset by the lower flow velocities in the b.l. The results are three-dimensional secondary flows.

The b.l. can in certain cases separate itself from the body (b.l. separation). This phenomenon arises in flow regions where the static pressure which is imposed by the main flow on the b.l. rises along the direction of flow. The main flow is then pushed away from the wall by the b.l. separation. A so-called dead water region is formed downstream of the breakaway point, in which a number of eddies arise. The flow velocities in this dead water region are disordered in respect of magnitude and direction; part of this dead water flows backwards (reverse flow). There is no significant frictional resistance in the separation path downstream of the breakaway point, but there is an increase in the so-called pressure resistance as a result of the dead water, which is far more significant than the decrease in frictional resistance. This means that the total flow resistance of the body increases considerably when there is a b.l. separation. Such flow separations should be avoided as far as possible by design measures and hydrodynamic streamlined devices (fittings, diffuser, diffuser device).

A particular type of flow separation is represented by the so-called separation bubble, which arises e.g. on flow profiles when the b.l. becomes turbulent immediately downstream of the laminar separation and then comes in contact again.

The b.l. plays an important part in flow through pipes (pipng). At the pipe inlet there is often a constant velocity distribution. A b.l. is formed at the wall, and its thickness increases with increasing distance downstream from the pipe inlet. The intact (i.e. as yet not contaminated by the effect of friction) core flow is accelerated thereby until the b.l. grows right up to the centralize of the pipe, after a so-called run-up length. Further downstream, the velocity profile of the pipe flow does not alter any further.

Bow-Thruster

*Bugstrahlruder, Querstrahler
Gouvernail à jet*

The b.t. is used as a manoeuvring aid on board ship at slow speeds, to be partially independent of the assistance of tugs. B.t.'s are useful for turning manoeuvres in narrow harbour basins when travelling through canals, when docking and casting off, etc.

B.t.'s are propeller pumps installed beneath the water line in a tunnel through the ship's hull, at right angles to the direction of travel, and usually in the bow (bow-thruster rudder). Large vessels often have two b.t.'s arranged one behind the other, because a single b.t. of the same performance would necessitate too large a diameter. Special ships may also be equipped with an additional stern thruster.

The propeller pumps a stream of water through the transverse tunnel either to port or starboard. This generates a reaction which displaces the ship in the opposite direction to the stream of water. The change of direction of the stream of water can be effected either by changing the direction of rotation of the propeller with the aid of a pole-changing motor (number of poles, asynchronous motor), or by using a variable pitch propeller rotating in one direction only (impeller blade pitch adjustment).

Fig. 1 illustrates a b.t. with reversal of direction of rotation. The compact unit is inserted from above into the transverse duct, and bolted on and sealed at the top flange. This facilitates dismantling and reassembly. If required, the b.t. can be fitted in a different way.

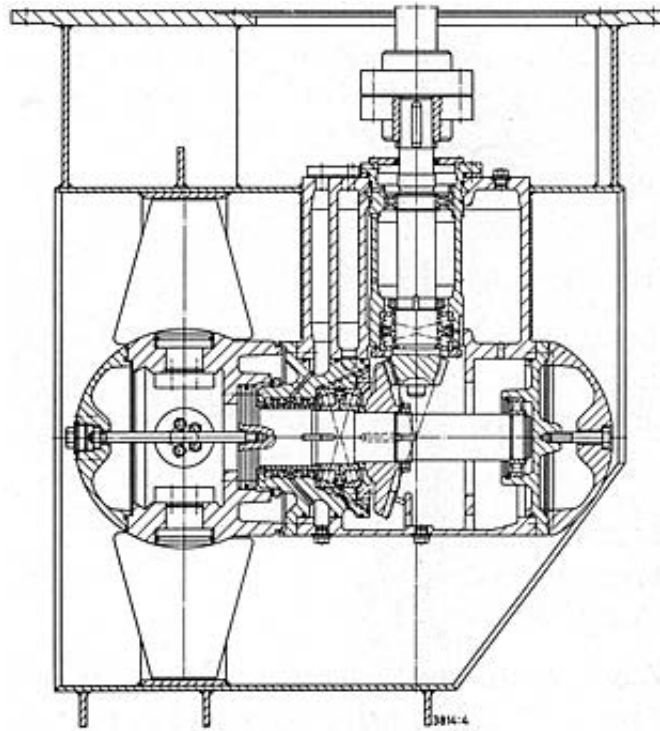


Fig. 1: Bow-thruster

The driving motor is either bolted directly onto the top flange (Fig. 2) or arranged higher up in the ship.

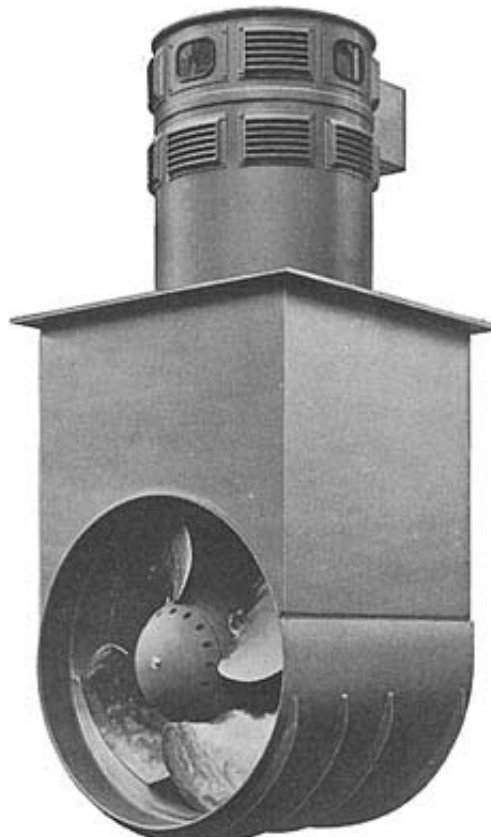


Fig. 2: Bow-thruster with motor

The torque is transmitted via bevel gears (Fig. 1). The pinion shaft and the propeller shaft are guided in oil-lubricated anti-friction bearings. The propeller shaft is sealed off by a number of radial sealing rings and an intermediate oil-filled sealing chamber at its exit from the gearbox casing. The propeller blades are inserted in the propeller hub (propeller pump), and their pitch can be altered if required after dismantling of the b.t.

Usual sizes of b.t.'s are:

tunnel (pipe) diameter	from 800 to 2400 mm,
transverse thrust	from 10 to 170 kN,
motor ratings	from 40 to 1100 kW.

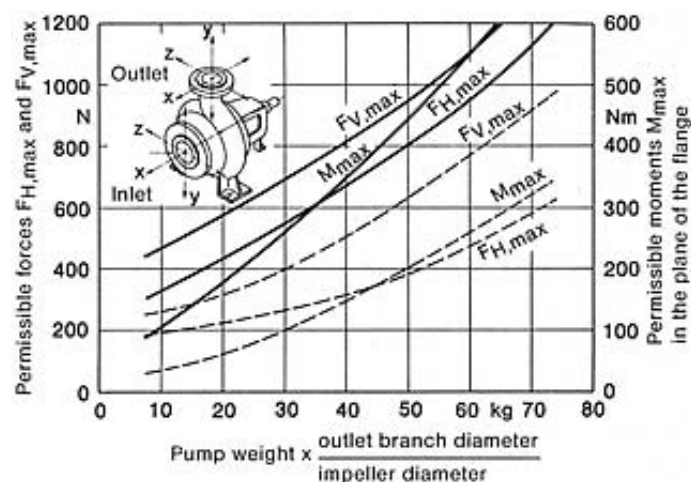
Branch Loading

Stützenbelastung

Charge de tubulure

Those centrifugal pumps mounted on the pump foundation are usually used as a fixed point for the attaching of pipes. Even if pipes are attached to the pump suction and discharge branch without residual stresses, one has forces and moments in the pipes known as b.1. depending on operating conditions (pressure, temperature). They lead to stresses and deflection of the pump casing, and most of all to coupling misalignment, where the quietness of centrifugal pump and the lifespan of the flexible materials in the shaft couplings are negatively affected. Therefore, a limit is placed on allowable b.1.

Since the combined loads for each pump branch are composed of three force and three moment components, it is not possible to obtain values for all combinations. Either one checks whether the plant pipe loading is acceptable using involved calculations, or one can try to make due with generally acceptable values published. Examples of such are: values for several types of pumps are found in the EUROPUMP guideline under "Forces and Moments allowed on Centrifugal Pump Flanges" (1986), or in API 610 and ISO 5199 under single stage chemical and refinery pumps. The illustration shows an example of allowable b.1. concerning a single stage volute casing pump, according to ISO 5199 (the solid line is for a pump mounted on a poured concrete slab; the dotted line is for a pump that is not mounted on a poured concrete base).



Allowable moments M_{\max} in the plane of the flange surface, and allowable forces $F_{H, \max}$ (in the x, z plane) and $F_{V, \max}$ (in the y direction), according to ISO 5199 for single stage volute casing pumps constructed from ferritic cast steel or nodular cast iron at room temperature (according to DIN 24256/ISO 2858). (Lower values are obtained for austenitic stainless steel or cast iron or for higher temperature service.)

By-pass

Bypass

By-pass

B.p., equivalent to a deviation or diversion circuit. The b.p. plays an important part in centrifugal pump technology in control and balancing device applications.

B.p. operation in connection with control applications means that a centrifugal pump is operated at a higher capacity (rate of flow) than that required as useful capacity in the piping (pumping plant).

The b.p. flow tapped off can either be returned directly to the pump suction branch from the pump discharge branch in a short loop, or it can be led through other equipment such as a condenser or cooler before being returned to the suction stream. There are three main reasons for adopting this type of b.p.:

1. operating behaviour reasons in order not to operate the pump at very reduced rates of flow,
- 2.

control reasons (control), in cases where the shaft power curve falls off as the capacity increases (propeller pump, peripheral pump),

3. thermodynamic reasons, in order to prevent overheating of the fluid pumped at part loads (operating behaviour). The b.p. flow is led off e.g. by a nonreturn valve for minimum flow device (valves and fittings), which is preferably fitted direct on the outlet branch of high pressure and super pressure pumps (boiler feed pump).

The b.p. plays an equally important part in axial thrust compensation (balancing device), e.g. on boiler feed pumps.

C

Cable Gland Stuffing Box

Kabelstopfbuchse
Presse-étoupe de câble

see [Terminal Gland](#)

Cable Lead-In

Leitungseinführung
Passage de câble

A c.l.i. is the entrance of a connector cable into the terminal box, that is, the housing of the electromotor. It prevents moisture from entering the terminal box (Figs. 1 and 2). The c.l.i. is not to be confused with the [terminal gland](#).

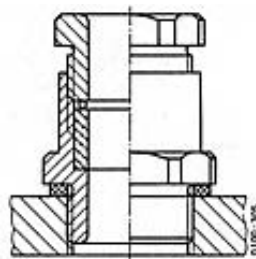


Fig. 1: Cable lead-in according to DIN 46255

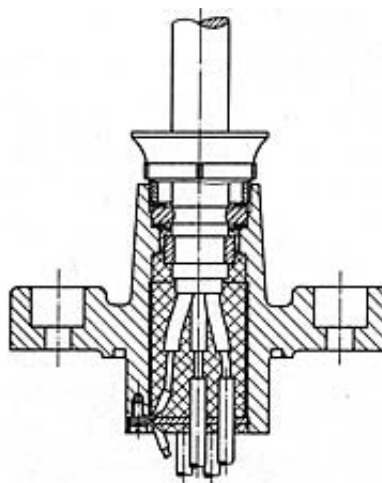


Fig. 2: Watertight cable lead-in in a submersible motor pump

Canned Motor Pump

Spaltrohrmotorpumpe
Électro-pompe à stator chemisé

C.m.p.'s are [centrifugal pumps](#) designated after their drive. The canned motor is a special type of submersible motor or of [wet rotor motor](#) (see Fig. funder wet rotor motor for illustration of a c.m.p.). The stator winding of the canned motor is protected against the fluid pumped by a cylindrical tube, the so-called can (Fig. 2 under chemical

pump). This mode of construction avoids stator winding insulation and corrosion problems but complicates the removal of heat from the stator winding, and has an adverse effect on the efficiency, on account of the eddy current losses in the can and the increase in width of the gap between rotor and stator; furthermore, the adequate strength of the thin walled can becomes problematic at the higher pressure levels.

C.m.p.'s are grandness pumps made of materials of high corrosion resistance; they operate entirely maintenance-free. Like all wet rotor motor, they require special care during assembly, erection and commissioning, to ensure that the internal spaces of the motor are reliably vented, and that solid foreign bodies in the fluid pumped are kept away from the underwater bearings (plain bearing); only in this way can a sufficient heat removal and perfect bearing lubrication be ensured. As to performance range and applications of c.m.p.'s, see under wet rotor motor and application fields for pumps.

Capacity

Förderstrom
Débit

The c. Q of a centrifugal pumps is the useful volume flow delivered at the outlet cross-section (pump discharge branch) of the pump.

Volume flows which are tapped upstream of the outlet cross-section of the pump for extraneous purposes (e.g. see by-pass) must be taken into account when calculating the c. of the pump. In the event of appreciable compressibility of the medium pumped, a conversion to the conditions prevailing at the pump suction branch must be made (pump output).

The SI unit of c. is $1 \text{ m}^3/\text{s}$ but in centrifugal pump technology the units $1 \text{ m}^3/\text{h}$ and 1 l/s are also commonly used. In conjunction with the throttling curve (characteristic curve) the following additional concepts of c. occur:

Optimum c. Q_{opt} : the c. at the operating point of optimal efficiency at the rotational speed and for the medium pumped specified in the supply contract.

Minimum c. Q_{min} : the minimum permissible c. at which the pump can operate continuously without suffering damage, at the rotational speed and with the medium pumped specified in the supply contract.

Maximum c. Q_{max} : the maximum permissible c. at which the pump can operate continuously without suffering damage, at the rotational speed and with the medium pumped specified in the supply contract.

Peak c. Q_{Sch} : the c. at the apex (relative maximum of the throttling curve) of at: unstable throttling curve.

Balance water flow Q_E .

Leakage loss Q_L .

Suction end volume flow: Q_s , the volume flow entering the pump suction branch.

Inlet volume flow Q_e , the volume flow entering the inlet cross-section of the plant.

Discharge end volume flow: Q_d , the volume flow leaving the pump discharge branch.

Outlet volume flow: Q_a , the volume flow leaving the outlet cross-section of the plant.

(Refer to Fig. 2 under head.)

Occasionally, a so-called pump delivery efficiency Q/Q_{opt} and a so-called plant delivery efficiency Q/Q_L are defined, where Q_L is the capacity specified in the supply contract.

The mass flow \dot{m} of the centrifugal pump is $\dot{m} = \rho \cdot Q$, with ρ density of pumped medium.

Capacity Measurement

Förderstrommessung
Mesure de débit

see [Measuring Technique](#)

Capital Investment

Investition
Investissement

see [Economics](#)

Capital Investment Servicing Costs

Kapitaldienstkosten
Coût de capitaux

see [Economics](#)

Carbonate Hardness

Karbonathärte
Durété carbonatée

see [Water Hardness](#)

Care of Centrifugal Pumps and Drives

Pflege von Kreiselpumpen und Antrieb
Maintenance des pompey centrifuges et de commandes

see [Maintenance](#)

Cargo Oil Pump

Ladeölpumpe
Pompe de chargement

C.o.p.'s are installed on board tankers, and are used to pump the oil out of the ship's tanks into the land installation tanks at the port of destination ([stripping system](#)). C.o.p.'s are often used also as ballast pumps for adjustment of the ship's draught and trim. The pump room is usually situated in the stern of the ship, between the tanks and the engine room (Fig. 1).

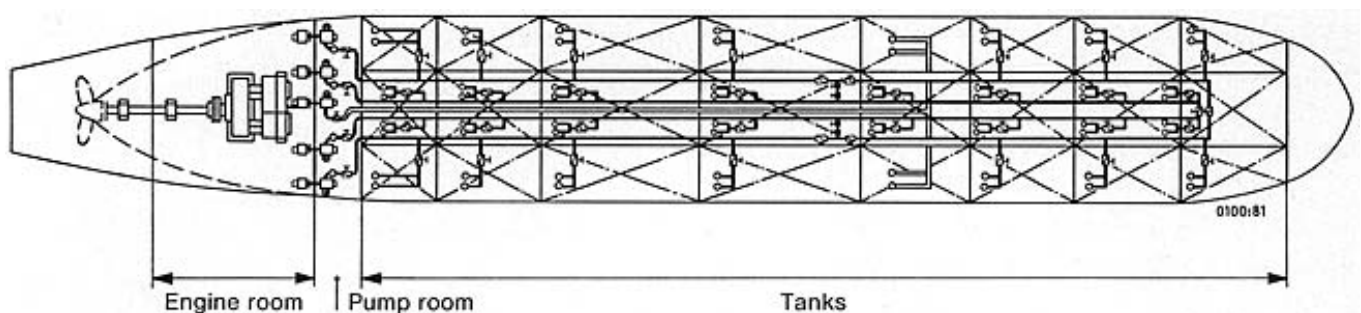


Fig. 1: Diagrammatic illustration of a tanker with suction piping system

C.o.p.'s are installed as low as possible. At the beginning of the discharging operation, the c.o.p.'s operate under a relatively high positive suction head (suction behaviour); when the oil level has sunk very low, they have to operate under suction lifts (suction behaviour) of up to 5 m or more. At this stage of pumping, there is often an ingress of air into the suction lines through the suction strainer heads which are no longer fully immersed in the medium. This air is removed by special evacuating devices before it penetrates into the pumps. C.o.p.'s are usually driven by steam turbines, but occasionally by electric motors or Diesel engines. Control by speed adjustment is generally used. The connecting shaft between the engine and pump room is led through a bulkhead fitted with a wall gland (shaft seals) as a safety precaution against fire and explosion risks.

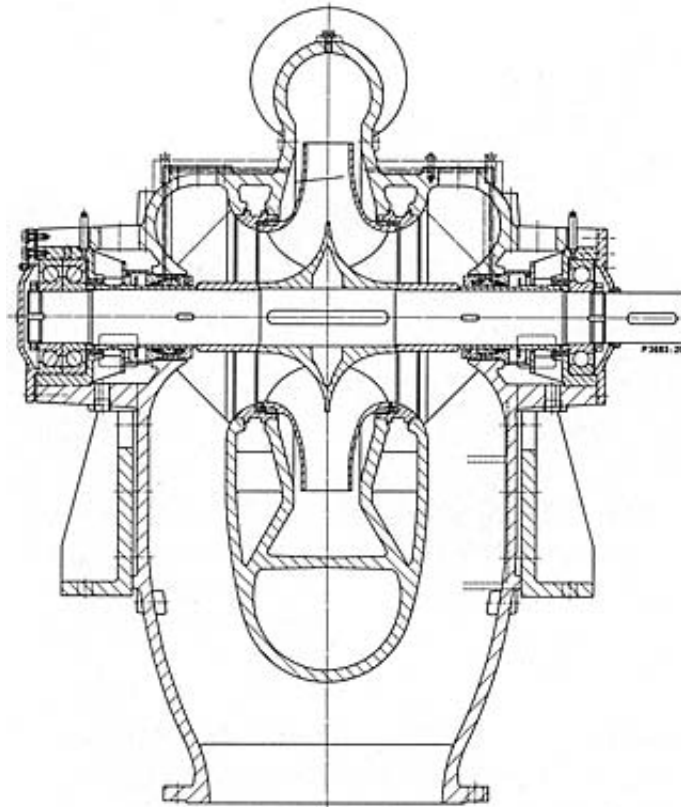


Fig. 2: Horizontal, axially split casing, double suction volute casing pump with radial impeller

The capacities per pump vary according to the size of the tanker from 1500 to 9000 m³/h.

The head averages 150 m. Apart from submersible pumps, which can only be used for small capacities, double suction pumps (multisuction pump) are generally adopted because of their good suction capability at the highest possible operating speeds. Axially split pump casings are generally preferred, both for horizontal and vertical pumps, although radially split casings seem to be gaining ground today for vertical pumps. Fig. 2 illustrates a horizontal axially split c.o.p. whilst Fig. 3 illustrates a vertical radially split c.o.p. of up-to-date design. The latter is a single stage pump fitted with an inducer upstream of the impeller. The casings are usually of red brass, with a double volute (pump casing; Fig. 6 under volute casing pump). The sturdy bearings (oil or grease-lubricated anti-friction bearings) are arranged close to the casing; mechanical seals are generally fitted as shaft seals.

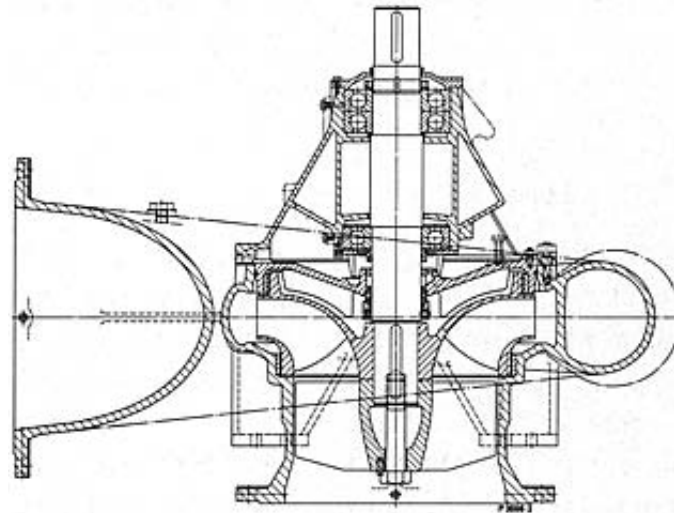


Fig. 3: Vertical, radially split casing, double suction volute casing pump with radial impeller

CARNOT's Shock Loss

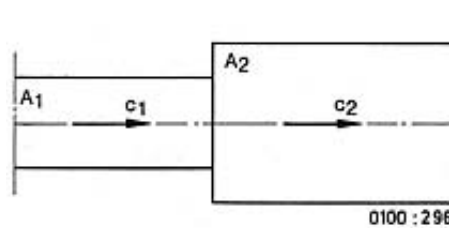
CARNOT'scher Stoßverlust
Perte par choc de CARNOT

With the sudden widening of a pipe diameter from A_1 to A_2 (illustration), the liquid does not enter the wider area as a cleanest, but mixes itself turbulently into the surrounding fluid. At the end of the transition area the liquid once again forms a constant stream of velocity c_2 . The pressure loss Δp due to mixing is called C.s.I. and is defined bathe conversation of momentum (fluid dynamics).

$$\Delta p = \frac{\rho}{2} (c_1 - c_2)^2$$

with

ρ density of the liquid.



Sudden widening of a pipe diameter

Cascade

Schaufelgitter
Grill d'aube

A c. is the regular arrangement of blade profiles (blade, flow profile) along a straight line (plane c.) or on a circle (cylindrical c.). On the co-axial cylindrical section through the axial impeller and diffuser of a propeller pump, we obtain the cylindrical c. of the stage, and if we develop (i.e. unroll) the cylindrical section into a plane, we obtain the plane c. of the stage. Fig. 1 illustrates this c. with its main geometric designations (on the impeller c.). From one cylindrical section to the next, through the stage of the propeller pump, different c.'s will be obtained. The shape of the flow lines in a plane c. (diffuser c.) is illustrated in Fig. 2.

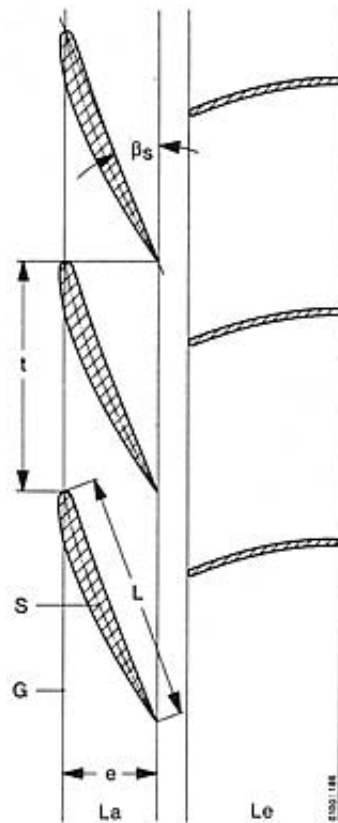


Fig. 1: Plane cascade

- S median line of blade profile;
- G cascade front;
- La impeller cascade;
- Le diffuser cascade;
- L length of blade profile;
- t blade pitch;
- β_s blade angle;
- e blade height

The description of the flow across a c. is frequently the object of theoretical and experimental investigations on axial and mixed flow fluid flow machines (fluid dynamics). Two main problems are involved here: 1. for a given c., the associated flow conditions are sought (direct problems; 2. for given flow conditions, the associated c. is sought (inverse problem). Singularities simulation via the aerofoil theory is applied on the basis of BETZ' fundamentals, according to which a c. may be regarded as an infinite series of individual vanes.

In practice, experimental findings are frequently relied on.

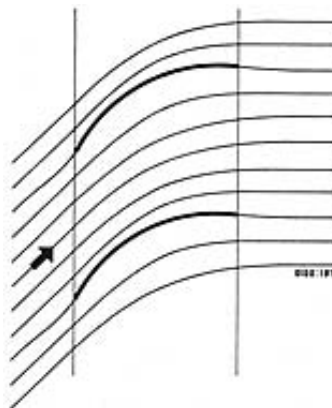


Fig. 2: Flow lines of a plane cascade (potential flow across a diffuser)

Cascade Flow

Gitterströmung

Écoulement à travers one mile d'aubes

see Fluid Dynamics

Casing

Gehäuse
Corps

see [Pump Casing](#)

Cathodic Protection

Kathodischer Schutz
Protection cathodique

see [Corrosion](#)

Cavitation

Kavitation
Cavitation

C., or the formation of cavities in a fluid, is a phenomenon involving the appearance and subsequent sudden collapse of vapour bubbles in a flow of liquid. Vapour bubbles are formed when the static pressure in the liquid sinks so low, e.g. as a result of an increasing absolute velocity (fluid dynamics) or corresponding change in inlet conditions (net positive suction head, suction behaviour), that without the application of external heat it attains the vapour pressure associated with the temperature of the liquid at that particular spot. The relationship between vapour pressure (vaporization pressure) and temperature is given in the Tables under vapour pressure.

If the static pressure subsequently rises again above the vapour pressure, along the flow path, the vapour bubbles collapse quite suddenly, and shock condensation occurs with great rapidity, in the nature of an implosion; if this implosion does not occur in the body of the flowing liquid but at the wall of a component guiding the flow, c. will result in material being eroded (see cavitation criterion under net positive suction head).

Before the occurrence of material disintegration, which incidentally does not necessarily arise in all cases of operation under cavitation conditions, c. makes itself noticed by an increase in the noise level, by rough running (quietness of centrifugal pumps), by a drop in pump efficiency and in head (cavitation criteria, see net positive suction head). In the case of propeller pumps, the head rises slightly at first with incipient c., before sinking under the effect of c., although more slowly than in the case of radial pumps. The typical cavitation noise in centrifugal pumps can best be compared with the noise made by gravel in a concrete mixer.

In order to enable easier observation and studios the phenomena occurring in and near the vapour bubble, bubbles are created artificially outside the walls (by means of a focused laser beam, ultrasonics etc.). Recent research in this field has indicated that the vapour bubble inverts at first when the implosion begins; then a water microjet is formed, directed towards the interior of the bubble, which pierces the opposite wall of the bubble. Slow-motion pictures (approx. 9 x 105 pictures per second) of the phenomenon indicate that, in the case of bubbles in close proximity to walls, this water microjet is always directed at the wall, which it strikes at high speed. This sequence of events, in combination with the fissured microstructure, the very fine pores, cracks and grooves in the wall surface, is considered as the mechanical cause of the disintegration of the material by c.

This type of material disintegration is intensified still further by a series of chemical actions which proceed at an increased pace during this arduous mechanical stressing.

In this connection, we need only mention here the very common occurrence of the disintegration by c. of the surface layers (protective layer) which protect the material, thus leading to increased corrosion, in conjunction with the oxygen entrained in the water. Because such surface layers are often of decisive importance in respect of the suitability of a given material in the presence of corrosive media (table of corrosion resistance), special attention must be paid to this aspect in conjunction with c.

Apart from this so-called vapour c., there is also a different c. called gas c. The difference is that in the case of vapour c., only the vapour of the fluid fills the cavity which is created, whereas in the case of gas c. various gases dissolved in the fluid diffuse out of the fluid into the vapour space. Gas c., like vapour c., can lead to a drop in pump efficiency and head, but it is not as dangerous as the latterly respect of the erosion of the material because the gas trapped in the cavitation bubble has a damping effect during the implosion, due to its compressibility. In the case of gas c., the formation of cavities is not necessarily linked with a pressure decrease down to the vapour pressure of the liquid pumped. As soon as the pressure drops below the gas saturation pressure, gas bubbles are formed in the liquid, and they form the nuclei of the vapour bubbles (gas content of pumped medium) when the pressure decreases further.

Cavitation wear (cavitation erosion). The impingement of the water microjets mentioned previously on solid walls causes violent pressure surges. The concentration of the forces on a narrowly limited attack surface can lead to the destruction of the material. Pitting becomes apparent after a certain period of incubation, first of all at spots of reduced resistance capability (graphite inclusions etc.). Very fine dimples or cavities are formed as a result, and these are exposed to the mechanical and chemical attack previously described. Pitting of the material which occurs has a honey-combed spongy appearance and structure. The amount of material sloughed away by c. on the pump internals can be determined e.g.:

- by geometric measurement
- by the loss of weight,
- by the amount of build-up metal deposited by welding,
- by the time taken to carry out repairs and reconditioning.

If it proves impossible to limit the effects of c. by design or operational measures (e.g. by more gradual transition sections, changes in the inlet conditions), or if it proves impossible to shift the collapse of the vapour bubbles away from the wall towards the centre of the flow path, then the wear caused by c. can only be reduced by selecting appropriate materials of construction (selection of materials).

Materials resistant to c. are those with a high fatigue strength, combined with a high corrosion resistance. For instance, if we give cast iron (GG-25) a cavitation-induced loss of weight index of 1.0, we have the following approximate graduation for other materials (see Table: typical cast materials for centrifugal pumps; under materials) in the direction of greater resistance to c.:

cast steel (GS-C 25)	Index 0.8
bronze (G-CuSn 10)	Index 0.5
cast chrome steel (G-X 20 Cr 14) multicomponent bronze	Index 0.2
(G-AlBz 10 Fe) chrome nickel steel	Index 0.1
(G-X 6 CrNi 18 9) NORIDUR®	Index 0.05
(G-X 3 CrNiMoCu 24 6)	Index 0.02

The above index values represent average values, arrived at by interpreting international literature and carrying out experimental studies in devices allowing a systematic cavitation generation, taking into account a number of uncertainty factors. It is not possible to consider them as absolute values for evaluations, because the individual index numbers are also highly dependent on the type of cavitation stressing.

Another factor of paramount importance for c. erosion is the chemical and electrochemical behaviour of the substances involved, viz. the fluid and the parent metal.

Cavitation Erosion

Kavitationsverschleiß

Usure par cavitation

see Cavitation

Cavitation Noise

Kavitationsgeräusch
Bruit de cavitation

see [Cavitation](#)

Cellar Drainage Pump

Kellerentwässerungspumpe
Pompe vide-cave

see [Drainage Pump](#)

Cellulose

Zellstoff
Pâte épaisse

see [Fibrous Material](#)

Center Conductor

Mittelleiter
Fil neutre

C.c., also known as neutral conductor, is the electrical conductor with the potential of the center; this is the point for which the sum after potential differences of the main conductors (outside wires) disappears in the case of symmetrical (balanced) loading of the three-phase system. The c.c. of a three-phase system is connected to the star point of the generator or transformer ([terminal designation](#), [three-phase current](#)).

In the case of unsymmetrical (unbalanced) loading, e.g. if an additional single phase user is connected side by side with the three-phase current users, a compensating current flows in the c.c.

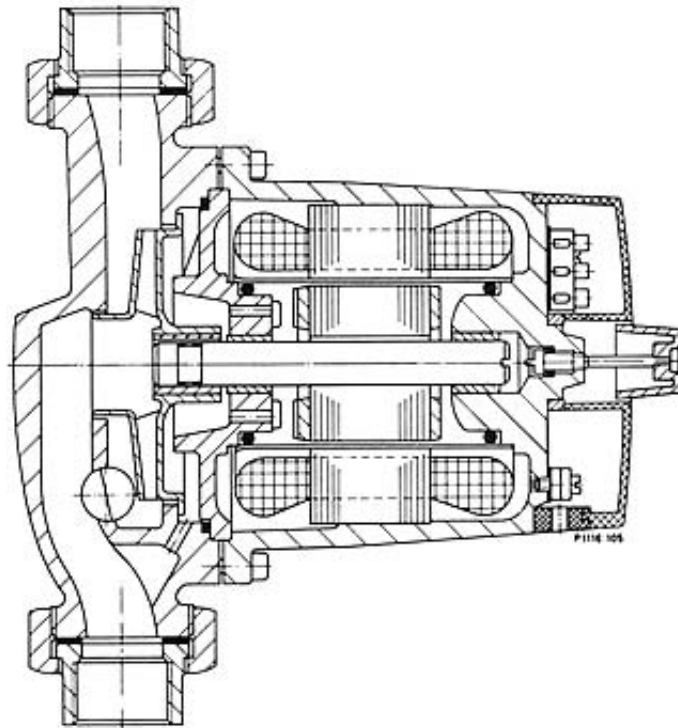
The c.c. can be earthed either rigidly or via a choke (earthed system or system with ground-fault neutralizer) or it can be insulated (insulated system).

Central Heating Circulating Pump

Heizungs-Umwälzpumpe
Pompe de chauffage

The c.h.c.p. maintains a forced circulation in hot water central heating systems. In many cases, the very small static head differences in the heating system are insufficient to create an adequate natural circulation velocity as a result of the differences in density, also called thermosiphon effect, which arise. This makes the provision of a c.h.c.p. necessary. Because of the forced circulation, the pipes can be of smaller bore than in the case of natural circulation systems, which makes for a lower plant cost. Small c.h.c.p.'s up to 2 kW approx. are equipped with canned motors ([canned motor pump](#), see illustration); this ensures absolute leak-tightness in the region of the c.h.c.p. Larger c.h.c.p.'s are constructed in the form of [close-coupled pumping sets](#) with mechanical seals or soft packed stuffing boxes ([shaft seals](#)). C.h.c.p.'s are also installed in the form of [twin pumping sets](#), with one c.h.c.p. acting as service pump and the other as standby. An appropriate electrical switching device ensures that the standby pump automatically starts up immediately after a failure of the service pump. The isolating valves ([valves and fittings](#)) necessary for the switchover are integrally built into the pumping set. The [twin pumping set](#) has only one pump suction and one pump discharge branch. C.h.c.p.'s are controlled either by switching on and off or by speed adjustment. Combinations of a mixing valve and a c.h.c.p. are also known; the heat requirement is sensed automatically either by room temperature sensor, or by an outside

sensor, or by a combination of both. In simple installations, the mixture ratio of boiler headwater and system return water is adjusted by hand.



Central heating circulating pump with canned motor

Centrifugal Pump

Kreiselpumpe

Pompe centrifuge

The c.p. is a work-producing machine according to the direction of the energy flow, a fluid flow machine according to the nature of the energy conversion, and a hydraulic fluid flow machine according to the nature of the fluid. The many varieties of c.p.'s are described fully under the headings pump types and application fields for pumps. The characteristic magnitudes of a c.p. include capacity, head, suction behaviour, flow velocity, total head, pressure, elevation (geodetic altitude), power (shaft power), efficiency (pump efficiency), rotational speed, specific speed, etc.

Centrifugal Pump Plant

Kreiselpumpenanlage

Installation de pompe centrifuge

see Pumping Plant

Change in Cross-Section

Querschnittänderung

Variation de section

see Pressure Loss

Channel Vortex

Kanalwirbel
Tourbillon relatif

see [Fluid Dynamics](#)

Characteristic

Charakteristik
Course caractéristique

see [Characteristic Curve](#)

Characteristic Curve

Kennlinie
Courbe caractéristique

The c.c.'s of centrifugal pumps plot the course of the following magnitudes over capacity Q : head H (throttling curve), shaft power P , pump efficiency η and NPSH value (net positive suction head) of the centrifugal pump.

The shape of the c.c. depends primarily on the pump type (impeller, pump casing, specific speed) (Figs. 1 to 4). Secondary influences, such as the cavitation, manufacturing tolerance, size and physical properties of the medium pumped (viscosity, hydrotransport, pulp pumping) are not taken into account in these diagrams.

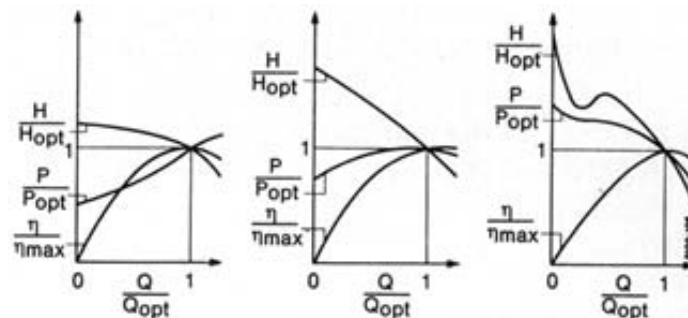


Fig. 1: Characteristic curves of centrifugal pumps of various specific speeds. Curves plotted in percentage ratios to the optimum point in each case

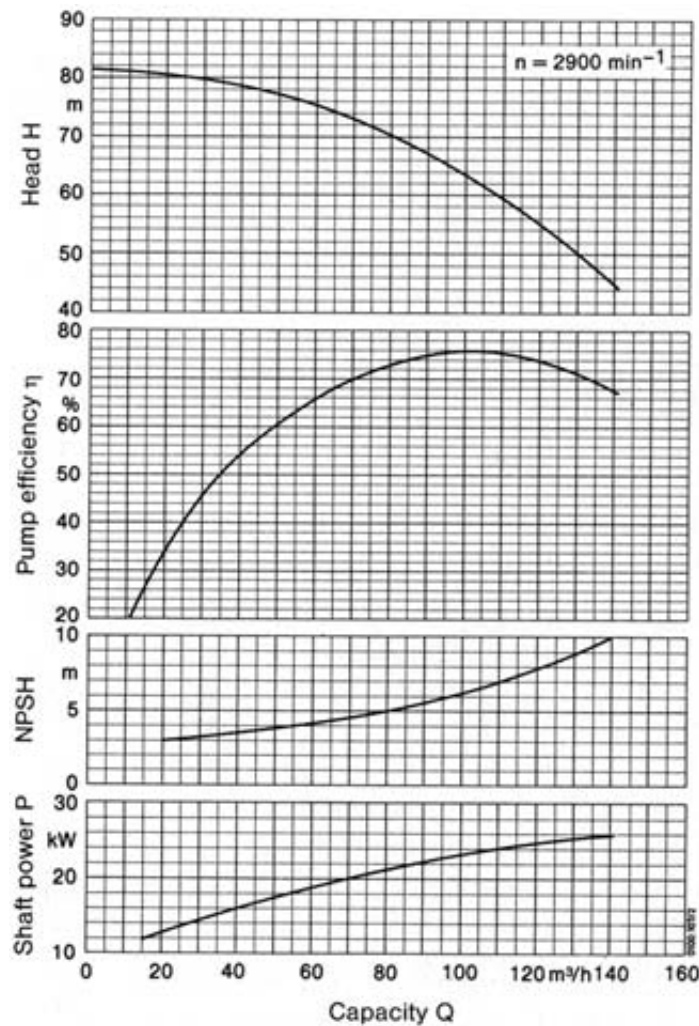


Fig. 2: Characteristic curves of a radial centrifugal pump, specific speed $n_q \approx 20 \text{ min}^{-1}$

In Fig. 1, the c.c.'s $H(Q)$, $P(Q)$, and $\eta(Q)$ are plotted in relation to the specific speed, i.e. for various impeller types. As the specific speed increases, the slope of the QH c.c. becomes steeper (for DIN definition of a flat and steep throttling curve, see manufacturing tolerance) The efficiency curve of low specific speed centrifugal pumps is relatively flat, whereas in the case of higher specific speed pumps the curve becomes peaky (more pointed). The shaft power curve of low specific speed pumps has its minimum value at $Q = 0$ (shut-off point), whereas in the case of high specific speed pumps, the shaft power is maximum at pump shut-off point.

Fig. 1 is intended to illustrate the qualitative pattern of the c.c.'s, whereas Figs. 2 to 4 reproduce the actual quantitative patterns of the c.c.'s of manufactured centrifugal pumps of various specific speeds.

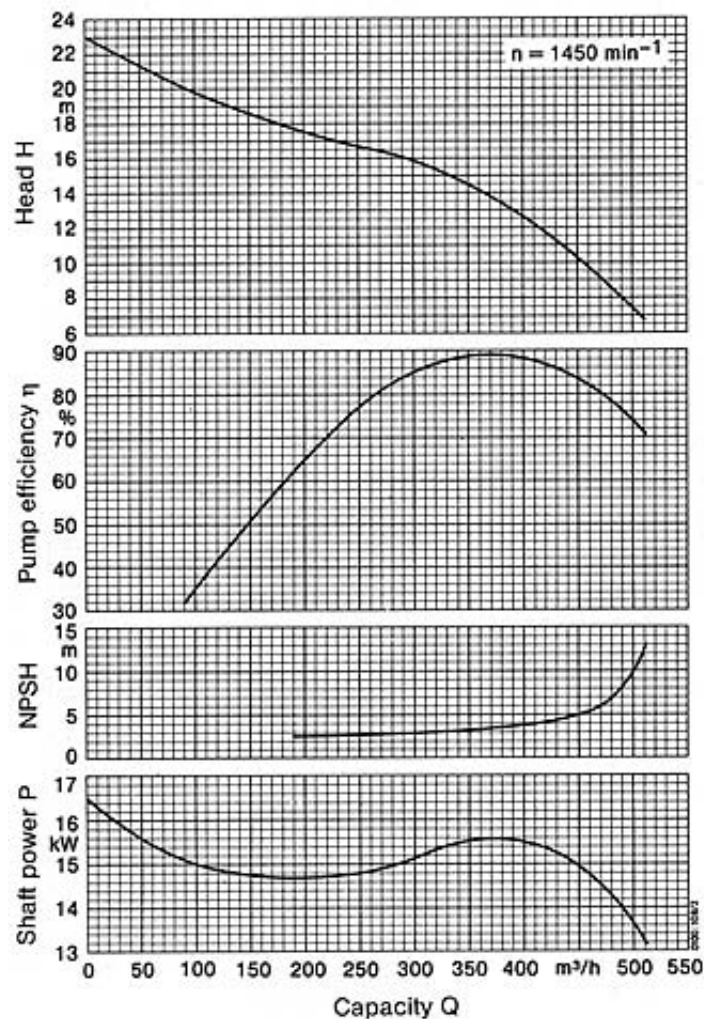


Fig. 3: Characteristic curves of a mixed flow centrifugal pump, specific speed $n_q = 80 \text{ min}^{-1}$

For the normal operating range of centrifugal pumps (n , Q and H all positive), it will be sufficient to plot the c.c. in the first quadrant of the Q - H system of coordinates (Figs. 1 to 4). The operating points which are not normally travelled through are situated in the other three-quadrants. These include e.g. the operation of a centrifugal pump as a turbine (turbine operation of centrifugal pumps), the behaviour of a centrifugal pump on failure of the drive, or the starting of a centrifugal pump (starting torque) in reverse rotation. The predetermination of the curves beyond the normal operating range of the centrifugal pump is by no means solely the outcome of theoretical considerations. It is necessary to resort to experimental results to solve this problem. These experimentally established curves represent the complete characteristic (fourquadrant performance chart) of centrifugal pumps, and they will depend on the pump type; for instance, Fig. 5 illustrates the complete characteristic of a double suction centrifugal pump with $n_q = 35 \text{ min}^{-1}$, according to STEPANOFF. Of the various alternative diagrammatic representations it became apparent that the clearest overall picture would be obtained by plotting the rotational speed (relative rotational speed n/n_N) Over the capacity (relative capacity Q/Q_{opt}) with the head H and the torque T (Starting torque) of the centrifugal pump as parameters. All magnitudes are specified as percentages of their design duty values (including negative values), in order to facilitate the transmission of the results to all similar pumps.

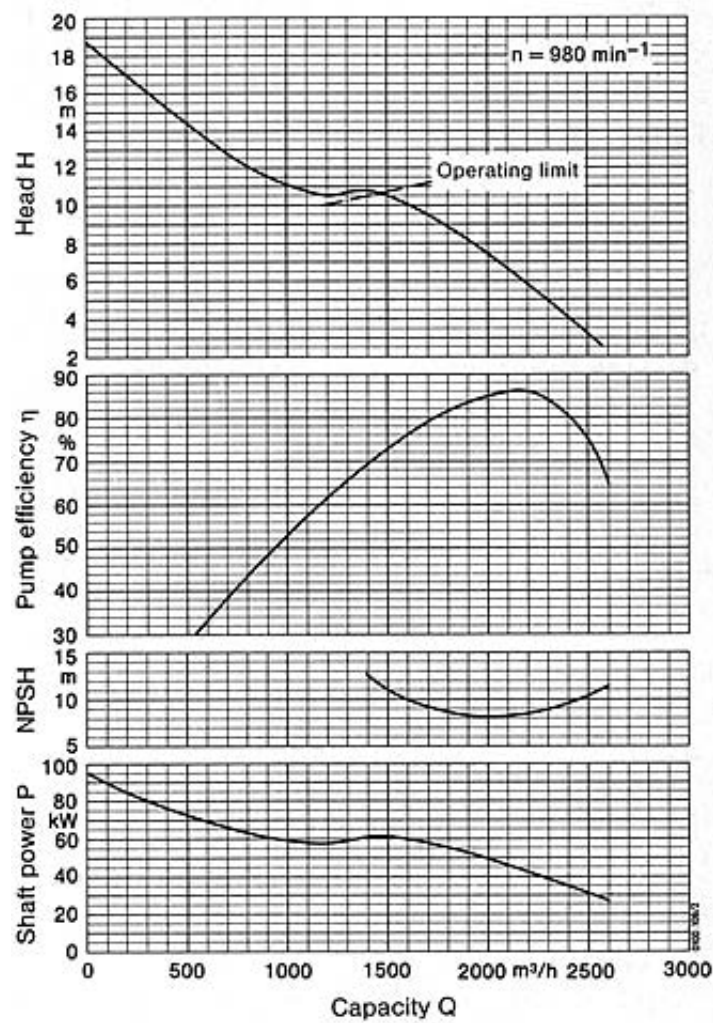


Fig. 4. Characteristic curves of an axial pump, specific speed $n_q \approx 200 \text{ min}^{-1}$

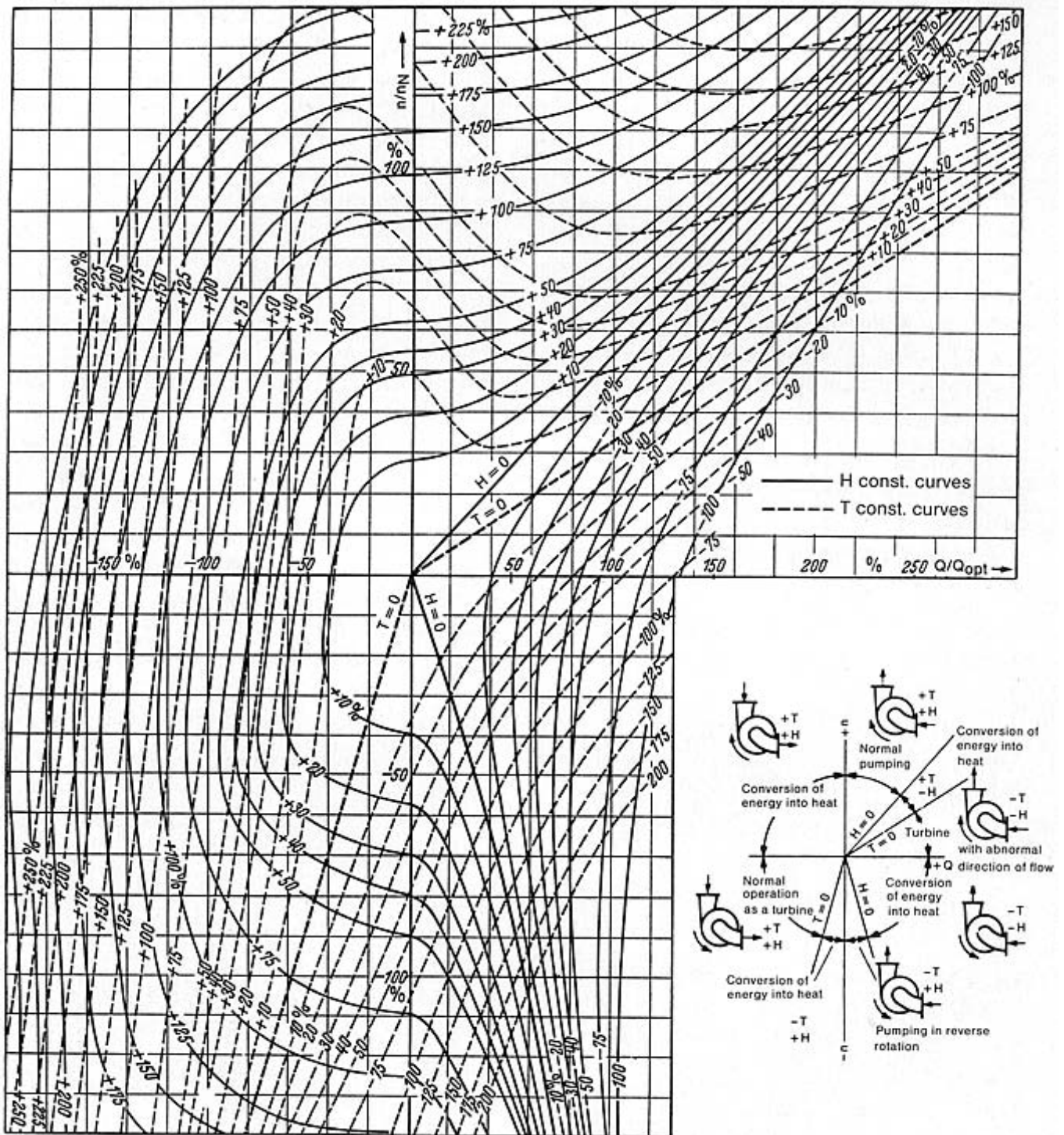


Fig. 5: Complete characteristic of a double suction centrifugal pump with $n_Q = 35 \text{ min}^{-1}$ according to STEPANOFF.

- $Q > 0$ inflow through suction branch,
- $Q < 0$ inflow through discharge branch,
- $H > 0$ discharge head at discharge branch higher than suction head at suction branch,
- $H < 0$ discharge head at discharge branch lower than suction head at suction branch,
- $n > 0$ direction of rotation as for normal pump operation,
- $n < 0$ direction of rotation as for normal turbine operation,
- $T > 0$ shaft torque in direction of rotation as for normal pump operation,
- $T < 0$ shaft torque in direction of rotation opposite to that of normal pump operation

Characteristic Number

Kennzahl

Nombre caractéristique

Various c.n.'s are used to characterize the operating behaviour and the construction type of fluid flow machines. For the derivation of the c.n. from the individual physical entities, see description under model laws. The unit of c.n. is 1. The most important c.n.'s of centrifugal pumps are:

a) C.n.'s which characterize the operating behaviour:

- Φ flow coefficient,
- Ψ pressure coefficient,
- λ performance coefficient.

b) C.n.'s which characterize the pump type:

- σ running speed coefficient,
- δ diameter coefficient.

The *flow coefficient*, also referred to as a capacity, delivery or volume coefficient in the technical literature, characterizes the capacity Q and is defined as

$$\Phi = \frac{v_m}{u}$$

where

v_m is the component in the meridian plane of the absolute velocity (velocity triangle),

u is the circumferential velocity of the impeller.

When the capacity varies at constant rotational speed of the pump, $v_m \sim Q$ and therefore $\Phi \sim Q$. The flow coefficient Φ is therefore indicative of the abscissa (analogous to Q) on throttling curves (characteristic curve) plotted in nondimensional representation. Φ can be related either to the inlet diameter of the blades (in which case $\Phi_1 = v_{1,m}/u_1$) or to the outlet diameter of the blades (in which case $\Phi_2 = v_{2,m}/u_2$).

Because of the association with the pressure coefficient, it is preferable to adopt the flow coefficient related to the impeller outlet, with the notation

$$\Phi = \frac{v_{2,m}}{u_2}$$

The pressure coefficient characterizes the head H of the pump and is defined as

$$\Psi = \frac{2g \cdot H}{u_{2,a}^2}$$

where

$u_{2,a}$ circumferential velocity of impeller outlet (index 2) at the extreme tip diameter (index a) of the blade, and

g gravitational constant.

When the head varies at constant rotational speed of the pump, $\Psi \sim H$. The pressure coefficient Ψ is therefore indicative of the ordinate (analogous to H) on throttling curves plotted in non-dimensional representation. If we introduce the specific energy Y of the pump, we have

$$\Psi = \frac{Y}{u_{2,a}^2/2}$$

The *performance coefficient* characterizes the shaft power (or power input) P of the pump, and if we introduce the pump output $P_Q = \rho \cdot g \cdot Q \cdot H$ and the shaft power $P = P_Q / \eta$, we obtain

$$\lambda = \frac{P}{\frac{\rho}{2} \cdot u_2^3 \cdot A_2} \quad \text{or} \quad \lambda = \frac{\varphi \cdot \psi}{\eta}$$

where

ρ density of pumped medium,
 η pump efficiency, and
 A_2 characteristic area with $A_2 = Q/v_{2,m}$.

The *running speed coefficient*, which is similar to the specific speed n_q , is an entity which characterizes the impeller shape of an hydraulically ideal pump construction;

$$\sigma = \frac{\varphi^{1/2}}{\psi^{3/4}}$$

With φ and ψ referring to the point of best efficiency.

If n_q is given, the running speed coefficient is calculated by means of the numerical value equation below:

$$\sigma = \frac{n_q}{157.8}$$

where n_q is expressed in min^{-1} .

We can plot the running speed coefficient σ in function of the *diameter coefficient*

$$\sigma = \frac{\psi^{1/4}}{\varphi^{1/2}}$$

for the optimum values Q_{opt} and H_{opt} . Of all centrifugal pumps at their respective nominal speed of rotation n_N , and all the points thus plotted on a $\delta - \sigma$ system of coordinates for hydraulically ideal centrifugal pumps will be found to group themselves around a hyperbolashaped curve, the so-called CORDIER curve. Therefore the running speed coefficient and the pressure coefficient play an important part right from the start in the design calculations of pumps, for the purpose of establishing the optimal pump type to be selected from the point of view of efficiency.

Chemical Pump

Chemiepumpe
Pompe pour l'industrie chimique

C.p.'s are pumps designed for handling cold or hot liquids used in the chemical, petrochemical and food-stuffs industries, also in the offside zones of refineries and in high temperature heating plants (vapour pressure). Typical media handled by c.p.'s may be corrosive, highly volatile, explosive, toxic, clean to slightly contaminated or very expensive.

The c.p. components in contact with the fluid pumped are manufactured from corrosion-resistant metallic or non-metallic materials, or are fitted with durable linings or coatings of rubber (Fig. 1), plastic or enamel.

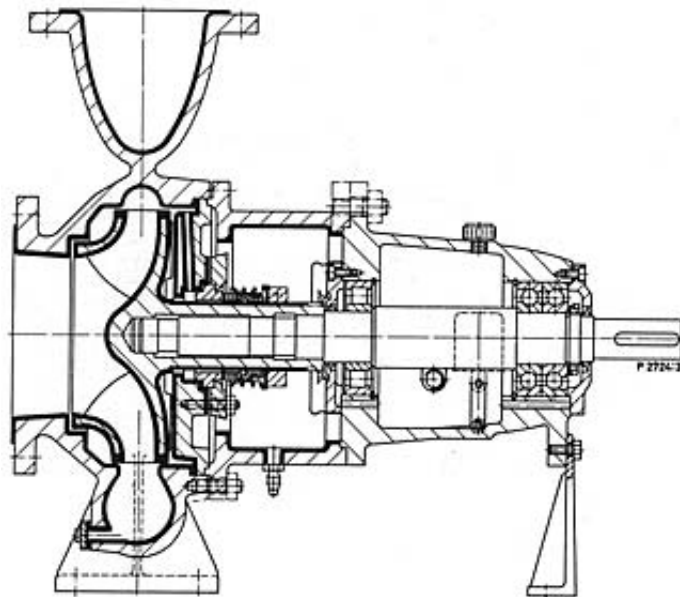


Fig. 1: Standard chemical pump with rubber lining

Suitable shaft seals ensure that no fluid can leak out of the pump, either during operation or shutdown. Canned motor pumps (Fig. 2), also known as glandless pumps or pumps without shaft seal (wet rotor motor), represent a special construction type without a shaft seal.

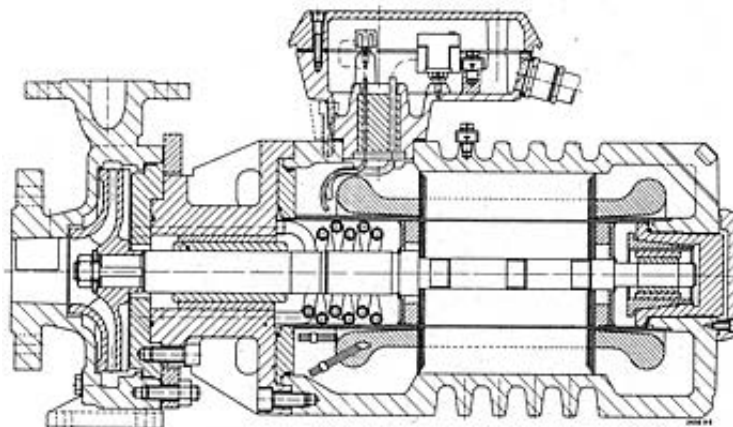


Fig. 2: Chemical pump, canned motor pump

In recent years pumps with magnetic couplings (Fig. 3) have been manufactured as glandless pumps. As opposed to canned motor pumps, a standard electric motor is used in these models. Disadvantages of these pumps versus canned motor pumps are: greater losses in power transfer and less safety in case of a rupture of the can; in the pump with magnetic couplings the leakage will make its way to the outside, while the canned motor pump retains the leakage in its pressure sealed motor housing. Thus, double-walled cans with a leakage indicator are occasionally used in these pumps.

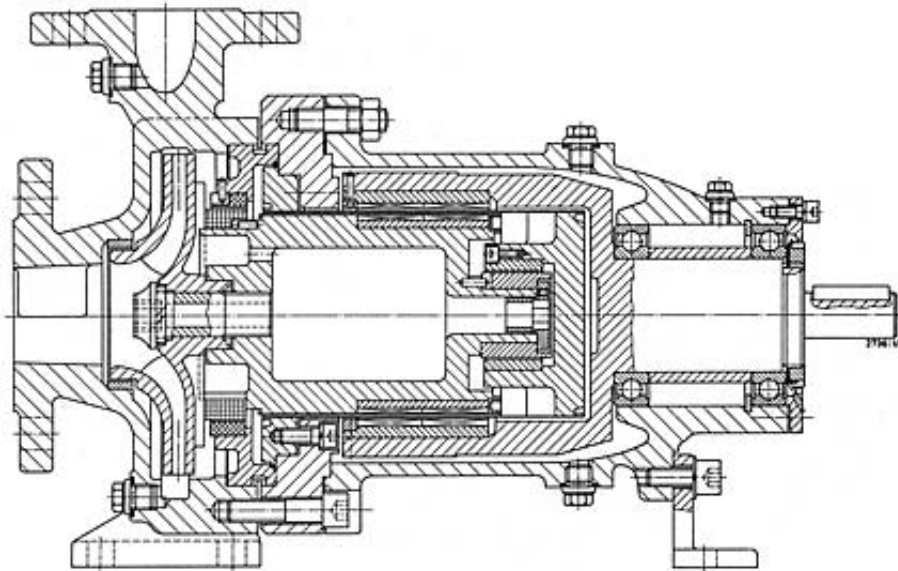


Fig. 3: Chemical pump with drive over permanent magnetic coupling

The leakage-free hydrodynamic shaft seal has proved itself when compared to conventional seals, especially for liquids which require an expensive auxiliary seal water system. An auxiliary impeller keeps the shaft opening free of fluid during operation, whereas another seal, a shutdown seal, keeps the shaft opening free of fluid during pump shutdown (Fig. 7 under shaft seals).

C.p.'s can be either horizontal or vertical, or of submersible pump (shaft sump pumps) for installation inside vessels (Fig. 4).

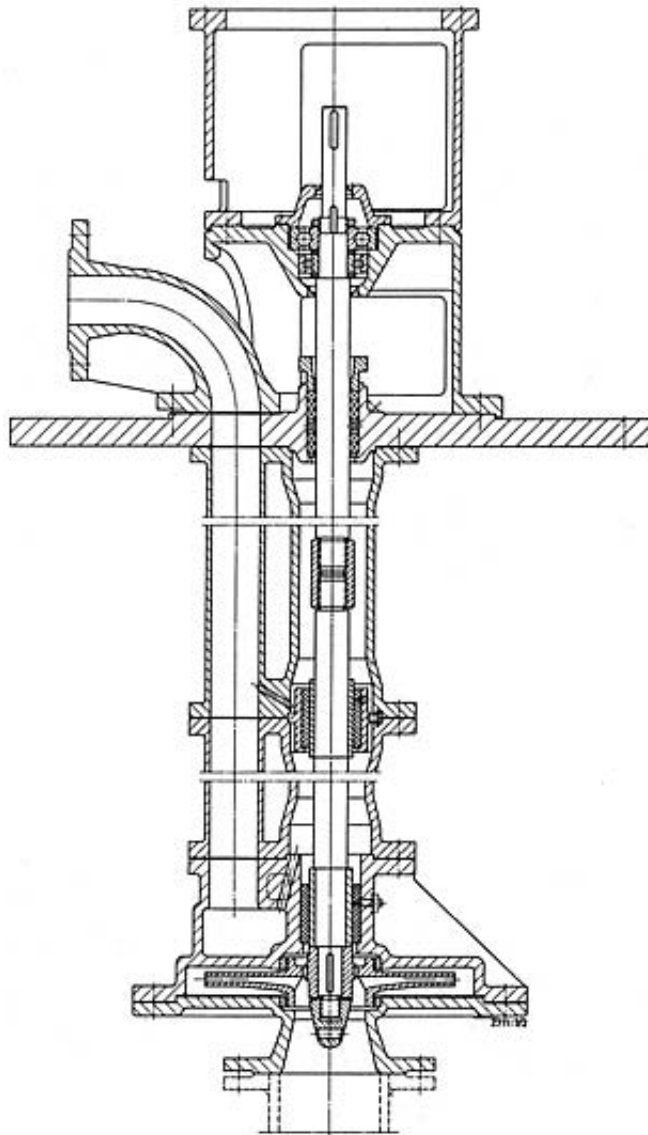


Fig. 4: Chemical pump for use as submersible pump

In the chemical and petrochemical industries, above all standard chemical pumps according to DIN 24256 or ISO 2858 specifications are being installed (Fig. 5). In the USA, a corresponding standard pump series is specified in ANSI B 73.1. The installation dimensions and hydraulic performances of these pumps are standardized, so that existing pumps can be replaced by others irrespective of manufacture. These pumps are of process type design; which enables the bearings, shaft seal and impeller to be dismantled as a unit without disconnecting the pipings from the casing (pump casing); if a spacer type coupling is fitted, the motor can also remain in situ on the baseplate, so that there is no need to re-align the pumping set (pump and motor) after re-assembly.

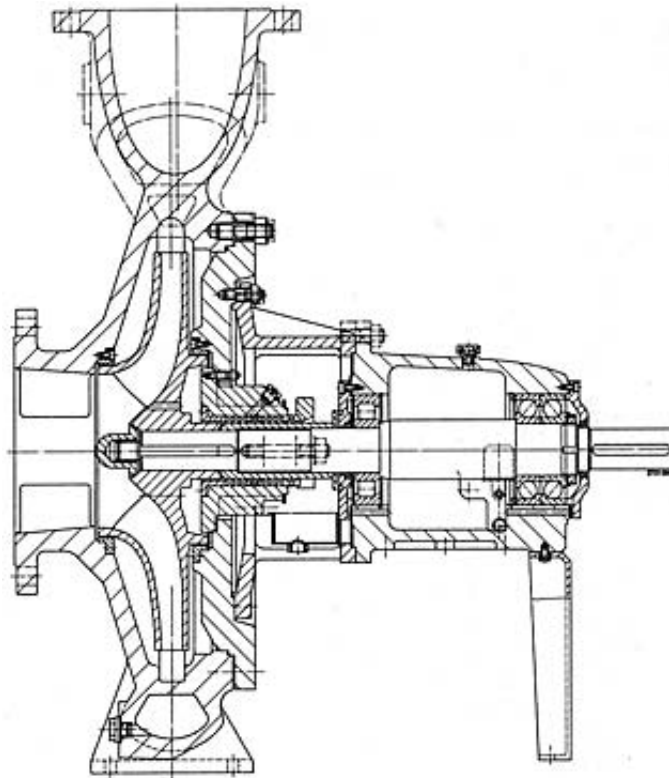


Fig. 5: Standardized chemical pump according to DIN 24256 and ISO 2858

C.p.'s are mainly single stage volute casing pumps (Fig. 5). but sometimes two stage, and are generally non-self-priming (self-priming pump). A special design feature is a low NPSH requirement (net positive suction head) of such pumps. To cater for special operating conditions, special types (variants) with heating jackets (Fig. 12 under pump casing), intensively cooled casing covers or coolable bearing brackets are provided.

C.p.'s according to DIN 24256 or ISO 2858 have specific demands to meet in accordance with various specifications such as, for example, ISO 5199, VDMA 24297, and DIN 24295, which target such problems as allowable vibrations, deformation of shafts, noise, pipe force and moments (branch loading), safety requirements, and foundation-less installation of centrifugal pumps. Regarding nameplates of horizontal c.p.'s: with drive motor see standard ISO 3661/ DIN 24259.

Circular Arc Vane

Kreisbogenschaufel
Aube en arc de cercle

see Blade

Circulating Pump

Umwälzpumpe
Pompe de circulation

C.p.'s are centrifugal pumps designed to generate a forced circulation in a closed system, e.g. in a central heating installation (central heating circulating pump), in a forced circulation boiler or in a reactor circuit (reactor pump), but also in open systems such as swimming pool filter systems (Fig. 1). The construction of the c.p. is determined by the frequently very high temperature of the medium pumped and by the very low head in relation to the system pressure, which corresponds to the loss of head (pressure loss) in the circulation system. The following types of c.p.'s are to be found:

1. *C.p.'s with shaft seal*. The pump shaft is sealed against the full closed system pressure by means of cooled soft packed stuffing boxes or mechanical seals; therefore the thrust bearing (anti-friction bearing) must have a particularly robust design to absorb the high static axial thrusts (Fig. 2). This type of construction is economical for system pressures up to 100 bar approx., as a general rule; for higher system pressures, grandness c.p.'s are used. C.p.'s with a shaft seal are generally horizontal pumps driven by an electric motor (drive) or a steam turbine.

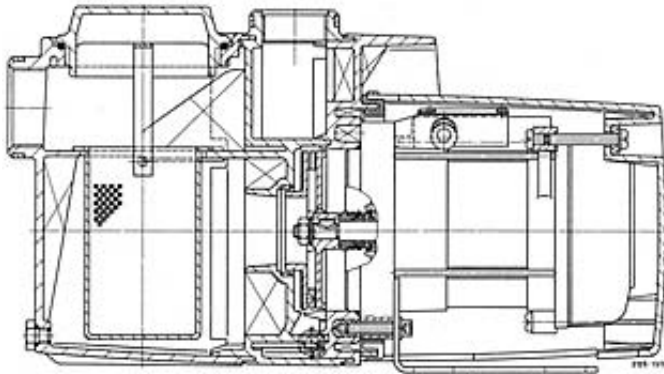


Fig. 1: Self-priming plastic circulating pump for swimming pool filter systems

2.

Glandless c.p.'s (Fig. 3). In these pumps, which are usually vertical pumps driven by a wet rotor motor with or without a can (canned motor pump), the pump and electric motor are arranged in a common pressure-tight casing with a thermal barrier cooled by a fluid from an outside source arranged between the pump and the motor. The bearings (plain bearing) are lubricated by the product pumped, the system pressure is practically unlimited and the total absence of leakage makes these pumps suitable for handling dangerous or expensive media. If a canned motor is used as drive, the complete internal surface of the c.p. can be made of corrosion-resistant materials (corrosion).

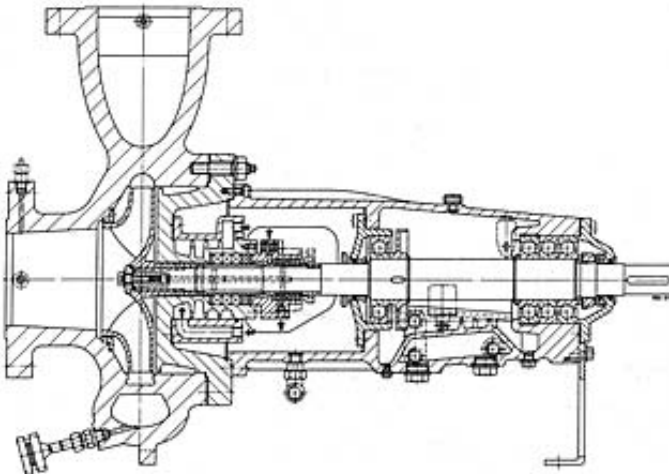


Fig. 2: Circulating pump with cooled soft packed stuffing box for forced-circulation boiler plants

3. *Special c.p. constructions*. Such pumps are built in accordance with the special application requirements, and they can have special impellers for handling mixtures of liquids and gases or suspensions, sealing systems incorporating an inert gas cushion or for other special requirements for nuclear reactor technology, heating devices for media which congeal rapidly, protective linings or coatings against abrasive wear (armoured pump, erosion), or they can be built in explosion-proof construction (explosion protection).

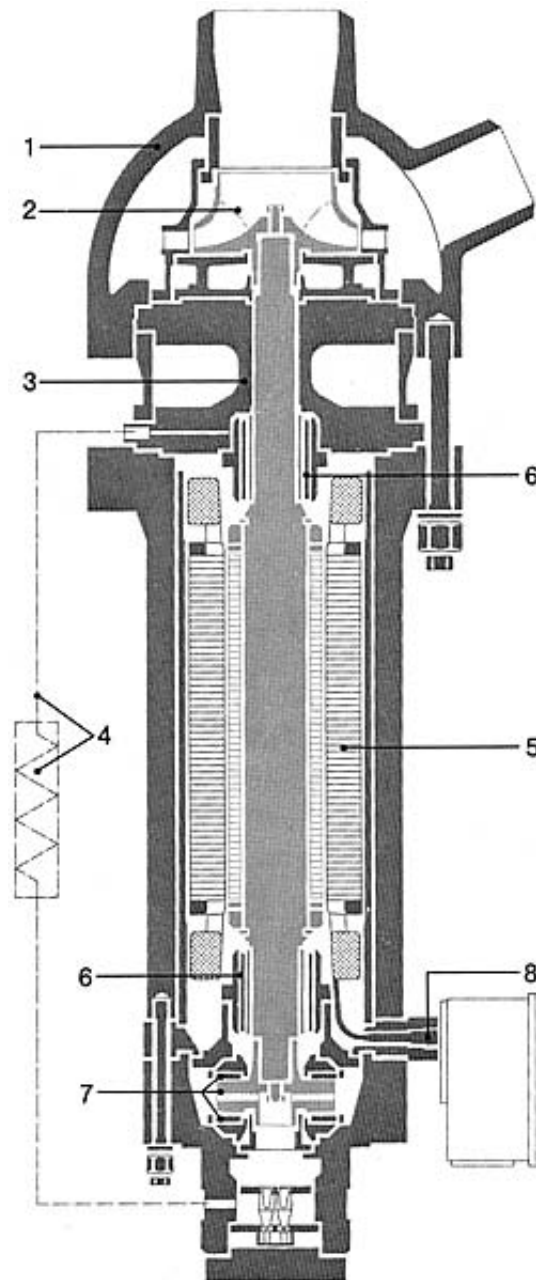


Fig. 3: Glandless circulating pumps with wet rotor motor and air-cooled heat barrier for steam power stations (cf. Fig. 2 under wet rotor motor)

1. pump casing;
2. pump impeller;
3. heat barrier;
4. high-pressure cooling circuit with heat exchanger;
5. asynchronous motor with plastic-insulated wet winding;
6. radial bearing;
7. thrust bearing;
8. cable glands

Circumferential Velocity

Umfangsgeschwindigkeit
Vitesse périphérique

The c.v. u is the velocity of a point particle executing a circular motion, e.g. a point particle of a rotating pump shaft or of a rotating impeller. The following relationship exists between the radius r of the point particle and the angular velocity ω (rotational speed): $u = r \cdot \omega$.

In fluid flow machine design, u , together with the relative velocity w and the absolute velocity v , forms the velocity triangle.

The c.v. has a marked influence in centrifugal pumps on the suction behaviour, the noise development and the strength of the rotating components. Normal c.v.'s of centrifugal pump impellers (at the outlet o.d.) range between 20 and 60 m/s, in exceptional cases up to 140 m/s.

All velocity components parallel to the c.v. are designated circumferential components of the flow.

Clean Water Pump

Reinwasserpumpe
Pompe d'eau pure

The c.w.p. is a water supply pump for pumping clean (or treated) water, e.g. drinking water, surface water (streams, rivers, ponds, sea), groundwater (wells, sources). In contrast we have dirty water pumps (sewage pump).

Clearance Gap Cavitation

Spaltkavitation
Cavitation marginale

In slit seals, particularly in the impeller clearance gap (clearance gap width) between the pump casing and the blade tips of open axial and mixed flow impellers (not fitted with an outer cover-plate), very high local flow velocities can arise as a result of the static pressure difference upstream and downstream of the clearance gap (clearance gap width), further encouraged by the sharp edges on the clearance gap. As a result, a correspondingly low static pressure is set up in the clearance gap, which may fall down to the vapour pressure of the fluid, even if the NPSH value (net positive suction head) of the pump is adequate. C.g.c. results from the partial evaporation of the fluid in the clearance gap. In the region of lower flow velocity downstream of the clearance gap, the vapour bubbles collapse (implode), a phenomenon which leads to the wellknown cavitation wear, particularly at the pump casing in the region of the impeller clearance gap. Some of the vapour bubbles are entrained by the fluid into the main stream, where they can act as cavitation nuclei and contribute to a premature development of cavitation.

Clearance Gap Loss in Centrifugal Pumps

Spaltverlust in Kreiselpumpen
Perte par fuit einterne d'une pompe centrifuge

Due to the difference in pressure upstream and downstream of the impeller (clearance gap pressure), part of pumped medium that has already been raised to the higher static pressure flows through the clearance gap (clearance gap flow) between the stationary and rotating portions of the pump. The volume flow in the pump discharge branch is lower than that of the impeller, with the difference amounting to the clearance gap flow. The loss of output in this respect is designated c.g.l.i.c.p., and it can arise between impeller and pump casing (clearance gap width), in slit seals, at balancing devices, and, in the case of multistage pumps, between the individual stages.

The relationship below is used to calculate the clearance gap flow Q_{sp}

$$Q_{sp} = \zeta_{sp} \cdot A_{sp} \cdot \sqrt{\frac{2 \cdot \Delta p_{sp}}{\rho}}$$

and the loss of output at the clearance gap is given by

$$P_{v.sp} = \frac{\rho \cdot g \cdot Q_{sp} \cdot H}{\eta_h}$$

with

ζ_{sp}	clearance flow coefficient,
A_{sp}	clearance gap area,
Δp_{sp}	difference in static pressures upstream, and downstream of the clearance gap,
g	<u>gravitational constant</u> ,
ρ	<u>density of pumped medium</u> ,
η_h	<u>hydraulic efficiency</u> ,
H	<u>head</u> .

The loss of output at the clearance gap is influenced by the REYNOLDS number of the clearance gap flow, the surface roughness and the configuration of the clearance gap. The loss of output at the clearance gap $P_{v.sp}$ related to the shaft power P , for geometrically similar pumps with geometrically similar clearance gaps, is independent of the pump size and of the circumferential velocity, but very markedly dependent on the specific speed n_q . For a simple cylindrical clearance gap ring (neck ring) at the impeller inlet, with the relative clearance gap width $s/D_{sp} = 0.002$ and a relative clearance gap ring width $b/D_{sp} = 0.133$, we obtain a curve approximately as plotted in Fig. 1. The increase in the relative clearance gap results, as shown in Fig. 2 in an approximately linear increase in the clearance gap losses. Fig. vindicates that the clearance gap loss at specific speeds $n_q > 30 \text{ min}^{-1}$ is negligible. Only for $n_q < 30 \text{ min}^{-1}$ is a complicated and expensive reduction of the clearance gap losses worth while, e.g. by narrowing the clearance gap (Fig. 2).

The c.g.l.i.c.p. can be further reduced through the use of threaded feed bushes (shaft seals).

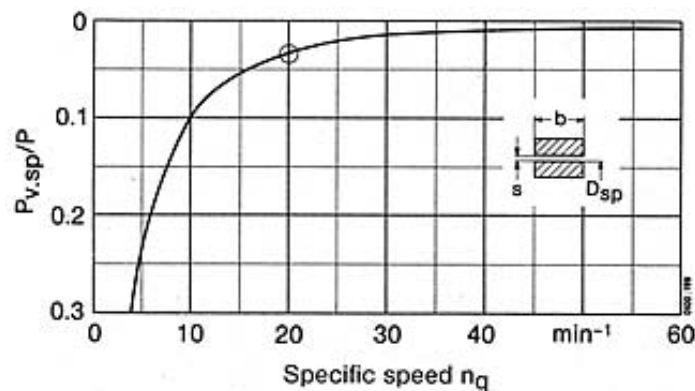


Fig. 1: Loss of output at clearance gap, divided by shaft power, in function of specific speed of pumps with radial impellers, plotted here for a smooth surface cylindrical clearance gap with $s/D_{sp} = 0.002$ and $b/D_{sp} = 0.133$

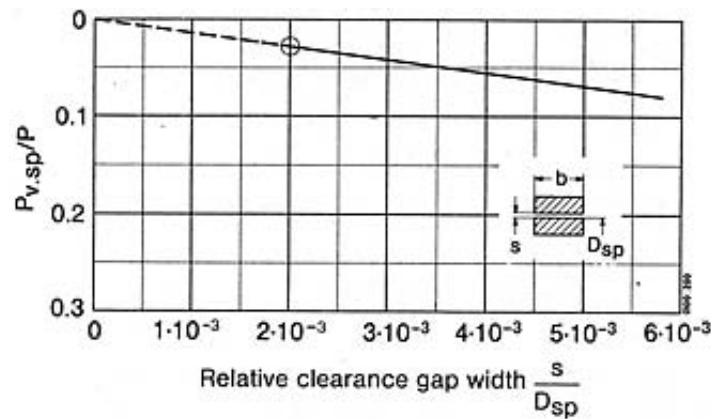


Fig. 2: Loss of output at clearance gap, divided by shaft power, for a volute casing pump with a specific speed $n_q = 20 \text{ min}^{-1}$ and a smooth surface cylindrical clearance gap with $b/D_{sp} = 0.133$, plotted in function of the relative clearance gap width

Clearance Gap Pressure

Spaltdruck

Pression dans une chicane

C.g.p. is the static pressure differential between the pressure at the discharge (or outlet) end of the impeller and at the suction (or inlet) end of the impeller, related to an impeller element along a flow line (impeller).

Clearance Gap Width

Spaltweite

Valeur de jeu

On slit seals (shaft seals) the c.g.w. usually, represents the distance (clearance gap) between the rotating and the stationary component (leakage loss, clearance gap loss in centrifugal pumps).

The c.g.w. has a special significance in the case of axial and mixed flow impellers of centrifugal pumps; in this case the c.g.w. represents the clearance gap between blade tip and casing wall. The c.g.w. has a considerable influence on the performance data of the pump. In the case of axial impellers, the c.g.w. usually amounts to 1 ‰ of the impeller diameter, with a minimum of 0.1 mm. Much larger c.g.w.'s have to be provided in the case of temperature fluctuations of the medium pumped and of the pump itself (e.g. during the starting process) because of the usually different thermal expansions of the pump components. The sizing of the c.g.w. is also dependent on the type of bearings, the permissible extent of clearance gap cavitation and the extent and nature of contamination of the medium pumped (abrasion, hydrotransport).

Clockwise Rotating Impeller

Rechtslaufrad

Roue à droite

see Impeller, Rotational Speed

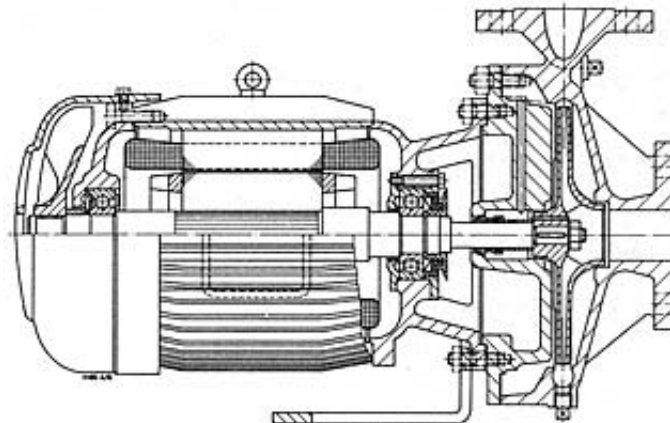
Close-coupled Pumping Set

Blockpumpe

Pompe monobloc

C.c.p.s.'s are characterized by a common or by rigidly coupled motor and pump shafts, and by a pump casing not standing on its own base but bolted onto the drive flange (see illustration). The special advantage of this mode of construction lies in the simple installation (installation of centrifugal pumped; the c.c.p.s. requires no alignment of the shafts or casings in relation to each other. As the forces and moments (branch loading) arising in the c.c.p.s. and in the piping have to be absorbed by the motor bearings or the motor casing respectively, this type of construction is limited to relatively small outputs.

An alternative to the c.c.p.s. is the closely-related flanged motor pump, which stands on the pump base instead of the motor feet. The flanged motor pump is capable of absorbing larger piping forces and moments than the c.c.p.s.



Close-coupled pumping set

Closed Loop Test

Ringleitungsversuch

Essai en conduite fermée

C.l.t. is the name given in centrifugal pump technology to a model test (model laws) or to a prototype test of pumps and pump components in a closed circuit (loop) on the pump test bed. In contrast to the c.l.t., we have the test in open circuit (open basin) (pump test bed).

Coat of Paint

Anstrich

Couche de peinture

see Surface Protection

Cock

Hahn

Robinet

see Valves and Fittings

Collecting Well

Sammelbrunnen
Puits collecteur

If individual wells have too low a yield (uneconomic pumping), the water from several boreholes is led into a c.w. via siphon lines (siphoning installation). By means of one or more centrifugal pumps, the water level in the c.w. is lowered in relation to the individual wells, and after evacuation of the siphon line (venting), water flows into the c.w.

Compensator

Kompensator
Compensateur

see Piping

Concrete Casing Pump

Betongehäusepumpe
Pompe à corps en béton

see Tubular Casing Pump, Volute Casing Pump

Condensate Pump

Kondensatpumpe
Pompe à condensé

The c.p. is a centrifugal pump designated in accordance with the nature of the medium pumped. The role of the c.p. is to pump out under vacuum the water in a condenser (condensate) which has condensed there from steam. In condensers equipped with a hotwell or receiver beneath the condenser, the c.p. delivers the condensate to a reservoir (e.g. the headwater tank), and in this case it is also known as a condensate return pump; in closed circuit it delivers the condensate direct into the boiler feed pump, via a low pressure preheated (economizer).

Fig. 1 illustrates the open condensate or feedwater circuit in a conventional steam power station, starting from the condenser, via the c.p. to the headwater reservoir, the boiler feed pump, the boiler and the steam turbine.

The maximum exhaust steam mass flow from the steam turbine determines the capacity of the c.p. The head is made up of the static difference in elevations between the water level in the condenser and the water level in the headwater reservoir, the difference in static pressure heads in the headwater reservoir and in the condenser, and the pressure losses (losses of head) in the flow on the suction side from the condenser to the c.p. and on the discharge side from the pump discharge branch to the feedwater reservoir, including any loss of head through a condensate preheater in the discharge piping.

The construction of the c.p. is governed by the fact that the vapour pressure of the water reigns approximately at the suction end of the c.p. (e.g. approx. 56.2 mbar for pure water at 35°C), and that there is only a very low positive suction head (suction behaviour) available. The suction behaviour requirements are very stringent therefore, and they are met by multistage vertical can-type pumps with a double suction first stage (Fig. 2), or by ring section vertical can-type pumps (multistage pump) with an intermediate extraction (Fig. 3) for the desalination of the condensate (if necessary); two stage vertical pumps with an inducer (Fig. 4) are also used. A relatively high suction head (net positive suction head, suction behaviour) can be achieved by the low-lying pump suction branch in the housing. To reduce the NPSH value (net positive suction head), suction impellers are provided, which were specifically designed to get an improved suction behaviour. An additional inducer can also be installed in single suction impellers.

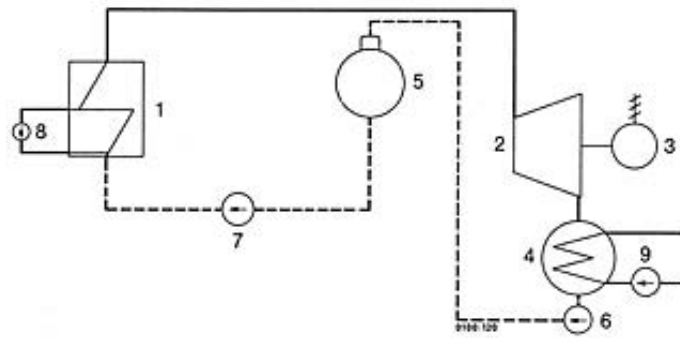


Fig. 1: Condensate and headwater circuit
 1 boiler; 2 turbine; 3 generator; 4 condenser; 5 feedwater reservoir; 6 condensate;
 7 boiler feed pump; 8 circulating pump; 9 main cooling water pump

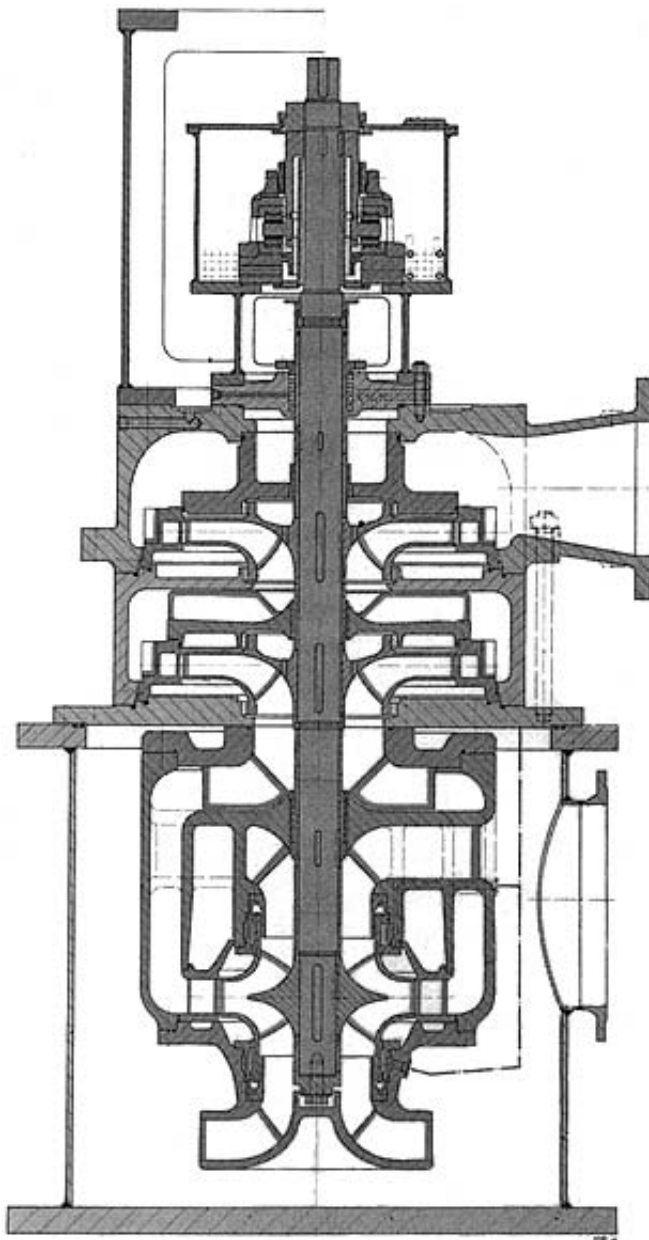


Fig. 2: Vertical condensate pump (vertical can-type pump) with double suction first stage. Installation above floor

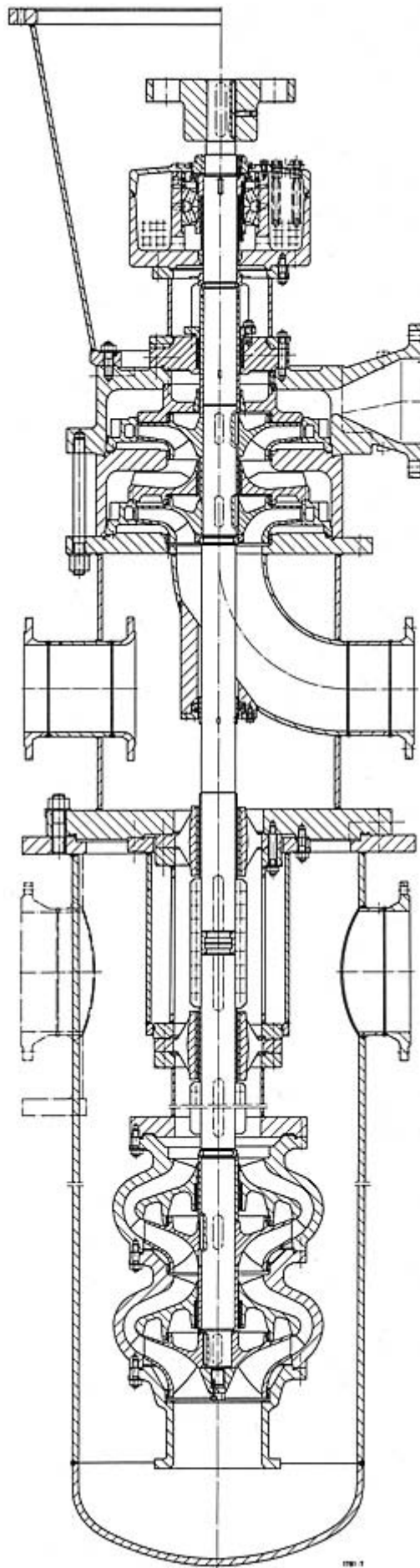


Fig. 3: Vertical condensate pump (vertical can-type pump) with intermediate extraction after the second stage. After having passed through the desalination plant, the condensate is led back to the third stage of the pump. Installation below floor

A double suction impeller (multisuction pump) decreases the NPSH value by about a factor of 1.6, when compared to a single suction impeller. At capacities over about 150 l/s (per half impeller) one must watch the amount of cavitation, the speed relationships and the length of cavitation bubble trail (net positive suction head), since stronger cavitation will occur. One measure of the intensity of the cavitation is v_a , the speed at which material is removed (for a given material):

$$v_a \sim w^a \cdot L_{B1}^b \cdot m^c$$

where

w flow velocity,
 L_{B1} length of cavitation bubble trail,
 m influence of the pump size (affinity laws).

The size of the exponents is dependent on the design concept of a suction impeller. The values lie in the range of:

$4 \leq a \leq 8$,
 $2 \leq b \leq 4$,
 $c \approx 3$.

From this one concludes that different designs are found in large c.p.'s than in small ones. Since the speed at the blade inlet is difficult to influence at a particular flow speed, one must attempt to reduce the length of the cavitation bubble trail.

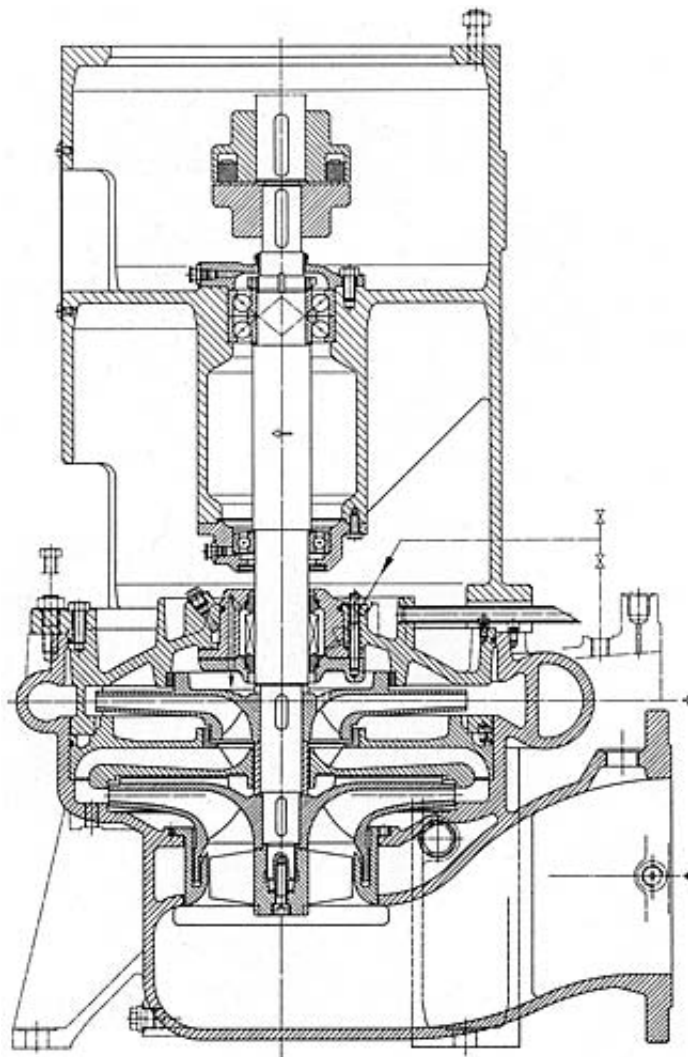


Fig. 4: Vertical two stage marine condensate pump with inducer

The shaft seal of the c.p. must be vacuum-tight. This applies equally to the standby pump which is kept in readiness for instant start-up. On horizontal pumps, the pump suction branch is arranged axially (end suction), thus dispensing with the necessity for a shaft seal at the suction end, and the suction end bearing (plain bearing) is arranged inside the suction branch, supported on radial ribs and lubricated by the condensate pumped. Sometimes this bearing is constructed in the form of a hydrostatic bearing.

Three-phase motors with squirrel-cage rotors (asynchronous motor) are generally used as drives of c.p.'s, if they are suitable from the control point of view. The following control possibilities (control) are available for the adaptation of the pump to the fluctuating turbine loads and for the prevention of dry running of the c.p.:

1. changing the throttling curve (characteristic curve) by altering the pump speed (speed adjustment, control);
2. alteration of the plant characteristic by throttling (throttling control, control), by means of a control valve in the discharge line of the c.p.;
3. alteration of the plant characteristic by returning the excess flow to the condenser (by-pass adjustment, control);
4. hanging the throttling curve (characteristic curve) by the automatic adaptation of the rate of flow (capacity) to the positive suction head (suction behaviour); this type of control, based on incipient cavitation is known as selfregulation.

The *self-regulation* of c.p.'s exploits the change in the characteristic curve $H(Q)$, or pump throttling curve, due to flashing of part of the condensate either upstream of or in the first stage, thus reducing the stage head $H(Q)$ by a certain amount, depending on the degree of vapour blocking (H_{cav} , Fig. 5). Depending on the water level $H_{z,geo}$, the resultant (cavitation-dependent, cavitation) break-away curve $H_{cav}(Q)$, intersects the system characteristic curve $H_A(Q)$ at the operating point.

The self-regulation of c.p.'s imposes very arduous requirements. particularly on the first stage of the pump, due to the severe cavitation strain. This is the main reason why this type of regulation is no longer adopted on the larger c.p. units which are being increasingly used today.

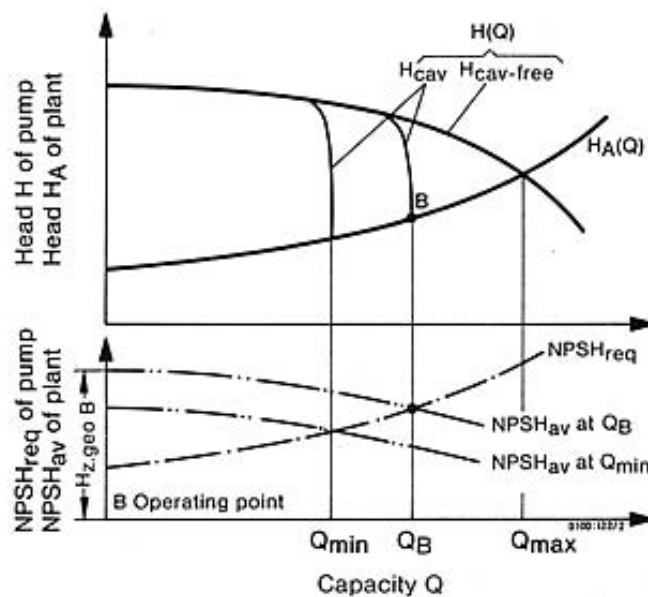


Fig. 5: Throttling curves $H(Q)$, $NPSH_{req}$ of condensate pump, and system characteristic curve $H_A(Q)$ and $NPSH_{av}$ illustrating the operating of a self-regulated condensate pump

Congruence Law

Kongruenzgesetz
Loi de la congruence

see [Affinity Law](#)

Constant Efficiency Curves

Muschelkurven

Courbes ovoïdes

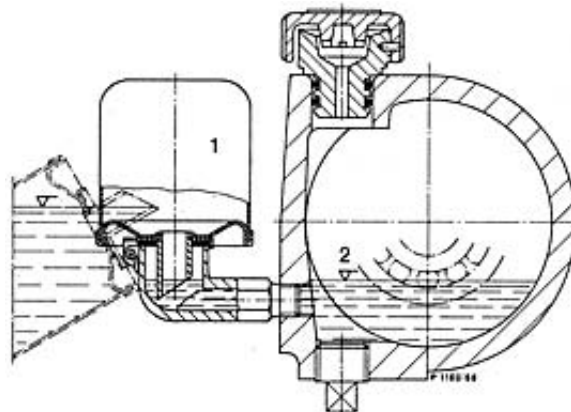
see [Performance Chart](#)

Constant Level Oiler

Ölstandsregler

Graisseur à niveau constant

C.l.o. is used to keep the quantity of lubricating oil under constant control, e.g. for the lubrication of [anti-friction bearings](#). As long as there is oil in the reservoir of the c.l.o. (see illustration), the oil level in the bearing housing will be automatically maintained at the correct level for safe lubrication during service. The reservoir should always be kept at least one-third full to ensure an adequate oil level over along period of operation. The chain-dotted lines on the picture show the oil position of the reservoir when topping up of same is carried out.



Constant level oiler 1 reservoir; 2 oil level

Continuity Equation

Kontinuitätsgleichung

Équation de continuité

see [Fluid Dynamics](#)

Control

Regelung

Réglage

The [centrifugal pump](#) and the [pumping plant](#) are two systems connected in series. Fig. 1: The QH curve (throttling curve, [characteristic curve](#)) results from the function of the centrifugal pump, i.e. a [head](#) H determined by the [pump type](#) in function of the associated [capacity](#) Q . The flow through the piping, on the other hand causes a [pressure loss](#) which increases practically with the square of the capacity, and is defined by the system characteristic curve ([head](#) H_A of plant). At $H = H_A$ ([head](#)) the [capacity](#) adjusts itself to that corresponding to B (Fig. 1). If the [system characteristic curve](#) becomes steeper due to throttling, i.e. $H'_A > H_A$ (governing by throttling), the flow through the [piping](#) is retarded decreasing the flow velocity and consequently the capacity reduced until anew equilibrium $H =$

H'_A is reached, this time at a lower capacity corresponding to B' . This equilibrium behaviour is made use of to positively c. the capacity:

1. by changing the system (piping) characteristic curve, *control by throttling* ($Q_{\text{Pump}} = Q_{\text{Plant}}; H_{\text{Pump}} > H_{\text{Plant}}$)
2. by changing the pump QH curve, *speed adjustment. regulation by inlet-vortex, impeller blade pitch adjustment* ($Q_{\text{Pump}} = Q_{\text{Plant}}; H_{\text{Pump}} = H_{\text{Plant}}$)
3. by changing the continuity, *by-pass adjustment (by-pass)* ($Q_{\text{Pump}} > Q_{\text{Plant}}; H_{\text{Pump}} = H_{\text{Plant}}$).

These c. methods 1 to 3 are discussed in greater detail below.

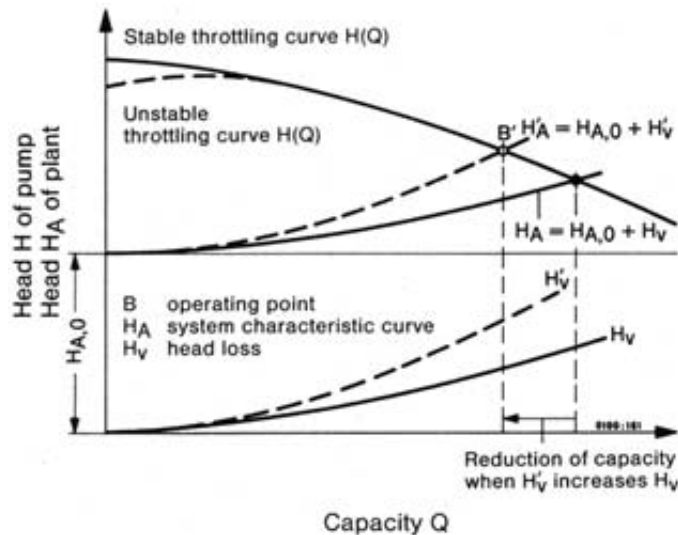


Fig. 1: Characteristic curves of centrifugal pump and of system .

Method 1. The system (piping) characteristic curve is altered by means of globe or gate valves etc. (valves and fittings) incorporated in the piping, which affect the pressure loss (Fig. 2). The increase in flow losses $H'_v > H_v$ and the corresponding change in the system (piping) characteristic curve causes it to intersect the QH line at a lower capacity. This throttling is a pure loss c. involving high operating costs, because the head generated by the pump (at low capacities) is only partially required by the plant (piping), and the balance is converted into wasteful forms of energy. Throttling is mainly used on radial pumps (pump types) which are the ones best suited to tolerate this method of operation over their entire characteristic curves, due to their hydraulic behaviour (operating behaviour of centrifugal pumps), particularly in cases when rates of flow which differ markedly from the optimum capacity are not used for a prolonged period. This mode of c. is mainly used on the smaller centrifugal pumps. It only makes sense when the shaft power decreases as the capacity decreases, which is the case for radial pumps and for some mixed flow pumps (characteristic curve). Throttling is a cheap form of c. from the point of view of investment costs, but its economics in operation should be investigated before it is adopted, particularly in the case of high shaft powers and prolonged operating periods under these conditions. The adoption of throttling is indicated in cases where the pump shut-off head (head) of a centrifugal pump must remain unaltered, e.g. for control reasons (switching on and off) by means of pressure-dependent control devices. Instead of, or in support of a throttling valve (valves and fittings) a fixed orifice plate (throttling plate) is often installed in the piping for long-term throttling. This mode of throttling like any other should always take place on the discharge side of the pump, to avoid cavitation in the centrifugal pump (net positive suction head).

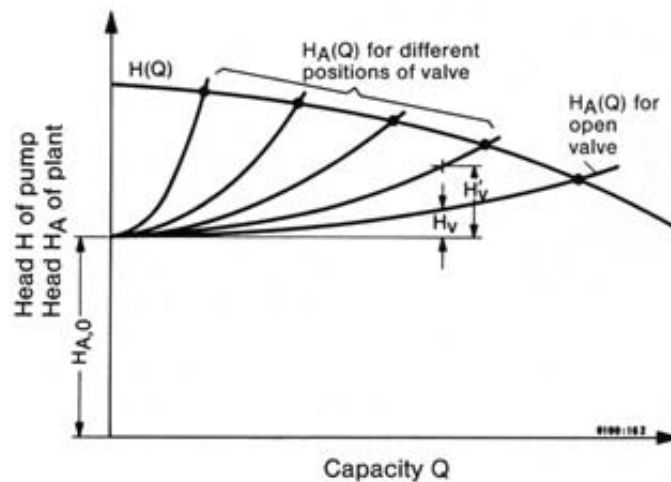
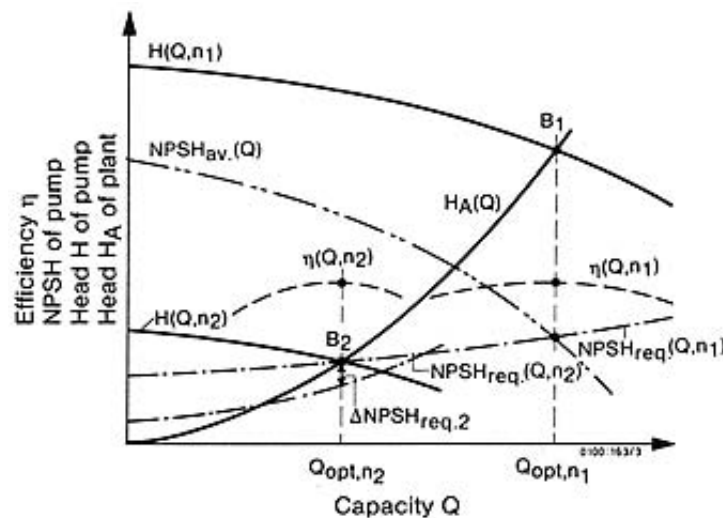


Fig. 2: Capacity conditions control by throttling.

A different system characteristic curve applies to every position of the throttling valve

Method 2. The QH curve of the centrifugal pump (throttling curve, characteristic curve) can be altered in the following ways: by changing the rotational speed; by changing the approach flow to the impeller by means of regulation by inlet-vortex (cooling water pump); by changing the centrifugal pump geometry within the impeller, e.g. by impeller blade pitch adjustment (propeller pump, impeller), or within the diffuser by pitch adjustment of the guide vanes (infrequently used); by partial covering over of the impeller outlet of radial impellers by control rings under part load conditions (infrequently used); by cutdown the impeller as a once-only adaptation to the operating point conditions (a process that cannot be reversed); by cavitation-conditioned self-regulation of the centrifugal pump (Fig. 5 under condensate pump).

Fig. 3: Characteristic curves of pump $H(Q)$, $\eta(Q)$

and $NPSH_{req}(Q)$ for speed adjustment ($n_1 = n$; $n_2 = n/2$)
in conjunction with system characteristic curves
 $H_A(Q)$ (with $H_{A,0} = 0$) and $NPSH_{av}(Q)$

All the c. methods described under point 2. attempt to generate only as much head in the pump as required by the system (piping) characteristic curve at the desired capacity. This type of c. of centrifugal pumps is economic from the operating cost point of view.

The QH curve (throttling curve) of a centrifugal pump changes with the rotational speed n in accordance with the relationship $Q \sim n$, $H \sim n^2$ (model laws). As shown in Fig. 3, all the points of the throttling curve (Fig. 2 under performance chart) move along parabolas when the speed is changed e.g. B_1 moves along the parabola to B_2 . On system characteristic curves which pass through the origin of coordinates $H_A = 0$, $Q_A = 0$, the operating point of optimum efficiency can move along the system characteristic curve in such a way that the pump will always operate at its capacity Q_{opt} of optimum efficiency. The higher $H_{A,0}$ (Fig. 4), the more the pump will move into the region of poor part load efficiencies when being regulated down to smaller capacities, and into the region of poor overload efficiencies when being regulated up to higher capacities. Nevertheless, even in this case only the

required head is generated by means of speed adjustment. In respect of energy consumption, this is the most economical form of c., and also the most sparing as regards stressing of the pump. When the speed is reduced, there is always a safety margin in respect of NPSH value (net positive suction head) on the suction side of the pump (Fig. 3). Speed adjustment is usually effected via the drive, i.e. by using a steam turbine, gas turbine, i.c. engine (e.g. a Diesel engine) or an electric motor (direct current motor, asynchronous motor) as drive, and more rarely via the gearbox (hydraulic torque converter, fluid coupling).

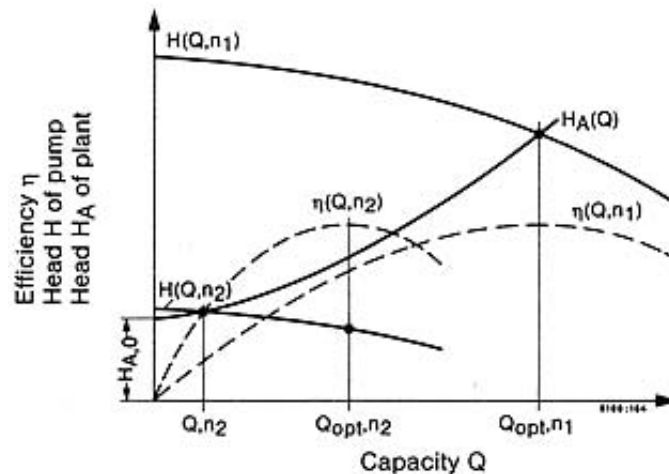


Fig. 4: Characteristic $H(Q)$ and system characteristic curve $H_A(Q)$ when $H_{A,0} > 0$

As a general rule, the approach flow (inlet conditions) to a centrifugal pump impeller is swirl-free or irrotational (vortex flow), i.e. $\alpha = 90^\circ$ (velocity triangle). In Fig. 10 under cooling water pump, the QH curve for 90° pitch adjustment angle applies to inviscid approach flow. A rotational swirl in the same direction as the impeller rotation leads to a lowering of the QH curve and the shaft power on all centrifugal pumps. Whereas this is hardly noticeable on normal radial impellers, the change in approach flow has a much more marked effect on mixed flow and axial pumps. Therefore pre-rotational swirl adjustment represents an effective method of changing the QH curve and saving power at the same time on high specific speed mixed flow pumps; the pre-rotational swirl adjustment device with inlet guide vanes (see Fig. 1 under cooling water pump) allows infinitely variable adjustment.

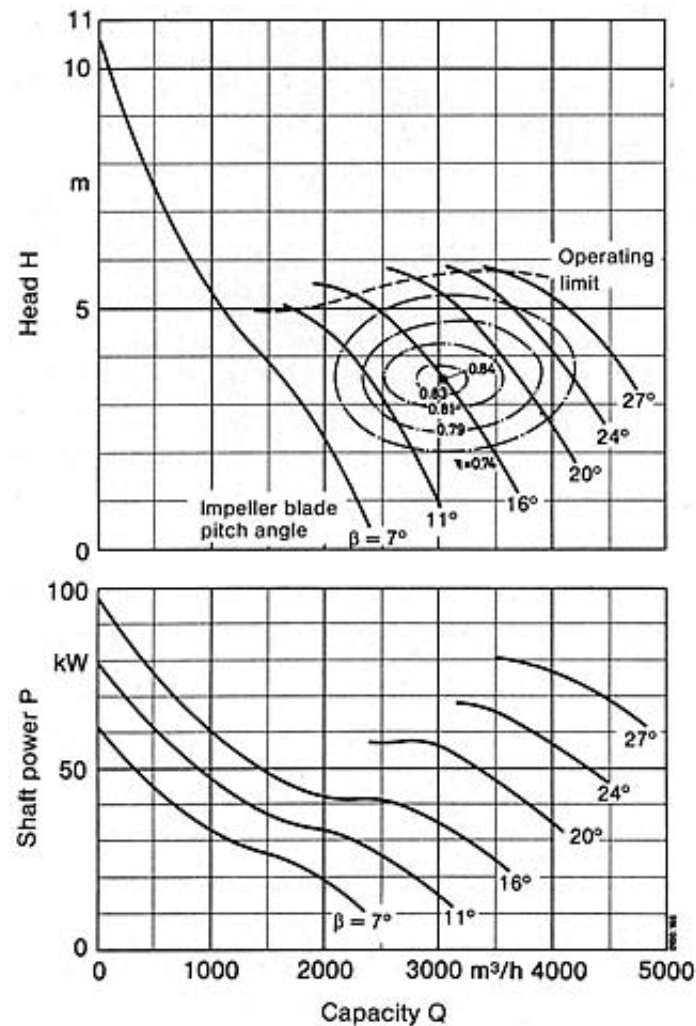


Fig. 5: Alteration of characteristic curves by impeller blade pitch adjustment

In the case of propeller pumps, the stepless c. of QH characteristic curves is successfully effected by means of impeller blade pitch adjustment. Fig. 5 illustrates the performance chart of a variable pitch blade propeller pump.

Much simpler, but not nearly as effective in saving energy, is the more rarely used pitch adjustment of the guide vanes on radial centrifugal pumps (Fig. 6).

Attempts have also been made to exploit a partial delivery of the impeller for c. purposes, either by shifting the impeller axially, or by partial covering over of its outlet by means of a control ring (Fig. 7).

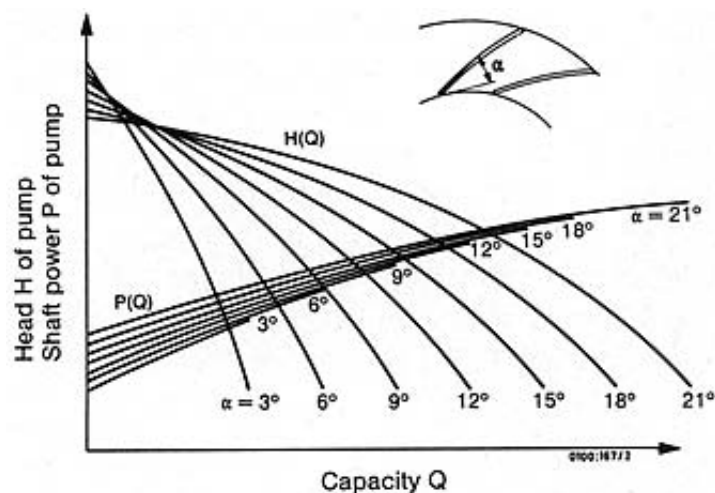


Fig. 6: Changing the characteristic curves of radial centrifugal pumps by varying the pitch angle of the guide vane ($n_Q = 26 \text{ min}^{-1}$)

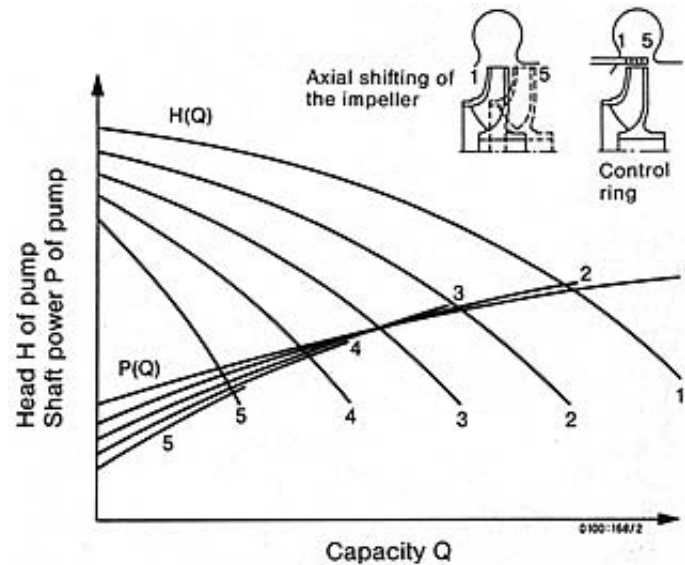


Fig. 7: Changing the characteristic curves by partial covering over of the impeller outlet width ($n_q = 26 \text{ min}^{-1}$)

Cutdown of impellers or cutting back of impeller vane tips (Fig. 8) can only be interpreted in the widest sense as a c. method; it really represents an initial effort at adaptation of a centrifugal pump to a pumping plant.

Method 3. The method of c. by which unequal capacities pass through the centrifugal pump and the plant (piping) is known as by-pass adjustment (by-pass) (Fig. 9). This mode of c. is only of value on pumps with a shaft power characteristic curve which falls with rising capacity, e.g. on propeller pumps and peripheral pumps.

The cavitation-conditioned self-regulation of a centrifugal pump is described under condensate pump.

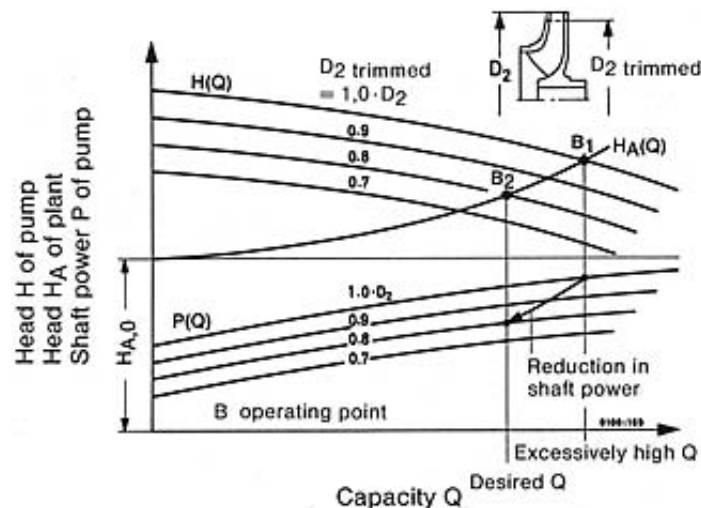


Fig. 8: Changing the characteristic curves by cutdown the impeller

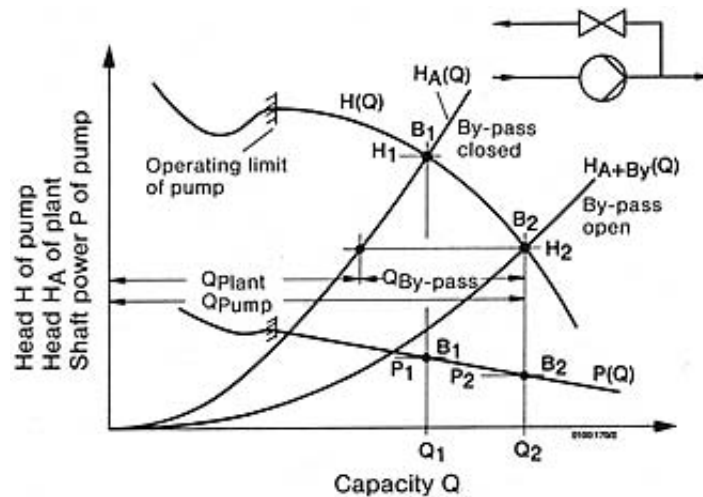


Fig. 9: Capacity control via by-pass

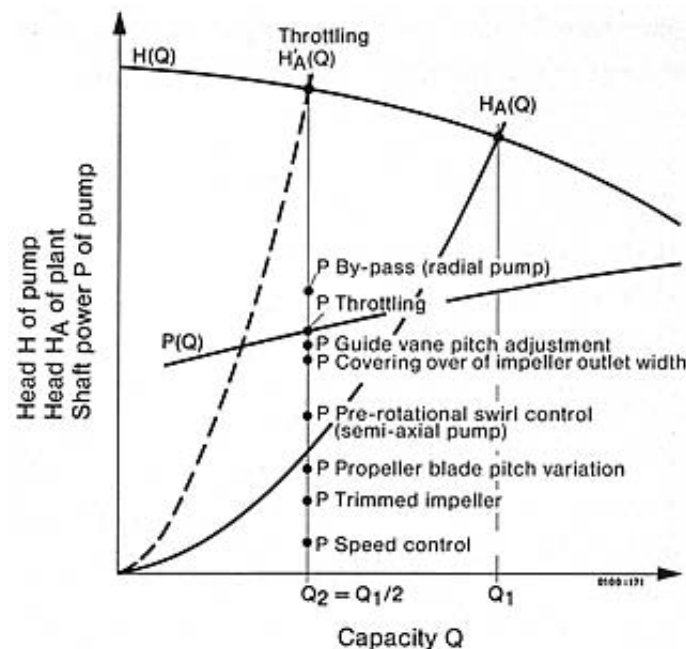


Fig. 10: Shaft power comparison for the various control methods on different pump types

Advantages and disadvantages of the various c. methods. In Fig. 10, the various methods of c. are compared qualitatively in respect of shaft power (shaft power is dependent on the characteristic curves of centrifugal pumps and the specific design; therefore, this example cannot be looked upon in a very general manner). In the example, the capacity is reduced by c. from Q to $Q/2$. By-pass adjustment on a radial pump (as opposed to by-pass in propeller pumps) absorbs the most shaft power. Then in gradually decreasing requirements of shaft power, we have among the most widely-used c. methods control by throttling, regulation by inlet-vortex, impeller blade pitch adjustment (variable pitch propeller) and rotational speed adjustment.

Fig. 11 shows a quantitative comparison of the energy efficiencies η_R of six different c. methods of radial flow pumps:

$$\eta_R = \frac{E_A}{E_P \cdot \eta_{Gr,opt}}$$

where

E_A yearly energy requirements of the system,
 E_p yearly energy requirements of the pump, and
 $\eta_{G_{r,opt}}$ optimum efficiency of the centrifugal pump unit (efficiency).

This energy efficiency η_R is displayed as dependent upon the relative static system head $h_{A,0}$:

$$h_{A,0} = \frac{H_{A,0}}{H_B}$$

where

$H_{A,0}$ static system head (system characteristic curve),
 H_B head at the operating point

as well as the c. requirement δ that gives the frequency and intensity of the capacity c.: When $\delta = 0$, it is not regulated (there is no energy loss due to c.); at $\delta \approx 1$, the capacities adjusted down to its minimum (valves and fittings), (energy loss due to c. is at a maximum). At $\delta = 0.5$, each capacity between 0 and Q_{opt} is equally likely throughout the year. Fig. 11 is valid for a radial flow pump with a specific speed on $n'_q = 40$; the curves at $n'_q = 20$ lie only slightly higher. The methods of capacity c. are identified as:

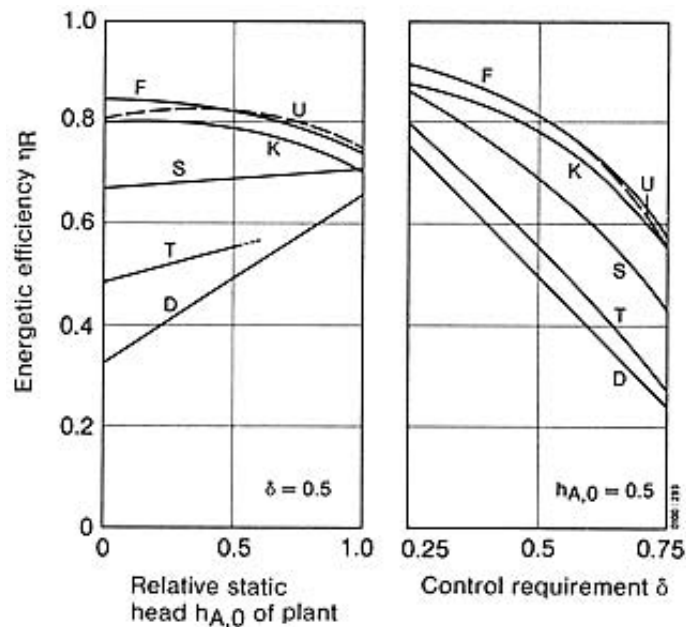


Fig. 11: Energy efficiency η_R , of various methods of capacity control in terms of the parameters listed in the text

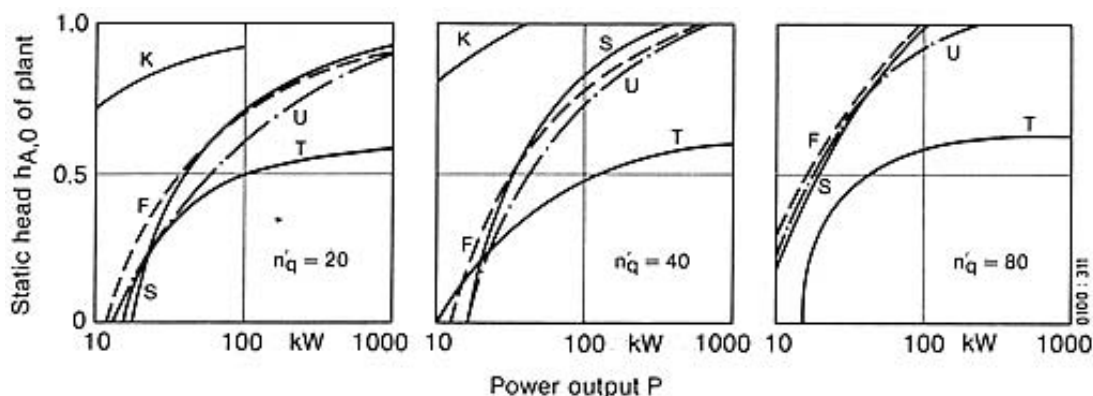


Fig. 12: Comparison of the different control methods by a graphical presentation as compared to throttling under the influence of the parameters listed in the text. Below the curves the paybackperiods are less than two years

- F speed adjustment through frequency change,
- K speed adjustment through variable speed V-belt drive,
- U speed adjustment through heterodyne gearbox,
- S speed adjustment through hydrodynamics slip coupling,
- D throttling through use of a throttle valve,
- T throttling through use of an energy recovery turbine.

The most energy is saved when using stepless variable speed adjustment.

The disadvantages of energy-saving c. lie in the higher initial costs, as opposed to throttling. A calculation of the economics will give the determining price factors for individual c. methods.

Fig. 12 shows the comparison of various c. systems on radial flow pumps in a graphic demonstration of the amortization time, over a period of two years (economics), as opposed to throttling

- n'_q specific speed,
- P shaft power in kW, and
- $h_{A,0}$ static component of the system head.

The period of depreciation was calculated with the interest rate set at 8%/year and with specific energy costs of 0.20 DM/kWh, and is valid for a c. requirement of $\delta = 0.5$. When relying less on control ($\delta < 0.5$), the curves are pushed down and to the right, and when c. is more important ($\delta > 0.5$) the curves are pushed up and to the left.

Coolant Pump

Kühlmittelpumpe
Pompe de refroidissement

see Reactor Pump

Cooling Water Pump

Kühlwasserpumpe
Pompe d'eau de refroidissement

The duty of a c.w.p. is to supply heat exchangers with cooling water; the capacity will vary according to the heat flow to be removed, and the required head will be determined by the nature of the cooling system.

A distinction is made between "wet" and "dry" cooling processes.

In "wet cooling" the air which absorbs the heat is in direct contact with the water (which is the cooling medium), whilst in "dry cooling" the cooling water and the air are not in direct contact.

Wet cooling

1. Cooling by fresh water. The cooling water is drawn from a river, lake or the sea, forced through the heat exchanger by the pumps and returned to the river, lake or sea at the point of extraction.
2. Cooling via a cooling tower. The cooling water is drawn once only from a river or lake and led into a collecting basin beneath the cooling tower. From the basin it is drawn by pumps, forced through the heat exchanger, and returned to the basin via the cooling tower. Water lost by evaporation or leakage is made up.
3. Cooling by fresh water and via a cooling tower. Depending on the amount of heat that can be discharged into the river or lake, the cooling system is switched over from fresh water operation to combined fresh water and cooling tower operation. The c.w.p. must be able to generate the head necessary both for fresh water operation and cooling tower operation.

If this is not realizable for hydraulic reasons, so that the preconditions for an economic operation under both sets of operating data do not exist, the following solutions can be adopted:

- a) The cooling water circuit is subdivided into a fresh water cooling circuit and a cooling tower circuit. This means that the water is not returned immediately to the river after passing through the heat exchanger, but is led to the cooling tower pumps. The latter pump the water via the cooling towers before returning it, cooled down, either to the river (outlet cooling) or back to the c.w.p.
- b) A speed-controlled motor (control) or a pole-changing motor is used as drive for the c.w.p. (number of poles).
- c) A speed-changing gear is fitted, which allows speed adjustment (control) to be effected while the pump is running.

Dry cooling

1. Direct air cooling. This process uses air exclusively for the cooling and condensation of the condensate in a closed circuit. No c.w.p.'s are used in this case.
2. Indirect air cooling. In this process, the cooling water is led through a dry cooling tower by a circulating c.w.p. and then led to a mixing condenser.

Head and capacity ranges of c.w.p. 's. The heads required range from 5 to 15 m as a general rule, in the case of fresh water operation, and up to 30 m in the case of cooling tower operation.

The capacity will depend on the cooling process, the type of heat exchanger adopted, and on whether the power station is nuclear or fossil-fuelled.

Guideline values for

- | | |
|---|-------------------------------|
| -fresh water cooling: conventional power station | 100-120 m ³ /h MW, |
| nuclear power station | 140-160 m ³ /h MW; |
| -indirect air cooling: conventional power station | 80-100 m ³ /h MW, |
| nuclear power station | 120-140 m ³ /h MW. |

Types of impellers. The required range of heads from 5 to 30 m is covered by three types of impellers:

For heads up to 10 m, an axial propeller is usually adopted (specific speed $nq > 160 \text{ min}^{-1}$).

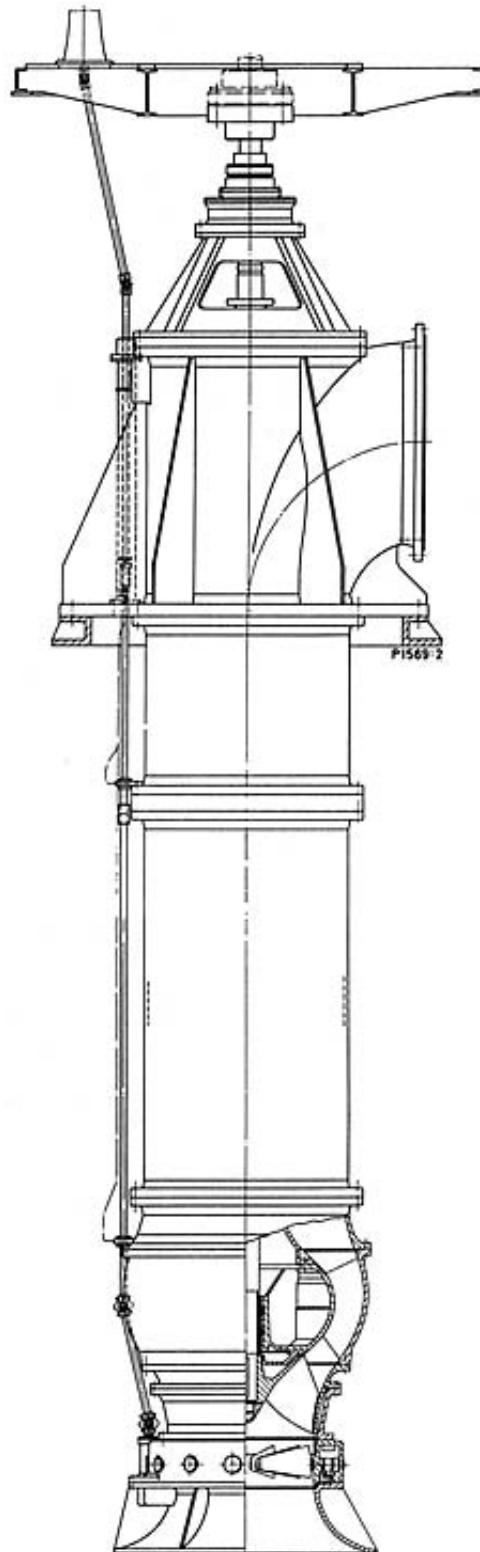


Fig. 1: Helical impeller tubular casing pump with inlet-vortex regulation device.

For heads from 10 to 20 m, semi-axial propellers can be adopted ($100 \text{ min}^{-1} < n_q < 160 \text{ min}^{-1}$).

For heads from 10 to 30 m, diagonal impellers (also known as helical or mixed flow impellers) are used ($70 \text{ min}^{-1} < n_q < 200 \text{ min}^{-1}$).

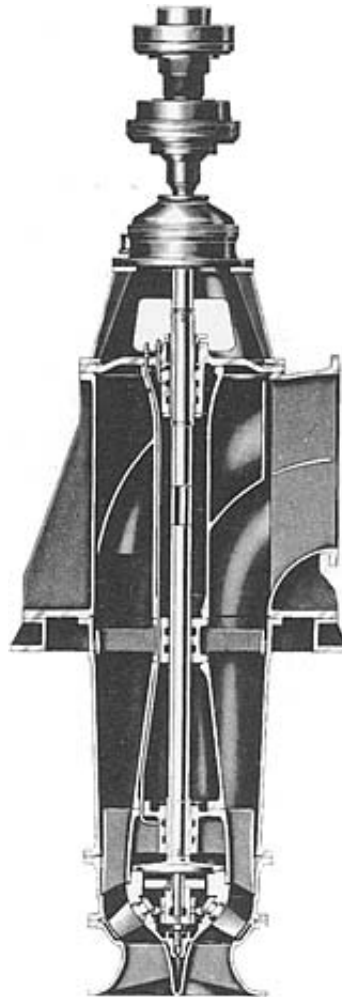


Fig. 2: Tubular casing pump with semi-axial propeller
(with blade pitch adjustment)

All three types can of course be used also in multistage version, thus enlarging the head range. The multistage version is of course more expensive, and is only adopted if the controllability of the rate of flow demands a particular pump type.

Pump types. Vertical tubular casing pumps or vertical volute casing pumps are usually adopted as c.w.p.'s, and they are built entirely of metallic materials (Figs. 1 to 4). Increasing use is also made of submersible motor pumps (illustrated in Fig. 5 with a helical impeller, impeller).

From sizes DN 1200 to 1400 (nominal diameter) in the case of volute casing pumps, and sizes DN 1600 to 1800 in the case of tubular casing pumps, concrete volute casings or concrete tubular casings are sometimes adopted for economic reasons. In such pumps, the volute casing or the tubular casing (pump casing) consists partly or wholly of concrete (Figs. 6 and 7).

In the case of tubular casing pumps, the impeller is either an axial (Fig. 3) or a semi-axial (Fig. 2) propeller, or a helical impeller (Fig. 1); in the case of volute casing pumps, the impeller is a semi-axial propeller or a helical impeller (Fig. 4). On board ship, c.w.p.'s (multisuction pumps) are sometimes also fitted with double suction radial impellers (Fig. 8) in conjunction with a volute casing (pump casing).

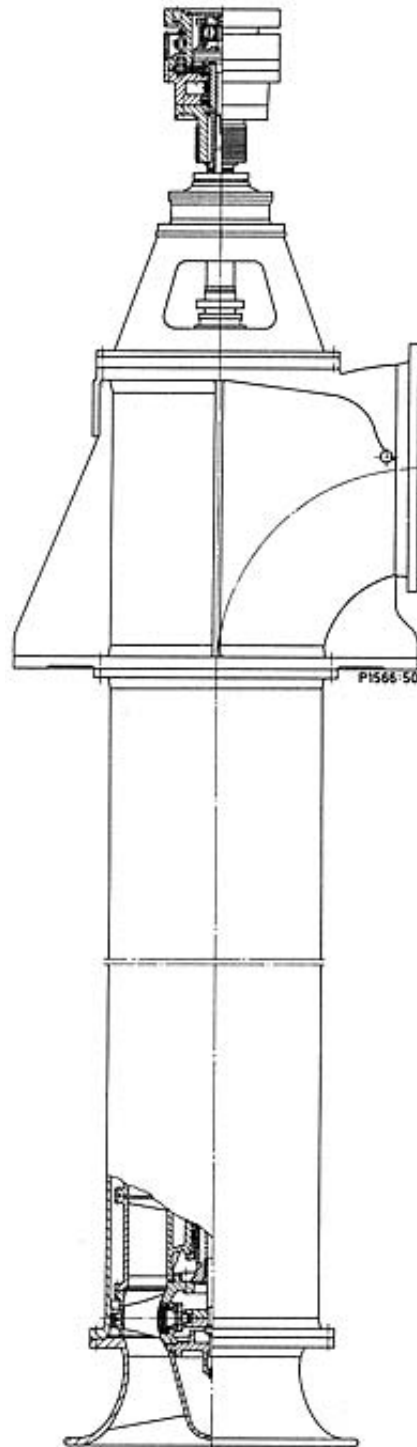


Fig. 3: Tubular casing pump with axial propeller
(with blade pitch adjustment)

Control of c.w.p. 's. A. control system is needed if:

- the capacity at constant head has to be adjusted to varying conditions in the heat exchanger (e.g. if the turbine is running at part load),
- a constant capacity (rate of flow) is required, but the head of the pump varies (e.g. fluctuations of the water level on the suction side, or change-over from fresh water to cooling tower operation),
- one pump fails, and the remaining pumps have to provide 100% of the required cooling water flow between them.

The following methods of c.w.p. control can be adopted:

1. *Throttling control.* The head of the plant is increased by partial closure of a throttling device (valve, butterfly, valves and fittings) in the discharge line, and consequently the capacity is reduced in accordance with the characteristic curve. There are limits to this type of control. The shaft power of the pump on pumps with specific speeds n_q above 100 min^{-1} increases with decreasing capacity, and may exceed the motor rating limit. In addition, on all the pump types described here, the flow separates from the blading when a certain part load capacity is reached (operating behaviour), and the pump runs rough and noisily (quietness of centrifugal pumps), a mode of operation which must be avoided at all costs for any appreciable length of time. Of all the control methods described here, control by throttling produces the highest losses. If used at all, it is confined to very small units.

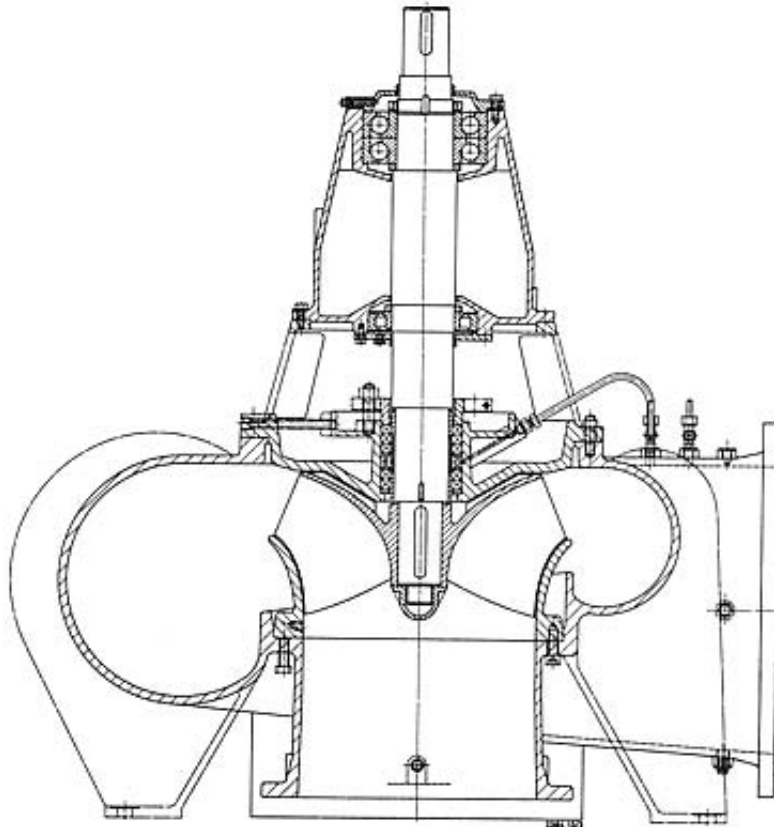


Fig. 4: Volute casing pump with helical impeller

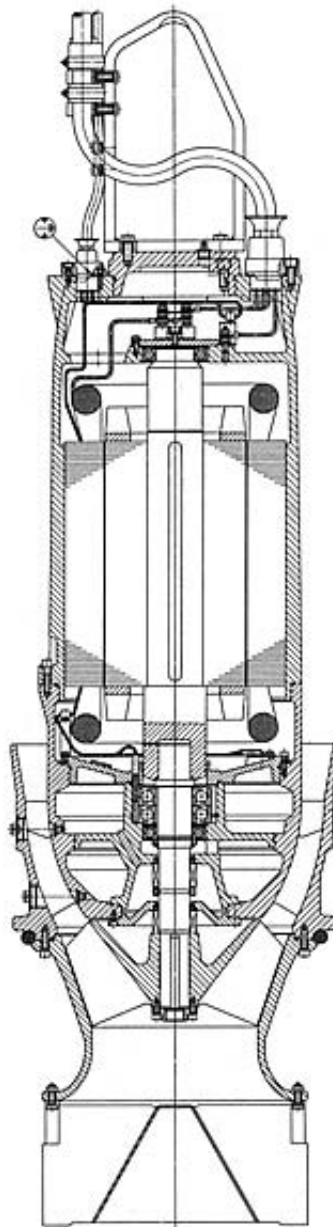


Fig. 5: Submersible motor pump with helical impeller

2. *Speed adjustment.* In the case of speed adjustment (control), the rotational speed of the pump is altered. The head, capacity and shaft power of the pump all adjust themselves in accordance with the affinity law. However, the greater the geodetic fraction $H_{A,0}$ (see Fig. 4 under control) of the system head H_A (system characteristic curve), the more the operating point departs from its optimum at reduced speed. In other words, it shifts toward part load (operating behaviour of centrifugal pumps) and, hence, toward the cavitation limit. The rotational speed is either changed by means of a speed-control gear which can be shifted while the pump is running, or the driver speed is controlled directly. Electric motors are frequently used as drives, and direct current motors are seldom employed for this application, because of the high shaft powers required by c.w.p.'s. In the case of three-phase motors in the higher power range, electric speed adjustment by, say, thyristor (electrical switchgear) has recently been, getting less expensive and is therefore being used with increasing frequency for c.w.p.'s.

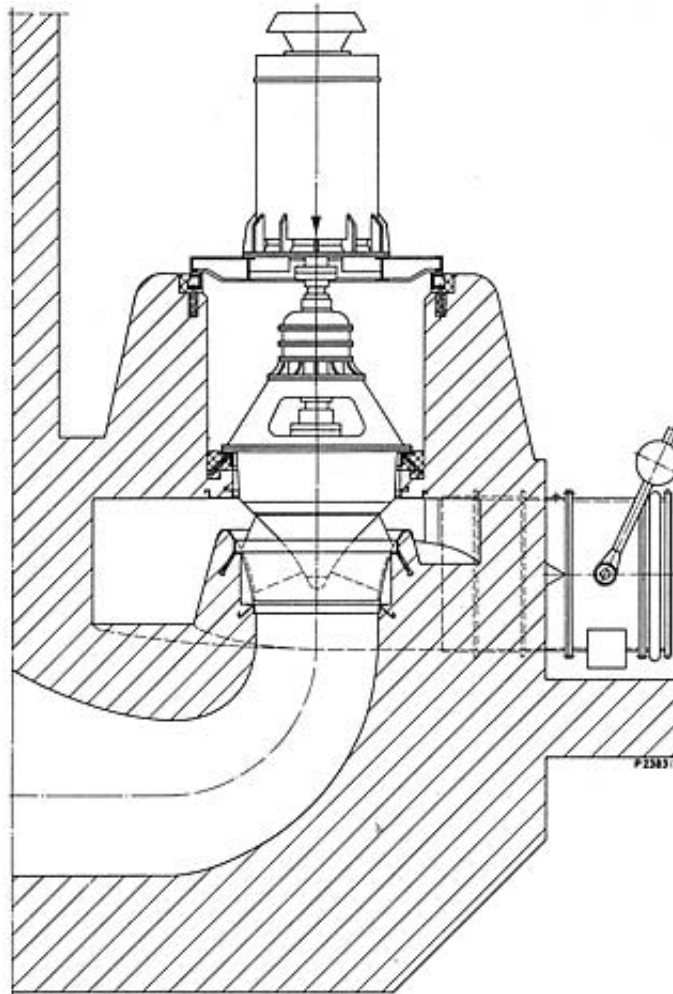


Fig. 6: Concrete volute casing pump

3. *Control by impeller blade pitch adjustment*. This method of control is feasible on pumps fitted with axial and semi-axial propellers. The blade pitch is adjusted while the pump is running. The performance chart of such pumps at all the possible pitch adjustment angles usually exhibits elliptically shaped efficiency curves with an almost horizontal principal axis (Fig. 9). This governs the field of application of this type of control: if the requirement is for large variations in capacity combined with relatively small variations in head, at an almost constant efficiency, this method of control is best.

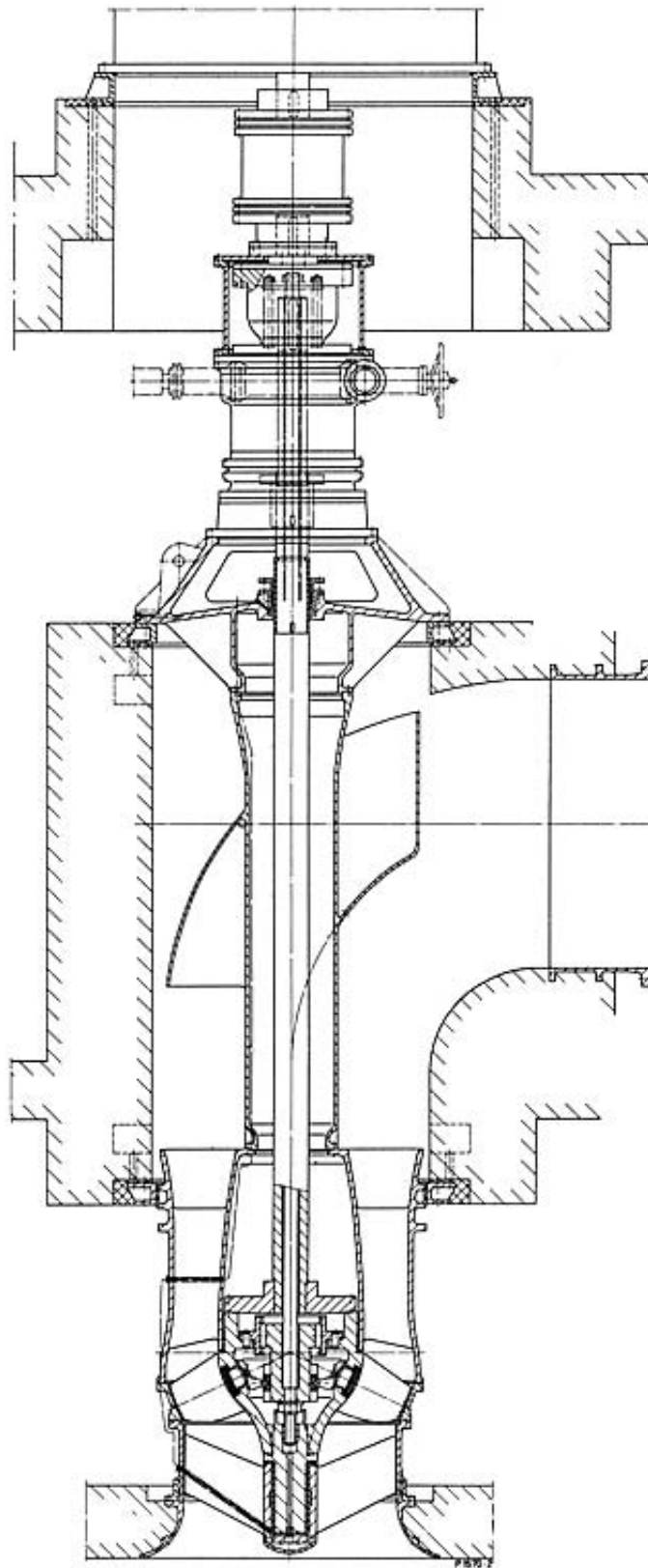


Fig. 7: Concrete tubular casing pump with adjustable semi-axial propeller

4. *Regulation by inlet-vortex.* This method of control is mainly applied to helical impeller pumps, but seldom to propeller pumps. It involves the control of the flow upstream of the impeller, by inducing a flow with a pre-rotational swirl in the same direction as the impeller rotation for control towards low capacities, and in the opposite direction for high capacities.

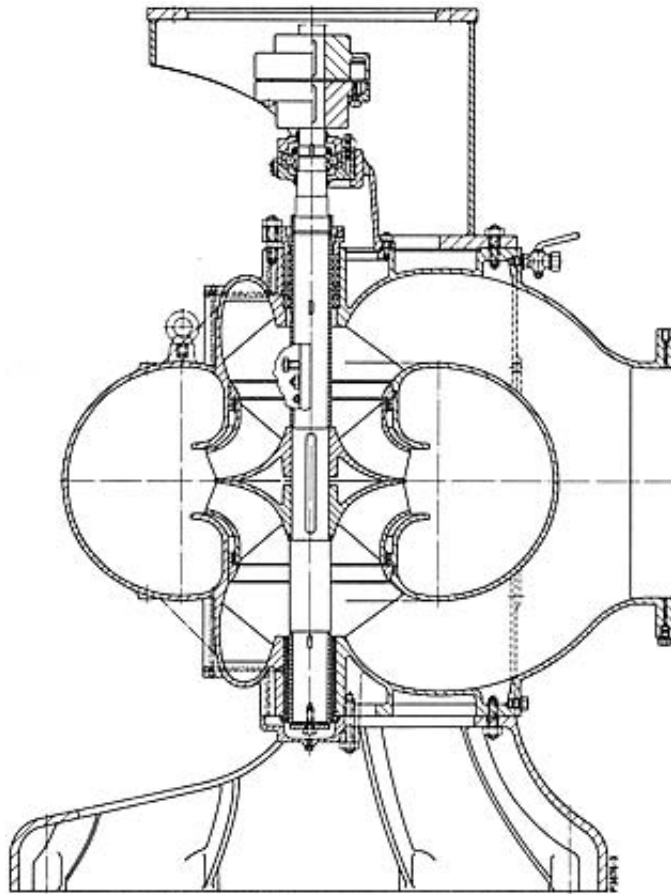


Fig. 8: Double suction circular casing pump

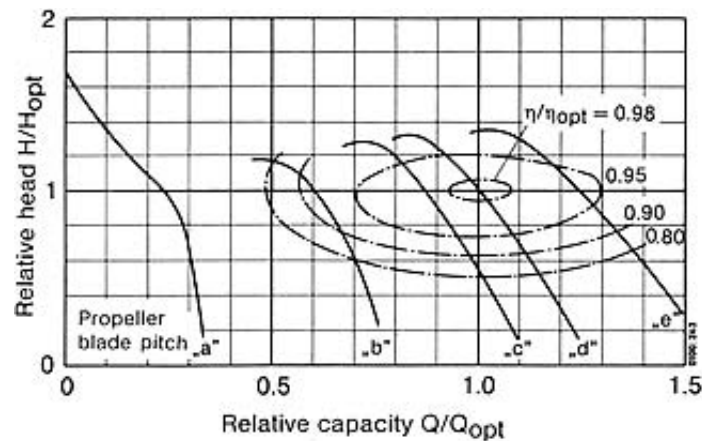


Fig. 9: Typical performance chart of an axial propeller pump with control by propeller blade pitch adjustment

This is effected by means of a stationary Inlet guide cascade, with blade patch angle adjustment (Fig 1)

The performance chart of such pumps at all the possible Inlet guide vane angles exhibits usually elliptically-shaped efficiency curves. The principal axes of these ellipses run roughly parallel to the pump characteristic curves (Fig 10), and differ in this respect from the ellipses of pumps with impeller blade patch adjustment, which run nearly horizontally.

This governs the field of application of this type of control which is to be given preference when the requirement is for large variations in head combined with relatively small variations in capacity, and optimal pump efficiencies

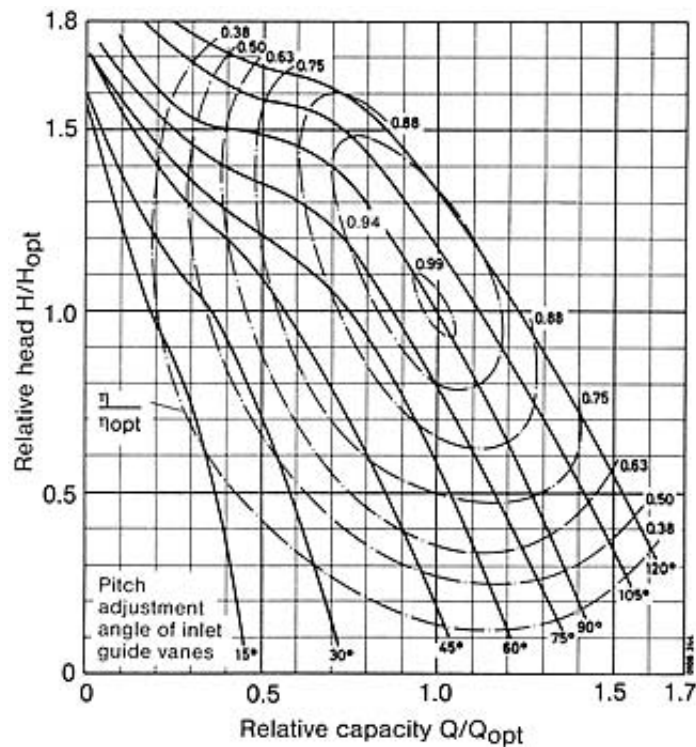


Fig. 10 Typical performance chart of a helical impeller tubular casing pump with pre-rotational swirl control

Copper Conductor

Kupferleitung
Conducteur en cuivre

see [Voltage Drop](#)

Correction

Aufwertung
Correction

see [Efficiency Re-Evaluation](#)

Corrosion

Korrosion
Corrosion

C is the reaction of a metal with its environment according to DIN 50900 and VDI guideline 3822. In most cases the reaction concerned is electrochemical and can only take place in electrolytes. The c reaction can be divided into two parallel semi-reactions: the anodic oxidation (dissolution of the metal) and the cathodic reduction (the ions in the electrolyte).

When a c. reaction leads to a break-down of a part or an entire system, one talks about a c. damage ([damage](#)). Different types of damage result from various types of c. The most common types of c and their manifestations are:

1 - Without mechanical loads:

Uniform c. (see Fig. 1) is a form of c. that attacks the entire surface uniformly, whereas

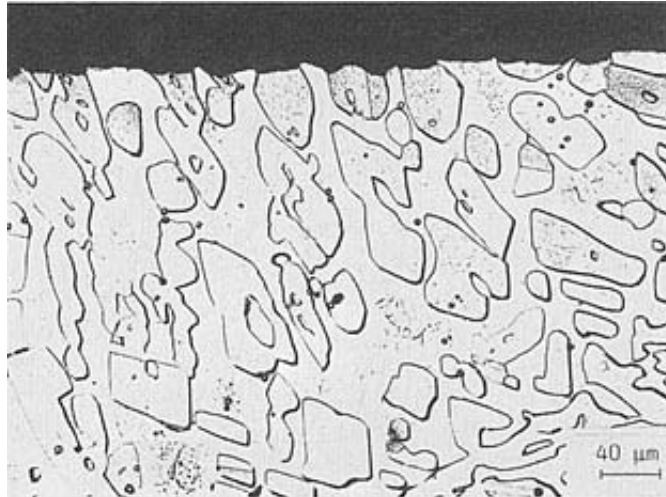


Fig 1 Uniform attack (micro graph)

Shallow pit c. (Fig 2) is characterized by localized attack of the surface and is caused by the formation of galvanic elements (see DIN 50 900, Part 2).

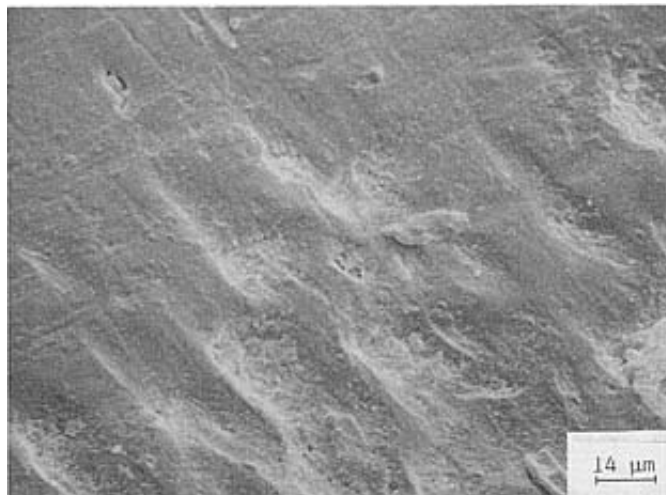


Fig. 2 Shallow pits (SEM picture; SEM = surface electron microscope)

Pitting c. is a very localized formation of small pits (see Fig 3), again caused by local galvanic elements. Pitting can occur in liquids with chlorides especially on all stainless steels and on aluminium alloys.

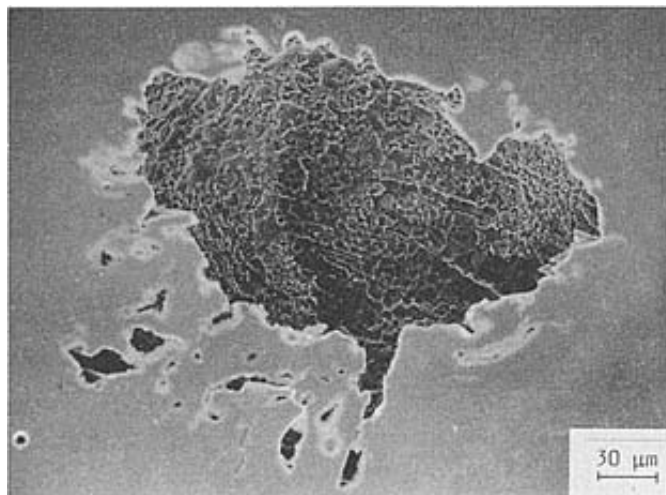


Fig. 3 Pits (SEM picture)

Crevice c. is found in small crevices between metals of same type or between a metallic and a non-metallic materials. It is caused by chloride ions which can concentrate in the crevice or by depletion of oxygen in the crevice and prevent oxidation-preventing Oxides from protecting the metal.

Galvanized c. is caused by the difference in the c. potentials of different materials, where the less noble metal is anodically dissolved and the more noble forms the cathodic surface for the reaction. The c. rate depends on the difference in c. potentials between the two metals, and the ratio of surface areas of the two materials. Therefore, contact between small anodes and very cathodes especially when the difference in the c. potentials is very large should be avoided not to enhance galvanic c.

Selective c. is a type of c. where only certain parts of the material structure are corroded, e. g. certain phases or areas along grain boundaries (Figs 4 and 5). Examples of selective c. are (spongiöse) c., intergranular c., dezincification and dealumination.

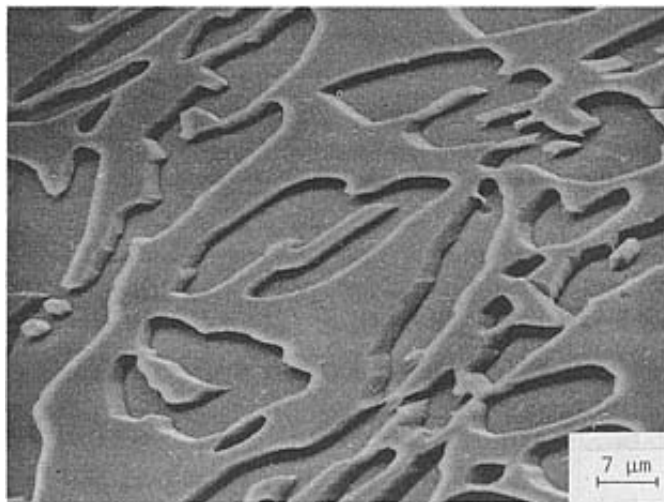


Fig. 4 Manifestation of selective corrosion (SEM picture)

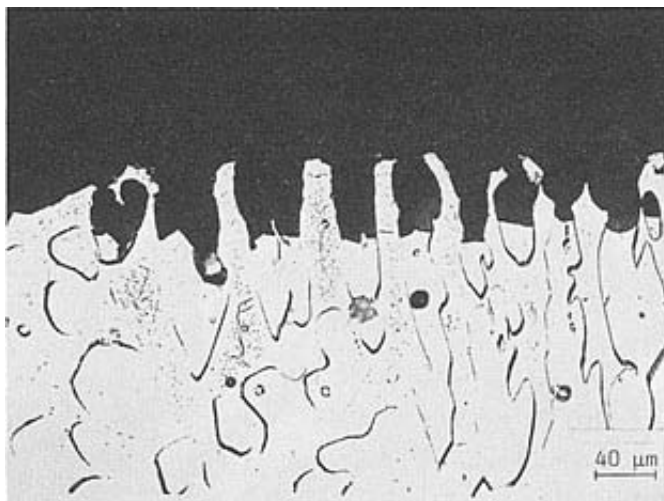


Fig. 5 Manifestation of selective corrosion (micro graph).

Graphitic c. (Fig. 6) is an example for selective c. of cast iron caused by the dissolution of ferrite and permutite due to the lack of protective layers.

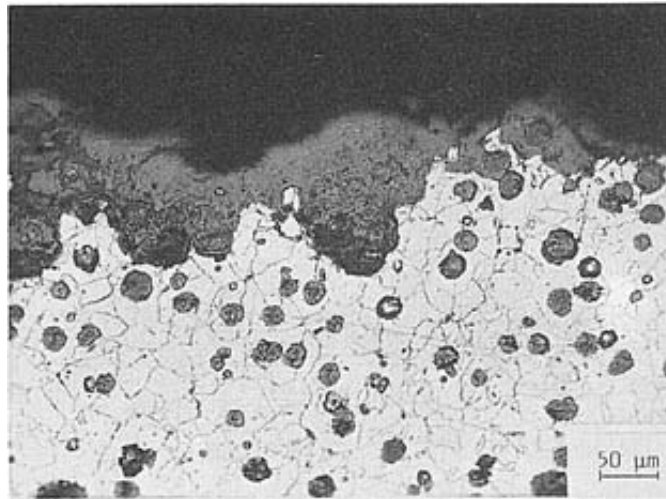


Fig. 6 Spongioid (micro graph)

The original shape of the part preserved by the graphite skeleton filled with c. products.

Intergranular c. (Fig. 7) is the preferred attack of the areas along grain boundaries. In stainless steels it is caused by chromium depletion due to carbides precipitating on the grain boundaries. In some cases, intergranular c. can lead to a total disintegration of the material (grain disintegration).

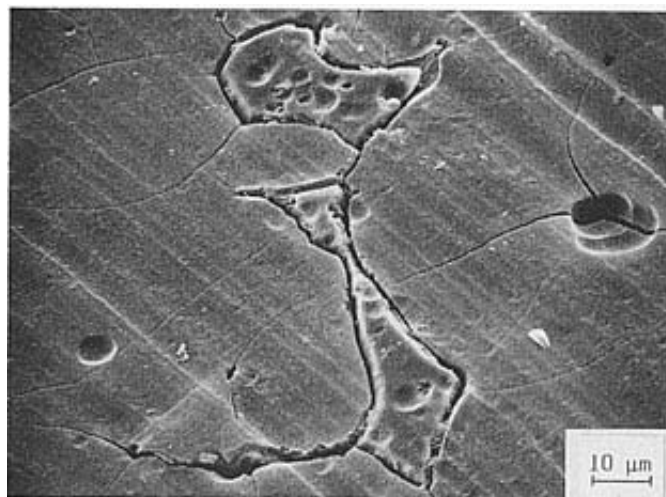


Fig. 7: Manifestation of intergranular corrosion (SEM picture)

Dezincation and dealumination is the selective c. of zing- and aluminium-rich phases in brasses and bronzes.

Microbiological c. is caused by microorganisms, for example, by sulfate-reducing bacteria.

C. during shut-down period can only occur stagnant liquids during shut-down periods of a plant.

2 - With mechanical loads;

Erosion-c. (Fig. 8) is a combination of mechanical wear (abrasion) and c., where normally c. is enhanced by the abrasion of protective layers from the metal, e.g. in solids containing fluids.

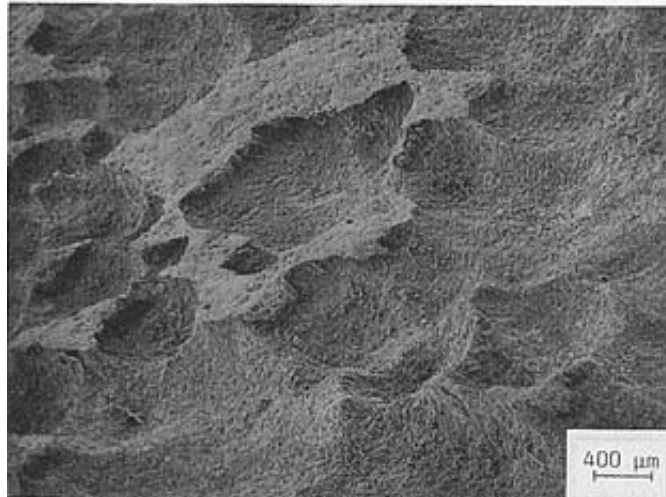


Fig. 8: Loss of material by corrosion-erosion (SEM picture)

Cavitation c. (Fig. 9) is a combination of cavitation and c., where the c. is either initiated or accelerated by local destruction on protective layers by cavitation.



Fig. 9: Loss of material by cavitation corrosion (extrem close-up)

Fretting-c. is caused by mechanical wear of two metal parts due to friction, which damages the protective or passiv layers on the materials in an aggressive liquid.

Stress c. cracking (see Figs. 10 and 11) occurs as inter- or transgranular cracks in materials when it is exposed to specific liquids and static or dynamic low-cycle mechanical stresses. The brittle fracture of the material without any visible c. products is one of the features of stress c. cracking. The stress necessary for the formation of cracking can also be inherent in the material caused e.g. by surface deformation, work hardening etc. Crack initiation and propagation can be caused by electrolytic (anodic) metal dissolution, or be hydrogen or strain induced.

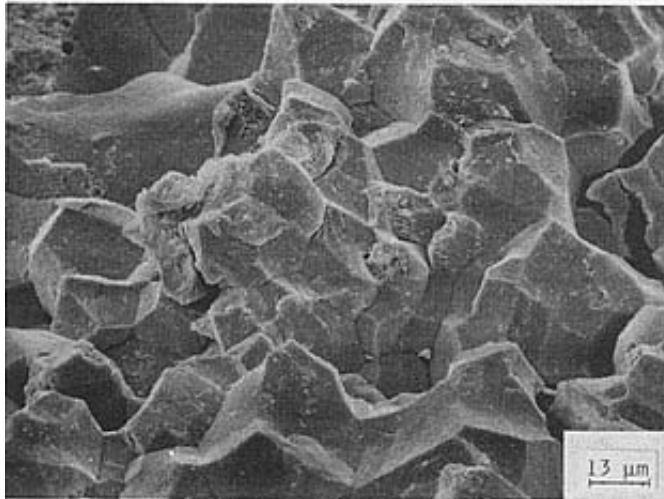


Fig. 10: Corrosion cracks by intergranular stress corrosion cracking (SEM picture)

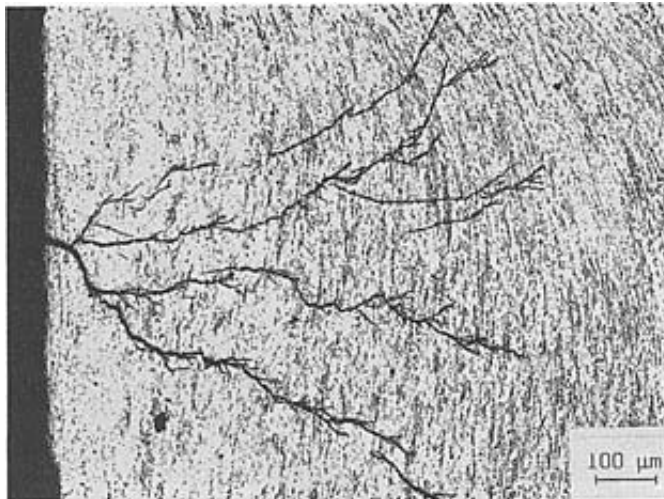


Fig. 11: Corrosion cracks by intergranular stress corrosion cracking (micro graph)

C. fatigue (see Fig. 12) is a cracking mechanism caused by cyclic stresses and a corrosive liquid. The cracks are mostly transgranular.

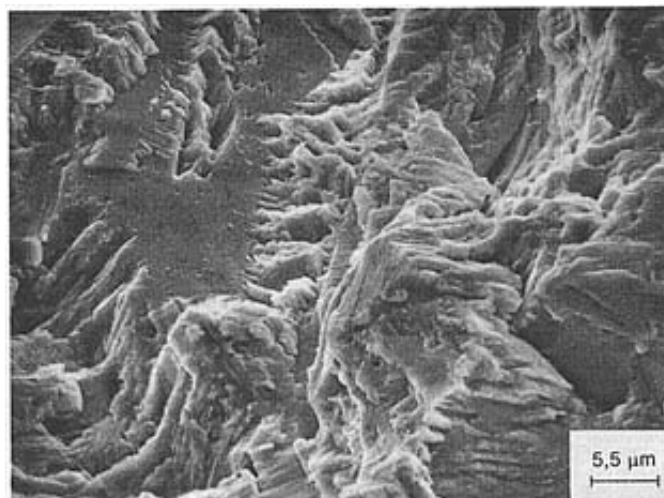


Fig. 12: Corrosion cracks by corrosion fatigue (SEM picture)

Further explanation;

A *c. cell* is galvanic cell that consists of a cathode and an anode, which are metallic and electrolytically connected. This element can be caused by: a combination of different metals (galvanic c.), different phases in one material (selective c.), local aeration, and local ion concentration (crevice and pitting c.). Therefore, less noble metals, phases etc. act as anodes, where metal dissolution occurs and noble metals act as cathodes, at which cations from the electrolyte are reduced. Even on homogeneous metal surface local anodes or cathodes can be formed by locally different aeration or deposits of c. products.

Electrode potential is the electrical potential or other electronic conductors of a metal in an electrolytic solution. The electrode potential can only be measured as a voltage against a reference electrode (see Fig. 13).

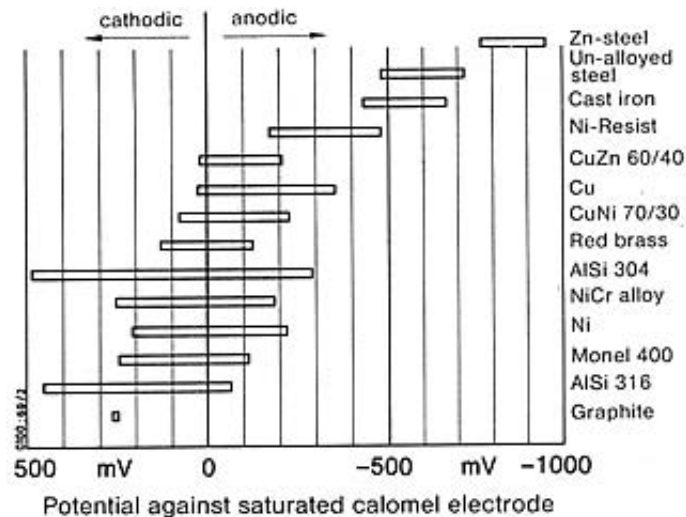


Fig. 13: Practical electrochemical series for seawater

A *reference electrode* is an electrode that keeps its electrode potential constant even when exposed to outside voltages. The potential of the reference electrode always refers to the potential of the standard hydrogen electrode.

C. potential (rest potential) is the potential a c. cell shows without any outside currents.

Pitting potential is the critical potential where pits start to grow constantly in number and size.

Repassivation potential is the potential below which pits stop to grow and start to repassivate, i.e. form a passive layer in the bottom of the pit.

Passivation is the transition of a metal from the active to the passive state (passivity). Passivation can be caused by electrochemical or by chemical reactions.

Passivity occurs when the anodic metal dissolution decreases to very low current densities, e.g. by shifting the electrode potential to more noble values or by increasing the concentration of oxidizing species in the solution. Due to this, a thin oxide layer is formed on the metal, which (passive layer, protective layer) prevents further metal dissolution. In this case, less noble metals and alloys show very good c. resistance (table of corrosion resistance).

C. protection can actually be activated by c. when uniform protective layers of reaction products are formed on the metal surface (protective layer). C. protection can also be achieved by keeping the metal separated from the corrosive medium by protective coatings, or by electrochemical means (for example: cathodic protection).

Cathodic protection is an electrochemical means of protection. The metal surface is polarized at a potential, where anodic metal dissolution is negligible, and local types of c. are avoided as well. Cathodic protection (= a polarization of the metallic surface to a non-precious potential) can be achieved by two methods:

1. The metal to be protected is electrically connected with a second, less noble metal and both are immersed in the corrosive medium. Therefore, a c. cell is formed where the less noble metal dissolves, anodically - and has to be renewed once at a time - whereas the noble part is protected cathodically.
2. The metal to be protected is connected to the negative pole of d.c. source. The inert anode, the corrosive medium and the part then form a complete electrical circuit with the external d.c. supply.

Corrosion Protection

Korrosionsschutz
Protection contre la corrosion

see [Corrosion](#)

Corrosion Rate

Korrosionsgeschwindigkeit
Vitesse à la corrosion

see [Corrosion](#)

Corrosion Resistance

Korrosionsbeständigkeit
Résistance à la corrosion

see [Table of Corrosion Resistance](#)

Corrosiveness

Aggressivität
Agressivité

see [Table of Corrosion Resistance](#)

Cos φ

Cos φ
Cos φ

see [Power Factor cos \$\varphi\$](#)

Costs

Kosten
Coût

see [Economics](#)

Counterclockwise Rotating Impeller

Linkslaufrad
Roue à gauche

see [Impeller](#), [Rotational Speed](#)

Coupling

Kupplung
Accouplement

see [Shaft Coupling](#)

Coupling Alignment

Ausrichten von Kupplungen
Alignement d'accouplement

C.a. is absolutely essential to ensure a mechanically trouble-free running of the pumping set (consisting of centrifugal pump and driver), and to avoid damage to the transmission elements. This involves first of all the maintaining of the prescribed gap a (Fig. 1) between the coupling halves ([shaft coupling](#)). Then the shaft centralizes must be in straight alignment with one another at the coupling (no kinking). Any lateral or height misalignment (lack of concentricity) (Fig. 3) and any angular misalignment (lack of parallelism) (Fig. 4) must be compensated.

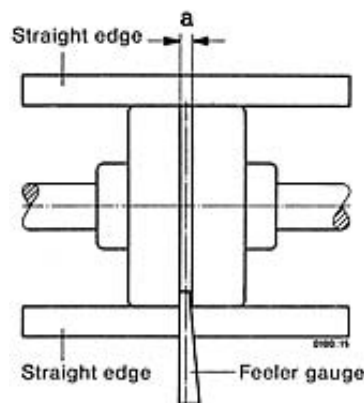


Fig. 1: Flexible coupling alignment with the aid of a straight edge and feeler gauge

Alignment is checked preferably with the aid of a straight edge and feeler gauge (Fig. 1). A coupling is correctly aligned if the straight edge laid across both coupling halves parallel to the shaft maintains the same distance from the shaft at all points around the periphery. In addition, the axial gap between the coupling halves should remain the same at all points around the periphery. Alignment can be effected accurately and quickly with the aid of an alignment jig (Fig. 2). The required degree of accuracy of c.a. will depend mainly on the type of coupling. See Performance Chart and on the [rotational speed](#).

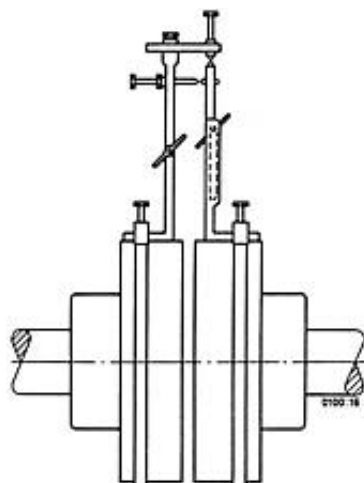


Fig. 2: Flexible coupling alignment with the aid of an alignment jig

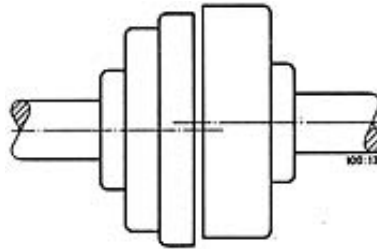


Fig. 3: Lateral or height misalignment

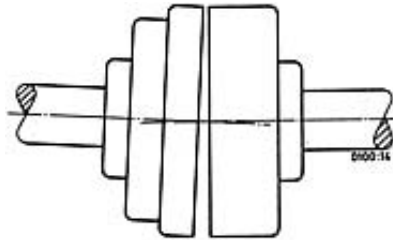


Fig. 4: Angular misalignment

In the case of c.a. of hot water pumps, the special instructions of the pump manufacturer must be observed on account of the thermal deformations which occur.

After completion of c.a., it is advisable to dowel the pump and driver on the baseplate or foundation (pump foundation) to prevent any shifting of the units during operation.

Coverage Charts

Raster

Grille de sélection

See Performance Chart

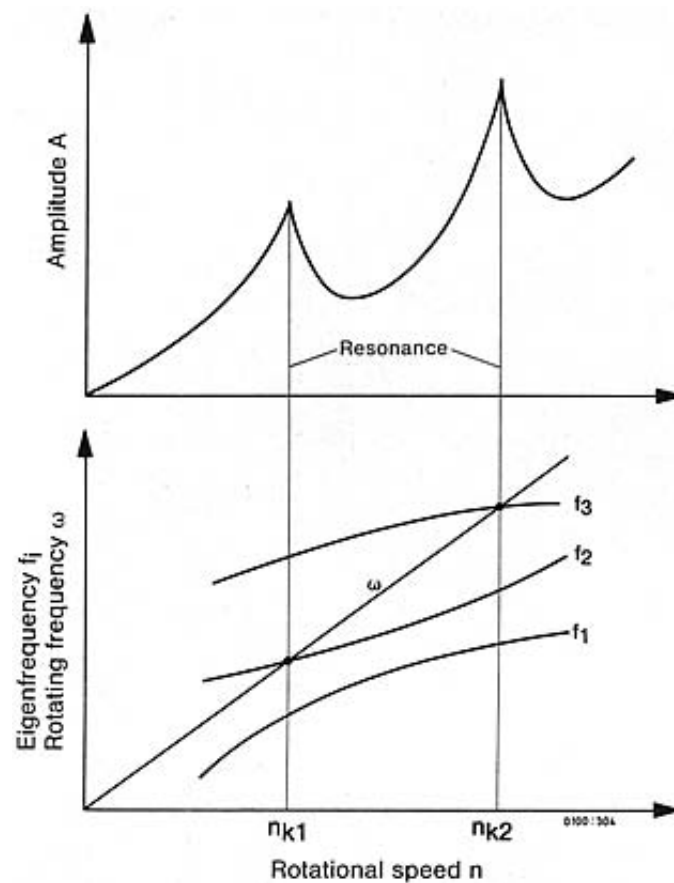
Critical Speed of Rotation

Kritische Drehzahl

Vitesse critique

With the lateral c.s.o.r., n_k , one signifies the resonance condition of rotating machinery, where the angular velocity, ω (rotational speed), coincides with a resonance frequency of the rotor system in bending or any natural frequency $f_1, f_2 \dots$ in the entire system (illustration).

Furthermore, the c.s.o.r. is also associated with the torsional critical speed, where the frequency of a pulsating torque is equaled the torsional resonance frequency of the rotor system.



Schematic view of the vibration amplitude of a rotor versus the speed.
 The equation of angular velocity $\omega = \pi n/30$ with one of the natural frequented f_i determines the critical speed of rotation n_k (n in min^{-1} , ω and f in s^{-1})

Cryogenic Pump

Tiefteuturpumpe
 Pompe cryogénique

see [Liquefied Gas Pump](#)

Current Requirement of Electric Motors

Stromaufnahme von Elektromotoren
 Intensité absorbée par un moteur électrique

Table 1 lists the average current consumptions J_1 in A for standard three-phase squirrel cage rotor motors (asynchronous motor) and for direct current motors, corresponding to the motor outputs (drive rating), motor speeds, voltages and power factors.

Table 1: Average current consumption of standard three-phase squirrel-cage rotor motors and of direct current motors														
Output of motor	Three-phase squirrel-cage rotor motor												Direct current	
	n = 3000 min ⁻¹			n = 1500 min ⁻¹			n = 1000 min ⁻¹			n = 750 min ⁻¹			n = 1400 min ⁻¹	
	J ₁ in A			J ₁ in A			J ₁ in A			J ₁ in A			J ₁ in A	
kW	220 V	380 V	cos φ	220 V	320 V	cos φ	220 V	380 V	cos φ	220 V	380 V	cos φ	220 V	440 V
0.33	1.55	0.9	0.80	1.6	0.95	0.76	1.75	1.0	0.70	2.0	1.15	0.62	2.0	1.1
0.5	2.25	1.3	0.83	2.35	1.35	0.79	2.55	1.5	0.73	2.85	1.65	0.65	2.7	1.4
0.8	3.3	1.9	0.85	3.4	2.0	0.80	3.7	2.1	0.75	4.1	2.4	0.68	4.6	2.3
1.1	4.4	2.5	0.87	4.6	2.7	0.81	4.9	2.8	0.77	5.4	3.1	0.7	6.4	3.2
1.5	5.8	3.3	0.88	6.0	3.5	0.82	6.4	3.7	0.78	6.9	4.0	0.72	8.5	4.3
2.2	8.1	4.7	0.88	8.3	4.8	0.83	8.8	5.1	0.79	9.5	5.5	0.73	12.5	6.3
3	11	6.5	0.89	11.3	6.5	0.83	12	6.9	0.81	13	7.5	0.75	17	8.5
4	14.5	8.5	0.89	14.8	8.5	0.84	15.6	9.0	0.81	16.5	9.5	0.77	23	11.5
5.5	19.5	11.5	0.89	20	11.5	0.84	21	12	0.82	22	12.5	0.79	30	15
7.5	26.5	15.5	0.89	26.8	15.5	0.85	28	16	0.83	29.5	17	0.81	42	21
11	38	22	0.89	38.5	22	0.86	40	23	0.85	41	23.5	0.83	60	30
15	52	30	0.90	52	30	0.87	54	31	0.85	55	32	0.84	78	39
22	75	43	0.90	75	43	0.88	77	45	0.87	78	46	0.85	115	58
30	100	58	0.90	100	58	0.89	103	59	0.88	104	60	0.86	155	77
40	132	77	0.90	132	77	0.89	135	78	0.88	137	79	0.87	210	105
50	165	95	0.90	163	94	0.90	166	96	0.88	169	98	0.87	260	130
64	206	119	0.91	206	119	0.90	210	121	0.89	213	123	0.88	325	160
80	256	148	0.91	256	148	0.90	260	150	0.89	265	153	0.88	400	200
100	320	185	0.91	320	185	0.91	325	187	0.89	330	190	0.88	500	250
125	395	228	0.91	395	228	0.91	400	230	0.90	405	234	0.89	625	310

Table 2 lists the corresponding values for the submersible motors (asynchronous motor), for 2 and 4 pole motors and for a three-phase current supply frequency $f = 50$ Hz.

Submersible motors have a somewhat higher current consumption than dry three-phase squirrel cage rotor motors, on account of the fluid friction and of the increased gaps. In the case of canned motors (wet rotor motor), there is an even higher average current consumption, on account of the additional losses in the can.

Table 2: Average current consumption of three-phase submersible motors (2 poles, $n = 3000 \text{ min}^{-1}$; 4 poles $n = 1500 \text{ min}^{-1}$)

Output of motor	Three-phase submersible motor $f = 50 \text{ Hz}$					
	$n = 1500 \text{ min}^{-1}$			$n = 3000 \text{ min}^{-1}$		
	$J_1 \text{ in A}$			$J_1 \text{ in A}$		
kW	220 V	380 V	$\cos \Phi$	220 V	380 V	$\cos \Phi$
0.74	4.1	2.4	0.85			
1.1	5.1	2.9	0.86			
1.5	6.7	3.8	0.86			
2.2	9.4	5.4	0.86			
3.0	12.3	7.1	0.86			
3.7	16	9.3	0.86			
5.5	23.3	13.4	0.86	23.5	13.5	0.85
7.5	30.5	17.5	0.86	30.5	18	0.85
9.2	37.5	21.5	0.86	36	21	0.85
11	43.5	25	0.86	42.5	24.5	0.85
15	58.5	33.5	0.86	58.5	34	0.85
16.5	65	38	0.87	70.5	41	0.84
25	87.5	51	0.87	91	52.5	0.84
30	107	62	0.87	111	64.5	0.84
37	131	75.5	0.87	139	80	0.84
45.5	161	93.5	0.87	169	96	0.84
55	195	113	0.87	203	117	0.84
62.5	222	128	0.87	230	133	0.84
74	260	151	0.88	270	155	0.84
92	321	187	0.88	330	190	0.84
110		222	0.88		226	0.84

Cutdown of Impellers

Abdrehen von Laufrädern
Rognage de roues

The shape of the throttling curve (characteristic curve) of a centrifugal pump operating at constant rotational speed does not permit a simultaneous reduction of the capacity Q and head H by throttling (control). If it is intended to reduce Q and H permanently without any change of rotational speed, a reduction of the outer diameter of the impeller (impeller) is a relatively simple and hydraulically very effective way of achieving the desired objective, and this can be done on a lathe. The c.o.i. results in a reduction of the circumferential velocity at the impeller outlet of the centrifugal pump. The c.o.i. also results in a change of blade length (blade), in a change of impeller width at outlet, and in a change of discharge angle of the blades; therefore the effect of this diameter reduction will vary according to the type of impeller involved.

If the diameter reduction is kept within reasonable limits, so that a mutual overlap of the blades still remains, the relationship between the impeller diameter D , the head H and the capacity Q of the full diameter impeller (index x) and the corresponding values of the trimmed impeller (index y) can be expressed as follows:

$$\frac{Q_x}{Q_y} \approx \frac{H_x}{H_y} \approx \left(\frac{D_x}{D_y}\right)^2$$

The Q and H pairs of values with the indices x and y are situated on a straight line passing through the origin of the QH diagram (see illustration).

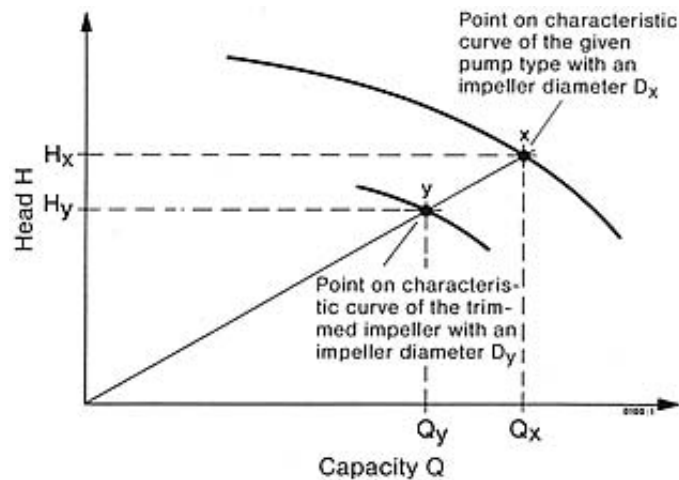


Diagram for determination of cutdown diameter

Impellers of low specific speed (up to $n_q = 25 \text{ min}^{-1}$ approx.) can be relatively extensively machined down in diameter without impairing the pump efficiency (efficiency), whilst impellers of relatively high specific speed suffer an appreciable reduction in efficiency. The cutting back of impeller vane tips has a similar effect to the c.o.i.

Cutting Back of Impeller Vane Tips

Ausdrehen von Laufrädern

Rognage de roues limité aux ailettes

In the case of centrifugal pumps fitted with a diffuser, if the reasons described under cutdown of impellers apply, it is usual to cut back the impeller vane tips only and to leave the impeller shrouds intact, so as not to impair the flow into the diffuser; this procedure is called "cutting back" (of impeller vane tips) as opposed to "cutdown" (of impellers). The diameter to which the vane tips of the impeller should be cut back can be estimated with the aid of the formula and graph given under cutdown of impellers.

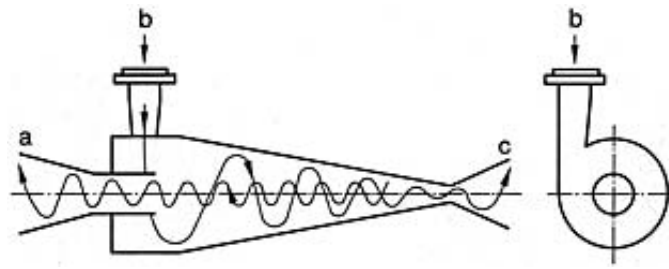
Cyclone

Zyklon

Cyclone

The c. (also hydrocyclone) is a separator device using centrifugal force, which can remove light and heavy solids, as well as gases, from a liquid. After the tangential entry of the pressurized, contaminated liquid into the cylindrical chamber of the c., a centrifugal force field is created so that, for example, heavy solids, such as sand, will gather near the outer wall, from where they are led away by a small, funnel-shaped pening. The clean liquid can then be removed from the center of the c. (illustration).

C.'s are used in centrifugal pumps to clean lightly contaminated liquids, without high costs, for use as bearing lubricant (plain bearing) or as quench fluid for seals (shaft seals).



Cyclone

a outlet of cleaned water; b inlet; c outlet of contaminated water.

D

Damage

Schaden
Dommage

D. is a change in a part that affects its foreseen and planned function, or does not permit it to perform its function (see VDI guideline 3822). D. *assessments* encompass a systematic examination and testing to determine the causes and effects of d. Aside from construction errors (such as defective castings), d. related to the applications are common.

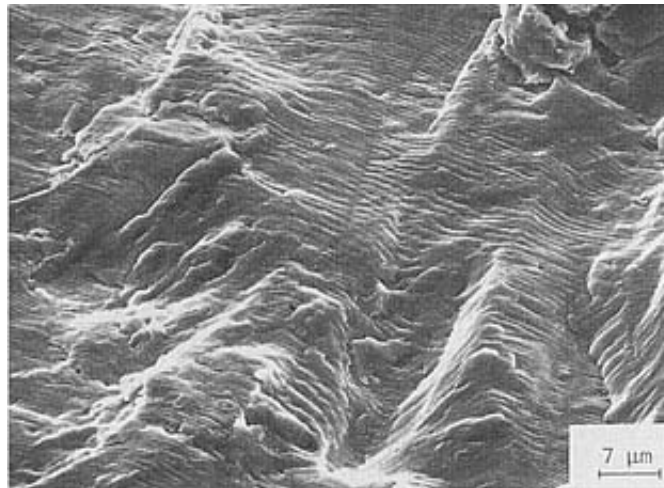


Fig. 1: Fracture surface of a reverse bending fatigue failure

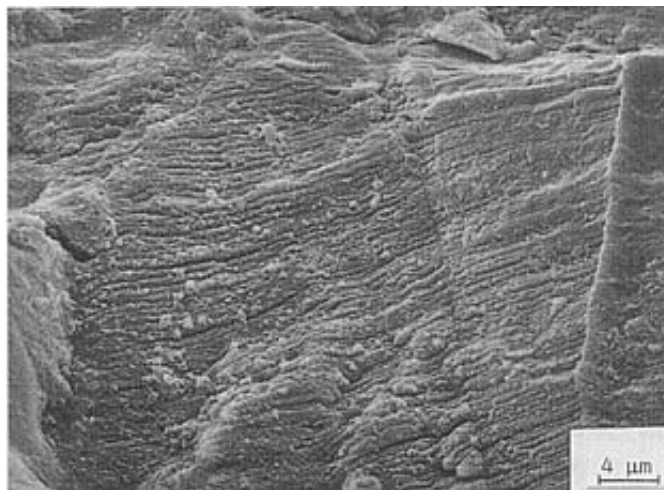


Fig. 2: Fracture surface of a reverse bending fatigue failure

1. Important examples of d. occur through mechanical means during standard operation, especially breaks of all types. One can differentiate between ductile or brittle (such as tension, compression, bending, or torsion) failures that can result from internal vibrations. Figs. lands show examples of cracks caused by reverse bending (photographed by electron-microscope). D. characteristics, causes and results are described in VDI guideline 3822, Page 2.
2. D. can also result from corrosion from aggressive media. When parts are worn by corrosion and mechanical stress, stress corrosion cracking can cause d. D. characteristics, causes and results can be found in VDI guideline 3822, Page 3 (corrosion).
3. Further d. can occur on centrifugal pumps due to wear of moving parts with close clearances or fluid velocity.

Decreased Output

Minderleistung

Moins-value en pulsance

This concept plays an important part in the calculation of a radial or mixed flow centrifugal pump impeller which is designed to produce a given head H at a given capacity Q . Starting with an infinite number (subscript ∞) of very thin blades on the impeller, we would in theory (subscript th) obtain in an ideal, i.e. frictionless, fluid a head $H_{th.\infty}$, and a pump output $\rho \cdot g \cdot Q \cdot H_{th.\infty}$, with ρ density of pumped medium and g gravitational constant. Because of the influence of the finite number of blades (subscript z), the flow lines no longer follow the blade profiles. Regions of higher pressure and lower flow velocity arise on the leading face of the blade, when the number of blades is finite, and regions of lower pressure and higher flow velocity on the back face of the blade; consequently the flow lines are deflected to the extent that the head is reduced from $H_{th.\infty}$ to $H_{th.z}$, and the flow output is reduced to $\rho \cdot g \cdot Q \cdot H_{th.z}$. The d.o. is usually taken to be the difference in outputs $\rho \cdot g \cdot Q(H_{th.\infty} - H_{th.z})$, and this difference amounts to between 25 and 60 % of the pump output in the case of conventional radial impellers. The ratio $H_{th.\infty} / H_{th.z}$ is given by the so-called reduction in output factor $(1 + p)$, which encompasses in empirical form the effect of the number of blades, impeller geometry, impeller diameter, blade exit angle and also the effect of the guide device downstream of the impeller.

The relationship between the heads $H_{th.z}$ and H is given by the hydraulic efficiency

$$\frac{H}{H_{th.z}}$$

The calculation of an impeller according to the d.o. theory pre-supposes a great deal of accumulated experience. This type of calculation is being superseded to an ever increasing extent in centrifugal pump technology by the introduction of electronic data processing equipment, which employs strictly analytical processes (e.g. finite element process, singularities process). For the calculation of axial impellers, the d.o. method has already been superseded quite a long time ago by calculation according to the aerofoil theory.

Deep Well Suction Device

Tiefsaugevorrichtung

Dispositif d'aspiration en puits profond

If the water level on the suction side lies too low for aspiration by a normal pump, centrifugal pumps equipped with a d.w.s.d. are used (particularly in the U.S.A.) for pumping relatively small water flows up to 10 m³/h from small bore wells where the water level is up to 40 m below ground level. The advantages of such an arrangement are: no moving parts inside the borehole, and installation of the centrifugal pump at any desired location. Disadvantages include produced overall efficiencies the pump cum jet booster in comparison with borehole or underwater motor pumps (submersible motor pump).

The following symbols are used:

Q, H	<u>capacity</u> and <u>head</u> of <u>centrifugal pump</u> ,
Q_T, H_T	capacity and head of motive water to jet booster,
Q_n, H_n	useful capacity and head of jet booster,
H_{tot}	overall total head,
H_s	permissible <u>suction head</u> of pump,
T	depth of water surface beneath pump centralize,
H_{VS}, H_{VT}	loss of head in rising main and inlet booster feed pipe respectively.

Mode of operation

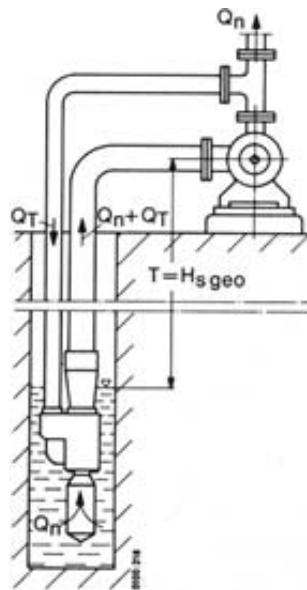


Fig. 1: Centrifugal pump with jet booster device for lifting from great depths

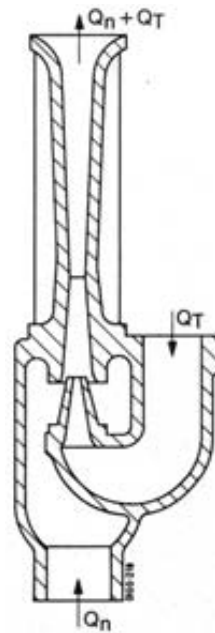


Fig. 2: Jet booster for lifting from great depths

A partial flow Q_T is tapped from the total capacity Q of the discharge line (pumping plant) of a normal centrifugal pump installed above ground level (Fig. 1), and led as motive water to the jet booster (jet pump) which is installed approx. 1 m beneath the lowest water level likely to arise. The jet booster (Fig. 2) aspirates the useful capacity Q_n and discharges the total capacity $Q_n + Q_T$, assisted by the suction action of the pump, into the pump suction branch. After passing through the pump, the circulation capacity Q_T is again tapped off, and the useful capacity Q_n is delivered to the consumer. The installation can be sized as follows:

$$H_n = T + H_{VS} - H_S,$$

$$H_T = H_{tot} - H_{VS}.$$

The values H_n and H_T can be calculated as an approximation from the above, as T and H_{tot} are known. The ratio H_T/H_n is formed, and the corresponding value for Q_T/Q_n is obtained from Fig.3. As Q_n is also given, both Q_T and the approximate design data of the centrifugal pump can be obtained from the equations

$$Q = Q_n + Q_T,$$

$$H = H_{tot} - H_n.$$

When these data are known, a suitable pump type can be selected, and the bore of the jet booster feed pipe and of the rising main can be determined. Thus the values of H_S , H_{VS} and H_{VT} (pressure loss) are now known, and the whole calculation can be repeated more accurately.

Example: Given $Q_n = 5 \text{ m}^3/\text{h}$ $H_{tot} = 80 \text{ m}$, $T = 20 \text{ m}$, $H_{VS} = 4 \text{ m}$, $H_S = 8 \text{ m}$.

$H_n = 20 + 4 - 8 = 16 \text{ m}$,
 $H_T = 80 - 4 = 76 \text{ m}$, thus $H_T/H_n = 76/16 = 4.75$;
 from Fig. 3 we obtain $Q_T/H_n = 0.95$,
 thus $Q_T = 0.95 \cdot 5 = 4.75 \text{ m}^3/\text{h}$;
 $Q = 5.0 + 4.75 = 9.75 \text{ m}^3/\text{h}$ and
 $H = 80 - 16 = 64 \text{ m}$.

The bore and losses of head in the rising main and in the motive water feed pipe can be determined from the above values, and then the calculation can be repeated more accurately. In most cases the approximate calculation will be found to be adequate.

The curve in accordance with Fig. 3 plots the average relationship between motive and useful water flows in jet boosters. The more accurate pattern of this curve will depend on the type of jet pump used. When planning a centrifugal pump cum d.w.s.d. installation, it is advisable to carry out an accurate economics calculation, because jet devices or water jet pumps have a relatively low efficiency in comparison with other centrifugal pump types which can be used for such applications (borehole pump, submersible motor pump, underwater motor pump).

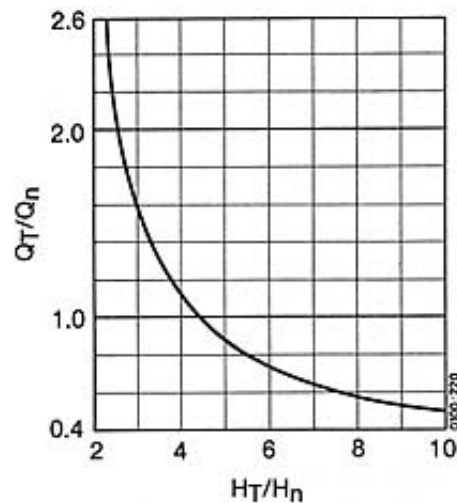


Fig. 3: Calculation diagram for jet booster devices

Degree ENGLER

ENGLER-Grad
Degré ENGLER

see Viscosity

Degree of Reaction

Reaktionsgrad
Degré de réaction

D.o.r., symbol r_{th} , is a ratio magnitude used for stages of fluid flow machines, defining the ratio of the clearance gap pressure head to the head (on centrifugal pumps) or to the fall head (on turbines). The d.o.r. can be calculated from the velocity triangles of an impeller element (fundamental equation, fluid dynamics) and is given by

$$r_{th} = 1 - \frac{v_2^2 - v_1^2}{2 \cdot \Delta(u \cdot v_u)}$$

with

v absolute velocity at impeller inlet (subscript 1) and at impeller outlet (subscript 2),
 $\Delta(u \cdot v_u)$ see fundamental equation fluid dynamics.

Usual d.o.r.'s of centrifugal pumps vary between 0.5 and 1.0, and in exceptional cases on propeller pumps also above 1.0. The d.o.r. gives an indication how to split up the static pressure (pressure) at the impeller and at the stage. A d.o.r. of 0 means there is no increase in static pressure within the impeller ("constant pressure impeller"), whereas a d.o.r. of 1 means that the static pressure increase in the stage takes place solely in the impeller.

Delivery Coefficient

Lieferzahl

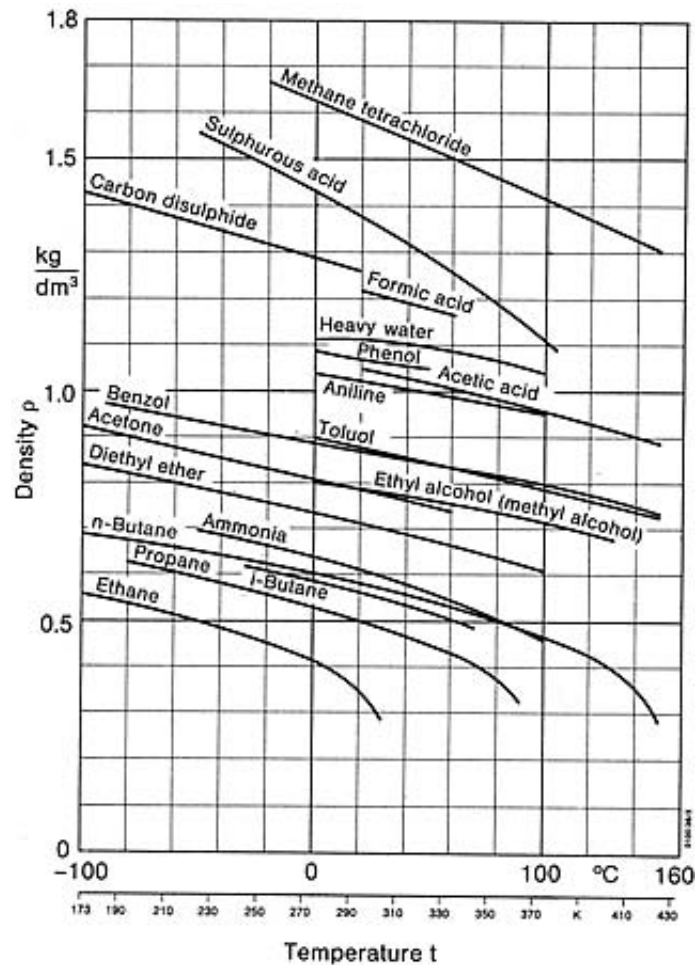
Coefficient de débit

see [Characteristic Number](#)

Density of Pumped Media

Dichte von Fördermedien

Masse volumique du liquide pompé



Density of various fluids in function of temperature (see for enlarged diagram)

The density ρ of a pumped medium (or substance) is the ratio of the mass m in a given volume V of the medium pumped to the magnitude of said volume:

$$\rho = \frac{m}{V}$$

with

m in kg, and

V in m^3 or dm^3 .

The Table and the illustration give the density of various pumped media; the density of water in function of the temperature is given in Table 1 under vapour pressure (see [Table 1](#) and in the [diagram](#)).

The d.o.p.m. defined above is not identical to the pulp density, which is an important concept in the cellulose and paper industry (pulp pumping).

Deposits in Pumps

Ablagerungen in Pumpen

Dépôts dans pompes

As distinct from protective coatings, d.i.p. (incrustations) are an undesirable phenomenon. One can distinguish between:

- deposits of corrosion products;
 - deposits of salts (predominantly carbonates, sulphates and phosphates) resulting from an imbalance in the chemical equilibrium (water hardness, pH-value);
 - deposits caused by precipitation of salts in solution, due to vapour formation in the water (particularly in shaft seals), and accumulations of undissolved substances, from the colloidal to the coarse dispersive;
 - deposits of sand (erosion). Growths (algae, bacteria) in clarification water, brackish water and seawater can also be classed under the heading "deposits" in the wider sense.
-

Design Duty Point

Auslegungspunkt, Berechnungspunkt

Point de calcul

As a general rule, the following data are given for the hydraulic design (calculation) of a centrifugal pump: capacity Q, head H and rotational speed n. These values give the d.d.p. of the centrifugal pump on the QH curve. The actual operating point of the manufactured centrifugal pump often differs from the d.d.p. uncertainties in determining the true system characteristic curve, properties of the medium pumped which differ from those assumed in the calculation, control of the centrifugal pump governed by the plant conditions, manufacturing tolerance, calculation uncertainty, performance chart screen of pump series, etc.).

The objective is very often to ensure that the d.d.p. lies as close as possible to the operating point of maximum efficiency or to the operating point of best suction behaviour. If the actual flow conditions differ appreciably from those of the d.d.p., there will be a drop in efficiency at part load and overload operation (>>operating behaviour) accompanied by disturbances due to vibrations and cavitation.

Table: Density ρ of various liquids at atmospheric pressure

Centrifugal Pump Lexicon																				KSB	
Temperature		Ethane C ₂ H ₆	Acetone (CH ₃) ₂ CO	Ammonia NH ₃	Ethyl alcohol C ₂ H ₅ OH	n-Butane C ₄ H ₁₀	i-Butane C ₄ H ₁₀	Benzol C ₆ H ₆	Aniline C ₆ H ₅ NH ₂	Ether C ₂ H ₅ OC ₂ H ₅	Formic acid CH ₂ O ₂	Acetic acid C ₂ H ₄ O ₂	n-Propane C ₃ H ₈	Phenol C ₆ H ₅ OH	Methanol CH ₃ O	Sulphurous acid H ₂ SO ₃	Carbon disulphide CS ₂	Toluene C ₇ H ₈	Carbon tetrachloride CCl ₄	Heavy water D ₂ O	
t	T	density ρ in kg/dm ³																			
°C	K																				
-100	173	0.5589	0.920			0.6900				0.842							1.432				
-90	183	0.5479				0.6827												0.9697			
-80	193	0.5367				0.6744							0.6240					0.9604			
-70	203	0.5250				0.6663							0.6134					0.9509			
-60	213	0.5125				0.6577							0.6025					0.9419			
-50	223	0.4993	0.868	0.695		0.6492				0.790			0.5910			1.555	1.362	0.9327			
-40	233	0.4850	0.855			0.6400							0.5793					0.9234			
-30	243	0.4700				0.6306	0.6156						0.5680			1.509		0.9141			
-20	253	0.4526	0.832			0.6210	0.6052						0.5555					0.9049	1.670		
-10	263	0.4339				0.6107	0.5940						0.5430			1.460		0.8956			
± 0	273	0.4117	0.812	0.636	0.8080	0.6008	0.5835	0.9001	1.039	0.736			0.5300	1.092	0.810	1.435	1.292	0.8863	1.630	(1.105)	
10	283	0.3865			0.7990	0.5898	0.5718	0.8920					0.5160		0.801			0.8769		1.107	
20	293	0.3502	0.791	0.609	0.7902	0.5788	0.5590	0.8790	1.022	0.714	1.220	1.049	0.5015	1.071	0.792	1.380	1.262	0.8677	1.585	1.105	
30	303	0.2860			0.7815	0.5665	0.5462	0.8675					0.4860		0.783			0.8583			
40	313		0.765		0.7726	0.5546	0.5340	0.8576			1.192	1.028	0.4690		0.774			0.8489	1.545	1.100	
50	323		0.756	0.561	0.7634	0.5422	0.5198	0.8460	0.996	0.676	1.184	1.018	0.4500	1.050	0.765			0.8395			
60	333		0.740		0.7546	0.5284	0.5052	0.8357			1.169	1.003	0.4328		0.755			0.8301	1.505	1.090	
70	343				0.7452	0.5148	0.4900	0.8248					0.4090		0.746			0.8205			
80	353				0.7357	0.5003		0.8145				0.980	0.3764		0.736			0.8110	1.460	1.070	
90	363				0.7260	0.4848		0.8041					0.3230		0.725			0.8012			
100	373			0.458	0.7158	0.4680		0.7927	0.951	0.611		0.960			0.714	1.110		0.7914	1.420	1.040	
110	383				0.7048	0.4492		0.7809							0.702			0.7813			
120	393				0.6927	0.4272		0.7692							0.691			0.7710			
130	403				0.6791	0.4003		0.7568							0.678			0.7608			
140	413					0.3620		0.7440										0.7501			
150	423					0.2900		0.7310		0.518		0.896						0.7392	1.310		

Diagonal Impeller

Diagonalrad
Roue hélicocentrifuge

see [Impeller](#)

Diameter Coefficient

Durchmesserzahl
Coefficient de diamètre

see [Characteristic Number](#)

Differential Pressure

Differenzdruck
Pression différentielle

see [Pressure](#)

Diffuser

Diffusor, Leitrad
Diffuseur

The distinguishing characteristic of a d. is a flow path with increasing cross-sectional area in the direction of flow, in a closed duct or channel. In centrifugal pump technology, d.'s are very frequently used on the discharge side of the casing of [volute casing pumps](#), ring section pumps (>>pump casing) and [multistage centrifugal pumps](#); they are also used as a component of [piping runs](#) ([hub diffuser](#)).

The main purpose of a d. is to transform a flow of a given [flow velocity](#) and given static [pressure](#) into a flow of lower velocity and higher static pressure with as small a loss as possible ([BERNOULLI equation](#), [fluid dynamics](#), [pressure loss](#)). The boundary layers in a d. are exposed to flow separation to a high degree ([boundary layer](#)). Consequently, certain welldefined angular d. divergences must not be exceeded; in the circular section d.'s generally adopted in centrifugal pumps, this critical angular divergence (taper angle) is of the order of 8 to 10°; if guiding devices are fitted, or in the case of [vortex flows](#) or of considerable throttling downstream of the d., these angles can be exceeded without risk of flow separation.

The d. also plays a role when connecting pipe sections of different [nominal diameter](#). In this case, a d. must be selected of adequate length so as not to exceed the critical divergence, and thus avoid additional pipe friction losses and pulsating flow separation in the d.; but in certain cases a sudden transition from a small bore pipe to a larger bore pipe may prove more advantageous than installing a d., both for cost and for hydraulic reasons ([CARNOT d.](#), [fluid dynamics](#), [CARNOT's shock loss](#), Table 4 under pressure loss, form I and II).

A nozzle ([standard nozzle](#), [entry nozzle](#)) is used to accomplish the opposite effect to the d. The d. as a specific component of a [centrifugal pump](#) is a [diffuser device](#) fixed to the [pump casing](#), in the shape of an axial, mixed flow, radial or onion-shaped rim of blades. If there is a d. in a centrifugal pump (diffuser pump), it is usually arranged downstream of the [impeller](#) (follow-up d.), although in the case of [control](#) by pre-rotational swirl adjustment there is a d., or rim of inlet guide vanes, upstream of the impeller, and one downstream, both fulfilling different functions. The function of the downstream d. is to deflect the [vortex flow](#) at the exit from the [impeller](#) into a vortex-free flow as far as possible, and with minimum loss; during this process, if we consider one and the same [flow line](#) of an axial d., the magnitude of the [absolute velocity](#) v is reduced whilst the static [pressure](#) increases. In the case of [multistage pumps](#), it may be of advantage in some cases to leave a small amount of residual swirl (rotating in the same direction) in the flow, in order to improve the [suction behaviour](#) of the following [stage](#).

The number of guide vanes (blades) in a d. should be different from the number of impeller blades. In order to avoid interference vibrations, nor should these numbers have a common divisor. Occasionally a vaneless radial diffuser is arranged downstream of radial impellers, which is sometimes called guide ring despite the fact that it does not provide directional guidance of the flow (in the sense of a d.). The absolute velocity is retarded in this ring, and the static pressure increases; there is also no danger of clogging when contaminated media are pumped.

Adjustable pitch guide vanes in d.'s are a rarity in centrifugal pumps, with the exception of the inlet guide vanes used for control by pre-rotational swirl adjustment.

Diffuser Device

*Leitvorrichtung
Organe diffuseur*

D.d. are those hydraulic parts in centrifugal pumps that alter the vortex flow before or after the impeller. When used before the impeller they are referred to as pre-rotational swirl controller (control) to increase the vortex. D.d.'s behind the impeller work like diffusers and change the kinetic energy into pressure energy.

Depending on the pump type, the d.d. can be a radial, axial or semi-axial diffuser, or even a volute casing or annular casing (pump casing).

Direct Current

*Gleichstrom
Courant continu*

D.c. is an electric current which keeps flowing in one direction. D. c. energy can be converted into chemical energy in accumulators, and reconverted from chemical energy.

Normally shunt-wound and compound-wound generators are used as D.c. generators in D.c. power plants. The operating voltages are 110 V 220 V and 440 V. The voltages at the generator terminals are usually some 5% higher, and Amount to 115 V, 230 V and 460 V.

Three-wire installations have 2 X 220 V between the neutral wire Mp (center conductor) and the outer wires L+ and L-, (terminal designation). In three-wire systems, two D.c. generators connected in series can be used, with the neutral wire connected to the junction line between the two generators. If only one generator is used, it must be equipped with two slip rings connected to opposite points on the armature winding, which feed a choke with an iron core. The center conductor is then connected to the neutral (central) point of the choke coil.

If both halves of the network are under equal load, only the magnetizing current flows through the choke. If the loads are unequal, the difference between the two outer conductor currents flows through the center conductor and through one half of the choke back to the generator armature. For a normal installation layout, a neutral wire current of 15% of the rated current is permitted.

D.c. systems are today being supplied with power in increasing numbers via semi-conductor rectifiers, mainly silicon rectifiers, from a three-phase current mains supply (three-phase current). For a constant d.c. voltage, the rectifier circuits are lined up with diodes (electrical switchgear), whereas for a variable d.c. voltage, either all the bridge branches are lined up with thyristors (fully-controlled bridge circuit) or one half is equipped with thyristors and the other half with diodes (semi-controlled bridge circuit).

Direct Current Face Plate Starter

*Gleichstrom-Flachbahnanlasser
Démarreur à plots*

D.c.f.p.s. is an item of electrical switchgear for direct current motors, by means of which the starting process takes place through the variation of Ohmic resistances in the armature and exciter circuits. On d.c.f.p.s.'s, the contact studs of the sequence switch all lie in one plane. D.c.f.p.s.'s are being gradually superseded by curators, e.g.

thyristors ([electrical switchgear](#)).

Direct Current Motor

Gleichstrommotor

Moteur à courant continu

Despite the fact that public electricity supply networks today operate almost universally on [three-phase current](#), d.c.m.'s have maintained their established position in technology, e.g. as exciters, but mainly as drivers. D.c.m.'s supplied with power from the three-phase mains supply via up-to-date control and regulation techniques (e.g. thyristors, [control](#), [electrical switchgear](#)) fulfil important drive assignments, including the [drive](#) of [centrifugal pumps](#).

Table: Characteristics of direct current motors

	Shunt-wound motors	Compound-wound motors
Starting	Exciter circuit under full tension, the series-connected resistance in the Armature circuit is switched off in steps. Self-starter for automatic or remotecontrolled installations	
Starting torque	approx. 1.7 times full-load torque (if full mains voltage is available)	approx. 2.5 times full-load torque
Starting current	approx. twice full-load current	approx. 3.5 times full-load current
Speed behaviour	speed fluctuates very little even under fluctuating load conditions	
Regulation	speed increase by weakening of the field (loss-free)	
	up to 1.2 times without interposes above 1.2 times with interposes	up to 3 times with interposes
	speed decrease by series-connected resistances in the armature circuit (not loss-free): regulating starter.	
Reversal of direction of rotation	Only possible via commutation by reversing either the armature current only or the excitation current only. A simple crossing over of N and P (terminal designation) will be of no avail. On compound-wound machines the main current winding must also be reversed, if the exciter winding is reversed.	

The Table lists the main characteristics of d.c.m.'s. Series-wound motors are not included in this Table, since they are rarely used to drive pumps because of their very powerful starting torque, their harsh run-up to speed and an inadmissible speed increase under low-load conditions.

Direction of Rotation

Dressing

Sense de rotation

see [Rotational Speed](#)

Dirty Water Pump

Schmutzwasserpumpe
Pompe à eaux sales

see [Sewage Pump](#)

Discharge Casing

Druckgehäuse
Corps de refoulement

see [Pump Casing](#)

Discharge Elbow

Auslaufkrümmer
Coude de refoulement

D.e. is the discharge pipe elbow (elbow pressure loss) of a tubular casing pump.

Discharge Line

Druckleitung
Conduite de refoulement

see [Pumping Plant](#)

Discharge Loss

Auslaufverlust
Perte de charge à la sortie

D.1. in centrifugal pump technology refers to the following head loss:

$$H_{V, \text{ discharge}} = \zeta \frac{v^2}{2g}$$

where

ζ loss coefficient, $\zeta = 1$,

v flow velocity in the outlet cross-section of a component exposed to hydraulic flow, and

g gravitational constant.

Dock Pump

Dockpumpe
Pompe de dock

The d.p. is a centrifugal pump used for emptying dock installations (dry dock, floating dock). A distinction must be made between a dry dock pump and a floating dock pump.

Dry dock pump. The pumping station of a dry dock is usually arranged along one of the longitudinal walls of the dock, near the dock gate (Fig. 1). The water enters the intake chamber via trash racks arranged in the dock floor. For main bilge pump duty (usually two to four pumps) tubular casing pumps (mixed flow pump, capacities 3000 to 30000 m³/h) are generally used today, but for the construction and repair docks of giant tankers, large volute casing pumps with a concrete casing are often used nowadays (capacities from 30 000 to 50 000 m³/h). When pumping commences, the static head (head) is equal to zero, and only the pipe friction losses (pressure loss) have to be overcome. There exists simultaneously a very adequate positive suction head (suction behaviour) of 10 to 12 m, and no difficulties arise regarding the net positive suction head of the pump (cavitation). As the water level in the dock sinks progressively, the head of the installation increases, and the capacity of the pump decreases (characteristic curve). Towards the end of the emptying operation, the main bilge pumps are switched off in succession, and so-called "after-bilge pumps" are switched on; these are usually smaller tubular casing pumps fitted with helical impellers (impeller). When the dock is completely empty, drainage pumps take over the duty of draining the leakage water. Because coarse solids and other dirt are usually contained in this leakage water, due to the repair work going on, these drainage pumps are usually fitted with non-clogging impellers (impeller).

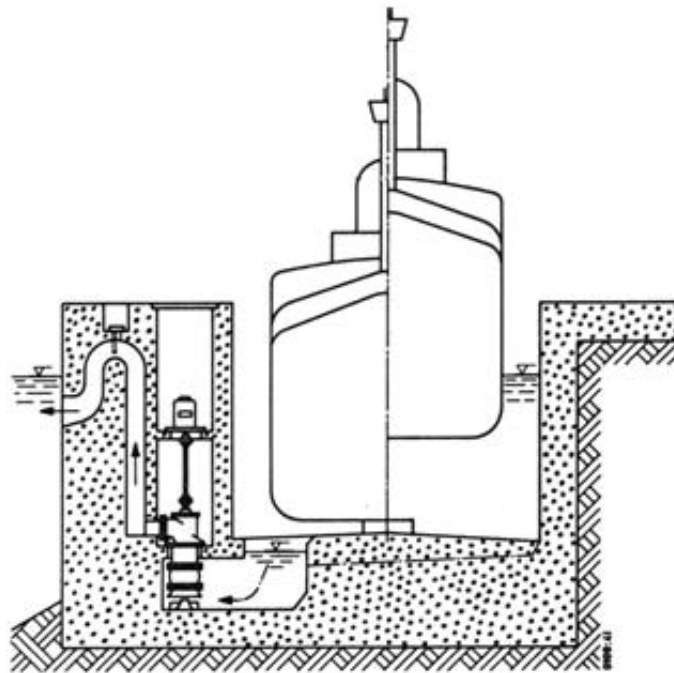


Fig. 1: Dry dock

Floating dock pump. In a floating dock, the bilge pumps (usually four to six pumps, sometimes more) are arranged in one leg of the U-shaped floating dock (Fig. 2). These pumps are vertical, usually radially split, multisuction pumps with a ring section or volute casing (pump casing) for capacities from approx. 500 to 3500 m³/h. The rotor is withdrawable in the upward direction. The pump shaft and the drive shaft which extends vertically upwards (open type) are guided in grease-lubricated plain bearings. The thrust bearing and driver are arranged in a watertight compartment a few metres above the pump.

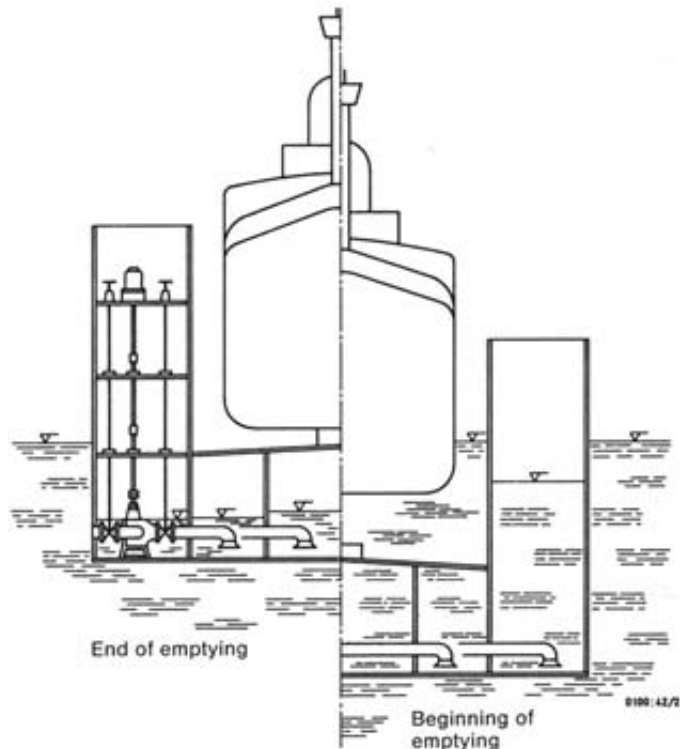


Fig. 2: Floating dock

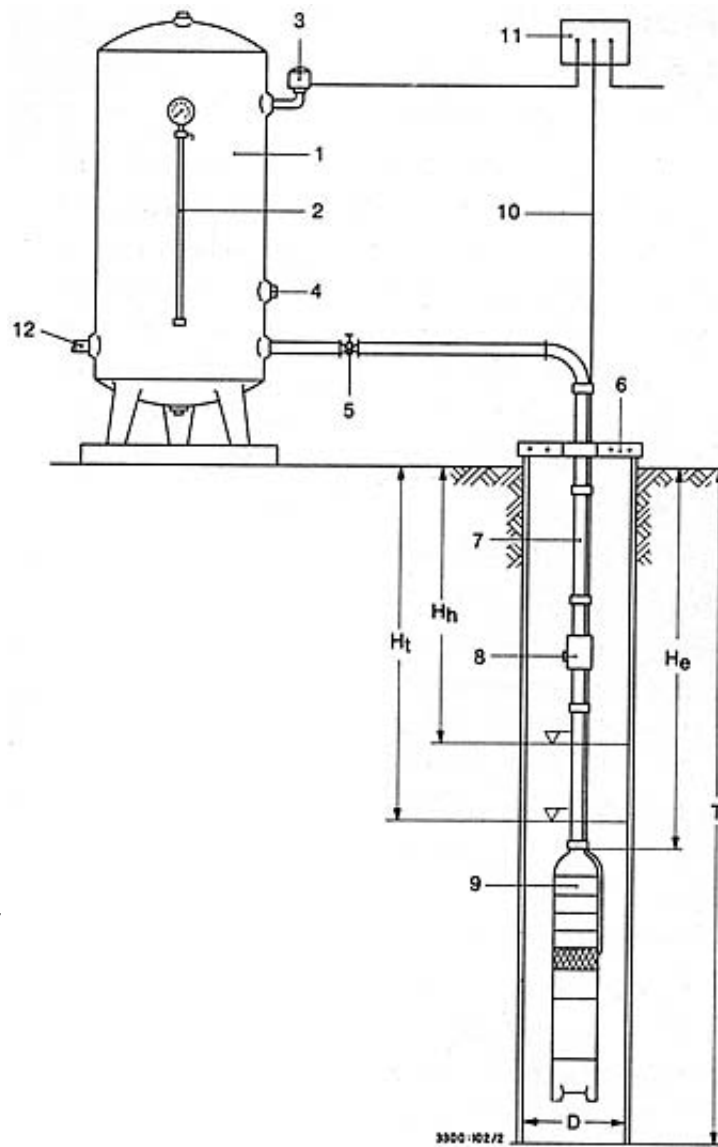
The complete floating dock is subdivided into separate compartments (Fig. 2), each of which can be flooded individually. When the dock is submerged, the pumps and their drive shafts are under water (wet installation). Each compartment is equipped with its own suction pipe, connected to its bilge pump via a shutoff valve (valves and fittings). All the pumps and shut-off valves can be remotely operated from the control desk, so controlling the water levels in each compartment and hence the inclination of the dock, both longitudinally and transversely. The compartments are pumped out to different water levels, according to weight distribution, for the emersion of the dock and of the docked ship. In contrast to dry docks, the head of the installation fluctuates by only a few metres.

Domestic Water Supply Plant

Hauswasserversorgungsanlage
Installation domestique d'adduction d'eau

D.w.s.p.'s are installed for the supply of individual houses or farms in bases where a well of adequate yield exists on the premises. Both horizontal and vertical pumps are used. The pumps must be equipped with a foot valve in the suction pipe (pumping plant); exceptions to this rule are underwater motor pumps, which are suspended beneath the lowest water level which occurs (see illustration).

Start-up and shutdown of the pumps are pressure-controlled. The air cushion in the pressure vessel can, in the case of operation of a underwater motor pump, be increased or kept constant with the aid of a combination of a vent valve and a nonreturn valve (valves and fittings). Any excess air in the pressure vessel is expelled by an air monitor (item 4). Small size d.w.s.p.'s are available in the form of fully-assembled, fully automatic compact plants ready for connections.



Domestic water supply plant
 with an underwater motor pump
 1 pressure vessel; 2 water level
 gauge; 3 pressure switch; 4 air
 monitor; 5 shut-off valve; 6
 suspension clip; 7 rising main;
 8 vent valve; 9 underwater motor
 pump; 10 electric cable; 11
 switchgear; 12 domestic water
 pipe;
 D i.d. of borehole;
 T depth of borehole;
 H_e installed depth;
 H_h water level at rest;
 H_t depressed water level

Double Channel Impeller

Zweikanalrad
Roue à deux canaux

see [Impeller](#)

Double Suction Impeller

Doppelströmiges Laufrad
Roue à double entrée, roue à double flux

see [Impeller](#)

Double Volute

Doppelspirale
Volute double

see [Pump Types](#)

Drainage Pump

Entwässerungspumpe
Pompe d'assèchement

D.p.'s are used for dewatering cellars and courtyards exposed to flooding or groundwater, and for the drainage of pits and deep roadway underpasses. These pumps are usually switched on and off automatically via float switches. The water pumped may contain dirt and solid particles; the permissible grain size of the latter will depend on the pump size and impeller fitted.

The conventional type of pump, rigidly connected to a flanged motor via a long suspension pipe (Fig. 1) is being replaced more and more by the submersible motor pump (Fig. 2). In the latter type, usually installed vertically, the motor situated above the pump (it is often oiltilled for better cooling) is sealed by mechanical seals and shaft sealing rings (shaft seals). The chamber situated between the seals arranged in pairs is filled with grease or oil.

Compact construction resulting in a small bulk makes submersible motor pump suitable as portable and mobile d.p.'s.

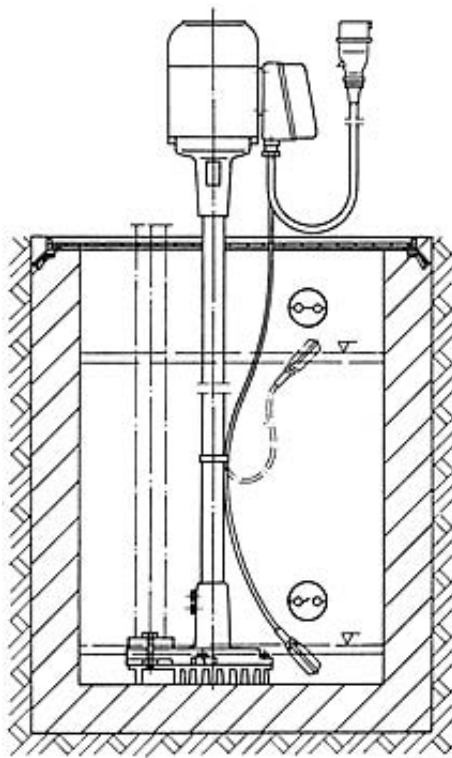


Fig. 1: Drainage pumps with dry-installed electric motor

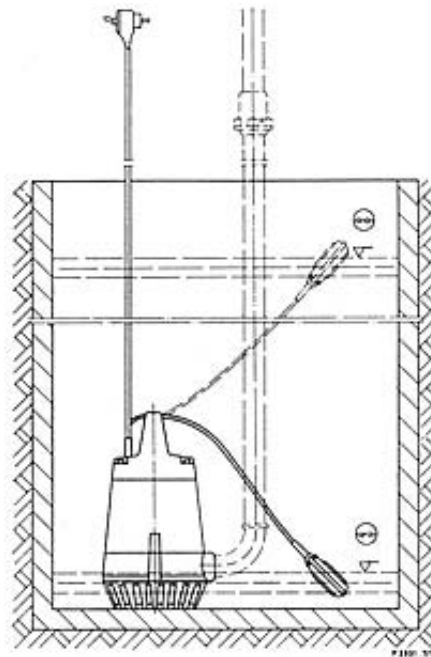


Fig. 2: Drainage pump of submersible motor pump type

Drive

Antrieb
Commande

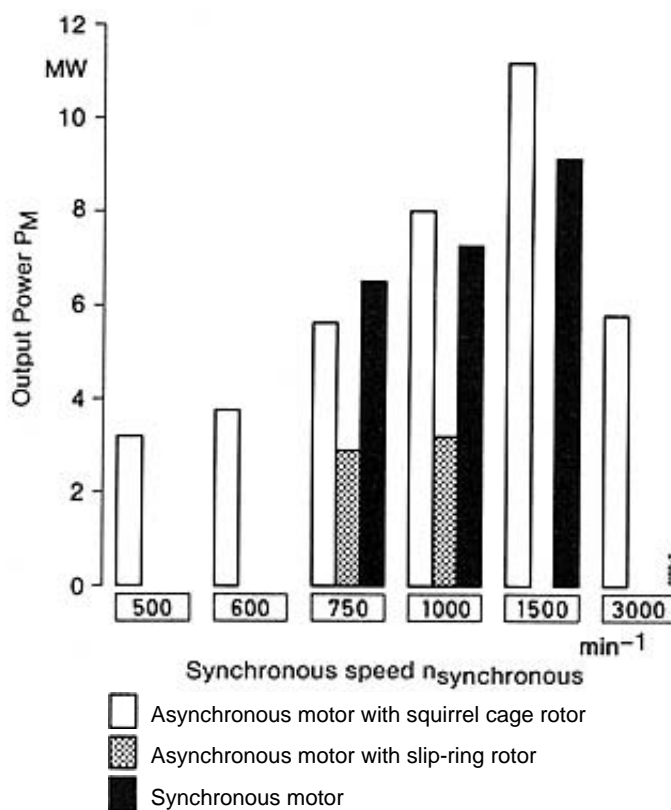
Most centrifugal pumps are driven by electric motors. Apart from electrical d.'s, centrifugal pumps are sometimes driven by reciprocating internal-combustion engines (e.g. Diesel engines), gas turbines, steam turbines, and more rarely by hydraulic turbines.

Electric motors and turbines generate a uniform torque, whereas reciprocating internal combustion engines generate a non-uniform (irregular) torque (see also positive displacement pump). The cyclic irregularity of the Diesel engine is compensated to a considerable extent by design measures (flywheel masses, number and arrangement of cylinders).

In the case of electrical d.'s, single phase a.c. motors (alternating current) with squirrel-cage rotors (asynchronous motor) are used almost exclusively in the low power range up to 1 kW approx.

In the medium and high power ranges, up to 8000 kW approx., three-phase motors (synchronous motor) with short-circuit rotors (squirrel-cage rotors) predominate. In the top power range of centrifugal pump d.'s, synchronous motors are sometimes used on account of the power factor improvement (compensation of reactive current) they offer and their relatively high motor efficiency; in storage power stations they serve as generators during turbine operation.

Typical power ranges of electric motors used as d.'s of centrifugal pumps are illustrated in the diagram.



Power ranges of electric motors driving centrifugal pumps, in-function of synchronous speed

Drive Rating

Antriebsleistung
Puissance moteur

PM is the available power, provided by the shaft coupling, of the drive, for which it is measured. The SI units of the d.r. are Watts (W), and with larger d.r.'s the practical units are 1 kW or 1 MW.

Serially produced pumps are usually coordinated with their drive motors; their d.r. is chosen in conjunction with the largest shaft power, within allowable limits; in the case of standard motors it is rounded up to the next larger nominal d.r. The rating printed on the name plate on the standard motors is not allowed to be exceeded by more than 3% during continuous use. Additional safety precautions against possible later wear or deposits is not usual for serial pumps, as their shaft power, a function of capacity at the upper limit of the operating range, generally changes only slightly (characteristic curve).

Customized centrifugal pumps are designed according to their operating point. Their d.r. must always be larger than the shaft power P of the pump, due to power increases through deviation of the design duty point, variable operating point, speed variations (affinity laws), changes in the density of the medium, manufacturing tolerance, wear or deposits in the pumps etc. To counter these influences one adds an additional 10 to 20%, according to power, for customized clean water pumps in accordance with the formula:

$$\frac{P_M}{P} = \frac{60 + 1.1 \cdot P^2}{50 + P^2}$$

with

P_M d.r. in kW,

P shaft power of pump in kW.

Greater margins to the ratings are possibly necessary for sewage pumps, for the starting process, for a change in the density of pumped medium, for operation of a system with a single pump in parallel operation, for influences from the pump's ambient conditions (tropical climate), foreign amounts of wear and for control of over synchronous speeds (synchronous speed).

Driving Flow

Treibstrom

Débit moteur

see Deep Well Suction Device

Dry Installation

Trockenaufstellung

Installation à l'air libre

The concept of d.i. is mainly used in conjunction with large vertical pumps. A d.i. centrifugal pump is installed in a dry area, and its pump casing is never flooded, wholly or partially, in contrast to wet installation pumps.

Dry Running

Trockenlauf

Marche à sec

D.r. is usually undesirable in a centrifugal pump; it occurs in the total absence of the liquid component of the medium pumped (absolute d.r.) or e.g. after an ingress of air in the suction pipe, or again if gas bubbles (formation of air pockets) attach themselves during normal operation to normally wetted rotating pump components (partial d.r.).

Under normal operation of a well-designed centrifugal pump, the liquid pumped completely fills the flow space inside the pump, including the close-clearance slit seals at the impellers and the packings or mechanical seals (shaft seals). The liquid helps to cool and lubricate the components in contact (mechanical seal), exercises a centering (centripetal) action in the throttling gaps of the impellers and shaft passages (multistage pump), so that e.g. long and slender ring section pumps are able to run without fouling of the rotor in the casing. In the absence of liquid, d.r. can occur in certain parts because of deficient cooling and centering action. The consequences are overheating, abrasion, seizure of the materials, vibrations and other phenomena which may in due course lead to the complete disintegration of the pump.

If the pump operator cannot avoid such instances of absolute or partial d.r., it may be necessary to design the centrifugal pump rather more expensively and elaborately; more robustly designed shafts will prevent fouling of the rotor in the casing, specially designed clearance gaps (slit seal) can be provided, mechanical seals or packings (shaft seals) should be supplied with lubricating or sealing liquid independently of the liquid pumped the same applies for the bearings inside the centrifugal pump which would normally be lubricated by the medium pumped

(plain bearing). Pumps fitted with a hydraulic axial thrust balancing device consisting of a balance disc and disc seat only must be equipped with an additional thrust bearing to prevent seizure. Finally the pump can be shut down in the event of incipient d.r. by an automatic protection device against d.r.

Self-priming pumps always require a certain liquid fill in order to carry out their self-priming operation. During this self-priming phase they operate under partial d.r. conditions.

Dynamic Head

Geschwindigkeitshöhe
Hauteur dynamique

D.h. is the kinetic energy portion of the head. According to DIN 24260 it is the kinetic energy of the fluid per unit weight. In the BERNOULLI equation (>>fluid dynamics) the d.h. is combined with the pressure head and the geodetic altitude and is given by:

$$\frac{v^2}{2g}$$

where

v flow velocity in a characteristic flow section, and
g gravitational constant.

Dynamic Pressure

Dynamischer Druck Staudruck
Pression dynamique

see Pressure

Dynamic Viscosity

Dynamische Zähigkeit
Viscosité dynamique

see Viscosity

E

Economics

Wirtschaftlichkeit

Rentabilité

The e. connected with the operation of a pump, a pumping plant is calculated by determining the annual operating costs of various alternatives, and the relevant interest payment and amortization costs of the plant values, i.e. of the machinery and buildings.

The operating costs K_E consist of the energy costs K_B in DM/year and the expenditure K_H for staff and auxiliary materials.

$$K_E = \frac{\rho \cdot g \cdot Q \cdot H \cdot z \cdot e}{3.6 \cdot 10^6 \cdot \eta_{Gr}}$$

with

- ρ density of pumped medium in kg/m³,
- g gravitational constant in m/s²,
- Q pump capacity in m³/h,
- H pump head in m,
- η_{Gr} efficiency of pumping set (group consisting of pump and drive),
- z number of annual operating hours in h/year,
- e specific energy costs in DM/kWh.

When one wants to replace a used pump (subscript 0) with a new one, with a $\Delta\eta$ better pump efficiency, the saved energy costs, ΔK_E , can be calculated with:

$$\Delta K_E = \frac{P_0 \cdot z \cdot e}{1 + \eta_0 / \Delta\eta}$$

where

- P_0 shaft power of the comparable pump in kW,
- η_0 pump efficiency of the comparable pump.

Personnel requirements are given by the degree of automation, the operating time and the maintenance cost of the system. The expenses K_H (in DM/year) vary from case to case.

The capital investment cost K_K (in DM/year) for interest and depreciation are separated into groups by machines, systems and buildings, and is calculated by:

$$K_K = I \cdot \left(\frac{1}{t_n} + \frac{p}{200} \right)$$

with

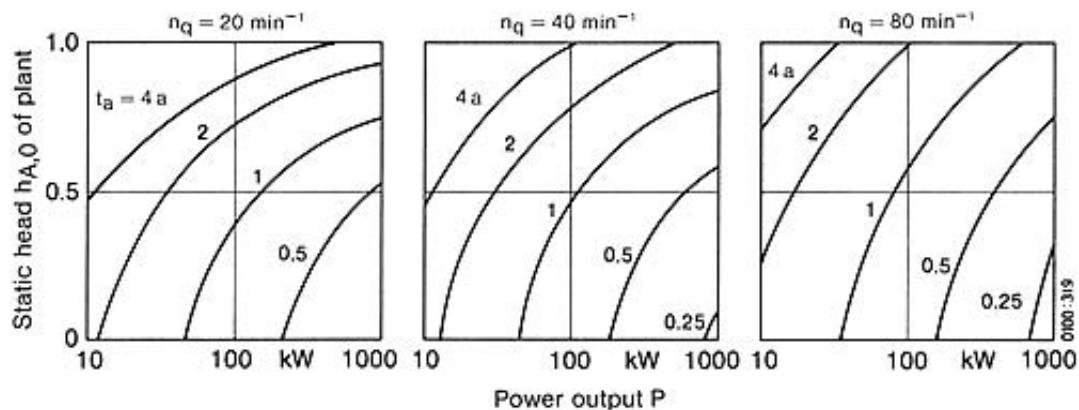
- I investments for buildings, machines, piping in DM,
- p interest rate in %/year,
- t_n useful economic life in years, which is approximately as follows:

- 40 years for buildings,
- 15 years for machineries,
- 40 to 60 years for pipelines.

If one sets the capital investment cost K_K , given by energy-saving investment I , equal to saved energy costs ΔK_E , one can then calculate the payback time t_a in years as a general measure for the e. of this investment:

$$t_a = \frac{1}{\frac{\Delta K_E}{I} - \frac{p}{200}}$$

The acceptable length for the payback time t_a depends on the economy: with new investments in pumps one can set a useful economic life often years, but with modernization in mind t_a should not exceed six years, although the technical life expectancy t_n can be much longer.



Payback time t_a as a function of various factors of influence in an economic viability calculation (cf. text)

- n_q = specific speed;
 P = drive rating at the operating point in kW;
 $h_{A,0}$ = static fraction of the system head $H_{A,0}$ referred to the head at the operating point

The example illustrated indicates a payback time of t_a in years after replacement of capacity adjustment by throttling (control) with available frequency adjustment with the calculation based on a large annual operating time and an interest rate of $p = 8 \text{ \%/year}$ and a specific energy cost $e = 0.20 \text{ DM/kWh}$ (see also Fig. 12 under control).

Experience shows that the following aspects generally apply to the sizing of a pumping plant or of a piping system:

1. high flow velocities reduce the capital cost of a plant, but also increase the energy costs and the rate of wear;
2. a smaller number of larger pumps often results in higher efficiencies and therefore lower energy costs; but if we include a standby unit, the capital costs are higher;
3. if the operating times are short, low capital costs must be aimed for, therefore the velocities should be selected as high as possible.
4. if the operating times are long, the energy costs become dominant, so that any measures aimed at reducing energy consumption. e.g. speed adjustment (control) would pay for themselves within a few years.

Effective Power

Wirkleistung
 Puissance réelle

see Power

Effective Pressure

Wirkdruck
Pression effective

see [Standard Nozzle](#), [Standard Orifice](#), [Standard Venturi Nozzle](#)

Efficiency

Wirkungsgrad
Rendement

E., standardized symbol η , is generally understood as a ratio magnitude characterizing the quality of a power transformation in energy technology, in particular in the case of work-producing machines (pumps, compressors, fans) and of prime movers (turbines): the η is the ratio of "useful power" to "expended power",

$$\eta = \frac{\text{useful power}}{\text{expended power}} = \frac{\text{expended power} - \text{power loss}}{\text{expended power}}$$

In centrifugal pump technology, the useful power, expended power and power loss assume different meanings, depending on the nature of the η . (according to DIN 24260):

Pump η :

useful power = pump output P_Q ,
expended power = shaft power P .

Mechanical η :

power loss = mechanical power loss P_m in the pump bearings (plain bearing, anti-friction bearing) and in the shaft seals,
expended power = shaft power P .
power

Internal η :

useful power = pump output P_Q ,
expended power = shaft power P minus mechanical power loss P_m in pump bearings and shaft seals.

Hydraulic η :

useful power = pump output P_Q ,
expended power = shaft power P minus power loss $P_{V.Rads.}$ caused by impeller side friction and mechanical power loss P_m in bearings and shaft seals.
power

E. of pumping set η_{Gr} :

useful power = pump output P_Q
expended power = power absorbed by the drive, measured at a location to be mutually agreed (e.g. at the terminal box of the electric motor, or at the start of an underwater cable).
power

Efficiency Re-Evaluation

Wirkungsgradaufwertung

Revalorisation de rendement

E.r.e. is a way of describing the fact that when two geometrically similar centrifugal pumps are compared, the bigger one, or the faster one, or the one operating on a medium of lower viscosity exhibits the higher pump efficiency it means that when passing e.g. from the test model to the prototype construction, the pump efficiency measured on the model must be memorized, in so far as the geometrical similitude has been maintained intact in all components (similarity conditions), including in relation to surface roughness and to clearance gap widths. Thus this change in efficiency is only a consequence of the change in REYNOLDS number (model laws) resulting from the change in pump size. Larger centrifugal pumps generally have higher REYNOLDS numbers, and this means, according to the laws of fluid dynamics that the flow losses are lower within certain limits, and therefore the internal efficiencies (efficiency) are higher.

Because it is impossible to achieve an exact geometrical similitude, there is an additional influence of the machine size present, which operates in the same direction as the REYNOLDS number. The influence of pump size on pump efficiency is of practical significance in cases where efficiency measurements (measuring technique, pump test bed) were carried out on reduced scale models of large pumps, and where an evaluation of the anticipated efficiency of the fullscale construction is sought from the efficiency measured on the model (this applies e.g. to the evaluation of the shaft power, if it exceeds the power that can be measured on the test beds or to pump casings made of concrete). There are as yet no universally valid rules for e.r.e. In any case the e.r.e. must be clearly defined and mutually agreed between purchaser and manufacturer before the model test takes place. An example of the approximation formulae for e.r.e. is given by the formulae of PFLEIDERER and ACKERET:

$$\frac{1 - \eta_{i2}}{1 - \eta_{i1}} = \left(\frac{Re_1}{Re_2} \right)^a \quad a \approx 0.1$$

$$\frac{1 - \eta_{i2}}{1 - \eta_{i1}} = \frac{1}{2} \left[1 + \left(\frac{Re_1}{Re_2} \right)^a \right] \quad a \approx 0.2$$

with

η_i internal efficiency,

Re REYNOLDS number (model laws).

The Re number is preferably calculated from the circumferential velocity at the impeller outlet and the impeller diameter.

The first of the two formulae quoted above (PFLEIDERER formula) was incorporated in the older editions (1968 ago) of the VDI rules for Centrifugal Pumps DIN 1944. Various Working Groups are at present dealing with the problem of e.r.e., e.g. Working Group No. 5 in IAHR (International Association for Hydraulic Research). Whose results are published from time to time.

Ejector

Ejektor

Éjecteur

see Deep Well Suction Device

Elbow

Krümmer

Coude

The flow through curved ducts is subject to additional losses (pressure loss) in comparison with the flow through straight ducts (flow in pipes). The boundary layer flow in an e. is subject to a pressure rise in the following zones: at the e. inlet at the outer radius of curvature, and at the e. outlet at the inner radius of curvature. At these points there is a thickening of the boundary layer, or a separation of the flow (boundary layer). In addition, due to the differing static pressures in the e. cross-section, a secondary flow is created, which superposes itself on the primary flow through the e. (flow-through, flow velocity) in such a way that a liquid particle, instead of passing through the e. along a simple curved path, in fact describes a much longer helical flow path (flow line).

Losses in e.'s (pressure loss) in a pumping plant can be reduced in the following ways: provision of smooth guide plates close to the inner side of the e. (in practice often constructed as vane cascade, Fig. 9 under pressure loss); increasing the radius of curvature (particularly effective on the inner side of the e.); decreasing the flow velocity (arranging a diffuser upstream of the e. and not combining same with the e.); reducing of the deflection angle. In the case of e.'s with sections which are not axially symmetrical (e.g. elliptical), the longer of the two section axes should be at right angles to the plane of curvature, in order to minimize the pressure loss.

In centrifugal pumps, in particular tubular casing pumps, e.'s may be used as intake elbow or discharge elbow.

Elbow Casing Pump

Krümmergehäusepumpe

Pompe à corps tubulaire coudé

see Propeller Pump

Electrical Circuits

Elektrische Schaltungen

Couplages électriques

E.c. for connecting of three-phase motors (asynchronous motor) to the mains comprise the star circuit (Fig. 1) and the delta circuit (Fig. 2).

Star circuit (Y):

$$U = U_{Str} \cdot \sqrt{3},$$

$$U_{Str} = U / \sqrt{3},$$

$$J = J_{Str}.$$

Fig. 1 illustrates the stator of a three-phase motor in star circuit. The starts of the winding strands U1, V1, W1 are connected to the conductors L1, L2, L3 of the mains (terminal designation). The ends U2, V2, W2 are joined together at the star point. The strand voltages U_{Str} (phase voltages) U1-U2, V1-V2, W1-W2 are equal to the star voltages of the mains. The currents J tapped from the mains have the same magnitude in the three respective windings. At any instant their sum is zero. The absorbed power is

$$\begin{aligned} P &= 3 \cdot U_{Str} \cdot J_{Str} \cdot \cos \varphi, \\ &= \sqrt{3} \cdot U \cdot J \cdot \cos \varphi. \end{aligned}$$

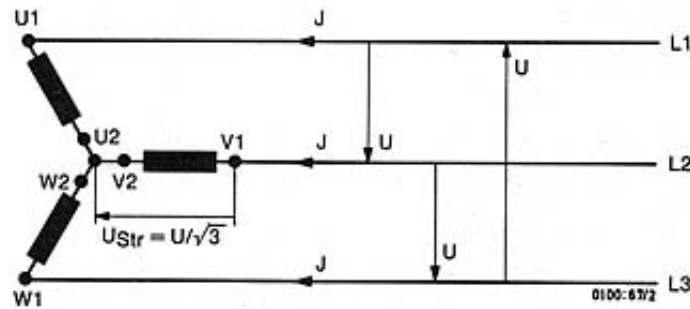


Fig. 1: Star circuit

Delta circuit (Δ):

$$\begin{aligned} U &= U_{Str} , \\ J &= J_{Str} \cdot \sqrt{3} , \\ J_{Str} &= J/\sqrt{3} \end{aligned}$$

Fig. 2 illustrates the delta-connected stator of a three-phase motor. The strand voltages U1-U2, V1-V2, W1-W2 are in this case equal to the line-to-line voltages. The currents J tapped from the mains are only the line-to-line currents, combined from the phase currents J_{Str} . Their sum is again zero at any instant. The absorbed power can be calculated from:

$$\begin{aligned} P &= 3 \cdot J_{Str} \cdot U_{Str} \cdot \cos \varphi , \\ &= \sqrt{3} \cdot J \cdot U \cdot \cos \varphi . \end{aligned}$$

The windings of three-phase motors can be connected to the mains conductors L1, L2, L3 in delta or star. The voltage indication on the motor rating plate of standard motors, e.g. 220/380 V signifies that the motor winding must be connected in delta for 220 V operating voltage, and in star for 380 V operating voltage. This means that *star-delta starting* of the motor (*star-delta starter circuit*) can only be effected at the lower of the two indicated voltages. If a motor for star-delta starting is to be connected to a 380 V network, it must be wound for 380/660 V. In order to simplify these indications, the voltage on motor rating plates is generally indicated today as 220 V Δ or 380V Δ respectively, instead of 220/380 V or 380/660 V respectively.

In the case of *direct-on-line starting*, the following points must be remembered: For an operating voltage of 220 V, a 220 V Δ motor should be delta-connected to the mains. For an operating voltage of 380 V, a 220 V Δ motor should be star-connected (380 V Υ), whereas a 380 V Δ motor should be delta-connected.

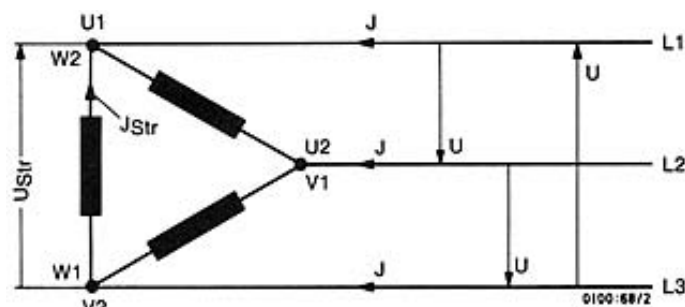


Fig. 2: Delta circuit

Electrical Switchgear

Elektrische Schaltgeräte
Appareillage électrique

E.s. consists of elements in energy distribution and control which close or connect, open, break or isolate electric circuits; e.s. conducts joins and isolates the turnover of energy and of signals, in electro-mechanical construction and in the form of semi-conductor switches. E.s should be selected in the form best suited to meet the duties and requirements at installation site, because in the case of switchings or faults in electric circuits, the transition from one steady state condition to another takes place in the form of transient effects. The reason is the presence by transient effects in energy accumulators (magnetic accumulators, e.g. chokes or coils, electrical accumulators, e.g. capacitors, cables). E.s. is subdivided into alternating current e.s. for voltages above 1 kV (in accordance with DIN VDE 0670) and into low-tension switchgear (in accordance with DIN VDE 0660).

- a) *Alternating current switchgear for voltages above 1 kV* is suitable for repeated switching on and off in the form of circuit-breakers, isolating switches, earthing switches, load isolators, fuses. The following types are used:

Circuit-breakers (of heavy-duty type) are switches with a switching capacity adequate to cope with the loadings which arise when switching electrical machinery on or off, particularly under short-circuit conditions. Single and multipole types are used, for the following duties: switching off the three-pole short-circuit, automatic reposing under short-circuit conditions, disconnection of a ground leakage on two phases, switching offender phase opposition of short-line faults, and of capacitive and small inductive currents.

Power isolating switches are circuit-breakers which create a visible separating gap during the switching off motion.

Isolating switches must only be disconnected with the circuit practically do-energized. They are air break circuit-breakers with a visible separation gap of a given insulating capacity, and they must meet special requirements with regard to the protection of plant and plant personnel. The separation gap must be easy to recognize in the switched off position of the isolating switch.

Earthing switches are operated with the circuit practically do-energized to earth and short-circuit switched off plant and machinery.

Load isolators are switches with a switching capacity corresponding to the loading which arises during the switching on and off of electrical equipment in undisturbed conditions.

Fuses interrupt current paths by the melting of certain components as a result of the heat generated by the current itself, when it exceeds a given value over a given period of time; they can disconnect short-circuits.

In accordance with DIN, the following values apply to circuit-breakers of heavy-duty type:

- Standardized rated currents in accordance with DIN 43 626 for 400, 630, 800, 1250, 1600 A, amongst others.
- Standardized rated (nominal) breaking capacity, in accordance with DIN 43 612 for 100, 150, 250, 350, 500 MVA, amongst others.

- b) In the case of *low-tension switchgear*, we have the following constructions and applications: switches (circuit-breakers), starters, controllers and resistors, regulators, plugs and sockets, fuses. These devices are used as follows:

Switches (circuit-breakers) are used for the arbitrary or automatic switching-on, -off or -over of current paths in main electrical circuits, for additional functions (e.g. command signals, annunciation, measurement, interlocking in auxiliary circuits) and for the indirect control of equipment in control current circuits.

Starters, controllers and resistors are used to start and run up equipment, particularly motors to normal operating condition.

Regulators are used for adaptation to the operating condition.

Plugs and sockets connect and interrupt current paths.

Fuses automatically break electric circuits in the event of over-current by melting of the current-conducting fusible conductor; as a general rule, fuses do not represent a perfect motor protection, they limit the current, and if correctly sized, they protect against undue thermal and dynamic short-circuit loading.

In accordance with DIN 43626, the following rated currents apply to low tension switch-gear: 6, 10, 16, 25, 40, 63, 100, 200, 400, 630 A.

The corresponding values for low tension quick-break fuse inserts (NH fuse inserts) are: 6, 10, 16, 20, 25, 36, 50, 63, 80, 100, 125, 160, 200, 224, 250, 300, 355, 425 A.

Low tension switches are subdivided according to the following features: mode of operation (setting switches, push button switches, locking switches); mode of actuation (hand switches, remote switches); their circuit breaking capacity (at alternating current for example isolating switches, load isolators, motor controllers, circuit-breakers); mode of arc extinguishing (e.g. pneumatic, magnetic arc quenching); application (protection, control, isolating, selector, limit, auxiliary twtchy; mode of installation or connection (e.g. mounted on a switchgear panel).

The various features in the subdivision are combined in the actual low tension switches.

Remote control switches are switches with a power drive (solenoid, pneumatic or motor drive). Remote actuation implies an arbitrarily effected change in position of a switch at any required distance by means of a power drive, an auxiliary tripping device or by breaking the auxiliary circuit of conductors or closed-circuit tripping devices.

Contactors are remote control switches with release force without a mechanical locking device, which are actuated and held in position by their drive, and are mainly used for high switching frequencies.

Load isolators are used to switch equipment and plant components (not motors) on and oft in the undisturbed condition, with nominal switching on and off capacities up to approx. twice the nominal (rated) current (e.g. master switch of a distribution system).

Motor controllers are sized in accordance with the starting current (starting process) of the motor, in respect of rated make and break capacity. Their breaking capacity amounts to eight times the rated motor current for motors up to 100 A rated current, and to six times the rated motor current (for $\cos \varphi = 0.4$) for motors above 100 A. The following further criteria must be taken into account when sizing motor controllers: continuous load rating, service life of the appliance, switching frequency etc.

Protective circuit-breakers (overload relays) protect plant components by automatic opening or closing in the event of excessive currents, overheating, fault voltages or fault currents (earth leakage), or under voltage in relation to built-in or built-on auxiliary services.

As regards *motor protection*, one can summarize the position as follows:

Motor protection is achieved by means of adjustable thermal over current releases with a time delay dependent on the current, with a bimetallic trip in each current path, heated by the operating current. The tripping is dependent on the current. The tripping delay times in relation to the starting currents of squirrel-cage rotors are determined in accordance with DIN VDE 0660. Fuses do not only provide protection against overloads but also against short-circuits.

Temperature sensors (e.g. semi-conductors) embedded in the winding ends of the motor are also used in motor protection devices for special requirements. The resistance of these sensors changes as the temperature rises, and their inertia is very small (this is an attractive feature in the case of steep temperature rises and blocked motor rotors).

Low tension circuit breakers are equipped with electro-magnetic overburdens releases (trips) for a short-circuit switching capacity at low switching frequency. They can be remotely switched off by means of tripping solenoids.

Semi-conductor switches are electronic switches for main circuits, they are equipped with thyristors and triacs as rapid switching elements. Their advantages include a very high switching frequency, a service life independent of the switching frequency, a practically unlimited and maintenance free continuous load rating, and a high switching speed. Against these advantages, they exhibit the following disadvantages: voltage drop, dissipation loss, inverse voltage, storing capacity, no potential separation, difficulty in recognizing faults, current carrying

capacity and resistance to short-circuits.

Thyristors are controllable, rectifying, silicon-base semi-conductor components (silicon-controlled rectifiers, SCR), that normally block the passage of voltage in both directions by reason of their high Ohmic resistance. If, however, a certain trigger voltage is applied to the emitter, the thyristor promptly act: like a semi-conducting diode, i.e. it lets the current pass in one direction. Thus, thyristors serve as electronic switches in such applications as rectifiers, a.c. - d.c. converters, frequency changers and speed adjustment (control) of three-phase motors (asynchronous motor).

Auxiliary compactors are auxiliary circuit push button switches with power drive, with or without time-delayed switching, also used as nonmeasuring electrical switching relays and contact multipliers.

Limit switches monitor physical magnitudes or operating conditions, and open or close when preset limits of main or auxiliary circuits are exceeded and/or not attained.

Monitors are limit switches which close a circuit if an upper limit value is exceeded, and/or open the same circuit if there is a drop below a preset lower limit, or vice-versa (temperature, pressure, rotational speed, flow monitors).

Control current circuits lead to the electrical actuating elements (e.g. coils on switches) and to the reporting centres where the control equipment is arranged.

In this connection, the concept of "control" is often used; it concerns the adaptation of the operating condition of an item of electrical equipment to changing operational requirements.

Relays are components or accessories of switchgear, according to DIN VDE 0660 and 0670, which under the influence of an electrical action magnitude control further devices through auxiliary circuit connecting devices via auxiliary current circuits, if need be with a preset time delay.

The "electrical action magnitude" may be an electrical magnitude or its derivation in function of time, or it may be a sum, difference, a product or quotient of several electrical magnitudes.

Measuring relays monitor an electrical magnitude and electrically actuate further devices if a given value of this action magnitude is exceeded, or if its value drops below a preset limit.

Relays in accordance with DIN VDE 0435 consist of an electrically influenced exciter component (one or more coils) and a contacted device. The relay designation is specified according to the exciting action magnitude, e.g. current relays, power relays. The advantages of these relays are: galvanic separation between input circuit and output circuit (coil/contact side), and on the contacted side, separation between the various compactor units and the individual contacts.

The following kinds of relays are used: auxiliary relays without time delay, which are executed in contact-amplifying and contact-multiplying forms (e.g. auxiliary transistor relays in so-called intrinsically safe circuits), time-delay relays used as time-delayed auxiliary relays without indication (signal), time limit relays equipped with a graduated time dial, accelerating relays with two or more operating steps.

The designation of the relay contacts refers to the excited condition of the exciter coil (opening, closing).

Electric Drive

Elektrischer Antrieb
Commande électrique

see Drive

Electric Motor

Electromotor
Moteur électrique

see [Drive](#)

Electric Power

Elektrische Leistung
Ouissance électrique

see [Power](#)

Electrochemical Series

Elektrochemische Spannungsreihe
Série électrochimique des tensions

see [Corrosion](#)

Emission

Emission
Émission

see [Immission Protection Act](#)

Energy

Energie
Énergie

E. in the physical sense is the ability to perform work. In the technical sense, e., work and quantity of heat are three entities of the same nature, defined by the same SI unit (unit), viz. 1 Joule (1 J = 1 N m = 1 W s). It is usual to specify thermal e. in J, mechanical e. in N m and electrical e. in W s. The electron volt (eV) is the unit of e. used in nuclear physics.

Enthalpy

Enthalpie, Wärmeinhalt
Enthalpie

E. is a thermodynamic variable of state, also called "heat content at constant pressure", $dh = c_p \cdot dT$ with

dh differential variation of e.,
 c_p specific heat of the medium at constant pressure,
 dT differential variation of temperature.

The e. of water in the saturated state at 273 K is by definition zero.

Entropy

Entropie
Entropie

E. is a thermodynamic variable of state,

$$ds = \frac{dQ}{T}$$

with

ds differential variation of e.,
 T thermodynamic temperature,
 dQ differential reversible quantity of adduced heat.

The e. of water in the saturation state at 273 K is by definition zero.

Entry Cone

Einlaufkegel
Cône de guidage

A. flow deflecting device anchored to the floor of the intake chamber immediately upstream of the entry nozzle. In order to prevent rotation of the flow around the e.c. (inlet conditions), radial guide vanes are often integrally cast with the e.c. (See illustration under intake chamber, types I to IV.)

Entry Nozzle

Einlaufdüse
Tulipe d'aspiration

E.n., also called suction bellmouth, is a nozzle-shaped inlet casing component (fittings), particularly on vertical tubular casing pumps (Fig. 1). The acceleration of the flow through the e.n. evens out irregularities in the velocity distribution. This equalization is specially important to ensure a uniform approach flow (inlet conditions) in high specific speed pumps (specific speed). In the case of an inlet flow accompanied by swirl (vortex flow), it is advisable to provide a cruciform shaped flow straightener (Fig. 1) in the e.n.; this straightens out the flow to a certain extent. On the other hand, if a inlet-vortex regulation device (control) is arranged downstream, there is no need to fit a flow straightener in the e.n. (Fig. 2).

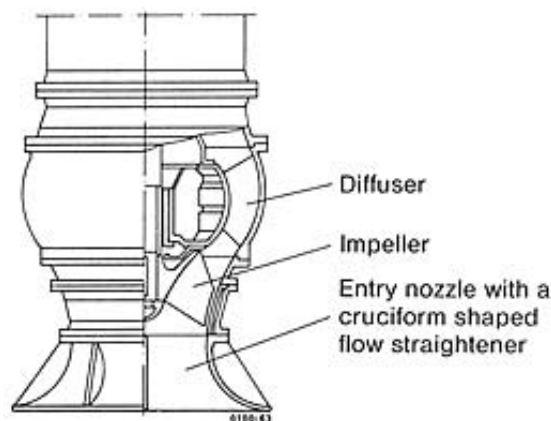


Fig. 1: High specific speed pump with entry nozzle and cruciform shaped flow straightener

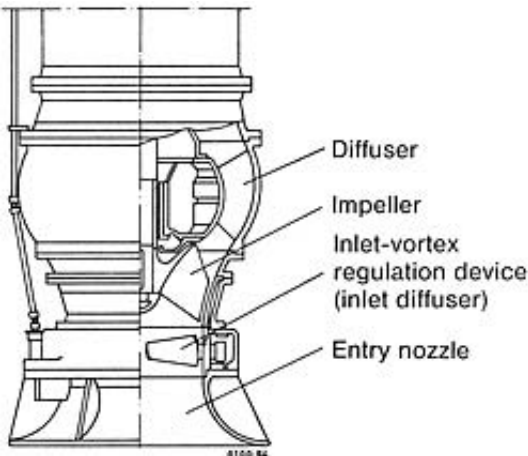


Fig. 2: High specific speed pump with entry nozzle and inlet-vortex regulation device

Entry Vortex

Einlaufwirbel
Tourbillon d'aspiration

see [Inlet Conditions](#)

Environmental Protection

Umweltschutz
Protection de l'environnement

E.p. is the collective concept for all measures and regulations designed to counteract the endangerment of the environment (immission protection act). E.p. is the subject of numerous laws and act: covering the various areas of environmental hazards. e.g.immission protection act; technical guidelines (abbreviation TA) for protection against noise; VDI guideline 2058 "Noise Accident Prevention Regulations" (UVV Lärm).

The technical guidelines for protection against noise (TA noise) contain regulations for the protection of the community or of the residents in the neighbourhood against noise radiated by plant and machinery subject to approval (authorization) in accordance with the immission protection act. According to these rules, the following immission guideline values for the noise pressure level (rating A) L_{pA} (noise in pumps and pumping installations, measuring technique) must not be exceeded:

a) Zones containing only industrial or trade plants and apartments of proprietors and directors of the plants, and of supervisory and emergency duty personnel:		70 dB
b) Zones containing mainly trade plants:	day-time	65 dB
	night-time	50 dB
c) Zones containing both trade plants and apartments, with no decisive preponderance of either trade plants or apartments:	day-time	60 dB
	night-time	45 dB
d) Zones containing a preponderance of residential apartments:	day-time	55 dB
	night-time	40 dB
e) Exclusively residential areas:	day-time	50 dB
	night-time	35 dB
f) Health resorts, spas, hospitals, nursing homes:	day-time	45 dB
	night-time	35 dB
g) Apartments within the plant itself:	day-time	40 dB
	night-time	30 dB

Night-time, defined for the purposes of e.p. comprises eight hours; starting at 22 hours and ending at six hours; in special circumstances, it can be displaced in time by up to one hour. VDI guideline 2058, Part 1 was elaborated for the purpose of evaluating work noise. It is in agreement with the TA noise regulations outlined above. Just as the TA noise regulations are intended to protect the neighbourhood, so is VDI guideline 2058, Part 2 intended to evaluate the danger to hearing (risk of causing deafness) in a noisy working environment. According to this guideline, a noise level of $L_{pA} = 90$ dB (guideline value) must not be exceeded. The VDI guideline 2058, Part 3 enables noise to be evaluated from the aspect of the demands it imposes on human beings at their place of work.

The "Noise" Accident Prevention Regulation is a legislative instrument for the protection of employed persons against the dangers of noise, binding both the employer and the insured persons. It was enacted by the Accident Insurance Underwriters (professional associations).

Equilibrium Condition

Gleichgewichtsbedingung
Condition d'équilibre

see [Boundary Layer](#)

Equipment Safety Act

Gerätesicherheitsgesetz, Maschinenschutzgesetz
Loi sur le matériel technique

E.s.a. is a short designation for the German law relating to technical machinery and working devices. This act stipulates that such devices shall only be put into service if they are constructed in a way that ensures safety of working and of operation, in other words they must conform with the generally accepted state of the art and satisfy the work protection and accident prevention regulations (UVV). It must be ensured that the user or operator is not exposed to any danger when using the working device in accordance with its appropriate function.

The e.s.a. have been in force since 1968, and do not contain any detailed provisions, but only define general objectives.

A number of prescriptions shows the way of achieving the goal of avoiding dangers caused by technical machinery and working devices: DIN 31000 contains general guidelines for the design of technical products from the safety aspect. Details of a number of very varied application fields are contained in technical regulations, amongst which we shall only cite the best known associations which have drawn up or published such regulations:

- Association of Professional Societies (VBG),
- German Standards Institute (DIN),
- Association of German Electrical Engineers (VDE),
- German Association of Gas and Water Engineering (DVGW),
- German Engineers' Association (VDI),
- Association of Technical Supervision Societies (VdTÜV),
- Working Community "Pressure Vessels" (AD).

The documents published by the above associations provide guidelines for the manufacture of machinery which will be safe from accidents. These guidelines relate to

- a) the functional design, such as mechanical strength, selection of materials, functional safety, electrical equipment, control etc.,
- b) design of the outer contours, such as surfaces, corners, stability, anti-slip protection etc.,
- c) protection against dangers which can arise in operation, such as noise, vibrations, heat, cold, dust, vapours, gases etc.,
- d) safety during transport.

Erosion

Erosion
Érosion

An attack on the materials in the pump by the fluid (one or more phases) is called e. in particular if the causes of the attack are not known. These possible causes are corrosion wear, or cavitation. Phrases to describe the attack on pump materials are:

- corrosive e. (corrosion),
 - erosive wear (abrasion, wear), or
 - cavitation-e. (cavitation, wear).
-

Error of Measurement

Meßfehler
Erreur de mesure

According to DIN standard 1319, an e.o.m. is the following difference:

indicated value minus true value.

The indicated value, i.e. the measurement result, is falsified by the imperfection of the object measured, the imperfection of the measuring instruments and of the measuring processes, by ambient influences and influences of the observers, and by alterations over a period of time of all these sources of error (see DIN 1319, Sheet 3).

In the case of acceptance tests on pumps, the e.o.m.'s are taken into account by the permissible uncertainties of measurement.

EULER Equation

EULER-Gleichung
Équation d'EULER

see Fluid Dynamics

Evacuation

Evakuieren
Évacuation

see Venting

Explosion Protection

Explosionsschutz
Protection antidéflagrante

Explosion is the self-propagation of flame in mixtures of combustible gases or vapours and air. Within certain concentration limits, ignition occurs with a minimum expenditure of energy.


Energy sources capable of triggering ignition include:

- electric sparks and arcs, which arise e.g. when opening or closing electric circuits,
- hot current carrying conductors (e.g. motor windings),
- mechanical sparks (e.g. if a rotating fan impeller fouls stationary parts).

Ignition sources can also be created by short-circuits, ruptures of cables, discharges of electrostatically charged plant components, lightning etc. According to government regulations (Elex V from 1.7.80), electrical equipment is only allowed to be used in areas threatened by possible explosion if a test certificate has been issued. For use in explosion zones 0, 10 and G, this applies only if the certificate explicitly authorizes the application in the zone in question.

Manufacturers of electrical equipment are therefore pressured to supply provisions for e.p. E.p. is not to be confused with types of protection for contact, foreign objects, and water spray.

The regulations for e.p. are found in: Electrical Equipment used in Areas of Possible Explosion - General Provisions in DIN EN 50014 with other complementary European standards. Also DIN 57 165, from 1.9.83, (mainly addressed to the user) which contains safety information concerning the above-mentioned Elex V and the regulations about inflammable liquids (VbF).

The general abbreviation for e.p. is "EEX" with the symbol being  with additional letters meaning:

- e for additional safety features,
- d pressure proof,
- p high pressure proof,
- i for self safety

and the following symbols denoting groups:

- I for firedamp endangered areas,
 - IIA to IIC for explosion endangered areas.
-

F

Faeces Pump

Fäkalienpumpe

Pompe d'évacuation de matières fécales

F.p., a pump designed to handle excrements, untreated sewage (crude sewage) usually in communal clarification plants, but also in large buildings and private dwellings where the local sewer cannot be directly connected to the main sewerage system because of lack of a sufficient natural gradient. The f.p. operates in a so-called faeces lifting plant. The f.p. must meet similar requirements as sewage pumps.

The f.p. must never clog, even if large amounts of solids are present in the sewage, and this is attained by fitting special impellers (single vane impeller) (Figs. 1 to 4). The axial thrust is balanced by back shroud blade. Impeller and casing wear rings with relatively close clearances (slit seal) and balance holes are unsuitable in the case of f.p.'s. Balance holes would soon become clogged and worn, because the sewage often contains very abrasive matter. Because of high wear, (wear) f.p.'s are in principle equipped with renewable wear plates, which can also be made of special wear-resistant material. depending on the amount of solids (hydrotransport) entrained.

The same arguments apply to the shaft protecting sleeve of horizontal pumps, which are usually sealed by soft-packed stuffing boxes fed with sealing water (shaft seals). F.p.'s are usually non-self-priming (self-priming pump).

F.p.'s are also built in the form of vertical submersible pumps (Fig. 2), submersible motor pumps (Fig. 3) and close-coupled pumping sets for the faeces lifting plants (Fig. 4).

If the fluid pumped contains large amounts of gas in solution, f.p.'s are sometimes fitted with torque-flow impellers (torque-flow pump) (Fig. 5).

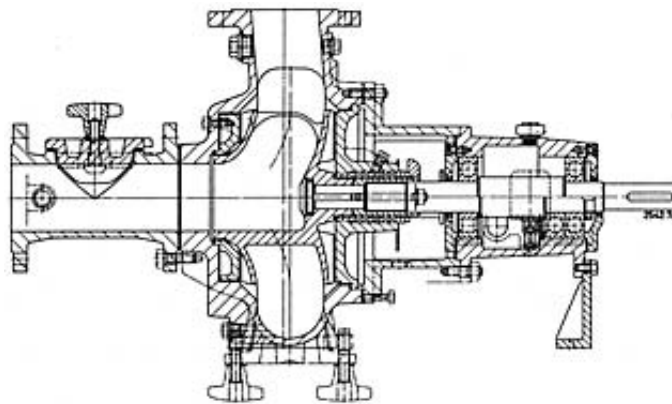


Fig. 1: Faeces pump with single vane impeller

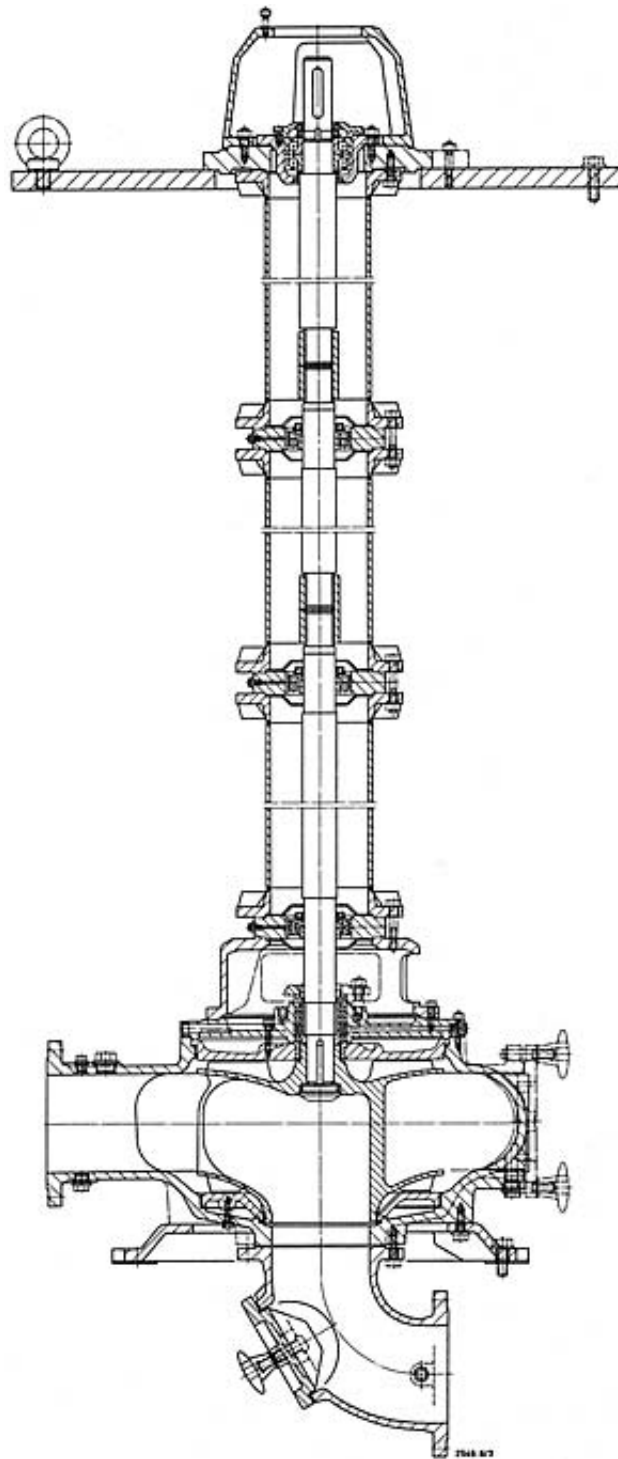


Fig. 2: Faeces pump of submersible pump type

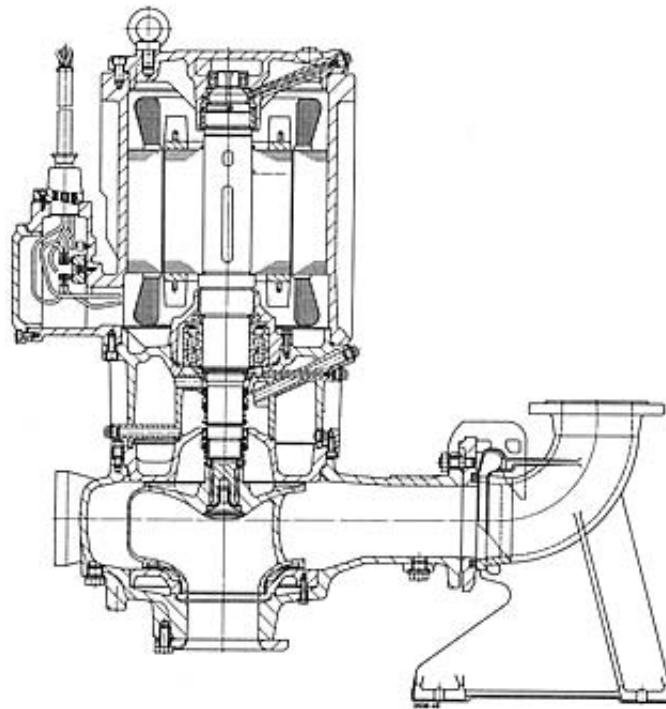


Fig. 3: Faeces pump of submersible motor pump type

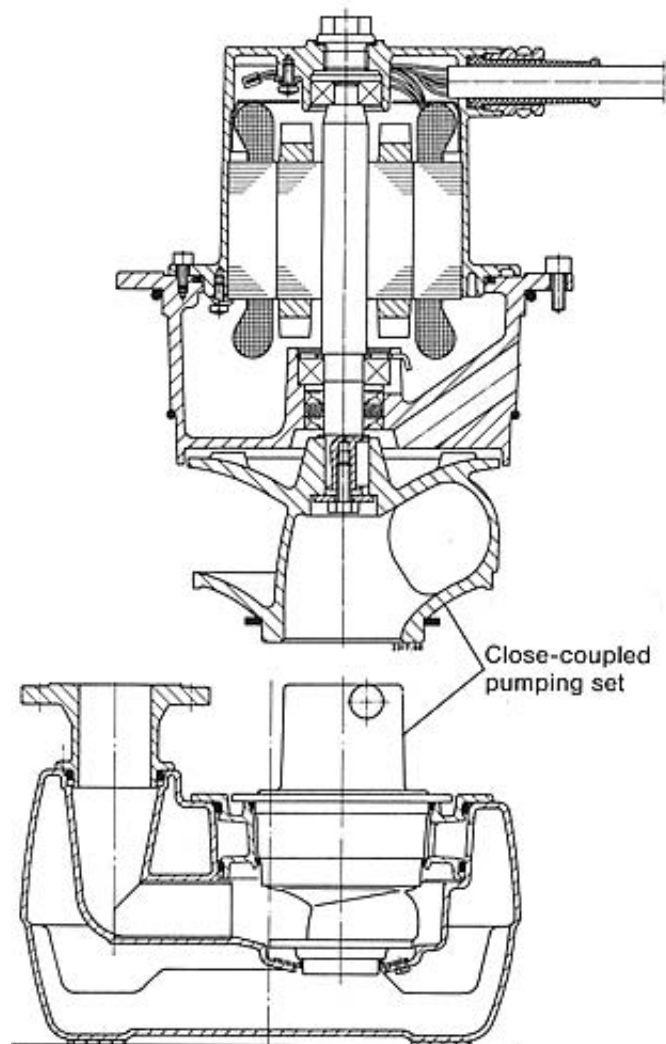


Fig. 4: Faeces pump in close-coupled design in a faeces lifting plant

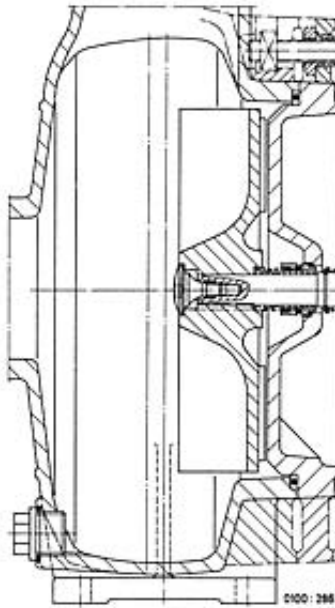


Fig. 5: Torque-flow impeller of a faeces and sewage pump

Feed Pump

Speisepumpe

Pompe Alimentaire

see [Boiler Feed Pump](#)

Fibrous Material

Faserstoff

Pâte épaisse

F.m., also called pulp ([pulp pump](#)) is the collective term used to describe cellulose and wood pulp ([pulp pumping](#)).

Filling-Up of Centrifugal Pumps

Auffüllen von Kreiselpumpen

Remplissage de pompes centrifuges

The suction pipe ([pumping plant](#)) of the non [self-priming pump](#), and the centrifugal pump itself must be primed with fluid before start-up ([starting process](#)). If the [impeller](#) or the [suction impeller](#) is not flooded when the pump is at rest, it is advisable to install a [foot valve](#) or a check valve ([valves and fittings](#)) beneath the lowest water level in the suction pipe. In this type of installation, which is limited to small [nominal diameter](#), the foot valve, the [pump casing](#), sundry [valves and fittings](#) and the [piping](#) must all be sized with an adequate safety margin against the highest static pressure loading which can arise. The reason for this is that the [foot valve](#) or the check valve may in certain circumstances suddenly snap shut when a reflux of fluid occurs when the pump is switched off and a pressure surge ([surge pressure](#)) will occur. In order to protect the pump, a further check valve is therefore often installed in the discharge line ([pumping plant](#)). Such an arrangement will however only operate satisfactorily if the check valve on the discharge side shuts before the valve on the suction side.

Non self-priming pumps which are not equipped with a foot valve or check valve in the suction pipe are primed before start-up by venting the suction pipe and the pump, smaller size pumps with foot valves are sometimes primed by means of a priming tundish with valve. Borehole shaft driven pumps are equipped with a foot valve despite their wet installation, if the shaft is guided in water-lubricated rubber bearings (plain bearing), and if some of these bearings are situated above the normal well water level of pumps installed at great depths. Such bearings require water lubrication when the pump is started up, and the foot valve ensures that all these bearings remain flooded at all times.

In the case of self-priming pumps, the self-priming device must be filled with the medium pumped before start-up.

Filter

Filter

Filtre

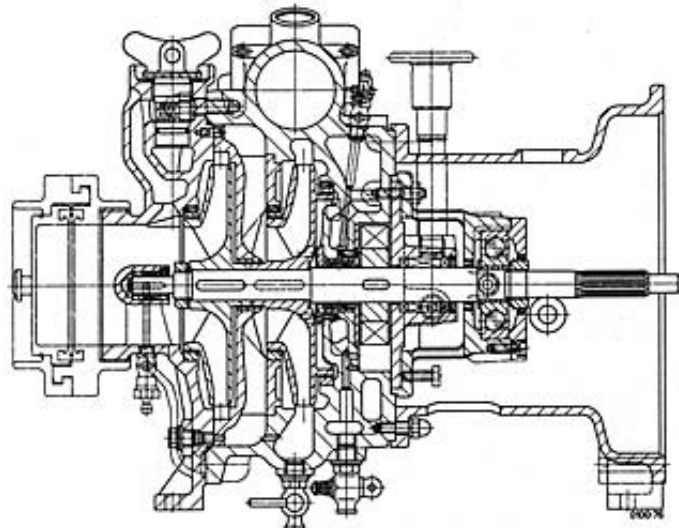
see Valves and Fittings

Fire-Fighting Pump

Feuerlöschpumpe

Pompe d'incendie

F.f.p. centrifugal pump used for pumping water for fire-fighting. Single and multistage, single suction centrifugal pumps are used for this application. Apart from stationary installations, e.g. hydrant plants in tall buildings and sprinkler plants (sprinkler pump), f.f.p.'s are of the self-priming type (self-priming pump) and are constructed in compact lightweight form. The illustration shows a typical f.f.p. for mounting on a portable fire-fighting appliance, with drive by an internal-combustion engine. On this pump, a water ring pump has been mounted on the common pump shaft within the common casing to act as self-priming stage; it rotates with the main pump. The size and performance data of f.f.p.'s are standardized in accordance with DIN 14 420, the associated suction and delivery hoses in accordance with DIN 14 810 and the couplings in accordance with DIN 14 811. The pressure losses in the hoses approximate the pipe friction losses in plastic pipes (pressure loss).



Fire-fighting pump for mounting on a portable firefighting appliance. Venting by means of a water ring pump

The drive is usually by an internal-combustion engine. This provides a simple speed control (rotational speed) via the fuel feed to the engine, according to the required capacities and heads, which vary widely in practice. Mobile f.f.p.'s are incorporated in fire engines (built-in or built-on) and in portable fire-fighting appliances. In the latter case, the pump is flanged direct onto the internal-combustion engine, and the unit is built into a carrying frame or cradle. Some of the more recent portable fire-fighting appliances (large appliances outside the standard) are equipped with gas turbine drives (improved performance combined with lower weight, wider choice of fuels). In such cases the carrying frame is often split into two parts, to make carrying easier, and the pump can be joined to or separated from its driver in a few simple manipulations. The drive of fire-engine mounted pumps is the main engine of the

vehicle itself, via an auxiliary shaft.

The following alternatives are used for venting (self-priming pump): water ring pump on a common shaft with the main pump, and constantly rotating with it (the self-priming phase is interrupted by means of a changeover cock valves and fittings); oil ring pump arranged next to the f.f.p. and driven off the latter via a friction wheel during the venting stage, ejector (deep well suction device), using a partial flow of exhaust gas from the internal-combustion engine or of compressed air from the compressor of the gas turbine (the venting process is interrupted by switching over a flap valve. valves and fittings); manually-operated air pump.

See under water jet for information on water pressure at the fire hose nozzle, jet rise heights and horizontal reach ranges.

Fittings

Formstücke

Raccords de tuyauterie

F. in a centrifugal pumping plant comprise any piping components which are used to effect a change of direction or to install a branch in the pipeline, and/or a transition from a given pipe cross-sectional area to a larger or smaller one. F. should be shaped to offer the least possible resistance to flow in order to minimize the pressure losses (head of plant), bearing in mind however the possible increase in manufacturing costs (economics). Figs. 1 to 6 illustrate commonly used f. in centrifugal pump technology.

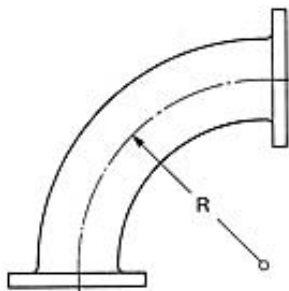


Fig. 1: 90° pipe bend

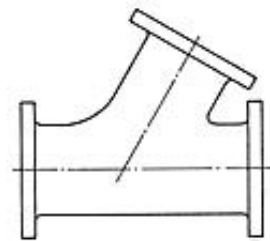


Fig. 2: Oblique T-piece

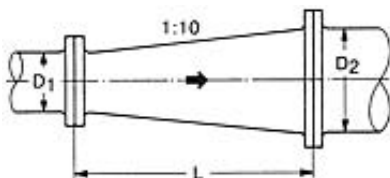


Fig. 3: Transition piece fitted as diffuser

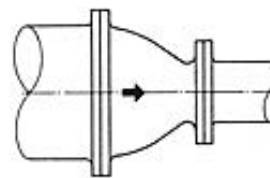


Fig. 4: Transition piece fitted as reducer (nozzle-shaped)



Fig. 5: Transition piece (reducer) fitted belly down to avoid air pockets

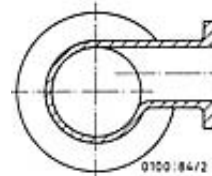


Fig. 6: Branch shaped to avoid air pockets

Fig. 1 illustrates a 90° pipe bend with radius R (centreline radius). Such pipe bends should have a radius of curvature $R > 2D + 100 \text{ mm}$ with D pipe diameter in mm, particularly if they are fitted immediately upstream of a pump suction branch. Pipe bends fabricated of cylindrical segments welded together should consist of at least six segments for a 90° bend.

Fig. 2 shows an oblique T-piece, which is preferable to a T-piece for reasons of reduced flow resistance.

Fig. 3 illustrates a diffuser type transition piece (diffuser). Its face to face length L should amount to $L = 5(D_2 - D_1)$ approx., (D = pipe diameter) when fitted as a divergent transition piece in relation to the direction of flow. The outlet diffusers discharging to atmosphere (e.g. in the case of land drainage installations land reclamation pump) should be sized for a discharge velocity (flow velocity)

$$v = 1.0 \text{ to } 1.5 \frac{\text{m}}{\text{s}}.$$

In contrast to a diffuser, the face to face length of a f. used as reducer can be much shorter. A nozzle-shaped reducer with favourable flow characteristics is illustrated in Fig. 4.

As illustrated in Figs. 5 and 6, eccentric reducers and branches should be installed in horizontal suction pipes to avoid the formation of air pockets.

For further f. see entry nozzle, intake chamber, intake elbow, and inlet conditions.

Flange Construction

Flanschausführung
Construction de bride

The geometric, functional and material construction of flanges, e.g. flanges on the casing of a centrifugal pump are always manufactured to a flange standard, in so far as such a standard exists for the operating conditions (pressure and temperature) to which the pump is subjected. The absence of such a standard is very much an exceptional case (e.g. for pressures above 400 bar).

Flanges are annular plates usually fitted on pipe ends, shaft ends, cylindrical bodies and split casings, designed to ensure a leak-tight separable connection (in pairs) between these components, by means of bolts. The main application of flanges is piping, where they serve to connect two lengths of piping together, or to connect piping components to the casing of an appliance, a valve or a machine (e.g. a pump). Flanges are usually cast on or welded on to the pipes or pipe nozzles of the casings.

A *flange standard* lays down the dimensions, surface finish, material, shape of sealing face, marking and technical conditions of supply of flanges for piping. Common national and international standards which apply to the suction and discharge branch flanges of centrifugal pumps include the following:

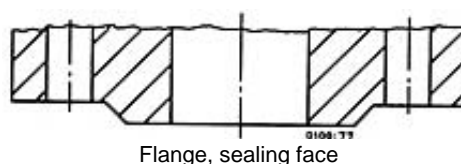
German standards:

- DIN 2532 to 2535 (cast iron flanges for pressure ratings PN 10, 16, 25 and 40);
- DIN 2543 to 2551 (cast steel flanges for pressure ratings PN 16, 25, 40, 64, 100, 160, 250, 320 and 400);
- DIN 2632 to 2638 (weldneck flanges for pressure ratings PN 10, 16, 25, 40, 64, 100 and 160);
- DIN 2628 (weldneck flanges for pressure rating PN 250);
- DIN 2629 (weldneck flanges for pressure rating PN 320);
- DIN 2627 (weldneck flanges for pressure rating PN 400).

The standards mentioned specify a raised face as sealing face for non-positive-locking gaskets (see illustration). For positive-locking gaskets there is a chaise of flanges with groove and tongue in accordance with DIN 2512, spigot and recess in accordance with DIN 2513 or 2514, for welded diaphragm gaskets in accordance with DIN 2695 with chamfer, for lenticular gaskets in accordance with DIN 2696 with machined groove.

American standards:

- ANSI B 16.1 (cast iron flanges of pressure classes 25, 125, 250 and 800);
- ANSI B 16.5 (cast steel and weldneck flanges of pressure classes 150, 300, 400, 600, 900, 1500 and 2500).



In the German flange standards, the numerical value of the nominal pressure in bar is equal to Flange, sealing face the max. applicable pressure at a reference temperature of 20°C. In contrast, in the American flange standards, the numerical value of the "class" (e.g. 400) of cast iron flanges is equal to the max. applicable pressure in lbs/in² at temperatures between 65 and 200°C, depending on the class; in the case of steel and cast steel flanges, it is equal

to the numerical value of the max. applicable pressure at temperatures between 350 and 650°C) depending on the material. At a temperature of 20°C, the numerical value of the max. applicable pressure, especially in the case of steel and cast steel flanges, lies appreciably higher than the numerical value which designates the class (the concept of pressure rating or nominal pressure is not generally used in American flange standards).

British standard:

- BS 4504 Part 1 (cast iron flanges for pressure ratings PN 2.5, 6, 10, 16 and 25; cast steel flanges for pressure ratings PN 16, 25, 40, 64, 100, 160, 250, 320 and 400; weldneck flanges for pressure ratings PN 2.5, 6, 10, 25, 40, 64, 100, 160, 250, 320 and 400).
-

Flat Belt Drive

Flachriementrieb
Transmission par courroie plate

see Belt Drive

Fat Gasket

Flachdichtung
Joint plat

see Seals

Float Valve

Schwimmerventil
Robinet à flotteur

see Vent Valve

Flood Drainage Pump

Hochwasserpumpe
Pompe d'évacuation des eaux d'inondation

see Drainage Pump, Land Reclamation Pump

Flow Coefficient

Durchflußzahl
Coefficient de débit

see Standard Nozzle, Standard Orifice, Standard Venturi Nozzle

Flow Controller

Flow controller
Contrôleur de débit

F.c., or flow monitor, is a device used on pumps for monitoring the cooling or lubricating fluid flows of plain bearings lubricated by the fluid pumped, or of mechanical seals (shaft seals), and very often on blandness chemical pumps. Flow monitors trigger a switching operation (switching on an alarm device or switching off the pump), if a preset volume flow is either exceeded or not attained, depending on the presetting.

The most widely used f.c.'s are based on one of the following principles:

- float principle,
- measurement of the impact pressure,
- measurement of the pressure differential across a nozzle (standard nozzle) or an orifice plate (standard orifice).

Float principle

A float is situated in a vertical tube with a divergent section in the upward direction, the direction of flow being from bottom to top. The float rises as the flow rate increases and reaching a certain height it triggers the switching sequence via a solenoid (Fig. 1) or via an inductive transmitter.

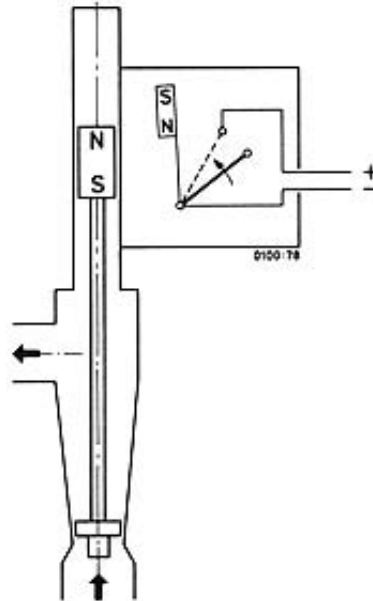


Fig. 1: Float-type flow controller

Impingement plate principle

The force exerted by the flow on an impingement plate arranged in a pipe is compensated by a spring. If the system is thrown out of equilibrium as a result of a change in the rate of flow, the switching sequence is triggered (Fig. 2).

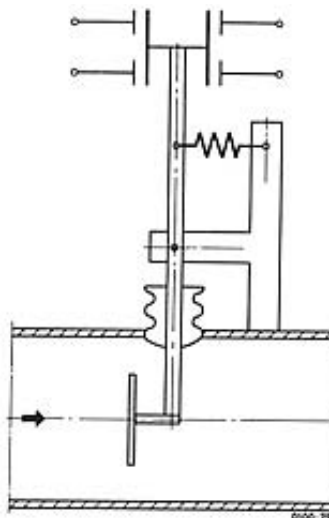


Fig. 2: Impingement plate type flow controller

Differential pressure principle

The pressure difference created by a nozzle or orifice plate incorporated in a pipe is converted into a force in a differential pressure measuring device (Barton cell); this force is compensated by springs. A change in the rate of flow throws the system out of equilibrium and triggers the switching sequence (Fig. 3).

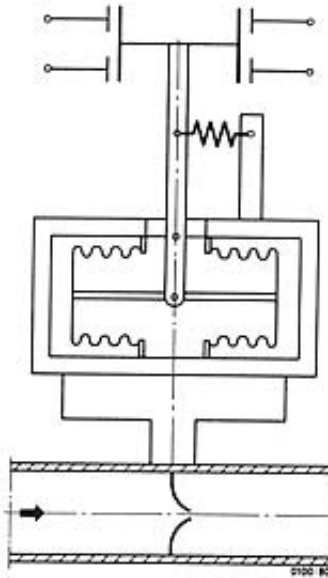


Fig. 3: Differential pressure type flow controller

Flow Line

Stromlinie

Ligne de courant

In a flow, the f.l. is a line, the local direction of which corresponds to the direction of the local flow velocity (absolute velocity, relative velocity). In a steady flow, in which a velocity independent of time exists at every point within the flow space, the f.l. describes the path of a liquid particle under consideration (flow path). F.l.'s exhibit no breaks and can never intersect one another, as otherwise there would be two different flow velocities at one and the same point (see Fig. 2 under cascade). As no normal components of the velocity can exist at a solid wall (e.g. the blade or the pump casing) it follows that all the body contours of the flow space are simultaneously f.l.'s. The BERNOLLI equation used in fluid dynamics is only valid in the strict sense for various points on a common f.l. The equation is however often used also for a flow space designated "thread of streams" or "filament of flown" which is encompassed by all the f.l.'s passing through a closed line. In the case of flow in pipes, the entire liquid content of the pipe is usually considered as a filament of flow (flow velocity). The circular projection of a f.l. into the meridian intersecting plane (section through the axis of rotation) of an impeller, for example, is referred to as the flux line.

Flow Measurement

Durchflußmessung

Mesure d'écoulement

see Measuring Technique

Flow Profile

Strömungsprofil

Profil d'écoulement

The f.p., also named blade profil (or blade form) results from the intersection of the flow surface (flux line) with the blade, e.g. of an impeller, a diffuser or a hydraulically ineffective support blade supporting a pump bearing. The F.p. is formed by adding the blade thickness equally on either side along the length of the median line (illustration under cascade), according to a prescribed distribution (e.g. NACA profiles). The progression pattern of the profile thickness can be obtained from a table of profiles, or it can be calculated from analytical functions (EDP installation). The calculation of the f.p. from analytical functions plays a specially important part in conjunction with the numerical control (NC) techniques used today. The blade profile of an impeller will depend to a large extent on the type (specific speed) of centrifugal pump involved. Radial impellers and most diffusers almost always have long thin blades which do not exhibit any distinctive profile. The thickness of these blades (with the exception of propeller blades) is governed largely by strength calculation considerations and also by manufacturing method considerations (casting, milling, welding, forging, plastic injection).

In the case of propeller blades (blade), f.p.'s with a prescribed thickness distribution and camber predominate (aerofoil theory). Slim profile shapes with the maximum thickness far behind the leading edge (so-called laminar profiles) possess favourable characteristics as regards the NPSH (net positive suction head) and the hydraulic efficiency at the design duty point of the pump. Thick profiles are less sensitive to approach flow under shock (shock loss, operating point).

F.p.'s are calculated more and more frequently today with the aid of finite element methods and singularity methods (imposition of sinks, sources and vortices) (aerofoil theory).

Flow Resistance

Durchflußwiderstand
Résistance à la circulation

see Pressure Loss

Flow Reversal

Strömungsumkehr
Inversé d'écoulement

see Surge Pressure

Flow Separation

Ablösung
Décollement

see Boundary Layer

Flow Separation Limit

Abreißgrenze
Limite de décollement de courant

see Operating Behaviour of Centrifugal Pumps

Flow Straightener

Gleichrichter
Redresseur

see Inlet Conditions

Flow-Through

Durchfluß

Débit dans une section

The f.t. can refer to a mass throughput, in which case it is called mass f.t., or to a volume throughput, in which case it is called volume f.t. or capacity.

Flow Velocity

Durchflußgeschwindigkeit Strömungsgeschwindigkeit

Vitesse de passage, vitesse d'écoulement

The f.v. symbol v , is the mean velocity in a given cross-section across which the medium flows, e.g. a pipe cross-section.

$$v = \frac{Q}{A}$$

where

Q volume flow (capacity) at the cross-sectional area A under consideration.

The SI unit of f.v. is m/s. See underlining for f.v. in pipes (guideline values for f.v.'s occurring in practice). Pressure losses are related to $v^2/2g$, where v is the f.v. in a characteristic flow section, usually the connection section through which the flow passes upstream or downstream of the component, and g gravitational constant.

Fluid

Fluid

Fluide

The word f. comes from the Latin "fluere" meaning to flow. F. is the generic term denoting gases, vapours, and liquids that follow the rheological equations above the yield point (DIN 1342). The laws of fluid dynamics are applicable to f.'s.

Fluid Coupling

Strömungskupplung

Coupleur hydraulique

The f.c. (see Fig. 1) consists of a pump impeller (on the drive shaft) and of a turbine wheel (on the driven shaft). Both impellers are surrounded by a common casing. The pump impeller delivers the fluid (usually a low viscosity oil) trapped in the casing to the turbine wheel, which rotates the driven shaft. In contrast to the hydraulic torque converter there are no guide vanes arranged between the pump (subscript P) and the turbine (subscript T) of the f.c. Because of the absence of a diffuser supported on the stationary casing, the input torque T_P of the f.c. is the same as the output torque T_T (torque):

$$T_P = T_T = T$$

With the powers $P_P = T \cdot \omega_P$ and $P_T = T \cdot \omega_T$, the efficiency of the f.c. is

$$\eta = \frac{P_T}{P_P} = \frac{n_T}{n_P} = v$$

with

v rotational speed ratio of turbine speed to pump speed.

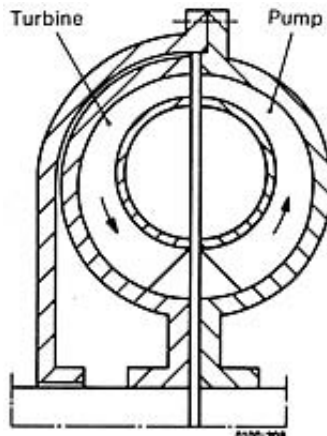


Fig. 1: Diagram of fluid coupling

Fig. 2 illustrates the characteristic curves of a f.c. for various pump rotational speeds n_P . The f.c. has a very high driving torque at the turbine speed $n_T = 0$, and for $n_T = n_P$ the torque $T = 0$. When power is transmitted, there is always a certain amount of slip, i.e. $n_T < n_P$. The turbine speed can be influenced by changing the volume of the oil fill V , e.g. with the aid of an adjustable fill tube, i.e. by altering the slip $1 - v$ (Fig. 3). The characteristic curves can be adapted to a considerable extent to the requirements of the driving and driven machines by various design modifications (Figs. 4 and 5). In combination with a gear box (gear drive for pumps), the f.c. is also referred to as a variable-speed coupling.

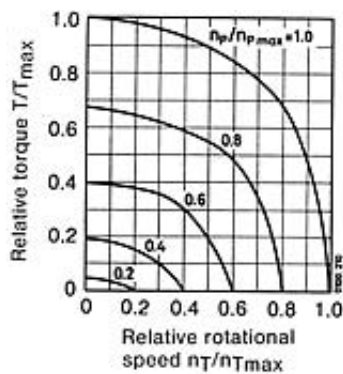


Fig. 2: Characteristic curves at different pump speeds

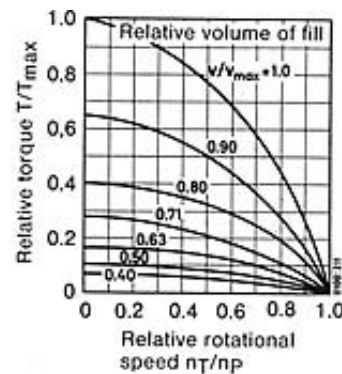


Fig. 3: Characteristic curves at different fill volumes

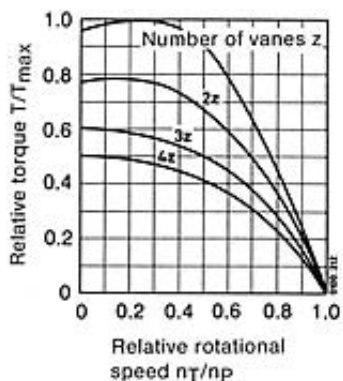


Fig. 4: Characteristic curves for different numbers of blades z

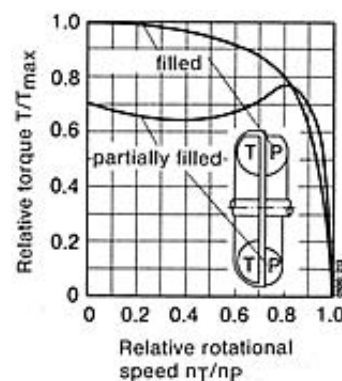


Fig. 5: Characteristic curves of fluid couplings with 'flattening' at the outer diameter and asymmetrical pump and turbine impeller shapes

The mechanical separation of the driving and the driven shafts damps down torque shocks and vibrations. A disadvantage is the reduction in efficiency, which can be quite considerable under certain conditions (heating up of the oil in the f.c.). That drawback can be alleviated by combining a f.c. with a hydraulic torque converter: in the lower speed and power range, the f.c. does the driving, in contrast to the upper-speed-range situation (80 to 100%), where the drive and output shafts are coupled together, thus transmitting most of the power with no losses due to slip etc., while a hydraulic torque converter enhances both the speed and the power (e.g. of boiler feed pumps) via the power-splitting capacity of a planetary gear.

Fluid Dynamics

Strömungslehre
Dynamique des fluides

F.d. is the science of movements of liquid and gaseous media (fluid). The scope of f.d. covers *hydro-* and partially *aerodynamics* of incompressible flows (negligible density alterations) and *gasdynamics* in which changes in density of considerable magnitude have to be taken into account. F.d. attempts on the one hand, to set up and solve equations describing flow phenomena on a purely theoretical basis and on the other hand to present in a suitable form and interpret empirically found relationships pertaining to select fluid flow problems.

Depending on the number of coordinates required, one can distinguish between one, *two* and *three-dimensional* flow phenomena, which can be either steady (steady flow), i.e. independent of time, or unsteady (unsteady flow), i.e. dependent on time. The theory described as "*filament of flow*" or "*thread of stream*" theory of one-dimensional flows deals with flows in a filament of flow limited on all sides by flow lines, and particularly with flows in pipes and closed ducts.

An important fundamental equation of f.d., which is valid in all cases, is the *continuity equation* which can be expressed as

$$\operatorname{div} \vec{v} = 0$$

in differential form for steady incompressible flows. or, in integral form: the product of flow cross-section A and flow velocity v (averaged over the cross-section) along the filament of flow remains constant:

$$A \cdot v = \text{constant.}$$

Furthermore, according to the *BERNOULLI equation*, the sum of the static pressure p, dynamic pressure ($\frac{\rho}{2} \cdot v^2$) and elevation term $\rho \cdot g \cdot z$ along a filament of flow in a frictionless flow is constant:

$$p + \frac{\rho}{2} v^2 + \rho \cdot g \cdot z = \text{constant}$$

with

ρ density of pumped medium,
g gravitational constant,
z geodetic altitude.

The extension of this formula to flows subjected to friction between two arbitrarily selected cross-sections A_x and A_y of a pumping plant requires the pressure losses and pressure changes caused by a pump or turbine to be taken into account:

$$\frac{p_x}{\rho \cdot g} + \frac{v_x^2}{2g} + z_x = \frac{p_y}{\rho \cdot g} + \frac{v_y^2}{2g} + z_y + H_{v,x,y} - H$$

with

g gravitational constant,

H head of a pump ($H > 0$) or head drop of a turbine ($H < 0$) between A_x and A_y , $H_{v,x,y}$ head loss (pressure loss) along the flow path from A_x to A_y , which is generally calculated according to

$$H_{v,x,y} = \zeta \frac{v_y^2}{2g}$$

with ζ loss coefficient.

The real two- and three-dimensional flows can generally be handled by the methods of potential flow, if the flow travels outside the boundary layer (at a sufficiently great distance from stationary walls). and can therefore be considered as quasi-frictionless. On the other hand, it is possible in only a few simple cases to obtain solutions valid for incompressible flow of viscous fluid from equations of motion (NAVIERSTOKES equations), which for homogenous flow result in the following:

$$\rho \frac{d\vec{v}}{dt} - \vec{F} + \text{grad } p - \eta \cdot \Delta \vec{v} = 0$$

with

\vec{v} vectorial flow velocity (absolute velocity),
 t time,
 \vec{F} vectorial field force (in the earth's gravitational field, weight force/volume),
 η dynamic viscosity,
 Δ delta operator for $\partial^2/\partial x^2 + \partial^2/\partial y^2 + \partial^2/\partial z^2$ in the x, y, z system of coordinates,
 grad p vector with x component $\partial p/\partial x$ and corresponding y and z components.

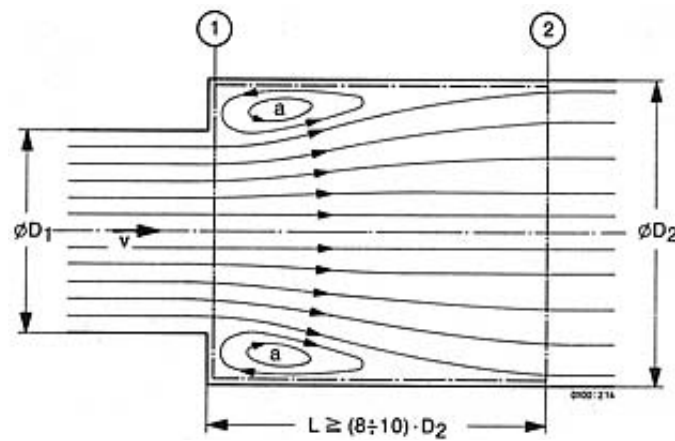
For sufficiently high REYNOLDS numbers (model laws) the general motion equations for the thin boundary layers present in proximity to stationary walls can however be greatly simplified and solved by the various methods of the boundary layer theory.

From the microstructure of the flow a distinction is made between laminar and turbulent flow. For the case of an incompressible frictionless flow, the general motion equation is simplified into the so-called EULER equation:

$$\rho \frac{d\vec{v}}{dt} - \vec{F} + \text{grad } p = 0$$

from which the BERNOULLI equation is derived for steady flows.

In the f.d. as applied to centrifugal pump technology (incompressible steady flow) the so-called *momentum theorem* plays an important practical role; this theorem states that all the forces acting from the outside on a liquid contained in a closed control space must be in equilibrium. The momentum theorem represents the integral form of the NAVIER-STOKES equation. As an example of application, we can consider the so-called CARNOT shock loss (impact loss) in an incompressible flow through a diffuser with an abrupt increase of cross-section (see illustration).



CARNOT shock diffuser (application of momentum theorem)

The diffuser has circular cross-sections; at the abrupt transition from the section $A_1 = \pi \cdot D_1^2 / 4$ to the larger cross-section $A_2 = \pi \cdot D_2^2 / 4$ a ring-shaped flow separation a (wake or dead water) is formed as a result of the inertia of the flowing mass. The flow only regains contact with the outer wall after a travel path L equal to at least 8 to 10 times the diameter D_2 . The control space is circumscribed in the illustration by the chain-dotted lines. The momentum theorem will now be applied to the outside forces acting in the direction of the flow velocity v on the fluid enclosed in the control space. The momentum forces F_J

$$F_J = \dot{m} \cdot v$$

with

\dot{m} mass flow

are constantly directed on the fluid enclosed in the control space. Other forces which are involved are: the pressure forces F_p

$$F_p = p \cdot A_p$$

with

p static pressure,

A_p surface of cross-section on which p acts,

and the friction forces F_v

$$F_v = \tau \cdot A_v$$

with

τ wall shear stress (wall friction),

A_v internal pipe wall surface on which τ acts.

Because of the wake (dead water) flow (boundary layer) at the pipe wall, F_v is negligible.

Therefore, according to the momentum theorem, the equilibrium of forces in the direction of the flow velocity v (positive direction) can be expressed as:

$$p_1 \cdot A_2 + \dot{m}_1 \cdot v_1 - \dot{m}_2 \cdot v_2 - p_2 \cdot A_2 = 0$$

with

$$A_2 = \pi \cdot D_2^2 / 4 \text{ and}$$

$$\dot{m}_1 = \dot{m}_2 = \rho \cdot v_2 \cdot A_2.$$

Therefore, according to the momentum theorem, the difference of static pressures becomes:

$$p_2 - p_1 = \rho \cdot v_2 \cdot (v_1 - v_2).$$

According to the BERNOLLI equation we obtain for a thread of stream, taking into account the pressure loss (head loss) $H_{v,1,2}$:

$$H_{v,1,2} = \frac{v_1^2 - v_2^2}{2g} + z_1 - z_2 - \frac{p_2 - p_1}{\rho \cdot g}.$$

If we now insert $p_2 - p_1$ as obtained from the momentum theorem. we obtain, with $z_1 = z_2$:

$$H_{v,1,2} = \frac{(v_1 + v_2) \cdot (v_1 - v_2)}{2g} - \frac{2 v_2 (v_1 - v_2)}{2g} = \frac{(v_1 - v_2)^2}{2g}$$

Because of the similarity between the above equation obtained from the momentum theorem with the equation for the loss of energy in a straight-line inelastic *impact* of two bodies, the pressure head loss obtained from the last-mentioned equation has been named CARNOT shock loss (or impact loss) in f.d.

A further important application of the momentum theorem leads to the so-called *fundamental equation of fluid flow machines*. If we subdivide the beading (blade) of an impeller into impeller elements, where one element (subscript E1) lies between two adjoining flow areas (flow line) of the relative flow of the impeller (relative velocity), the specific energy $Y = g \cdot H$ of the impeller element becomes

$$Y_{E1} = g \cdot H_{E1} = u_2 \cdot v_{2u} - u_1 \cdot v_{1u} = \sum_1^2 (u \cdot v_u)$$

with

u_1 circumferential velocity (velocity triangle) of impeller at inlet, related to the characteristic flow line of the impeller element,

u_2 circumferential velocity of impeller at outlet, related to the same flow line,

v_{1u} circumferential component of absolute velocity v at impeller inlet, related to the flow line pertaining to u_1 and u_2 ,

v_{2u} circumferential component of absolute velocity v at impeller outlet, related to the same flow line.

The fundamental equation of fluid flow machines has general validity, i.e. it is equally valid for any random impeller shape, and for centrifugal pumps and turbines. The equation is independent of the density of the pumped medium and can also be applied to cases where the density changes during its passage through the impeller e.g. to gaseous media and vapours.

With the aid of the cosine theorem, the fundamental equation for centrifugal pumps can be converted as follows:

$$g \cdot H_{E1} = \frac{w_1^2 - w_2^2}{2} + \frac{u_2^2 - u_1^2}{2} + \frac{v_2^2 - v_1^2}{2}$$

with

w relative velocity (velocity triangle),

u circumferential velocity,

v absolute velocity.

The following relationship exists between the specific energy $Y = g \cdot H$ and the pump output P_Q of the centrifugal pump:

$$P_Q = Y \cdot \dot{m} \quad \text{or} \quad P_Q = g \cdot H \cdot \rho \cdot Q$$

with

Q capacity through the impeller,
 ρ density of pumped medium,
 g gravitational constant
 H head,
 \dot{m} mass flow through the impeller.

If we apply this to the impeller element, we have the pump output of the element

$$P_{E1} = g \cdot H_{E1} \cdot \rho \cdot Q_{E1} = \left(\sum_i^2 (u \cdot v_u) \right) \cdot \dot{m}_{E1}$$

with

H_{E1} head of impeller element,
 Q_{E1} capacity through impeller element.

Thus finally we obtain the shaft power P of the centrifugal pump as

$$P = \sum_i^a P_{E1} + P_{v.Rads.} + P_m + P_{v.sp}$$

with

$\sum_i^a P_{E1}$ absorbed power of impeller elements, added up over all the flow lines between the inner (i) and outer (a) relative flow lines in the impeller,
 $P_{v.Rads.}$ power loss due to impeller side friction,
 P_m power loss (mechanical efficiency) due to friction in plain bearings, anti-friction bearings, shaft seals,
 $P_{v.sp}$ power loss due to clearance gap loss.

Fluid Flow Machine

Strömungsmaschine
Turbo machine

F.f.m.'s comprise in the main machines through which a fluid or a fluid charged with solid particles flows, and which are equipped with a bladed impeller for the conversion of mechanical energy and flow energy. In contrast to positive displacement machines (positive displacement pump), the flow deflection through the blades is the characteristic feature of f.f.m.'s in this energy exchange (see momentum theorem of fluid dynamics).

Depending on the direction of the exchange of energy, i.e. whether from the shaft of the f.f.m. to the flowing medium or vice-versa, a distinction is made among f.f.m.'s between "work producing machine's" (centrifugal pump, turbo-compressor fan, airscrew, ship's propeller) and "prime movers" (this rather imprecise expression in terms of physical science refers to drive engines like steam turbine, gas turbine, wind turbine and hydraulic turbine, in particular water turbine).

Flux Line

Flußlinie

Ligne de courant

F.l., circular projection of a flow line in the meridian section plane (section through the rotation axis of the impeller). If the f.l. is rotated around the rotation axis we obtain the so-called flux surface, which is important for the calculation of centrifugal pump impellers flow profile.

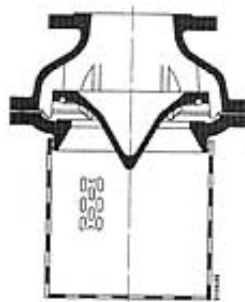
Foot Valve

Fußventil

Clapet de pied

F.v.'s are fitted at the bottom (mouth) of suction pipes (pumping plant), often in conjunction with suction strainer baskets, in order to prevent draining of the centrifugal pump after it has been switched off.

A type often used is the disc valve (see illustration) which has favourable flow characteristics (see Table 2 under pressure loss), and which can be arranged in groups for pipe sizes in excess of DN 350 (nominal diameter). The points listed under filling-up of centrifugal pumps should be observed when operating the foot valve.



Foot valve (disc valve)

Force

Kraft

Force

F., a vectorial (directional) magnitude which is the cause of the acceleration and/or deformation of a body. F. is not a basic magnitude in the "Système International d'Unités" (unit) which is mandatory for centrifugal pump technology (in contrast with former unit systems). The SI unit of f. is 1 N, which is the effective f. which imparts an acceleration of 1 m/s^2 to a body of 1 kg mass (unit).

Formation of Air Pockets

Luftsackbildung

Formation de poche d'air

The fluid pumped may entrain gas (usually air) in dissolved and undissolved (bubbles) form (gas content of pumped medium). Consequently large air bubbles tend to accumulate at suitable spots in the piping system or in the centrifugal pump itself, and form air pockets, which can seriously impair the pump performance. In order to avoid the f.o.a.p.'s, suction pipes should be laid with a rising slope towards the pump, and eccentric reducers and branch pieces, fitted belly down, should be used in horizontal suction pipes (Figs. 5 and 6 under fittings). If possible, each pump should have its own individual suction pipe, so that, when one pump only is operated, no air can penetrate via a shut-down pump into the suction pipe. Isolating valves (valves and fittings) in suction pipes should be fitted with their stem horizontal. The glands of the valve stems should preferably be connected to a

pressure water supply. If the discharge lines are of appreciable length, with rising and falling stretches of line, each apex should be provided with an automatic vent valve (float valve).

Foundation

Fundament
Fondation

see Pump Foundation

Free Passage

Feier Durchgang
Passage

see Impeller

Frequency

Frequenz
Fréquence

F. is the characteristic magnitude of a periodic phenomenon, e.g. a vibration or rotation (rotary frequency, but in this connection "rotational speed" is the term most commonly used). F. is defined as the reciprocal of the cycle time. The SI unit of f. is $s^{-1} = 1 \text{ Hz}$ (Hertz).

In electrical engineering, the mains f. is the number of oscillations or cycles of the alternating current or alternating voltage per second. In Europe, most supply networks have a f. of 50 Hz, with a very few exceptions of 42 Hz (individual networks in Italy). In North America and other parts of the world where the supply of energy has been dominated by American firms, the supply f. is 60 Hz, and 60 Hz is often found also on board ships. In power engineering, a f. of $16\frac{2}{3}$ Hz is used in certain instances (e.g. the German Federal Railways), apart from the usual f. of 50 Hz.

FROUDE-Number

FROUDE-Zahl
Nombre de FROUDE

see Model Laws

Fundamental Equation

Hauptgleichung
Équation fondamentale

see Fluid Dynamics

Fuse

Sicherung
Fusible de sécurité

Electrical f.'s are only a coarse protection (short circuit protection) against thermal overload; only protective motor cutout switches (electrical switchgear) or conductors with motor protection (bimetallic release) offer a true protection. According to VDE 0660, bimetallic releases (trips) should feature the following trip delay times for the release current expressed as a percentage overload of the set current:

Overload	5%	20%	50%
Release (trip) delay	> 2 hours	< 2 hours	< 2 minutes

When circuit breakers fitted with bimetallic releases without instantaneous short circuit trips are used, the following fuses should be adopted as short circuit protection: quick acting f.'s rated at approx. three times the rated current. slow f.'s rated at approx. twice the rated current. For starting under heavy load the above values should be increased by 50% approx. These rating rules apply to three-phase motors (asynchronous motor) from 1 to approx. 20 kW. The melting currents and fusing times of quick acting cartridge fuses (according to VDE 0635) are given in the Table below:

Rated amperage	Fusing time < 1 h	Fusing time > 1 h
6 and 10 A	2.1 X rated current	1.5 X rated current
15 and 25 A	1.75 X rated current	1.4 X rated current
35 A and over	1.6 X rated current	1.3 X rated current

G

Gas Content of Pumped Medium

Gasgehalt im Fördermedium

Contenance de gaz dans le liquide pompé

Only very exceptionally is the pumped medium a pure liquid. Both solid and gaseous substances are usually present as admixtures (two-phase flow).

Gases can be present in a liquid in two different forms. viz. in dissolved and undissolved form. In the dissolved form, the gas is distributed molecularly and attached to the liquid molecules as a result of the physical forces. Undissolved gas is present in a liquid in the form of bubbles. The content of a particular dissolved gas in a liquid does not exceed a precisely defined upper limit, depending on the type and condition of the liquid (e.g., apart from unstable supersaturated conditions, the max. volume percent of dissolved air in water of 20 °C at atmospheric conditions is approx. 2 %). whereas the content of undissolved gas has practically no upper limit. The content of undissolved gas is largely dependent on the type of motion reigning in the fluid. If the fluid is at rest, degasification usually proceeds rapidly, e.g. the bubbles rise to the surface of the liquid, and the dissolved gas content approaches its maximum value. A measuring device invented by VAN SLYKE is used to determine the dissolved gas content in a liquid; it separates the dissolved gas under vacuum and measures its volume (gas separation).

Dissolved gases in liquids have an effect on the suction behaviour of centrifugal pumps. A high g.c.o.p.m. leads to early incipient cavitation, i.e. it leads to a higher NPSH value (net positive suction head) (Fig. 1). Undissolved gas in bubble form influences the overall operating behaviour, and the extent to which gas can be entrained continuously varies widely according to the type, size and mode of operation of the pump concerned. Fig. 2 illustrates the changes in the characteristic curves $H(Q)$, $\eta(Q)$ and $P(Q)$ of a nonclogging pump (impeller) in function of the percentage the entrained air.

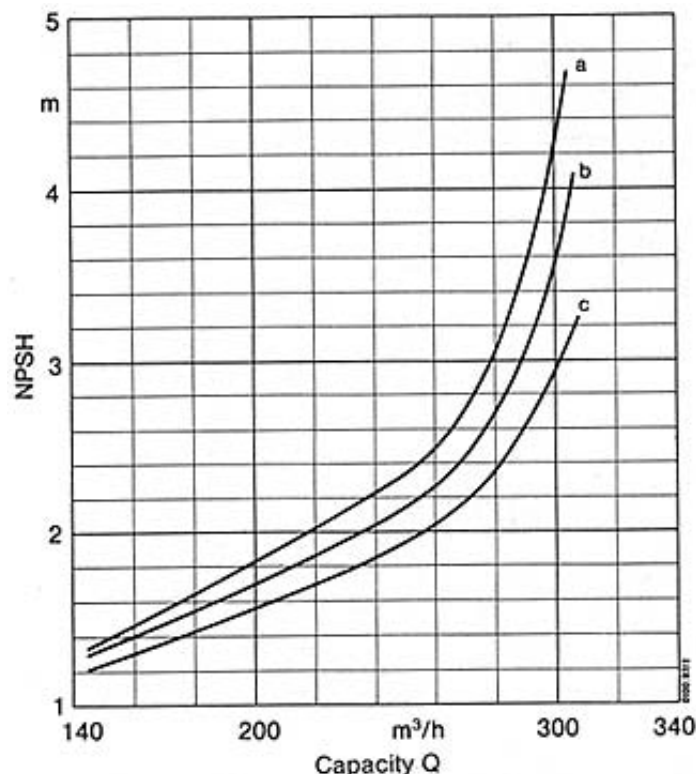


Fig. 1: Influence of dissolved air on suction behaviour of a radial centrifugal pump ($n = 1450 \text{ min}^{-1}$, $Q_{\text{opt}} = 210 \text{ m}^3/\text{h}$, impeller diameter $D = 404 \text{ mm}$)

a 1.9 to 1.6 percent by volume of air content,
 b 1.2 to 1.0 percent by volume of air content,
 c 0.5 to 0.4 percent by volume of air content
 (NPSH = net positive suction head)

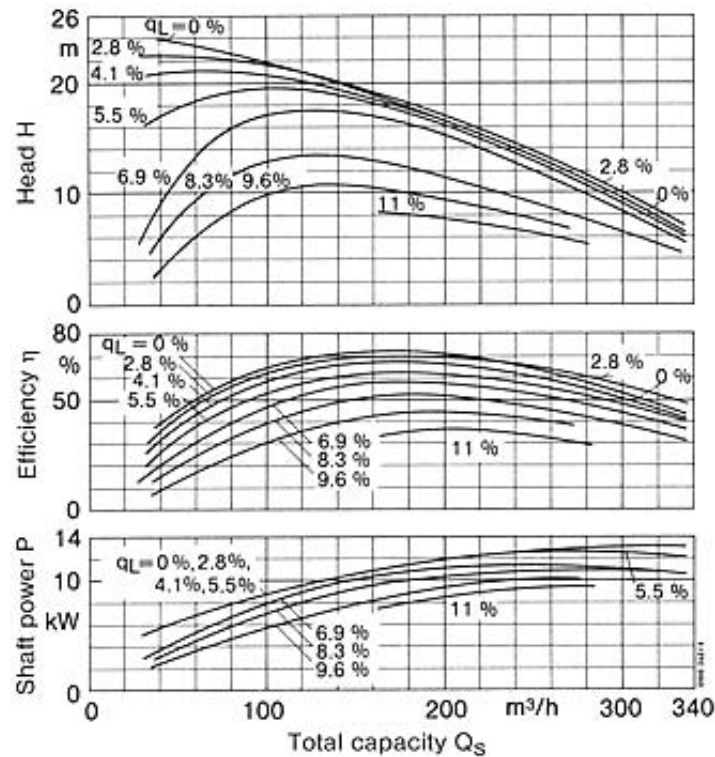


Fig. 2: Influence of undissolved air on the operating behaviour of a non-clogging impeller pump handling profiltered sewage (open three-passagge impeller, $D = 250 \text{ mm}$, $n = 1450 \text{ min}^{-1}$)

q_L = air volume/mixture volume (at suction branch) in %

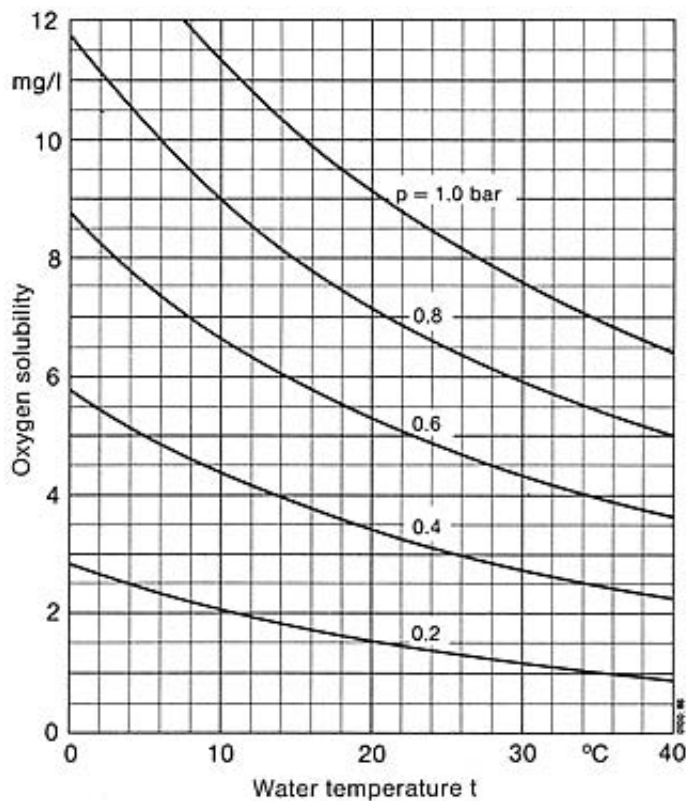
Q_s = Volume flow (suction end)

(see enlarged diagram)

Gas Separation

Gasabscheidung

Séparation de gaz



Solubility of oxygen in water in function of water pressure and water temperature

The solubility of air in water depends on the temperature and pressure of the latter (gas content of pumped medium). As the solubility decreases (see illustration) with decreasing pressure (and increasing temperature), air comes out of solution in the suction pipes (pumping plant) of pumps, and in the lines of siphoning installations (formation of air pockets). The gases which have been released out of solution accumulate at the apex of the line and have to be pumped out to prevent a breaking off of the water column. On average, the volume Q_{sL} which has to be sucked off per hour amounts to

$$Q_{sL} = \frac{X}{100} \cdot Q \cdot \left(\frac{p_1}{p_2} - 1 \right)$$

related to the suction condition, with

- X volume percent of air dissolved in water at the absolute pressure of 1 bar,
- Q capacity of pump or of siphoning installation in m³/h,
- p₁ absolute pressure in bar of the water at the beginning of the evacuation process (venting),
- absolute pressure in bar at the end of the evacuation process. For pumps operating on
- p₂ suction lift (suction behaviour), the absolute pressure at the suction end at capacity Q should be entered in the equation.

For siphoning installations we have

$$p_2 = p_B - 0.1 \left[e_s + \frac{v^2}{2g} (1 + \sum \zeta) \right]$$

with

e_s in m in accordance with illustration under siphoning installation,
 v flow velocity in m/s,
 $\sum \zeta$ sum of resistance coefficients (pressure loss),
 P_b barometric pressure (atmospheric pressure) in bar,

or in cases where the apex lies close to the collecting well

$$p_2 = p_b - 0.1 (e_s + e)$$

with

e_s and e in m in accordance with illustration under siphoning installation.

The volume of gas which is separated out of solution is preferably sucked away by an automatic evacuation plant (venting).

Media which separate large amounts of gas (e.g. sewage) also liberate gas in the low pressure zones of the impeller. This impairs the pumping performance, and open impellers with a large clearance gap width or torque-flow impellers (torque-flow pump) have given the best results in such cases. The gas must be separated and positively removed from the pump casing.

Gate Valve

Sehieber
Soupape à tiroir

see Valves and Fittings

Gearbox

Getriebe
Variateur de vitesse

see Gear Drive for Pumps

Gear Drive for Pumps

Zahnradgetriebe für Pumpen
Engrenage pour pompes

Optimal pump types (optimal in respect of construction costs and performance) can often only be achieved by selecting a pump rotational speed independent of the driver speed. Therefore g.d.'s f.p. have to be adopted, particularly for high powers. Depending on the required transmission ratio, single-step or multi-step gears are adopted (Figs. 1 and 2). If the pump impellers (or the single impeller) are attached directly on the driven shaft of the gearbox, then the gearbox and the work-producing machine form a compact unit (close-coupled pumping set, geared pump).

If high transmission ratios i are required, as well as an in-line arrangement of the drive and driven shafts. planetary gears (Fig. 4) or star gears (Fig. 3) are often adopted, as their rotationsymmetrical casing geometry fits in well with fluid flow machines and with electric motors (geared pump). This applies analogously to high step-down ratios (Fig. 3 under land reclamation pump).



Fig. 1: Single step gear, direction of rotation is reversed

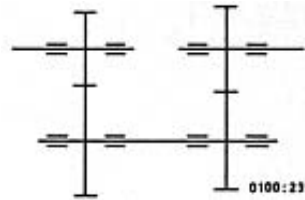
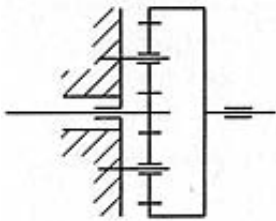
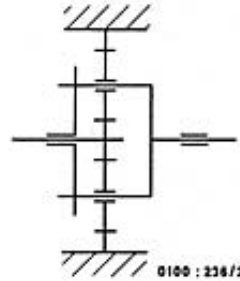


Fig. 2: Two step gear, direction of rotation unaltered

Fig. 3: Planetary gear with stationary pinion cage, direction of rotation is reversed-, $i = 2$ to 11 for 3 planet pinionsFig. 4: Planetary gear with rotating pinion cage, direction of rotation remains unaltered; $i = 3$ to 12 for three planet pinions

A high degree of manufacturing precision is essential, particularly if the circumferential velocities of the gear wheels are high, in order to reduce the noise level (noise in pumps and pumping installations). The Table below gives reference values for the ISO quality of the gear wheels.

Figs. 1 to 4 illustrate a few gearbox arrangements diagrammatically.

Table: Manufacturing quality of gear wheels

Circumferential velocity of gear wheel	ISO quality
0 to 4 m/s	8 to 9
> 4 to 12 m/s	6 to 7
> 12 to 60 m/s	4 to 5

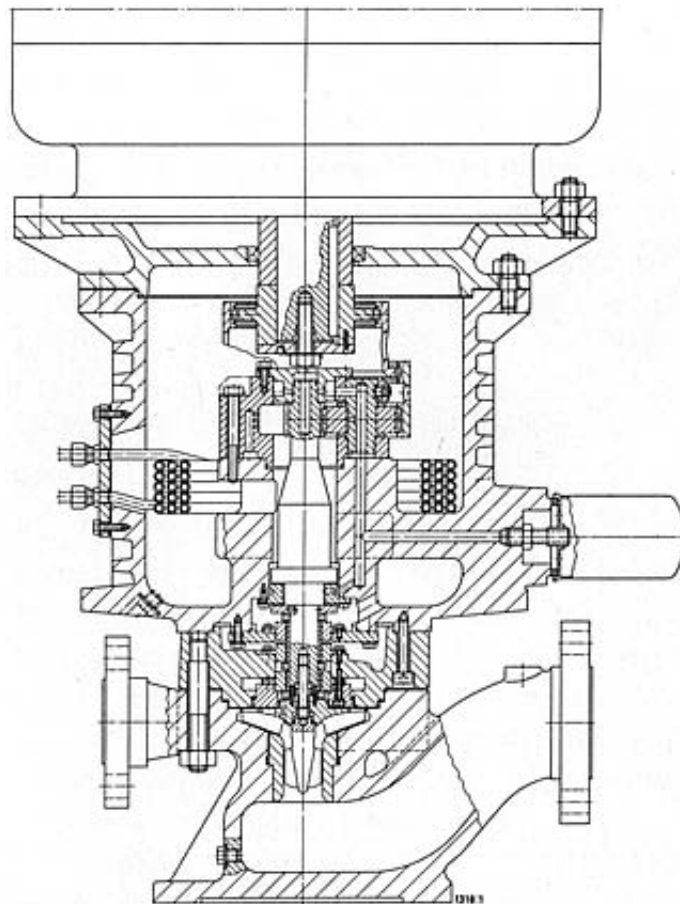
Geared Pump

Getriebepumpe

Pompe avec varlateur de vitesse incorporé

G.p., a centrifugal pump with a built-on or built-in gear drive between pump shaft and driver (drive).

A special variant of the g.p. is the geared highpressure pump which is used as high-velocity pump to produce high heads (up to 1000 m) at low capacities and with a high efficiency. The unusually high rotational speed for a centrifugal pump, of $15\,000\text{ min}^{-1}$ approx. requires a speed-increasing gear if a normal two or four pole electric motor is used as driver. The illustration shows such a pump with pump casing (inline type), open impeller with radial blades, two mechanical seals (shaft seals) for sealing off the fluid pumped and the gear oil, and a radial and an axial guide bearing for the sun wheel shaft of the speed increasing gear (torque division gear) arranged above. The motor is firmly flanged onto the gearbox casing and carries a coupling (shaft coupling) on its shaft stub end which has teeth meshing with the ring wheel, which in turn drives the star wheels of the gearbox.



Cleared high-pressure pump

The quality of manufacture, correct lubrication of the bearings and gears and proper maintenance of the lubricant (cooling and filtration) all represent important aspects for ensuring a low rate of wear of the gears.

The single stage process type design (the complete rotor can be pulled out in the upward direction) ensures a short downtime in the event of repairs. The inline construction (inline pump) greatly simplifies the foundation (only a very small one is required) and presents no alignment problems. The gear ratio graduations are numerous enough to ensure sufficiently small steps in rotational speeds, thanks to the module construction system used. As a general rule, several different impeller diameters with their associated diffusers can be accommodated in one casing size.

Usually the following items and magnitudes can be varied on a geared high-pressure pump: input speed, gear ratio, inducer, impeller size and trimming diameters of same, and diffuser size. This enables any desired operating point on the performance chart to be obtained economically in function of the inlet conditions at the suction end.

Gear Pump

Zahnradpumpe
Pompe à engrenages

see Positive Displacement Pump

Geodetic Altitude

Geodätische Höhe
Cote géodésique

Q.a. denotes the altitude z of a point on a centrifugal pump or pumping plant above a specific datum level BN (Figs. 1 and 2 under head, system characteristic curve). In the BERNOLLI equation (fluid dynamics) it is added to the pressure head and the dynamic head.

Geodetic Head

Geodätische Förderhöhe
Hauteur géométrique

see [Head](#)

Geodetic Suction Head

Geodätische Zulaufhöhe
Charge géométrique à l'aspiration

see [Suction Behaviour](#)

Geodetic Suction Lift

Geodätische Saughöhe
Hauteur géométrique d'aspiration

see [Suction Behaviour](#)

Gravitational Constant

Fallbeschleunigung
Accélération due à la pesanteur

G.c., symbol g , is the acceleration (which varies according to geographic location) caused by the earth's gravitational field. In centrifugal pump technology, it is usual to adopt the rounded-up figure for the pc. of:

$$g = 9.81 \text{ m/s}^2.$$

In accordance with DIN 1944 (edition 1968) (Acceptance Tests on Centrifugal Pumps), also DIN 24260 (edition 1986) (Centrifugal Pumps and Centrifugal Pumping Plants, Terms, Letter Symbols, Units), also ISO 2548 (edition 1973) (Centrifugal Pumps, Guidelines for Acceptance Test), this value is used in the definition equations for pressure head, dynamic head, head, net positive suction head (NPSH) etc.

On the earth's surface, the values of g.c. vary by less than 7 from one another.

In a weightless (non-gravitational) or almost weightless state (e.g. centrifugal pump in a satellite station) the head H of a centrifugal pump tends towards infinity because g tends towards approx. zero; therefore, in the case of any pump installed a long distance away from the earth, the finite specific energy Y should be adopted in lieu of the head H , i.e. the useful mechanical work transmitted by the pump onto the fluid, per unit mass of fluid pumped, which is independent of the g.c. The same considerations apply to centrifugal pumps installed in accelerated or retarded systems (e.g. pumps in a rocket).

Guarantee

Garantie
Garantie

The term g ., in connection with the acceptance test of a centrifugal pump usually implies the warranty of certain physical magnitudes and characteristics of the pump mutually agreed in the contract (warranty tolerance). In the DIN 1944 acceptance test codes, for instance, the concept of g . is used in the technical sense and designates the values laid down in the contract as basis for test verification. The concept of g . relates solely to the centrifugal pumps proper in this respect, and nothing is laid down or stated as to what rights and obligations ensue if the values laid down are not attained (see also ISO 2548 and 3555).

The specification of the magnitudes and characteristics of the centrifugal pump to be guaranteed will depend on the type and application of the pump. Every test designed to attain and verify with accuracy a g. costs a lesser or greater amount of money (acceptance test), and these costs should always be kept in a reasonable proportion to the purchase price of the pump.

In DIN 1944, three degrees of accuracy I, II and III are laid down for the g. and the acceptance tests (see Table 1); these degrees differ in respect of the extent of the g., the magnitude of the manufacturing tolerance and the extent and required degree of accuracy of the acceptance test (Table 1 under uncertainty of measurement). In order to satisfy the g. in accordance with the ISO acceptance test code, see details in Table 2 and overall tolerance (acceptance test codes for centrifugal pumps).

Table 1: Performance data and efficiency guarantees in accordance with DIN 1944 (October 1968)

Degree of accuracy		
III	II	I
Capacity guarantee for pumps with non-flat throttling curves H(Q) for n _L = constant		
$\frac{Q_L}{H_L} \cdot \left \frac{dH}{dQ} \right > 0.2$		
At the values specified in the supply contract for the rotational speed n _L , head H _L and for an adequate NPSH _{av} the capacity Q of a pump without speed control, as determined in the acceptance tests, must lie within the limits	At the values specified in the supply contract for the rotational speed n _L NPSH _{req} and head H _L , the capacity Q of a pump without speed control, as determined in the acceptance tests, must lie within	
0.95 and 1.15	0.95 and 1.10	0.95 and 1.05
of the capacity Q _L specified in the supply contract.	of the capacity Q _L specified in the supply contract.	
Number of guaranteed operating points: 1.	Number of guaranteed operating points: 1 or more.	
Head guarantee for pumps with flat throttling curves H(Q) for n _L = constant		
$\frac{Q_L}{H_L} \cdot \left \frac{dH}{dQ} \right \leq 0.2$		
At the values specified in the supply contract for the rotational speed n _L , capacity Q _L , and for an adequate NPSH _{av} , the head H of a pump without speed control, as determined in acceptance tests, must lie within the limits	At the values specified in the supply contract for the rotational speed n _L , NPSH _{req} and capacity Q _L , the head H of a pump without speed control, as determines in acceptance tests, must lie within the limits	
0.99 and 1.03	0.99 and 1.02	0.99 and 1.01
of the head H _L specified in the supply contract.	of the head H _L specified in the supply contract.	
Number of guaranteed operating points: 1.	Number of guaranteed operating points: 1 or more	
Efficiency guarantee (any efficiency which is not specifically designated as a guaranteed efficiency shall be understood to be an anticipated efficiency):		
No efficiency guarantee: At a mutually agreed head H _L the mutually agreed shaft power of the pump or the mutually agreed power of the drive shall not be exceeded.	At the values specified in the supply contract for the rotational speed n _L and NPSH _{req} the efficiency determined in acceptance tests must attain or exceed the guaranteed value.	

Table 2: Guarantees in accordance with ISO acceptance test code

Degree of accuracy ¹⁾	
C	B
Guarantee for capacity ²⁾: At the values specified in the supply contract for the head H_L and rotational speed n_L , the capacity Q must be situated between the limits	
0.93 and 1.07 ³⁾	0.96 and 1.04 ³⁾
of the capacity Q_L specified in the supply contract.	
Guarantee for head²⁾: At the values specified in the supply contract for the capacity Q_L and rotational speed n_L , the head H must be situated between the limits	
0.96 and 1.04 ³⁾	0.98 and 1.02 ³⁾
of the head H_L specified in the supply contract.	
Efficiency guarantee²⁾: The efficiency η at the intersection of the straight line joining the points $Q = 0$, $H = 0$ and Q_L , H_L with the measured throttling curve must be at least equal to	
95% ⁴⁾ ⁵⁾	97.2% ⁴⁾ ⁵⁾
of the guaranteed efficiency.	
1. 1) No guarantees shall be specified for degree of accuracy A (max. degree of accuracy) 2. If a test-value of the NPSH has been agreed in the contract, the specified guarantees must be attained at the agreed NPSH value 3. The values specified include the permissible uncertainty in measurement and the manufacturing tolerance	4. The values specified only take the permissible uncertainties in measurement into account 5. The values specified should be altered to 95.5 % and 97.5 % respectively in the case of the unit efficiency of close-coupled pumping units (sets) which incorporate an electric motor

Guide Vane

Leitschaufel
Aube directrice

see [Blade](#)

Guide Vane Pitch Adjustment

Leitschaufelverstellung
Réglage des aubes de diffuseur

see [Control](#)

H

Hand Pump

Handpumpe
Pompe à bras

H.p. is a positive displacement pump actuated by hand, e.g. a lever pump or a hand piston pump.

Hardness of Water

Härte eines Wassers
Dureté d'une eau

see Water Hardness

Head

Förderhöhe
Hauteur

An important energy concept (DIN 24260) in centrifugal pump technology; a distinction must be made between the h. H generated by the pump and the h. HA of the plant.

The h. of the pump is the pump output PQ transmitted by the pump to the product pumped, related to $\rho \cdot g \cdot Q$,

$$H = \frac{P_Q}{\rho \cdot g \cdot Q}$$

where

ρ density of pumped medium,
g gravitational constant,
H h. of the pump,
Q capacity.

Fig. 1 illustrates the concept of pump output PQ. The pump situated within the solid lines on the sketch (system limits) and the sum of the powers (positive input, negative output) must be equal to zero, i.e.

$$P + P_{Q-s} - P_{Q-d} - P_{v-i} - P_m = 0,$$

where

P shaft power of pump (shaft power),
 P_{Q-s} power input, adduced into pump suction branch s by the medium flowing into the pump,
 P_{Q-d} power output, led away at pump discharge branch by the medium flowing out of the pump,
 P_{v-i} power loss (mainly frictional power losses) arising within the pump between s and d, which does not however contribute to any increase of the power output (internal efficiency), and
 P_m mechanical power loss (mechanical efficiency) in the pump bearings (plain bearing, anti-friction bearing) and in the shaft seals.

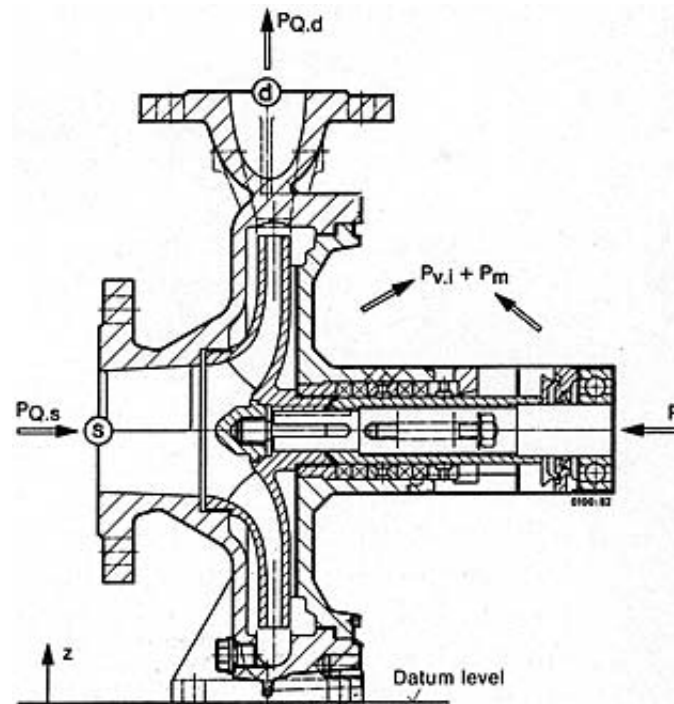


Fig. 1: Explanation of pump output concept
 $P_Q = P_{Q,d} - P_{Q,s} = P - P_{v,i} - P_m$

If we accept that the expression $P_{Q,d} - P_{Q,s}$ represents the pump output P_Q i.e.

$$P_Q = P_{Q,d} - P_{Q,s} ,$$

we obtain

$$P_Q = P - P_{v,i} - P_m$$

(internal efficiency) as useful mechanical output.

According to BERNOULLI (fluid dynamics) this useful output is:

$$P_Q = \left[(p_d - p_s) + \frac{\rho}{2} (v_d^2 - v_s^2) + \rho \cdot g(z_d - z_s) \right] \cdot Q$$

where

p static pressure,

v flow velocity,

z elevation (geodetic altitude).

Thus the h. of the pump becomes:

$$H = \frac{p_d - p_s}{\rho \cdot g} + \frac{v_d^2 - v_s^2}{2g} + z_d - z_s .$$

If the fluid pumped is appreciably compressible, the value of density ρ should by definition be taken as the arithmetic mean of density at the pump suction branch and density at the pump discharge branch:

$$\rho = \frac{\rho_d + \rho_s}{2}$$

The h . of the plant (Fig. 2) can be deduced similarly. taking the head losses H_v into account. It amounts to:

$$H_A = \frac{p_a - p_e}{\rho \cdot g} + \frac{v_a^2 - v_e^2}{2g} + z_a - z_e + H_{v,d,a} + H_{v,e,s}$$

where

H_A h . of the plant,

$H_{v,d,a}$ head loss in plant discharge line, from outlet cross-section of pump A_d to outlet cross-section of plant A_a , including any outlet losses and losses (pressure loss) caused by valves and fittings, appliances etc.,

$H_{v,e,s}$ head loss in plant suction lift line or overhead suction pipe, from the inlet cross-section A_e of the plant to the inlet cross-section A_s of the pump, including any inlet losses and losses (pressure loss) caused by valves and fittings, appliances etc.

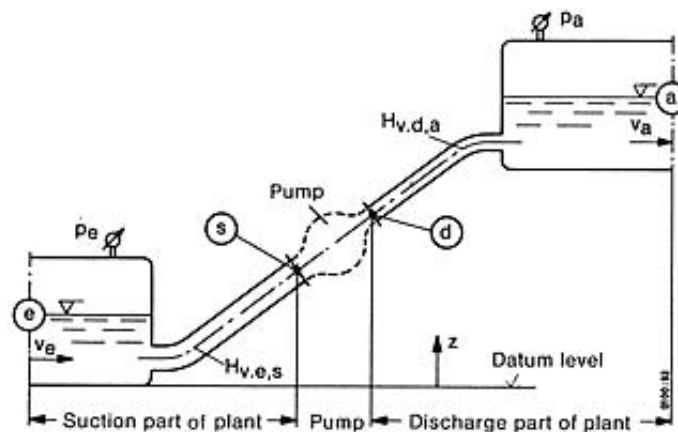


Fig. 2: explanation of magnitudes involved in the head H_A of the plant

In the h . of the plant, the expression geodetic head H_{geo} is sometimes used. H_{geo} designates the difference in altitude between the outlet cross-section A_a and the inlet cross-section A_e of the plant:

$$H_{geo} = Z_a - Z_e.$$

In steady state operation (rotational speed n = constant), the h . of the pump equals the h . of the plant:

$$H = H_A.$$

The SI unit of h . is 1 m.

The following concepts also occur in general in connection with the h . of the pump:

Optimum h . H_{opt} : h . of the pump at the operating point of optimal efficiency.

Upper limit h . H_{max} : max. permissible h . at which the pump can operate continuously without suffering damage.

Lower limit h . H_{min} : min. permissible h . at which the pump can operate without suffering damage.

Shut-off h . H_0 : for a capacity $Q = 0 \text{ m}^3/\text{s}$.

Peak h . H_{Sch} : h . at apex (relative maximum) of an unstable throttling curve.

Static h . $H_{A,0}$ (also called H_{stat}): that part of the system head (system characteristic curve) which does not depend on the capacity Q :

$$H_{A,0} = \frac{p_a - p_e}{\rho \cdot g} + H_{geo}.$$

Head Loss

Verlusthöhe

Perte d'énergie

The h.l. H_v , according to DIN 24260 (edition 1986), is the loss of mechanical energy between the inlet and outlet cross-section of a length of piping, including any entry and exit losses and losses caused by valves and fittings, appliances, etc. (pressure loss), related to the weight force (weight) of the fluid pumped. The SI unit of h.l. is 1 m.

Heat

Wärme

Chaleur

see Unit

Heating of Electric Motors

Erwärmung von Elektromotoren

Réchauffement des moteurs électriques

The losses in an electrical machine (drive) cause the active components to heat up, particularly the winding and the lamination packs. The permissible limit of this temperature-rise governed by the insulating material used. The thermal stability of insulants, their subdivision into insulant classes and the methods of determining excessive temperatures are dealt with in DIN VDE 0530 Part 1.

Any excessive winding temperature-rise will shorten the life of the insulation and may lead to an immediate breakdown of the electrical machine. The cause of such a temperature-rise may lie in faulty cooling or overloading.

Overloading results in excessive current requirement and consequently in copper losses rising according to the square law. By connecting the motor via a motor protection cutout switch with bimetallic relays (electrical switchgear), the winding can be protected from damage by overcurrent. The bimetal heats up to the same degree as the winding, and will automatically switch off the motor eventually if the overcurrent is small, but quite soon if the overcurrent is considerable. Motor protection cutout switches often also comprise magnetic quick-release trips or relays, which switch the motor off immediately in the event of a short-circuit and thus prevent serious damage. A comprehensive protection of the motor winding against other temperature-increasing influences such as insufficient or faulty cooling is provided by the so-called full motor protection. This involves the provision of temperature sensors embedded directly in the winding, which trigger a warning signal or switch the motor off when the permissible temperature is exceeded.

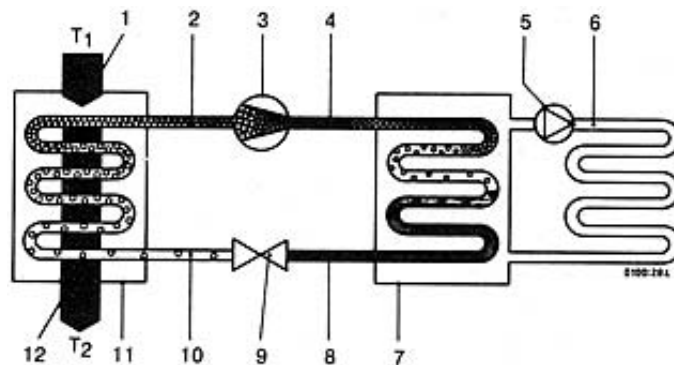
Heat Pump

Wärmepumpe

Pomps de chaleur

The h.p. is not a pump in the strict sense of the word, but a plant with the aid of which low temperature heat can be elevated ("pumped up") to a higher temperature level. The h.p. operates on the thermo-dynamic principle of a refrigeration plant. It allows the exploitation of the energy provided by the heat sources in our environment, including the sun, air, water and soil, and also the exploitation of waste heat from industrial processes.

H.p. plants as a general rule comprise two heat exchangers (condenser or liquefier, and evaporator), an expansion valve and a compressor. The circuit is primed with a liquid refrigerant which evaporates at low temperature and absorbs heat from its surroundings during this phase.



Example of a head pump system using ambient air as the heat source
 1 ambient air (T_1); 2 refrigerant in vapour phase; 3 compressor; 4 compressed refrigerant in vapour phase; 5 central heating circulating pump; 6 heating water; 7 liquefier (condenser); 8 liquid refrigerant; 9 expansion valve; 10 expanded refrigerant; 11 evaporator; 12 exhaust air (T_2 ; $T_2 < T_1$)

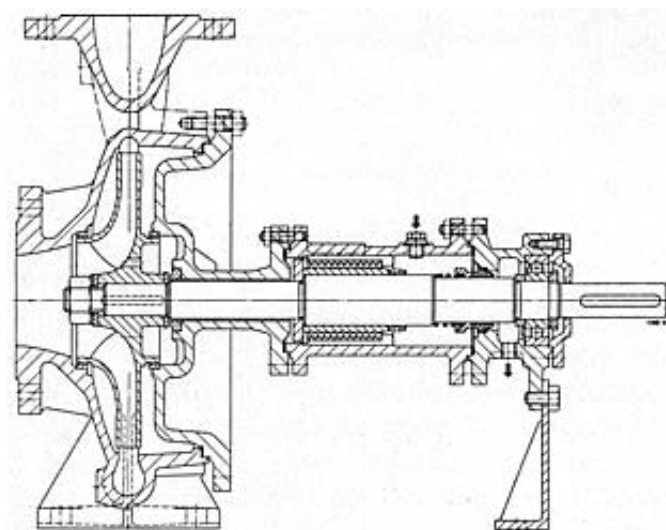
Then the compressor raises the medium to a higher pressure and temperature level, after which it is led through the liquefier (condenser) where it condenses. The heat released during condensation is transmitted to a heating circuit (central heating circulating pump). Finally the liquid refrigerant is expanded in the expansion valve and the cycle starts anew (see illustration).

The flow temperature which is capable of being attained economically in hot water heating systems or industrial water installations can amount to up to 55 °C H.p. plants have so far been installed mainly for room heating, industrial water heating and swimming pool water heating purposes, usually in conjunction with conventional heating plants (bivalent heating system).

Heat Transfer Pump

Wärmeträgerpumpe
Pompe à fluide caloporteur

The h.t.p. is a pump for the circulation of heat transfer media, usually of hydrocarbon base, e.g. for maintaining of the heat transfer medium circulation in a heating system. Since the temperature of heat transfer oil can reach up to 350 °C, a cooling section (typically air convection cooling) between the pump casing and the shaft seal, keeps the medium-lubricated plain bearing and standard mechanical seal (shaft seals) cool. The illustration shows the typical placement of bearings and shaft seals for a h.t.p.



Heat transfer pump

Helical Rotor Pump

Exzentrerschneckenpumpe
Pompe à vis exentree

see [Positive Displacement Pump](#)

High Pressure Pump

Hochdruckpumpe
Pompe à haute pression

H.p.p. is a [centrifugal pump](#) with a head situated between 200 and 1200 m (e.g. for [pressure boosting plants](#)).

In contrast we have [low pressure](#) [medium pressure](#) and [super pressure pumps](#).

High Speed Centrifugal Pump

Schleuderpumpe
Pompe centrifuge à grande vitesse

see [Geared Pump](#)

Hollow Vortex

Hohlwirbel
Vortex aéré

see [Inlet Conditions](#)

Horizontal Pump

Horizontalpumpe
Pompe horizontale

H.p. is a [centrifugal pump](#) with a horizontal shaft.

Hot Water Pump

Heißwasserpumpe
Pompe à eau surchauffée

H.w.p.'s are [centrifugal pumps](#) for pumping water at temperatures above 100 °C; they are used as [central heating system circulating pumps](#), as [boiler feed pumps](#) and boiler feedwater circulating pumps, also as [circulating pumps](#) in nuclear plants ([reactor pump](#)).

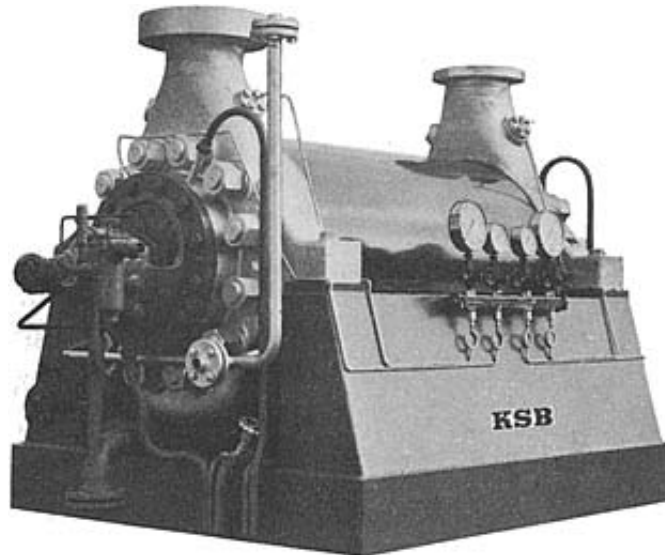
[Materials](#) frequently used for h.w.p.'s are, apart from cast iron and spheroidal graphite cast iron: cast steel, cast chrome steel and austenitic cast steel as well as appropriate wrought alloys for specific [pump casings](#).

In the event of sudden temperature fluctuations, considerable stresses arise in the tiebolts of ring section pumps ([multistage pump](#)) and in the barrels of [barrel pumps](#) ([boiler feed pump](#)). In order to prevent any mutual shifting of the pump and driver shaft centralizes under the influence of the high casing temperature, the pump feet are usually arranged at shaft centralize height. This pump foot arrangement is characteristic of h.w.p.'s (see illustration).

Depending on temperature and pressure conditions, the casing is sealed (seals) by O-rings, flat gaskets, spiral wound asbestos gaskets or metal-to-metal ground sealing faces.

The shaft seal must operate at a low temperature level. In order to achieve this, and to lead away the frictional heat arising during operation, it is necessary to provide cooling. Soft-packed stuffing boxes, mechanical seals or occasionally floating seals are used as shaft seals. The heat is led away via the cooling fluid in the stuffing box housing, in the case of soft-packed pumps; via a heat-exchanger in the case of mechanical seals with internal circulation; by injection of a cold medium from an outside source in the case of floating seals.

The shaft is guided in oil-lubricated anti-friction or plain bearings, which are cooled by air, water or oil.



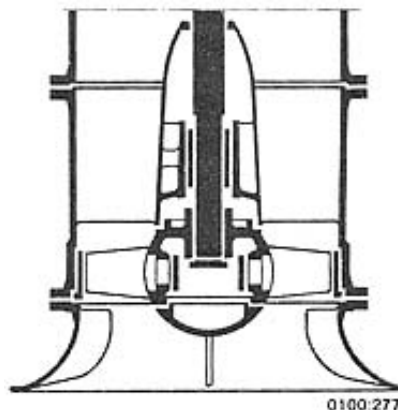
Hot water pump (boiler feed pump) in ring section design with baseplate return of balance water into the casing and instrumentation

Hub Diffuser

Nabendiffusor

Diffuseur à moyeu conique

On tubular casing pumps, particularly those of axial type, the flow velocity downstream of the impeller can be retarded with the aid of a h.d. (see illustration) amongst other methods. The enlargement of the free flow cross-section is effected by a gradual reduction of the hub diameter. H.d.'s are exposed to the danger of flow breakaway both at the inner hub contour and at the cylindrical outer contour (boundary layer, vortex flow diffuser).



Hub diffuser downstream of impeller of an axial pump

Hub Ratio

Nabenverhältnis

Rapport moyeu diamètre extérieur

The h.r. ν represents a characteristic geometric magnitude of the impeller of centrifugal pumps and is usually defined as follows:

$$\text{for axial impellers } \nu = \frac{D_i}{D_a}$$

$$\text{for non-axial impellers } \nu = \frac{D_{1,i}}{D_{1,a}}$$

with

D_i diameter of cylindrical hub,

D_a outer diameter of axial impeller,

$D_{1,i}$ hub diameter at inlet cross-section (suction mouth) of impeller,

$D_{1,a}$ outer diameter of impeller at inlet cross-section (suction mouth).

The choice of h.r. is closely linked to problems of optimal suction behaviour and efficiency at the various specific speeds. Other design aspects (e.g. when pumping sewage, or if adjustable or variable pitch blades which are highly stressed are involved) can influence the selection of the h.r.

Hydraulic Coupling

Hydraulische Kupplung

Coupleur hydraulique

see Fluid Coupling

Hydraulic Efficiency

Hydraulischer Wirkungsgrad

Rendement hydraulique

H.e., also known in technical literature on centrifugal pumps as "blade efficiency". The h.e.

η_h is the quotient of pump output P_Q over blade output P_{Sch} :

$$\eta_h = \frac{P_Q}{P_{Sch}}$$

with

$$P_{Sch} = P - P_{v.Lag/Dchtg/Rads},$$

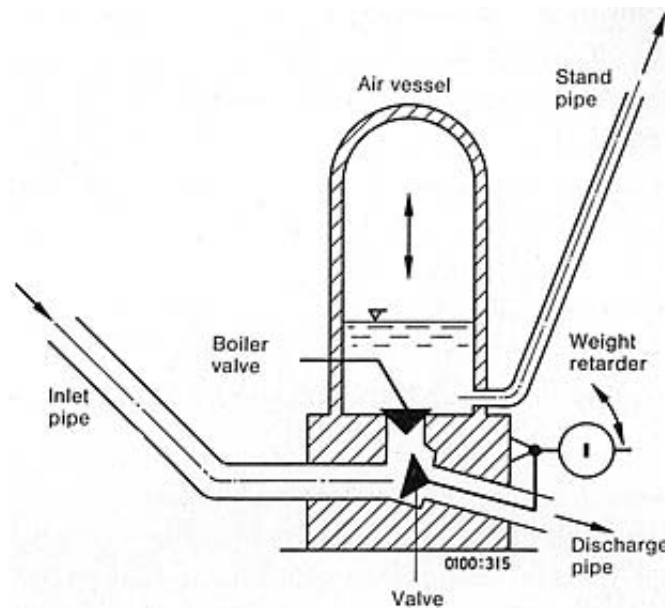
P shaft power of centrifugal pump,

$P_{v.Lag/Dchtg/Rads}$ mechanical power losses in the pump bearings (plain bearing, anti-friction bearing), in the shaft seals, and at the impeller shrouds (impeller side friction).

Hydraulic Ram

Hydraulischer Widder, Stoßheber
Bélier hydraulique

H.r. are relatively simple machines, with whose help water can be lifted to a higher static head or pressure level. The kinetic energy of flowing water in the supply line, retained by the sudden closing of the valve, is high enough to cause a pressure surge and to force part of the water into a pressure vessel supplying muser (illustration). After the opening of the valve, the water column is once again placed into motion, so that the process constantly repeats. This type of transport, although inflationary, uses no outside energy and, apart from the two valves, is completely maintenance-free. Therefore the h.r. is perfectly suited for irrigation purposes in developing countries.



Diagrammatic representation of a hydraulic ram

example:		inlet pipe	stand pipe	discharge pipe
head of the water	m	27	164	-
volume flow	m ³ /h	15	1.7	13.3

Hydraulic Torque Converter

Strömungsgetriebe
Transmission hydrodynamique

The h.t.c. (or speed transformer) is akin to the fluid coupling, and consists of a pump impeller (on the drive shaft), a turbine wheel (on the driven shaft) and a diffuser, the absorbed torque T_{Le} of which is supported on the casing. The following torque equilibrium exists between the input torque T_p (pump impeller torque) and the output torque T_T (turbine wheel torque):

$$T_p = T_T + T_{Le}.$$

The h.t.c. gets its name from the fact that there is a torque conversion from T_p into T_T . H.t.c.'s do not play an important part in the drive of centrifugal pumps (apart from pumped storage installations).

Hydrogen Corrosion

Wasserstoffkorrosion
Corrosion hydrogène

see [Corrosion](#)

Hydrogen Ion Exponent

Wasserstoffionen-Exponent
Exposant de concentration des ions d'hydrogène

see [pH-Value](#)

Hydrophor Plant

Hydrophoranlage
Installation d'hydrophore

see [Domestic Water Supply Plant](#)

Hydrotransport

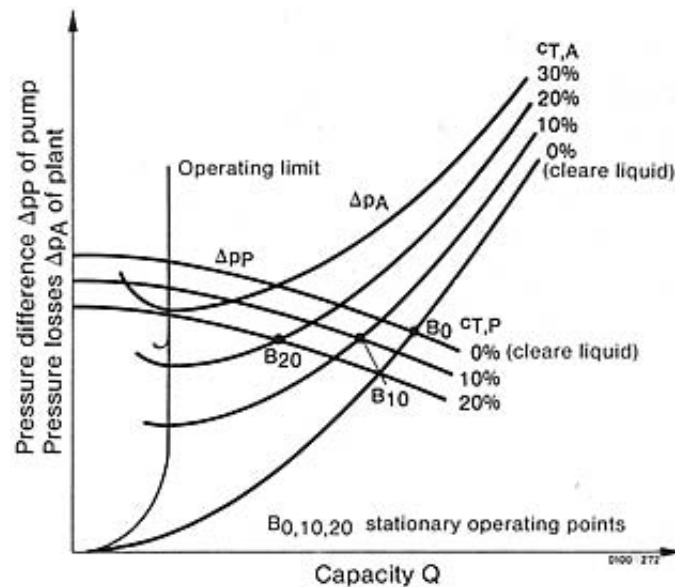
Feststofftransport
Hydrotransport

Fine-grained and coarse-grained solids can be conveyed in streams of liquid. This also applies to conveying through [piping](#). In contrast to clear liquids, it must be borne in mind that such mixtures (sludges, pulps, suspensions) have the tendency to separate out under the influence of gravity or of centrifugal forces. A coarse grain size and large differences in densities ([density](#)) between the solid and the carrier liquid will encourage separation, the extent of which can be gauged by the rate of sinking of a solid particle in the quiescent (still) liquid. Turbulence ([fluid dynamics](#), [pressure loss](#)) in the flow through the pipeline and pressure differences on the solid particles in the vicinity of the pipe wall ([boundary layer](#)) will tend to counteract this separating out process. For these reasons, the [flow velocity](#) as well as the grain size ([abrasion](#)) and the solids content in the liquid (concentration) in percent will determine the nature of the conveyance of the solids e.g.:

- sludge conveying (homogeneous mixture of very small particles of high concentration),
- suspension conveying (homogeneous mixture of low concentration),
- jump conveying (non-homogeneous mixture, with higher concentration at the bottom of horizontal pipes),
- skein conveying (first deposit layers in the of horizontal pipes and concentration at case the centre of vertical pipes),
- slug conveying (unsteady flow conveying).

If the permissible concentrations are exceeded, or if the minimum velocities cannot be attained, there is a risk of clogging of the [piping](#). These phenomena are however of lesser importance where the conveying of pulp suspensions ([pulp pumping](#)) is concerned.

Mixtures of solids and liquids are also capable of being pumped. [Wear](#) and economic considerations ([economics](#)) will determine the most suitable type of pump for this application. Fine grain size materials up to 3 mm diameter enable the adoption of piston pumps, on condition that the periodic pressure fluctuations of this type of pump do not exclude their use for other reasons ([positive displacement pump](#)). In the case of coarser grain sizes ([erosion](#)) it will be necessary to resort to [centrifugal pumps](#), which will however be exposed to a greater or lesser degree of [wear](#) in contact with solids. Special [pump types](#) ([armoured pump](#)). or the use of particularly suitable pump materials ([materials](#), [plain bearing](#)), whilst they cannot basically prevent wear can at least retard it to a considerable degree, making the application an economically viable one. In extreme cases it will be necessary to resort to charging devices designed in such a way that the [high pressure pump](#) handles pure liquids only, whilst the solids are fed into the pump discharge line ([pumping plant](#)) via sluices or tubular chamber feeders.



Pressure difference of pump and pressure losses of plant for various contents of solids (concentration c_T) in the liquid. The pump's pressure difference $\Delta p_p = f(c_T)$ may also increase with increasing concentration of high-density solids

The pressure loss caused by flowing mixtures of solids and liquids in pipelines (pipings) is calculated with the aid of the same formulae as those used for clear liquids; the pipe friction coefficient λ (pressure loss) in this case however also takes into account the additional frictional losses arising from the contact of the solid particles with each other and their contact along the pipe walls. The pressure losses which can be determined in this way for any given concentration c_T can be plotted as a function of the capacity Q (see illustration; it is the usual practice to plot the pressure differences rather than the pressure heads). The minimum value of these resistance curves also determines the flow velocity which must at least be maintained in order to prevent clogging.

Immission Protection Act

Immissionsschutzgesetz

Loi de défense de l'environnement

Immission, from the Latin "immittere" (= to send in) is a legal concept which relates to an influence or action (e.g. on a plot of land) from outside (e.g. from a neighbouring plot of land).

The i.p.a. of the German Federal Republic (B ImSchG) came into force on 1st April 1974. Its purpose is to protect mankind, animals, plants, buildings and other valuable objects from any damaging environmental influences and from any danger, serious disadvantages and serious nuisances, and also to prevent any such damaging environmental influences from arising.

Such damaging environmental influences within the meaning of the i.p.a. include: air pollution, noise, vibration, radioactive radiation and light and heat.

Within the framework of the B ImSchG, the concept of emission also arises, from the Latin "emittere" (= to send out). In this context, emissions include air pollution, noise, vibration, radioactive radiation, light, heat and similar phenomena which may have a damaging effect and which are emitted by a plant or a machine.

In accordance with the B ImSchG, the operators of plants and machines are obliged to keep any damaging environmental influences, serious disadvantages and nuisances away from the community in general and from the neighbourhood. The present state of technical development is adopted as the yardstick for the preventive measures against damaging environmental influences to be undertaken, in the sense of comparable processes, devices or modes of operation which have already been tried out and proved successful.

The i.p.a. is of special significance in the field of centrifugal pump technology in so far as protection against damaging noise and vibration is concerned (environmental protection), and also in connection with the cooling water supply problems (cooling water pump) of power stations (permissible temperature rise in rivers or other waters etc.).

Impeller

Laufrad

Roue

I is the rotating component of a centrifugal pump (or more generally of a fluid flow machine), equipped with blades. The i. converts mechanical energy (energy at the blades, efficiency, internal efficiency) into pump output with a greater or lesser efficiency by deflecting the flow at blades. Depending on the pattern of the flux lines in the i. (particularly in the region of the outer diameter of the i.), i.'s are subdivided into the following types;

- radial i.'s (Figs. 1, 2 and 7 to 14),
- mixed flow i.'s (Figs. 3, 4 and 6),
- axial i.'s (Fig. 5) and
- peripheral i.'s (Fig. 15).

To accommodate the blades, all i.'s are equipped with a back shroud, and in the case of closed i.'s also a front plate (impeller side friction), or, looked at differently, an inner coverplate, and, in the case of closed i.'s, also an outer coverplate. If the i. has no front (or outer) coverplate, it is classed as an open i. Fig. 6 illustrates a closed i. (a) in comparison with its corresponding version of an open i. (b); this figure also illustrates the basic difference between a single suction (a, b) and a double suction i. (c); in the case. of the latter (multisuction pump) the fluid flow approaches the i. axially from both sides.



Fig. 1: Radial impeller with pure radial blades, ram point impeller. S ram point (front view with front coverplate removed)

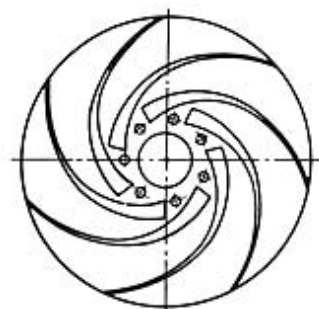
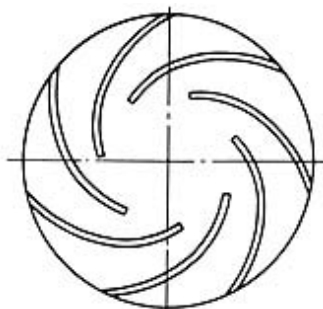


Fig. 2: Radial impeller with blades extending into the suction mouth (front view with front coverplate removed)

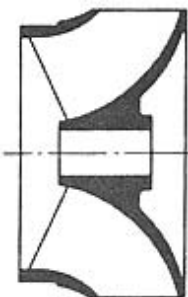


Fig. 3: Mixed flow impeller (helical impeller, diagonal impeller) (front view with front coverplate removed)

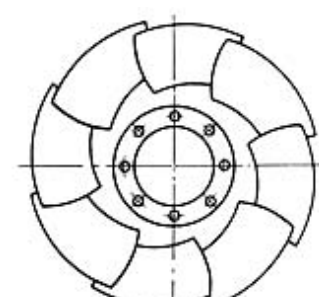
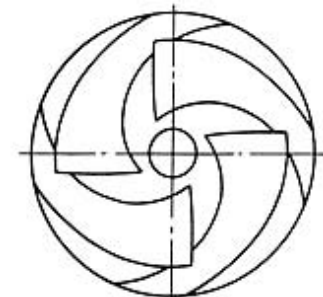


Fig. 4: Mixed flow impeller (semi-axial propeller)

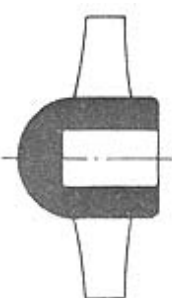


Fig. 5: Axial impeller (axial propeller)

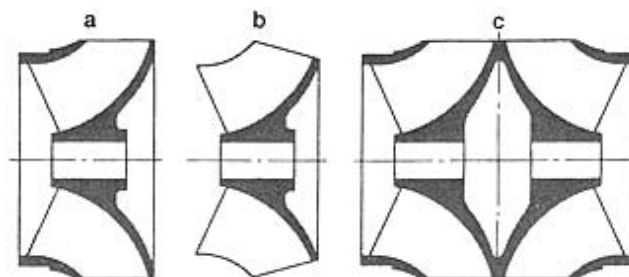
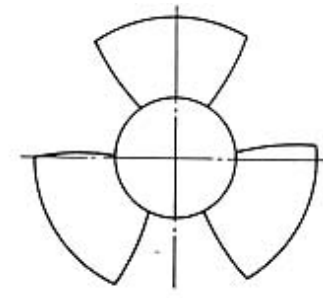


Fig. 6: Helical impeller illustrated to underline the difference between a closed and an open impeller, a single and a double suction impeller

- a) closed single suction impeller,
- b) open single suction impeller,
- c) closed double suction impeller

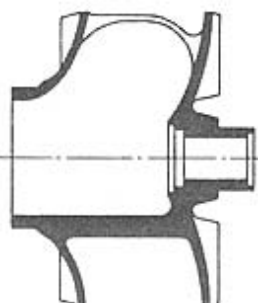


Fig. 7: Closed single vane impeller (front view with coverplate removed)

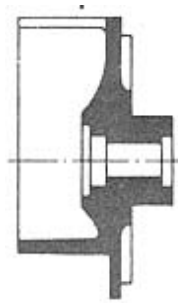
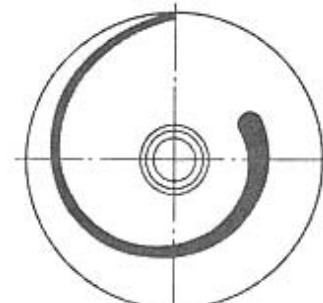
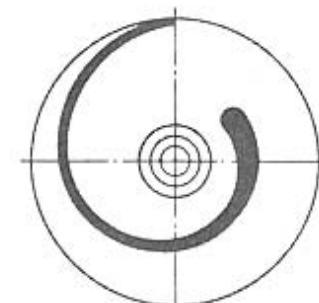


Fig. 8: Open single vane impeller



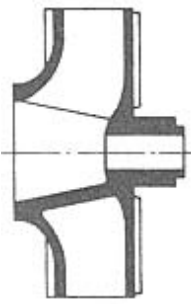


Fig. 9: Closed single-passage non-clogging impeller (front view with coverplate removed)

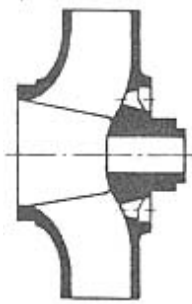


Fig. 10: Closed two-passage non-clogging impeller (front view with coverplate removed)

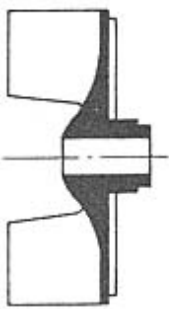
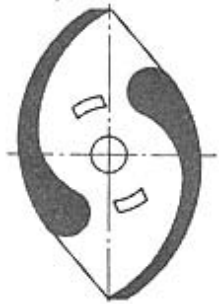


Fig. 11: Open two-passage non-clogging impeller with S-shaped blades

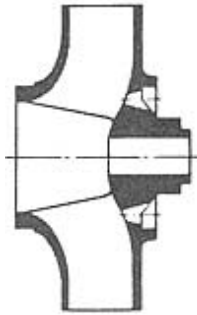
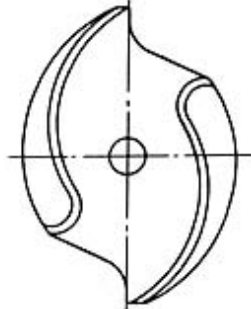


Fig. 12: Closed three-passage non-clogging impeller (front view with coverplate removed)

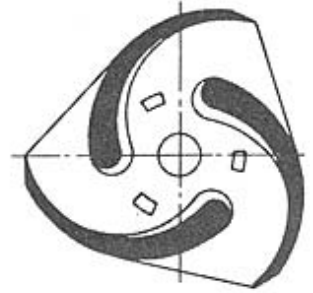


Fig. 13: Open three-passage non-clogging impeller with cylindrical blades

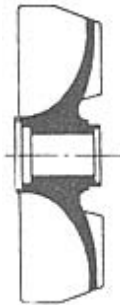
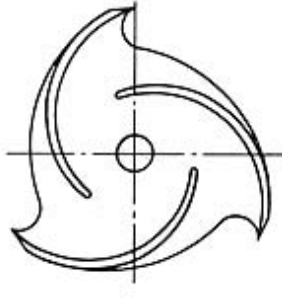


Fig. 14: Torque-flow impeller

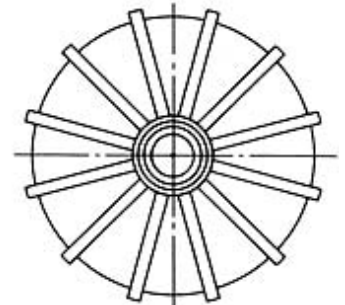




Fig. 15: Peripheral impeller

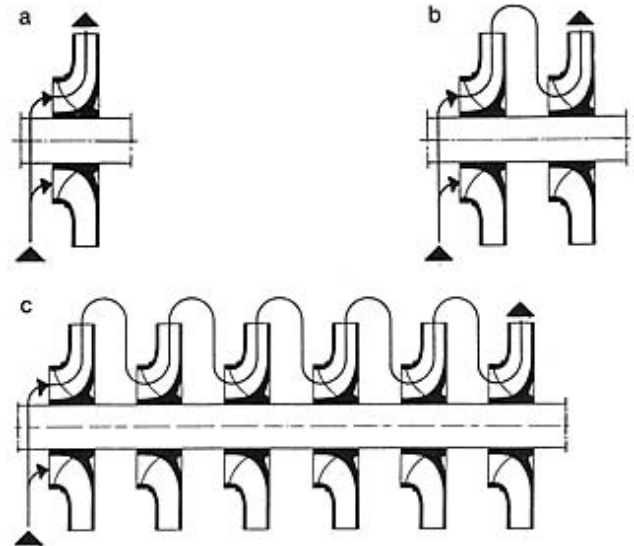
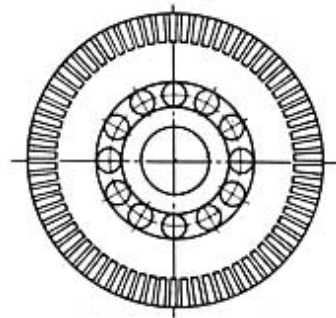


Fig. 16: Single suction impeller arrangement, with all impellers facing the same way

- a) single-stage;
- b) two stage;
- c) six stage

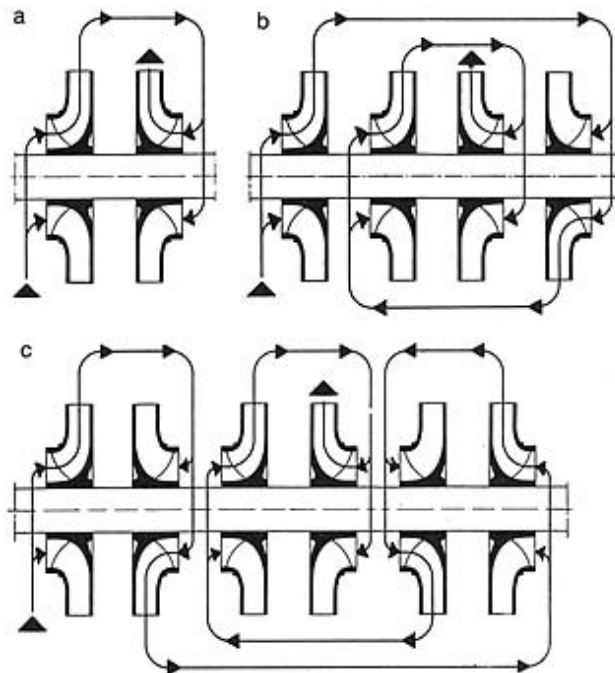


Fig. 17: Single suction impeller arrangement, with back-to-back or opposed impellers

- a) two stage, back-to back;
- b) four stage, cross-over;
- c) six stage, back-to back

In order to achieve optimal pump efficiencies and/or minimum NPSH values (net positive suction head), the i. must be provided with a given number of blades. If few blades are provided, the free or unimpeded flow cross-section through the i. is increased. This enables the i. to handle more or less contaminated liquids (sewage pump, pulp pump, hydrotransport). In practice, the number of blades of radial and mixed flow i.'s handling liquids containing sludge or solid admixtures is reduced to 1, 2 or 3 blades. Such i.'s, which can be of open or closed type, are

designated non-clogging i.'s or single vane i.'s as the case may be, see Figs. 7 to 13. Open non-clogging i.'s or single vane i.'s are used to handle liquids which liberate gas out of solution, whilst closed single vane i.'s (Fig. 7) are used to pump fluids containing very coarse solids. With regard to unimpeded flow, i.e. passage of maximum-diameter objects (e.g. in the food-processing industry: vegetables, fruit, beets), the decisive characteristic is the so-called ball passage or free, unrestricted i. passage, usually stated as a ball diameter (mm) or a rectangular dimension in the characteristic curve of non-clogging i.'s, single vane i.'s and torque-flow i.'s.

The blades of axial and semi-axial propellers (propeller pump) can be either fixed adjustable (when the pump is dismantled) or of variable pitch type (adjustable while the pump is running, impeller blade pitch adjustment). In the case of adjustable or variable pitch blades, the contour or profile of the pump casing and of the hub in the adjustment region is usually spherical. This ensures that the clearance gap width outside and inside at the hub remains constant for all blade pitch adjustment angles.

The torque-flow i. (Fig. 14) and the peripheral i. (Fig. 15) represent special type i.'s; they are fitted in torque-flow pumps and peripheral pumps respectively.

The question of which i. type should be selected for a given capacity Q at a given head H when designing a pump is an open question in the first place. The arbitrary selection of an axial, mixed flow, radial or peripheral i. is however drastically restricted by the fact that the rotational speed n and the size of the i. diameter D which will result from the selection must not turn out to be extreme. Thus the achievement of optimal pump efficiencies or stage efficiencies in the case of multistage pumps is tied up with certain well-defined i. types, depending on the specific speed:

radial i.'s	$n_q \approx$	to 60 min ⁻¹ ,
mixed flow i.'s	$n_q \approx$	60 to 160 min ⁻¹ ,
axial i.'s	$n_q \approx$	160 to 400 min ⁻¹ and over

Depending on the pattern of the flow path and the arrangement sequence of the i.'s on the pump shaft, the following i. arrangements are prevalent; single stage, multistage; single suction, multisuction; all i.'s facing the same way, or back-to-back or opposed. Typical i. arrangements are illustrated in Figs. 16 to 18 (multistage pump, multisuction pump).

In accordance with EUROPUMP-TERMINOLOGY and DIN 24250 a distinction is made between a left-handed and a right-handed impeller; accordingly, a right-handed impeller is one that rotates clockwise viewed in the direction of inflow.

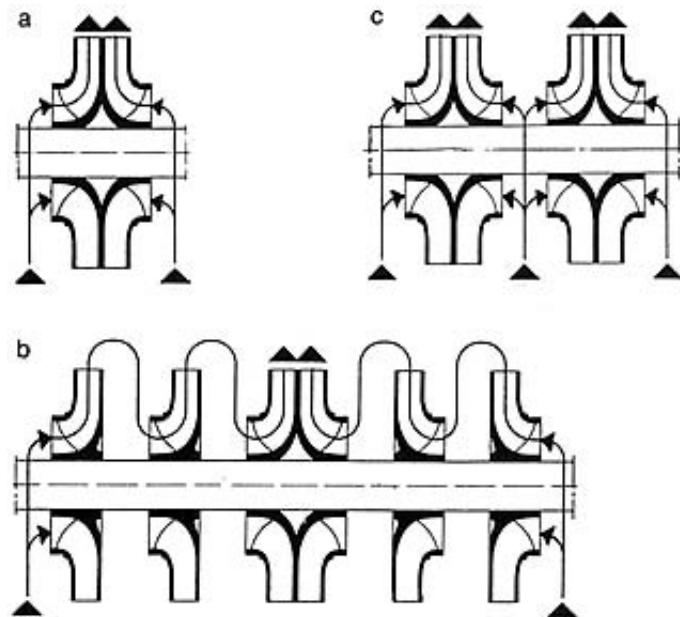


Fig. 18: Multisuction impeller arrangements, with back-to-back impellers

- a) double suction single stage;
- b) double suction, three stage;
- c) quadruple suction single stage

Impeller Blade Pitch Adjustment

Laufschaukelverstellung

Réglage des aubes de roue

I.b.p.a. is an internal device limited to propeller pumps fitted with axial or mixed flow impellers, which enables the pitch angle of the blades to be altered while the pump is running. Such an adjustment at constant rotational speed alters the capacity, head and shaft power (control); thanks to i.b.p.a. an optimum control with low losses can be achieved, but at the price of an expensive i.b.p.a. construction.

The prevalent method of i.b.p.a. in centrifugal pumps is by means of an axially sliding adjustment rod arranged inside the hollow pump shaft, which, in the case of large cooling water pumps has to absorb adjustment forces up to 600 kN and even higher in certain cases. The axial sliding motion of the adjustment rod is shifted axially via a mechanical screw thread drive (Fig. 1) or via a hydraulic piston. Manual adjustment via a reduction gear is sometimes found on smaller propeller pumps. The axial movement of the adjustment rod is converted into a rotating movement of the blade trunnions by the adjustment gear (Fig. 2) in the hub of the impeller (hub ratio), and hence of the blades themselves. Other known types of adjustment gear include those with a hydraulic motor or electric positioning motor built into the impeller hub for the blade pitch angle adjustment, and adjustment gears which convert a twisting motion of the adjustment rod into a twisting motion of the blade trunnions.

The forerunners of i.b.p.a. in centrifugal pumps are to be found in ship propeller design and in hydraulic turbine design.

In order to avoid clearance gaps (clearance gap width, clearance gap loss) which would reduce efficiency, the outer profile (flow profile) and the inner profile (hub profile) of the adjustable blade must lie on concentric spherical surfaces. This requirement governs the shape of the pump casing and of the impeller hub within the flow space of the impeller. This spherical shape which is necessary for design reasons is not always a hydraulically favourable shape. Blades which can only be adjusted in respect of pitch angle when the rotor is dismantled suffer from the same disadvantage.

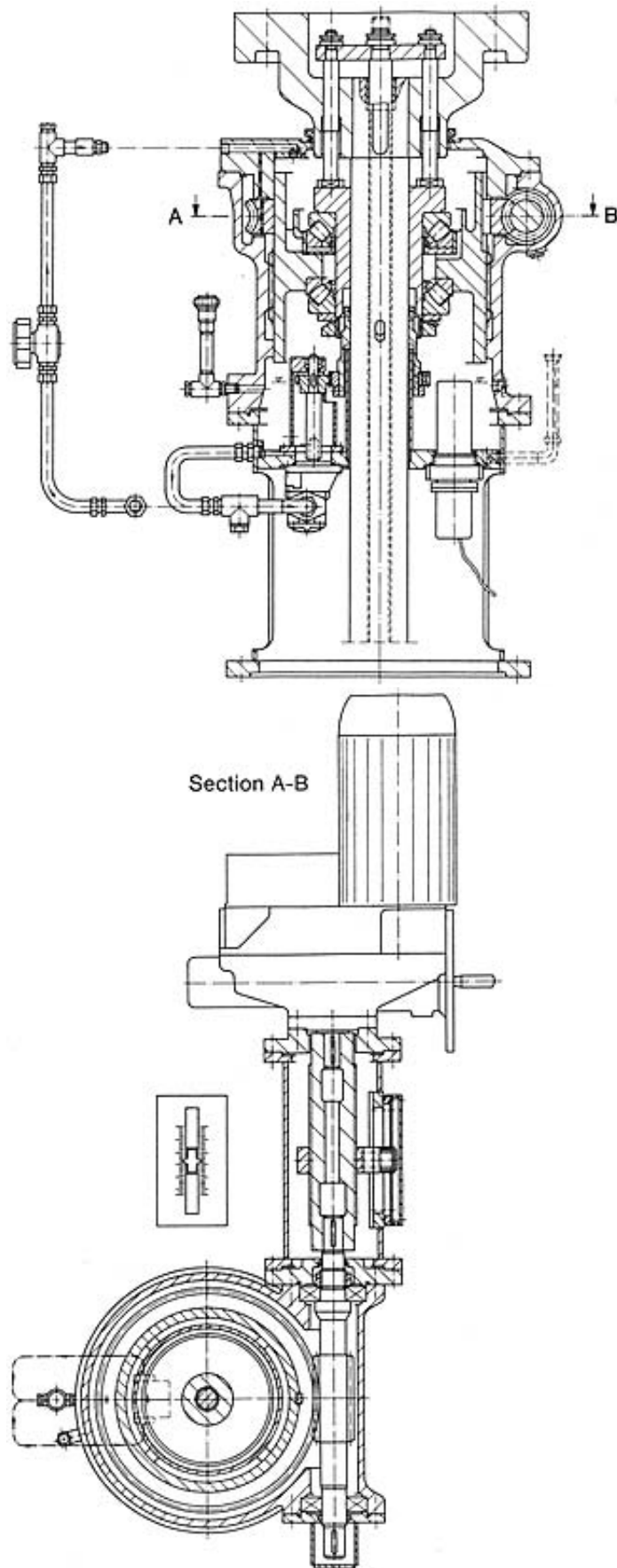


Fig. 1: Actuating gear for adjustment rod fitted inside the pump shaft

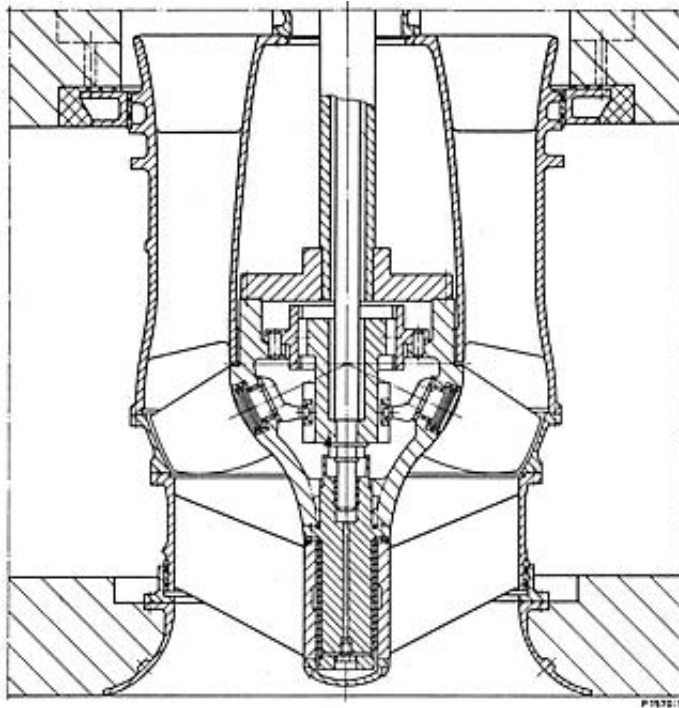


Fig. 2: Mechanical adjustment gear for pitch angle adjustment of impeller blades on a semi-axial propeller pump

Impeller Side Friction

Radseitenreibung

Frottement Flasques de roue

I.s.f. in centrifugal pump technology is the friction loss caused by the rotating medium between the impeller shrouds and the pump casing. The loss of power caused by i.s.f. is the dominating loss among the so-called external losses (bearing loss + seal loss + i.s.f.) (internal efficiency). Because these friction losses arise through rotation of the fluid between the outside surfaces of the impeller and the casing walls, the i.s.f. is greater for a closed impeller than for an open impeller with only one shroud.

The i.s.f. loss $P_{v.Rads.}$ is:

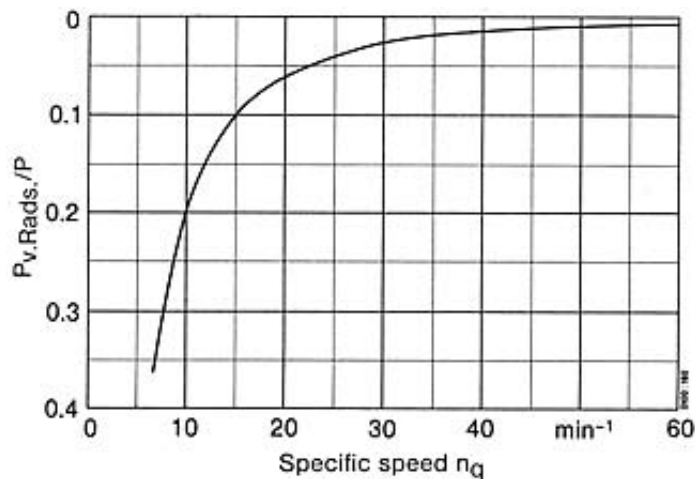
$$P_{v.Rads.} = c_M \cdot \rho \cdot (u_2^3 \cdot D_2^2 - u_1^3 \cdot D_1^2) = c_M \cdot \rho \cdot n^3 \cdot \left(\frac{\pi}{60}\right)^3 \cdot (D_2^5 - D_1^5)$$

with

c_M	moment coefficient,
ρ	density of pumped medium,
u_1, u_2	circumferential velocity of impeller at D_1 or D_2 ,
D_1	inside diameter of impeller shroud,
D_2	outer diameter of impeller,
n	rotational speed of impeller.

c_M is a coefficient which takes into account the influences of the REYNOLDS number (model laws), of the surface roughness and of the side space geometry. As the equations show $P_{v.Rads.}$ changes at constant circumferential velocity with the square of the impeller outer diameter, and at constant rotational speed with the fifth power of the impeller outer diameter. If the impeller outer diameter and the rotational speed are constant, the ratio of $P_{v.Rads.}$ over shaft power P is a function of the capacity Q , i.e. a function of the specific speed n_q (see illustration).

It has been demonstrated experimentally that the difference in surface roughness between a rough (cast) impeller disc and a polished one reduces the i.s.f. loss $P_{v.Rads.}$ by 30% approx. The influence of the side space geometry can also make a difference in the friction loss of up to 10%. The changes achieved by surface roughness and side space geometry should always be considered in conjunction with the specific speed. The illustration indicates that in the n_q region from 40 to 60 min^{-1} the losses causeway i.s.f. only amount to between 1.5% and 0.7% of P . On from $n_q = 30 \text{ min}^{-1}$ approx. downward, the friction loss increases rapidly with diminishing specific speed. $P_{v.Rads.}$ amounts to about 6% of P at $n_q = 20 \text{ min}^{-1}$ and to about 20% at $n_q = 10 \text{ min}^{-1}$. This means that the surface roughness and side space influences can hardly be measured for $n_q > 40 \text{ min}^{-1}$ but that a marked improvement at low specific speeds can be achieved by smooth impeller disc and casing surfaces. Because $P_{v.Rads.}$ increases with the fifth power of the impeller outer diameter, it will suffice to machine only the outside portions (from $0.7 \cdot D_2$ outward, approx.).



Power loss due to impeller side friction (related to shaft power) in function of specific speed

Impeller Vane

Laufschaufel
Aube de roue

see [Blade](#)

Incipient Cavitation

Incipient Cavitation
Cavitation naissante

see [Net Positive Suction Head](#)

Incrustation

Inkrustierung
Incrustation

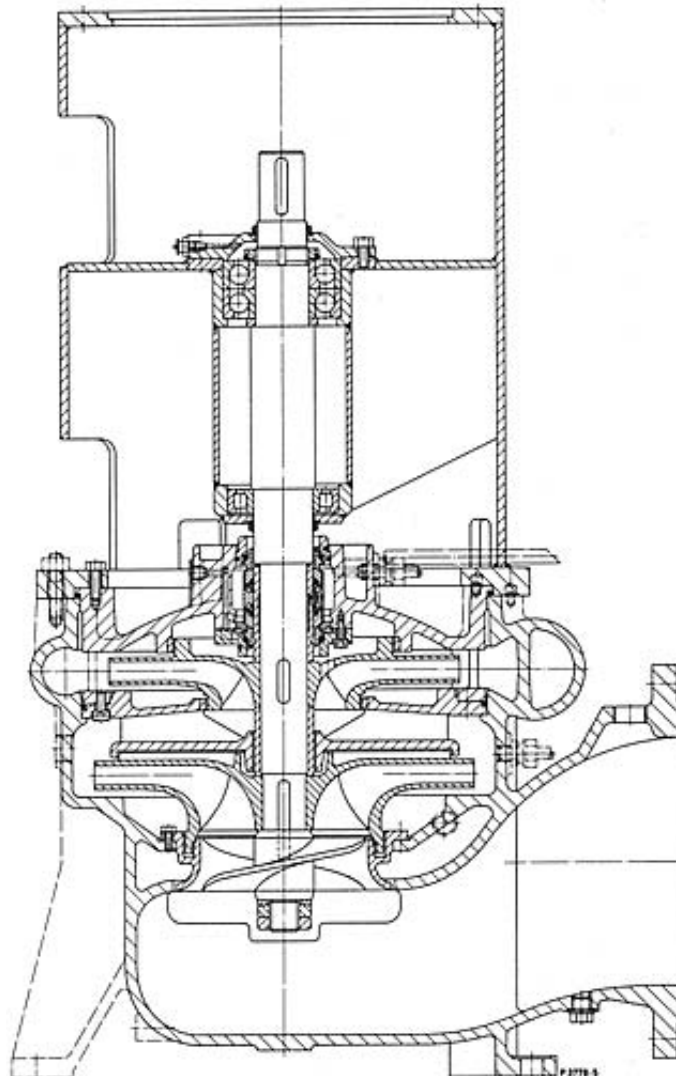
see [Pressure Loss](#)

Inducer

Inducer, Vorsatzläufer
Inducer

The i. is an axial impeller arranged immediately upstream of the first stage of a centrifugal pump, rotating at the same rotational speed as the pump impeller. The object of the i. is to reduce the NPSH value (net positive suction head) of the pump; this is accomplished by the i. increasing the static pressure upstream of the first stage impeller. The vortex flow (pre-swirl in the same direction as the impeller rotation, inlet conditions) downstream of the i. also plays a part in reducing the NPSH value of the pump (net positive suction head). The illustration shows a vertical volute casing pump fitted with an i., used as a general service pump on board ship.

In order to avoid blockage of the flow cross-sections in the i. by vapour bubbles (cavitation) on the pressure sides of the blades in the overload region (operating behaviour), the i. is preferably sized for 1.5 to 2.3 times the design capacity (design duty point) of the pump. Therefore the i. runs in the part load region for all the operating points of the pump which occur in practice, and supplies the necessary boosting pressure to the pump impeller (characteristic curve). The flow separation which may occur in certain circumstances at part load operation on the suction sides of the blades is far less objectionable than a flow separation on the pressure side, which, in the case of axial impellers usually leads to the collapse of the head of the pump.



Vertical volute casing pump (marine pump) with inducer

The head of the i. is an important magnitude in the evaluation of the pump efficiency of a pump fitted with an i. The i. operates under conditions of a considerable entry shock (impact) (shock loss at part load operation), and therefore at a lower efficiency than the pump impeller. The ratio of head of the i. to head of the pump impeller is important, because the deterioration in the efficiency of the complete unit (consisting of i. and pump) depends to a

considerable extent on this ratio. It follows that the provision of an i. on pumps of low specific speed, i.e. of high head is less detrimental to the efficiency than in the case of high specific speed pumps.

There are three main shapes of beading (blade) used on i.'s:

1. *helical surface*, in this case the head is generated solely by the incidence of the blades (used for i.'s with a low head);
2. *cambered blades* with various types of meridian line; the head is generated both by incidence and deflection;
3. *S-curved blades*, which shape the path of the flow deflection (vu path, velocity triangle, fundamental equation of fluid dynamics, vortex flow).

A small number of i. blades (two to three) have given the best results in practice, because of the lesser risk of blockage by vapour bubbles.

I.'s are exposed to very high stresses from cavitation, and must be manufactured from cavitation-resistant materials.

The following manufacturing methods are generally used for i.'s:

1. welding; steel plate blades pre-formed (bent) according to templates are welded onto a machined hub;
2. numerically controlled (NC) milling; in this case the three-dimensional (space) coordinates of the i., obtained by calculation, are processed in an EDP installation and fed to the NC machine tool in the shape of a punched tape;
3. copy milling; a roughly pre-cast i. is copy-milled from an accurate model;
4. casting; the i. is cast by the investment casting method.

Inlet Conditions

Zulaufbedingungen
Conditions d'aspiration

In order to ensure trouble-free running of a centrifugal pump, certain conditions relating to the disturbance-free approach flow to the impeller must be satisfied, amongst other things. Pumps of high specific speed (impeller) are more sensitive to disturbances in the approach flow than pumps of low specific speed. The transmission of energy from the blades to the medium pumped is based mainly on the centrifugal effect, which is not greatly susceptible to disturbances, in the case of low specific speed radial pumps, whereas in the case of high specific speed propeller pumps it is based on the flow deflection at the blades, which is highly susceptible to disturbances. Depending on the type of impeller involved, the conditions for disturbance-free approach flow must therefore be strictly complied with. Given the two assumptions that the approach flow is steady (fluid dynamics, steady flow) and that the NPSH value (net positive suction head) is adequate to avoid damaging cavitation in the pump, three important i.c.'s follow:

1. *Freedom from (rotational) swirl*. The swirl (vortex flow) at the impeller inlet, in so far as it is not created for the purpose of controlling the head (control) or improving the suction behaviour of the pump (inducer) represents a disturbance of the ideal approach flow to the impeller. Flows at the inlet cross-section of the pump which exhibit a swirl arise in most cases as a result of asymmetries in the approach (transverse approach flows, flows across an elbow, asymmetrical flow separation) and of part load eddies (operating behaviour). The axis of the vortex does not coincide with the axis of the pump in the case of pronounced asymmetries where the vortex is formed.

As a swirl in the same direction of rotation as that of the pump increases, i.e. as the peripheral components of the swirl flow become larger, the head and pump efficiency decrease at constant capacity. The reason is the reduced deflection effected by the impeller vanes in comparison with their capability at the design duty point. On the other hand, if there is an increasing swirl in the opposite direction to the direction of impeller rotation, the head will increase at constant capacity, to the point where the blades are overloaded (flow separation on the suction side of the blade, mechanical vibrations); the pump efficiency will drop even faster than in the case of a same-direction swirl.

The measurement of the "swirl" disturbance magnitude in the approach flow is effected by velocity measurements (measuring technique) with regard to both magnitude and direction, by means of probes or (mainly in model techniques in the case of concentric swirl distribution) by "rotameters" arranged in the suction branch, i.e.

speed-monitored paddle wheels of the size of the impellers. The rotameter has a very low frictional loss thanks to its bearing design, and its flat vanes (paddles) which have no angle of incidence in relation to a swirl-free flow intersect each other at the axis of rotation.

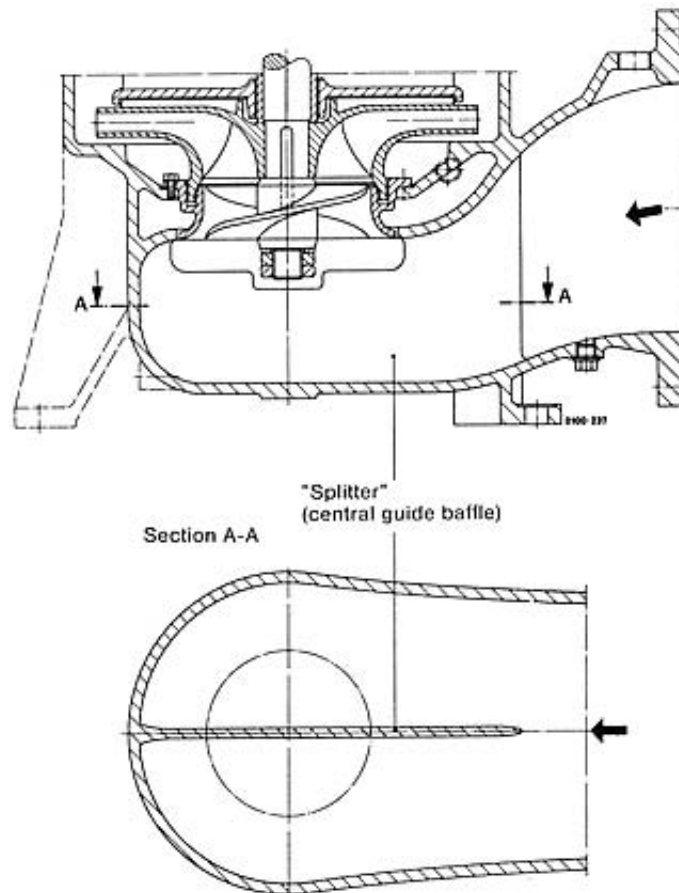


Fig. 1: Intake chamber upstream of radial centrifugal pump designed for disturbance-free approach flow to the impeller (here marine pump with inducer)

Honeycomb flow straighteners are best suited to reduce existing swirl components, but simple baffles in the shape of vanes (Fig. 3), cruciform shaped straighteners (Fig. 2; Fig. 1 under entry nozzle) and central longitudinal baffles ("splitters" Figs. 1 and 4) also produce good results. Complete absence of swirl at the approach to the pump is practically impossible to achieve.

On the basis of tests on pumps equipped with pre-rotational swirl control (specific speeds n_q from 70 to 200 min^{-1} approx.), peripheral components of the flow velocity at the outer edge of the suction branch which amount to less than 7 % of the corresponding axial components do not represent an appreciable disturbance.

The above percentage rate must be reduced in the case of pumps with specific speeds exceeding 200 min^{-1} , because of the need to comply more strictly with the i.c.'s.

2. Uniform velocity distribution. When designing a pump impeller, a uniform velocity distribution in the cylindrical portion of the suction pipe is generally taken for granted. This is understood as comprising all the profiles of the axial flow velocities from the rectangular profile to the profile for fully developed turbulent flow in the pipe (fluid dynamics). Distortions of the uniform velocity distribution pattern occur mainly as a result of flowing around obstacles (inserts) in the path of the flow (wake depressions), and from flow deviations and flow separation of all kinds (Fig. 2 under intake elbow).

The greater the deviation from uniform velocity distribution, the less likely is the pump capable of attaining the required performance data, because the individual blade is exposed to an approach flow under part load or overload conditions (operating behaviour) in the region of the velocity distortion. If the non-uniformity of velocity distribution is not rotation-symmetrical, mechanical vibrations will occur as a result of the unsteady flow along the blades. Resistances in the shape of screens or perforated plates arranged uniformly across the flow cross-section will exercise a smoothing action on a distorted velocity distribution (see VDI guideline 2040, Page 1); the same effect is achieved by an undisturbed length of piping of sufficient length.

If such a device causes the NPSH_{av} of the installation (net positive suction head) to sink to an unacceptable level, making cavitation more likely, another effective (but often very expensive) remedy consists in improving the approach flow space from the hydrodynamic aspect. Widening of this space will reduce the magnitude of the velocity peaks, and lengthening will reduce the wake depressions; deviations and flows around obstacles should be avoided or their effects minimized.

Fluctuations of the local axial velocity which are not rotation-symmetrical and which exceed the volumetric average value by more than 10% are generally considered unacceptable. In the case of pumps of very high specific speed ($n_q > 200 \text{ min}^{-1}$), fluctuations of the order of 5% may already prove to be unacceptably high.

3. Freedom from air-entraining entry vortices. The air-entraining entry vortex, also called hollow vortex, arises when aspirating from an open-surface reservoir, in the immediate vicinity of the surface of the water. The precondition for the formation of a hollow vortex is the local rotation of the medium at the surface (e.g. as a result of strong shear flows) and too low a flooding level of the entry nozzle by the medium pumped (Fig. 3). Starting with a funnel-shaped depression at the surface of the suction water, a tube-shaped air cavity forms instantaneously, extending from the surface to the pump impeller, and situated eccentrically in the suction pipe.

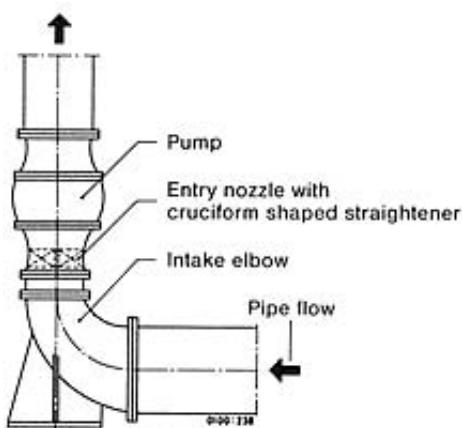


Fig. 2: Intake elbow for disturbance-free approach flow from a 90° flow path deflection

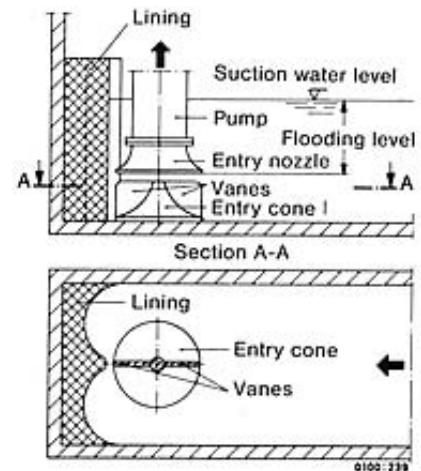


Fig. 3: Open, lined intake chamber

The disturbance effect of the air-entraining entry vortex on the pump performance data stems on the one hand from the absence of homogeneity of the pumped medium (non-attainment of the operating point, mechanical stressing of the impeller due to "chopping up" of the hollow vortex), and on the other hand from the induction of undesired swirl components (see freedom from swirl, point 1.). The asymmetry of the swirl components also leads to unsteady flow around the blades (mechanical vibrations).

The following measures can be taken to avoid the formation of air entraining entry vortices (intake chamber): increasing the depth of flooding of the entry nozzle (Fig. 3); covering over of the water surface exposed to air entraining vortices by means of a raft, floating balls, etc; installation of swirl-proventive baffles in the area of the suction water surface exposed to air entraining vortices. The freedom from air entraining entry vortices is the most important pre-condition for disturbance-free continuous operation of the pump.

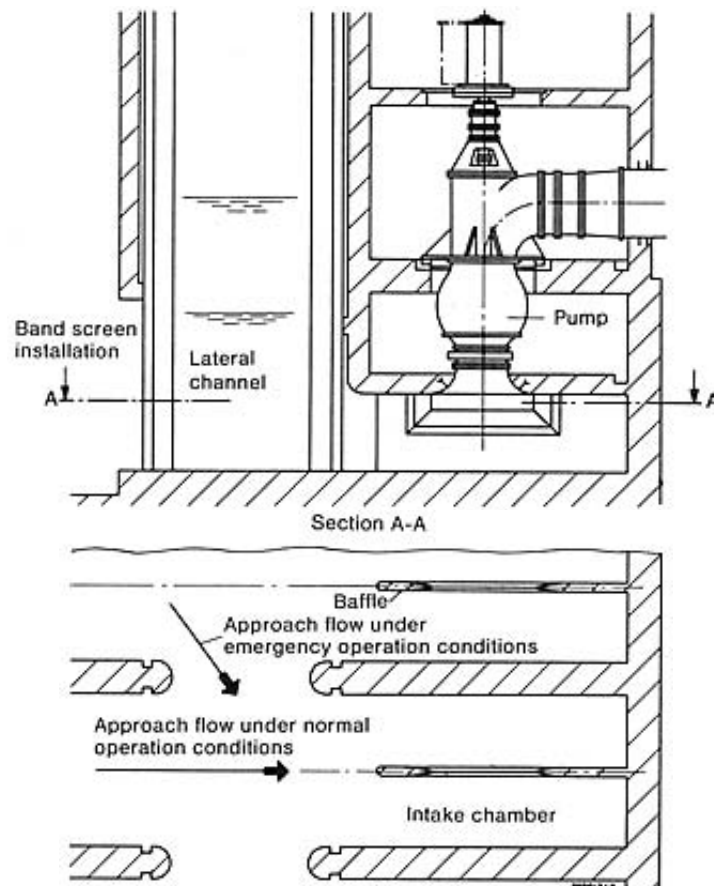


Fig. 4: Covered intake chamber with central baffle ("splitter")

Examples of a disturbance-free approach flow. Fig. 1 illustrates the shaping of the approach flow space designed to ensure a disturbance-free approach flow to the first stage impeller of a condensate pump. The approach flow compartment is relatively wide and long (deflection takes place at low velocities) and is in addition equipped with a longitudinal baffle ("splitter") designed to prevent the formation of appreciable swirl components.

Fig. 2 illustrates an intake elbow of circular cross-section for a tubular casing pump. A disturbance-free approach flow will be achieved if the velocity in the 90° bend (deflection zone) is increased to about two to four times the flow velocity in the pipe upstream of the elbow. Such intake elbows have also been successfully constructed of concrete with transitions from a rectangular cross-section to a circular cross-section (Fig. 6 under cooling water pump).

Figs. 3 and 4 illustrate recommended types of intake chambers (pump sump) for a disturbance-free approach flow to the pumps. The design illustrated in Fig. 4 ensures that the pump impeller will still receive a disturbance-free approach flow even if the flow into the intake chamber comes from one side (emergency operation in the event of failure of the traveling screen).

Inlet Cross-Section

*Eintrittsquerschnitt
Section d'entrée*

The i.c.s. of a pump is usually the flow cross-section at the pump suction branch. If there is no suction branch, the i.c.s. must be defined as a flow cross-section on the suction side with known static and hydraulic data (net positive suction head).

The i.c.s. of the plant must be mutually agreed, e.g. as a characteristic cross-section through a suction vessel or as a pipe cross-section on the suction side with known data (for examples, see DIN 24260).

Compare with outlet cross-section, see Fig. 2 under head.

Inline Pump

Inlinepumpe, Rohrleitungspumpe
Pompe in-line

I.p., centrifugal pump, the pump suction branch and pump discharge branch of which lie in a straight run of piping. Small i.p.'s are mounted in the form of piping pumps (see illustration). Wellknown i.p.types include: I.p.'s with screwed connection (screwed pump), i.p.'s with flanged connection (flanged pump) and i.p.'s with welded connection (welded-in pump).

For other forms of i.p.'s see Fig. 9 under pump casing or Fig. 1 under tank farm pump. The socalled U-turn pump represents an alternative to the i.p.; on this pump, the suction and discharge branches are arranged in parallel close to one another (Fig. 10 under pump casing).



Inline pump (hot water heating circulating pump)

Installation

Installation
Installation

see Installation of Centrifugal Pumps

Installation of Centrifugal Pumps

Aufstellung von Kreiselpumpen
Montage de pompes centrifuges

The concept "i.o.c.p." encompasses the arrangement of the pumping set (pump and drive) at site, together with all piping connections necessary for commissioning. This presupposes that the machine is so installed that all external forces and moments (branch loading) are safely carried by the foundation (pump foundation), supports, frames, plates, or by the piping itself.

Depending on the pump type, and frequently also on the application field for the pump, one can classify the type of installation into categories, viz.: installation on a foundation, or installation without foundation, wet installation or dry installation, installation indoors or in the open.

In the case of *installation on a foundation* (pump foundation) it is necessary to distinguish between horizontal and vertical shaft pumps.

Horizontal pumps and drives are supplied mounted on a combined baseplate (pump foundation) with the exception of close-coupled pumping sets. In the case of large pumping sets, or pumping sets with gearboxes or booster pumps, individual nameplates are used for the various units. The nameplates are placed on the foundation and the cavities between the two are filled in with grout, e.g. cement mortar; the foundation bolts should only be pulled tight after the grout has set firmly. This grouting-in provides the necessary rigidity for the baseplate to prevent any undue deformation or warping under load, e.g. by forces from the piping. Thereupon the couplings of the various units of the pumping set are aligned (coupling alignment), by placing shims under the feet of the machines on the nameplates; after adaptation and connection of the piping to the pump (without transmitting any stresses onto the latter), and after installation of the accessories (lubrication equipment, lubricating oil pump, filters valves and fittings etc.), an alignment check completes the installation of the pumping set.

Vertical pumps and drives are installed in a sequence similar to that for horizontal pumps, but there is no need to align the coupling if the pump and its driver are connected to one another by a drive stool or a lantern. In such cases, the pumping set is bolted onto the foundation by a foot flange. The drive stool determines the precise location of the driver.

Installation without a foundation is adopted in cases where the weights of the sets to be installed and the anticipated loads by the piping are limited, where the pump must be mobile or where vibrations against the floor must be insulated.

Submersible motor pumps, most inline pumps and grandness pumps (circulating pumps in particular) are installed without a foundation (the larger units are sometimes supported in addition on a simple support foundation).

Mobile pumps (e.g. fire-fighting pumps, mobile pipeline pumps) are attached directly, or via their baseplate, onto the mobile frame (vehicle or sled). This type of installation necessitates movable piping. In the case of large units, the baseplates must be designed rigid enough to keep any deformation within acceptable limits.

Portable pumps (e.g. garden pumps, cellar drainage or dewatering pumps) are designed in the form of close-coupled pumping sets which require no coupling alignment. They require no foundation, as they are always small pumps attached to flexible hoses and not to fixed piping. The pump casing (or the motor casing in certain instances) is designed for placing on any reasonably even and firm surface (base).

Foundationless nameplates are sometimes specified for acid pumps, to enable the easy removal of any corrosive leakage fluid from under the baseplate. Such nameplates must be designed rigid enough to keep any deformation within acceptable limits.

Particularly in the case of vertical pumps, a distinction must be drawn between wet installation, where the fluid pumped wets the outside of the pump casing as well as the inside, and dry installation, where the outside of the pump casing remains dry.

The advantage of wet installation resides in the lower installation and structural costs: the pump is submerged directly in the fluid to be pumped, e.g. in the case of submersible pumps (borehole shaft driven pump), the submersible motor pumps or most tubular casing pumps. The type of installation of tubular casing pumps can itself be subdivided into pumps where the weight of both pump and motor are carried either by a common floor level or by different floor levels of the structure, and into pumps with a discharge pipe placed above floor level or below floor level of the pump pit.

If it is necessary to carry out an external check of the pump (e.g. of the shaft seals) at regular intervals, the pump must be installed dry.

Dry installation has the advantage of allowing the accessible pump room to be used for other purposes as well (e.g. for the installation of other machines). This applies e.g. to most vertical marine pumps, whether pedestal-mounted (i.e. with the pump foot attached to a steel frame on the engine room floor, and with the pump and motor casings connected to one another by a drive stool or lantern) or bulk-headmounted, in which case the drive stool is attached to a bulkhead of the engine room. Dry installation is also adopted for volute casing pumps used in other application fields, e.g. in petrochemicals and sewage pumping (sewage pump), and for some tubular casing pumps.

Installation in the open, i.e. the installation of a pumping set with no protective building or roof surrounding it poses further requirements in addition to those described above, such as the effect of climatic conditions, including rain (protection against corrosion splashwater-protected motors), sunshine (unidirectional thermal expansion), frost, drift sand, wind forces etc. There are no generally applicable rules or aspects for installation in the open, because

conditions vary widely according to site.

Intake Chamber

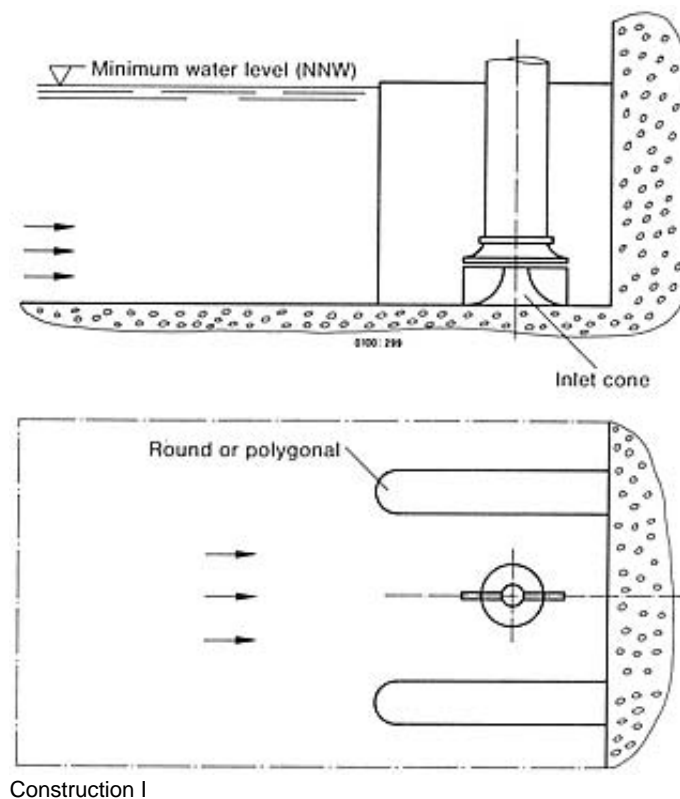
Einlaufkammer

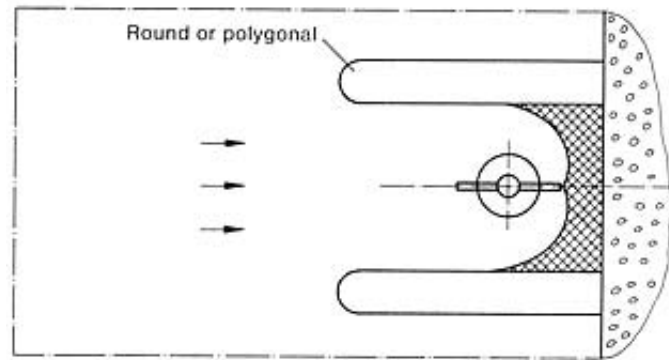
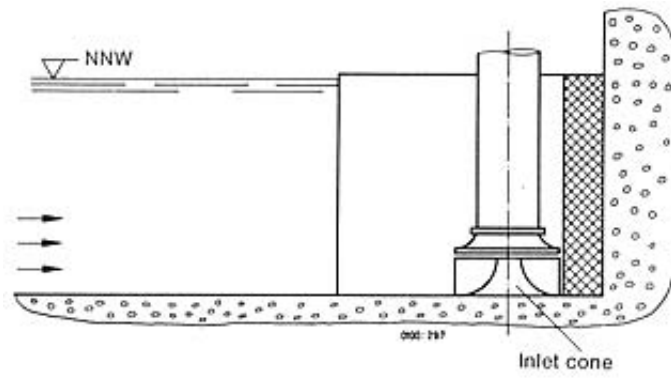
Chambre d'aspiration

The i.c. is a structure situated immediately upstream of the centrifugal pump, through which the fluid pumped, usually water, flows towards the pump. It allows an approach flow (inlet conditions) towards the centrifugal pump which is evenly balanced on all sides and is free of turbulence. Such a smooth approach flow, free of disturbance, is indispensable for high specific speed tubular casing pumps (specific speed) with propeller or mixed flow impeller (impeller), because such pumps react immediately to irregularities and disturbances in the approach flow. It only requires a very simple design of i.c. to avoid damage from vibrations and cavitation (cavitation) and a possible drop in pump output and efficiency of the pump, which would occur in the case of a disordered approach flow.

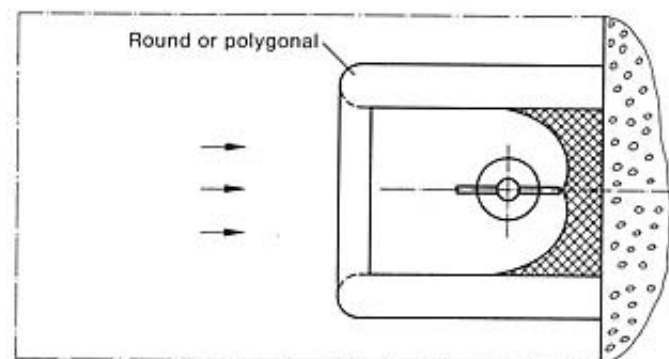
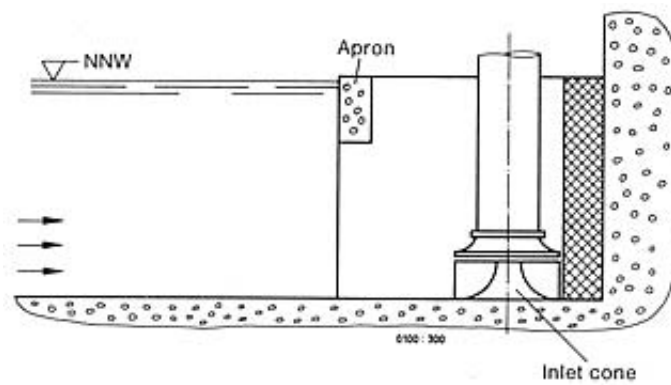
The risk of aspirating air whirls from the water surface (inlet conditions) is avoided by sufficient water levels in the i.c. The figure illustrates a choice of well-proven i.c. in four different designs. Given an identical capacity, construction shape I requires the highest minimum water level, construction shape IV the lowest. In this respect the required excavation depth depends on the construction shape of the i.c. construction shapes I, II, III; open i.c., suitable for axis parallel approach flow. The i.c. has a simple structural shape with a rectangular floor plan. In case of complicated inlet condition i.c. model tests are advisable.

A disturbance-free approach flow (inlet conditions) can also be achieved by means of intake elbows. The construction shape of an intake elbow is however more complicated than that of an i.c., and it is often necessary to excavate deeper in order to accommodate an elbow. The decision as to whether to provide an i.c. or an intake elbow requires a calculation of the economics. I.c.'s are often built for vertical-cooling water pumps ($n_q > 70 \text{ min}^{-1}$, specific speed). The operational reliability of the pumps is of paramount importance in power stations, in view of the operational availability of the power station; hence the i.c. is a structural unit which must be executed with great care. The simple type of i.c. is also used in irrigation and drainage stations, where it plays a conspicuous part in the economical design of the overall civil engineering structure. The i.c. is also often referred to as "pump sump".

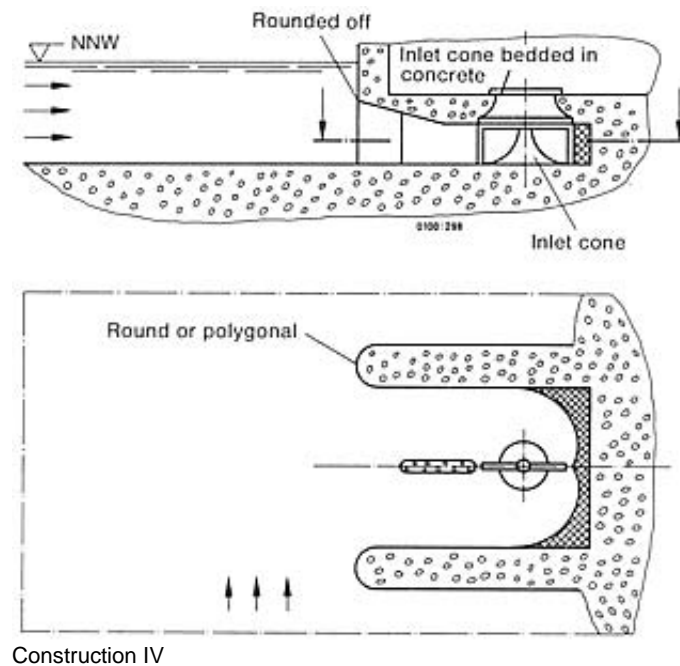




Construction II



Construction III



Four different types of intake chamber

Intake Elbow

Einlaufkrümmer
Coude d'aspiration

In many cases the medium flows into the volute casing pump through an i.e. (also called a suction elbow) of 90°. Usually cast or welded elbows with a constant diameter are put to this use. Impellers in pumps that run at higher specific speeds require a smooth flow of medium into the impeller eye (inlet conditions), that is often not possible behind these common elbows. Accelerating elbows, according to Fig. 1, in whose bends the medium will at least double its velocity, provide a satisfactorily smooth stream into the impeller eye (Fig. 2.). Even vertical tubular casing pumps can be equipped with these acceleration elbows (Fig. 2 under inlet conditions), however, intake chambers are preferred.

There is another problem that is found in double suction, single stage pumps. Due to the distortion of the flow pattern following a standard elbow, each half of the impeller receives flow of a different nature, which can lead to efficiency losses, cavitation, and insufficient quietness. An accelerating elbow can help here as well. An elbow with multiple turning vanes has the same results (Fig. 9 under pressure loss). Fig. 3 shows the top view of a horizontal double suction pump with such an i.e.

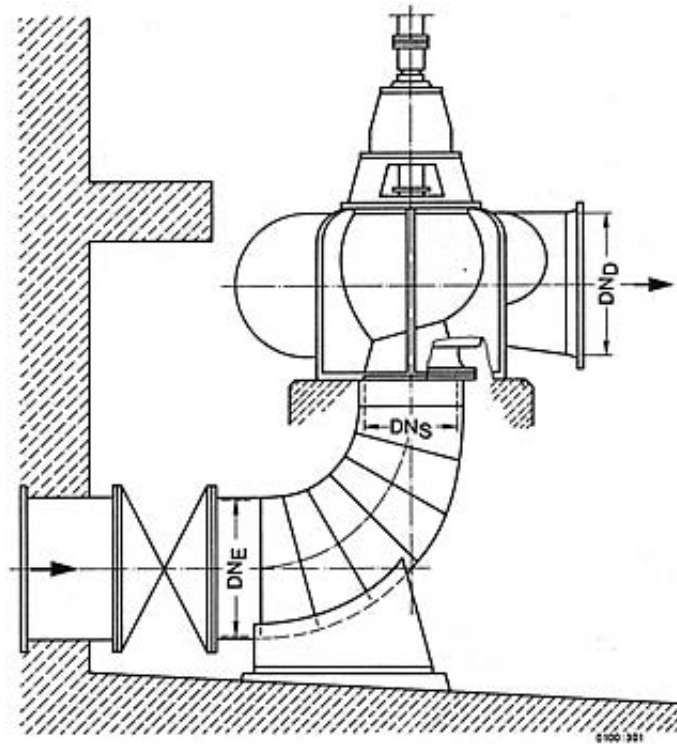


Fig. 1: Accelerating elbow in front of a vertical volute casing pump with high specific speed

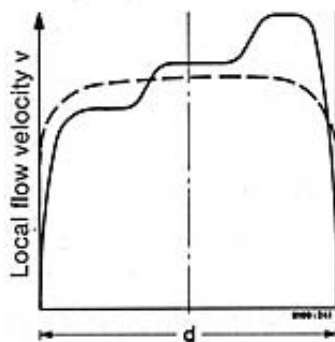


Fig. 2: Velocity profiles, — distorted, - - - smoothed

For large vertical tubular casing pumps i.e. 's are usually made of concrete, either with wooden footwork or with lost steel plate firework. In the first part of the i.e., the so-called intake funnel, the flow velocity is increased to 2 to 4 times its value at the entry, and in the deflection zone immediately downstream, the flow velocity is increased a further 2 to 4 times its value immediately upstream of the bend. These strong accelerations exert a smoothing or equalizing influence on distorted velocity profiles which have scoured as a result of disturbances upstream of the i.e. (Fig. 2, also Fig. 6 under cooling water pump).

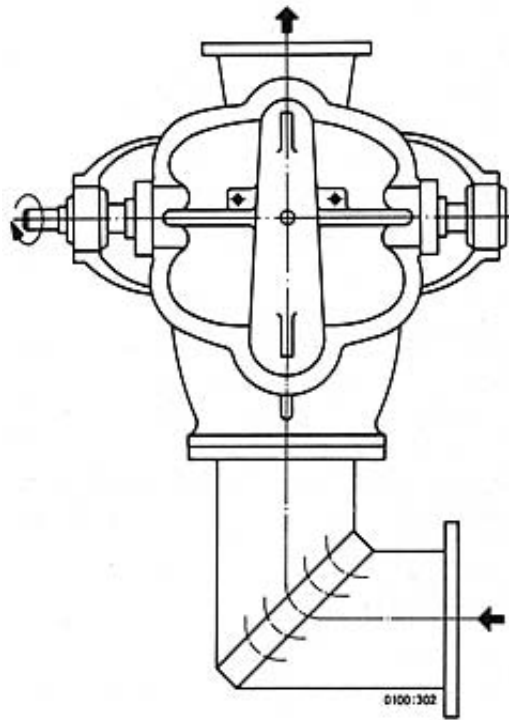


Fig. 3: Intake elbow with multiple turning vanes in front of a double suction horizontal volute casing pump (top view)

Interest Payment

Verzinsung

Païement des intérêts

see [Economics](#)

Internal Efficiency

Innerer Wirkungsgrad

Rendement interne

The i.e. η_i (efficiency) of a centrifugal pump (see DIN 24260) is the quotient of pump output P_Q over shaft power P minus the mechanical power losses P_m :

$$\eta_i = \frac{P_Q}{P - P_m}$$

P_m mainly comprises the frictional power losses of the pump bearings (plain bearing, anti-friction bearing) and of the shaft seals. The following relationship exists between the i.e., the pump efficiency η and the mechanical efficiency η_m :

$$\eta_i = \frac{\eta}{\eta_m}.$$

Efficiency re-evaluations usually relate only to the i.e.

The i.e. can also be determined on the basis of an analysis of the internal losses $P_{v,i}$ of the centrifugal pump (head). The power loss $P_{v,i}$ mainly comprises the following individual losses:

1. Losses resulting from the operation of balancing devices (axial thrust), e.g. balance discs, balance pistons, balance holes, back shroud blades.
2. Losses from impeller side friction, friction at the throttling rings and cylindrical surfaces at the impeller outlet.
3. Losses caused by internal circulation flows, in particular the secondary flows from the impeller discharge side through the throttling gap to the impeller suction side, or from the discharge side of the blades through the impeller gap (clearance gap loss, clearance gap width) to the suction side of the blades, also the reverse flows arising from flow separation (boundary layer) and from part load running (operational behaviour).
4. Losses scouring along the flow path of the fluid pumped through the impeller, diffuser and pump casing, i.e. internal fluid friction, wall friction, shock losses, momentum change losses (fluid dynamics).

With $P_Q = P - P_{i-v} - P_m$ (Fig. 1 under head) we have

$$\eta_i = \frac{P - P_{v,i} - P_m}{P - P_m}.$$

All internal losses are accompanied by a heating up of the pumped fluid amounting to the temperature difference ΔT :

$$\Delta T = \frac{g \cdot H}{c} \left(\frac{1}{\eta_i} - 1 \right)$$

where

g gravitational constant in m/s^2 ,
 H head in m,
 c specific heat in $\text{J}/(\text{kg} \cdot \text{K})$,
 ΔT temperature difference in K or $^{\circ}\text{C}$.

The heat radiation through the pump casing can be ignored in this context.

If the liquid is pumped several times in succession through the pump, as happens in test bed circuits (pump test bed), this heating up has a cumulative effect from one cycle to the next, in conjunction with the useful output (pump output) in the circuit (throttling device, piping etc.) which is also converted into heat, and cooling facilities must therefore be provided, if necessary.

Invested Capital

Anlagekapital
 Capital d'investissement

see Economics

Irrigation Pump

Bewässerungspumpe
 Pompe d'arrosage par irrigation

The duty of the i.p. is to lift water from a lower to a higher level, from which the water then flows through canals to the fields requiring irrigation (lift duty land reclamation pump), or to raise it to the required pressure head so that it can be sprinkled on the fields via piping systems (sprinkling).

The heads involved vary from 1 m approx. for normal lifting duty up to 40 m for sprinkling. In special cases, heads exceeding 100 m may be required. No general figures can be given for the capacity. This will vary according to the area of the region to be irrigated, the nature of the soil, the type of crop under cultivation and the climate. A very rough approximation of one to two litres per second and hectare can be given.

For heads up to 10 m, tubular casing pumps with axial propellers (impeller) are mainly used as i.p.'s (Fig. 1 under pump casing), nowadays often equipped with submersible motors (Fig. 1). For heads in excess of 10 m, either tubular casing pumps (Fig. 2) with helical impellers (impeller), or submersible motor pumps (as Fig. 1, but with helical impeller, see Fig. 5 under cooling water pump), or double suction (Fig. under multisuction pump) or single suction volute casing pumps (Fig. 1 under mixed flow pump) with helical impellers.

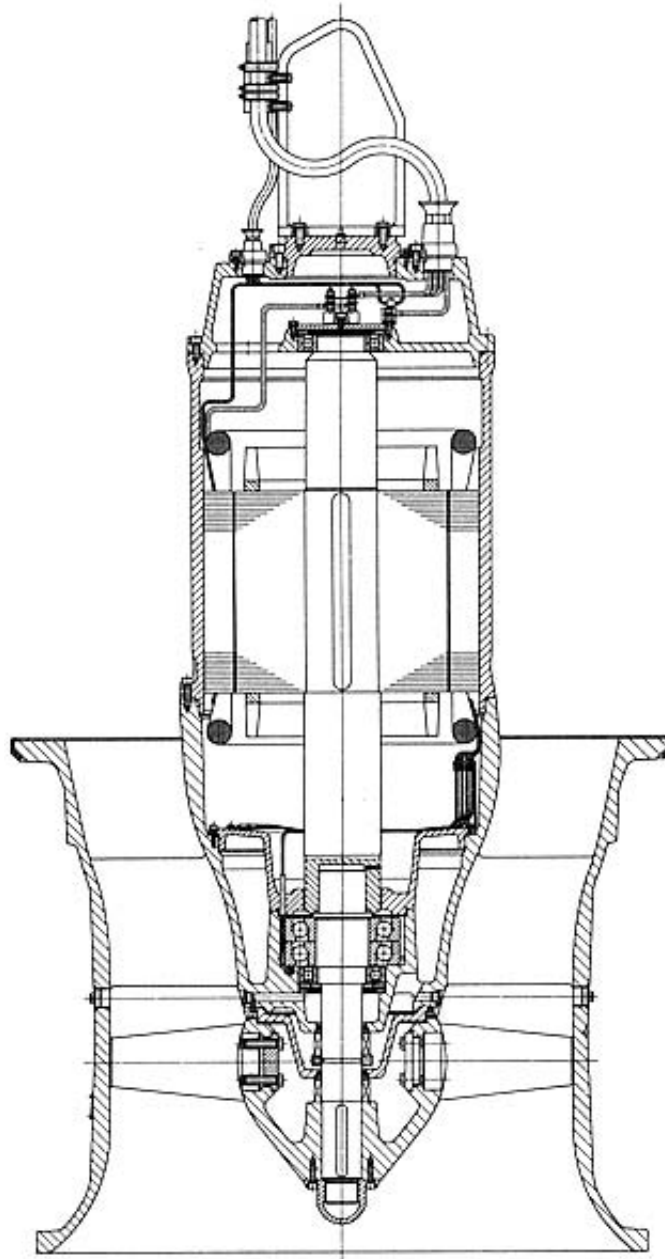


Fig. 1: Irrigation pump, maintenance-free submersible motor propeller pump with adjustable impeller vanes (compare Fig. 3 under land reclamation pump)

I.p.'s are normally not built as adjustable pumps. If it is desired to control the rate of flow (control) this is done most economically either by switching the pumps on and off or by means of a throttling valve in the discharge line (pumping plant).

Both horizontal and vertical pumps are used as i.p.'s. In the case of tubular casing pumps, the vertical arrangement predominates.

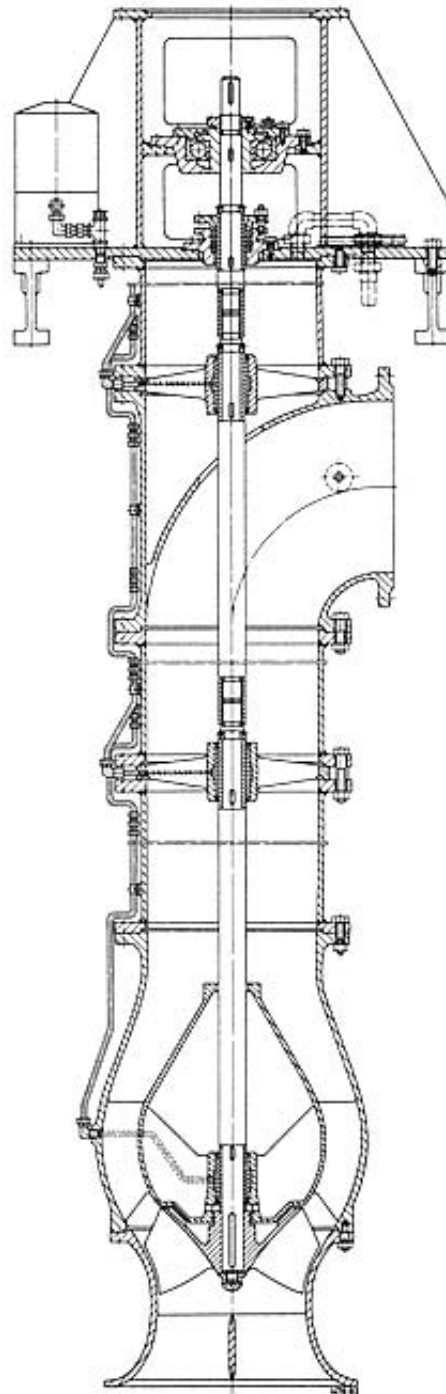


Fig. 2: Irrigation pump, tubular casing pump with helical impeller

J

Jet Pump

Strahlpumpe
Éjecteur

see Deep Well Suction Device

Jockey Pulley

Spannrolle
Poulie de tension

see Belt Drive

K

Kaplan Elbow

Kaplankrümmen
Coude de Kaplan

see [Intake Elbow](#), [Pressure Loss](#)

Kinematic Viscosity

Kinematische Viskosität
Viscosité cinématique

see [Viscosity](#)

L

Laminar Flow

Laminare Strömung
Écoulement laminaire

see [Fluid Dynamics](#)

Land Reclamation Pump

Schöpfwerkspumpe
Pompe d'épuisement

Drainage stations (Fig. 1) are pumping stations used for land drainage and/or irrigation. In coastal areas lying below the high water mark (dykes) drainage is the more prevalent duty. As a general rule, several small pumping stations spread over a catchment area supply one large main pumping station. When the gravity flow through the (lockable) dyke lock or sluiceway can no longer take place because of high water outside, the l.r.p. takes over.

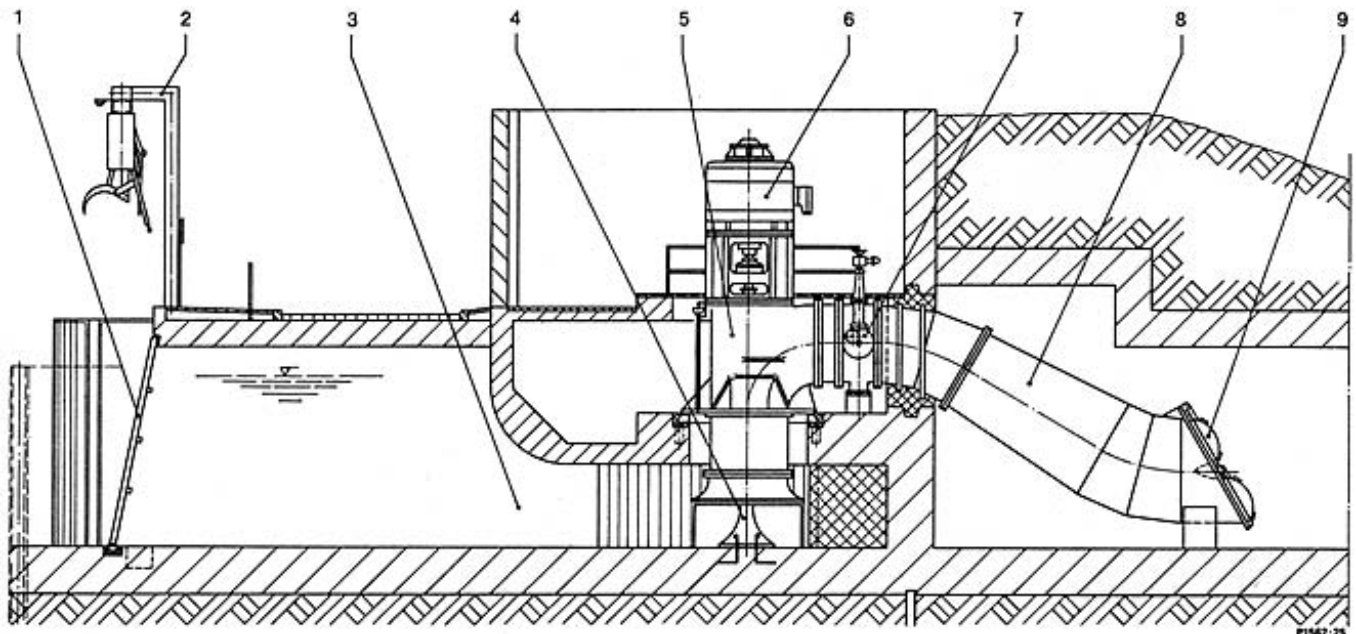


Fig. 1: Drainage station

1 trash rack) 2 trash rack cleaning machine; 3 intake chamber; 4 entry cone; 5 pump; 6 driving motor; 7 butterfly valve; 8 discharge line; 9 hollow float flap valve

In regions short of water, the irrigation duty predominates ([irrigation pump](#)). In many cases, a main pumping station situated next to a river or lake supplies several small intermediate stations via a system of ditches, and their l.r.p.'s in turn pump the water to the fields lying at a still higher level. More rarely we find a combination of both applications, in which the land is either drained or irrigated, depending on the season.

Because of the low [heads](#), [propeller](#) pumps with axial propellers are used almost exclusively. Most l.r.p.'s are installed vertically, a popular configuration consisting of a [submersible motor](#) pump (Fig. 1 under irrigation pump) in a pipe shaft (Fig. 2). In the past, [propeller pumps](#) installed at an inclination were in widespread use for large [capacities](#). Now, however, they have been extensively replaced by vertical l.r.p.'s with withdrawable rotors ([back pull out pump](#)).

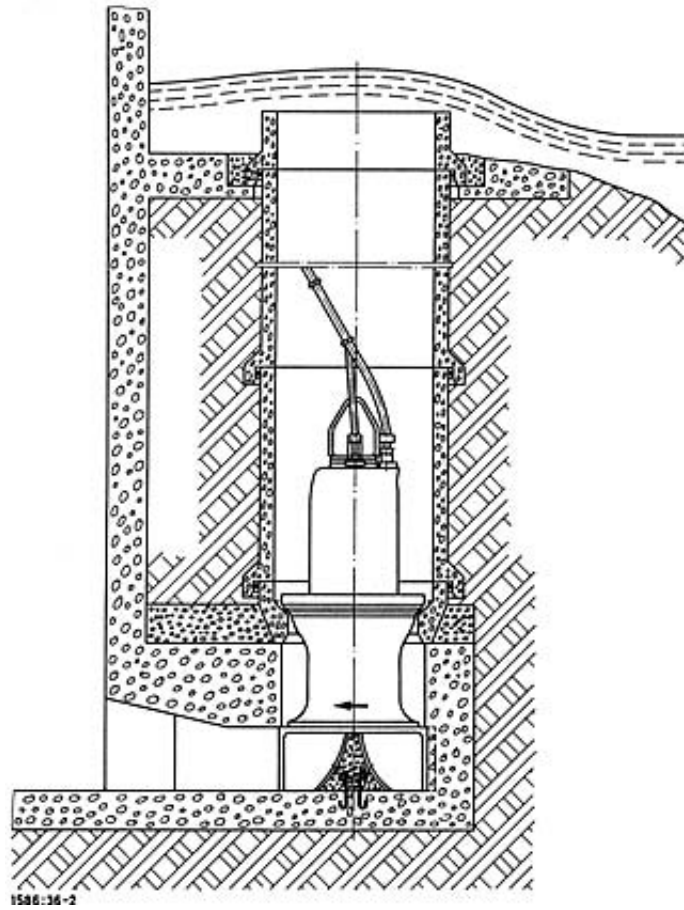


Fig. 2: Submersible motor propeller pump installed in the pipe pit as land reclamation pump

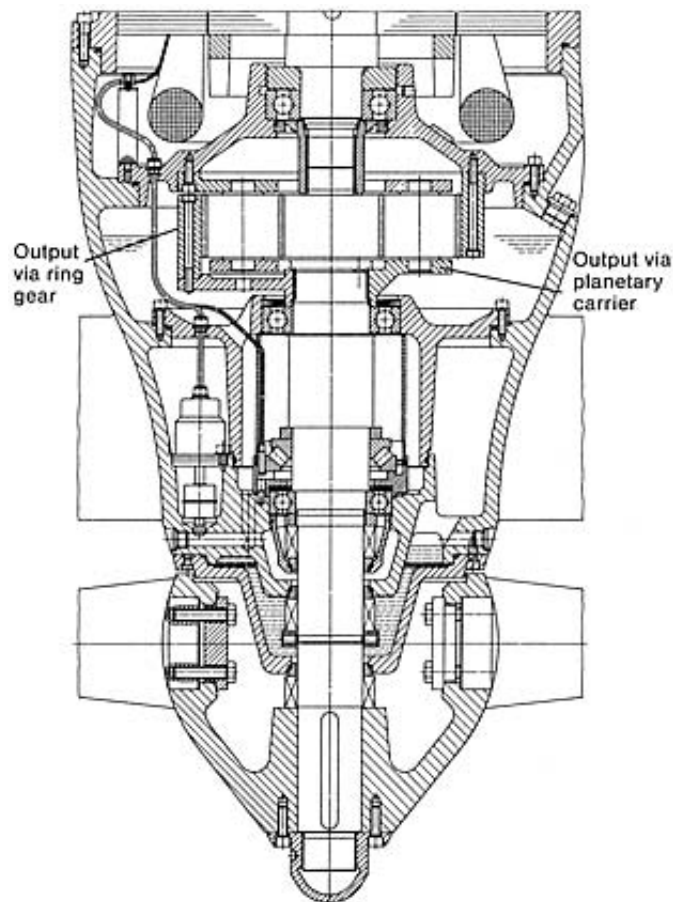


Fig. 3: Submersible motor propeller pump with integrated planetary gear. The two sides of the figure show different gear designs

L.r.p.'s are usually electrically driven (drive), either direct or via a gearbox, depending on size, this also applies to submersible motor pumps (Fig. 3). In remote regions, a Diesel engine drive is often used. Adaptation to changing capacities and changing heads (caused e.g. by high water or changing tides) is facilitated by a Diesel engine drive with speed control. In the case of pumps driven by electric motors, the larger units are often equipped with variable pitch propellers (impeller blade pitch adjustment), sometimes combined with speed adjustment (control) by pole changing motors or frequency adjustment. The data of conventional l.r.p.'s are: capacity Q from 500 to 60 000 m³/h and greater, head H from 0.5 to 8 m, nominal diameter DN of outlet branch from 300 to 2400 mm and larger.

Leakage Loss

Leckverlust

Perte fuite

L.I. is the volume flow QL escaping through the shaft seals.

Leak-Tightness

Dichtheit

Étanchéité

see Valves and Fittings

Lift-Type Check Valve

Rückschlagventil
Soupape anti-retour

see [Valves and Fittings](#)

Liquefied Gas Pump

Flüssiggaspumpe
Pompe pour gaz liquéfiés

The l.g.p is a centrifugal pump used for pumping liquefied hydrocarbons such as propane, butane, propylene and ethylene (refinery products in the distillation of crude oil), which come under the general heading of LPG (liquefied petroleum gas), and which are pumped in the temperature range of minus 104 °C l.g.p.'s are also used to pump liquefied natural gas, which consists mainly of methane and is generally designated as LNG (liquefied natural gas) and is pumped in the temperature range of minus 161 °C.

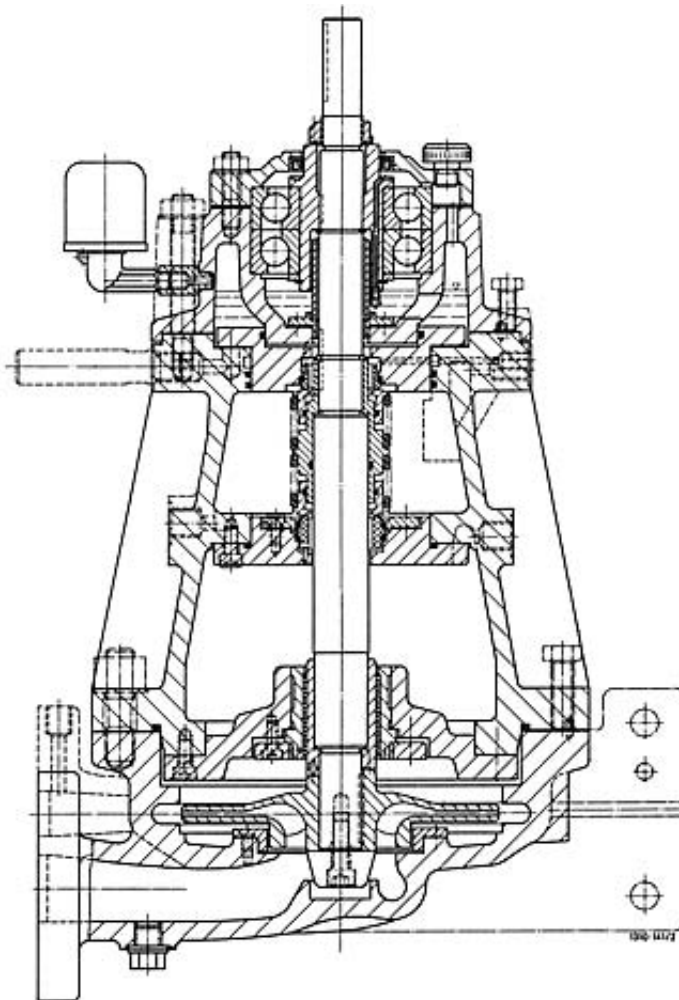


Fig. 1: Vertical refinery process pump (liquefied gas pump)

Depending on the application, various pump types are used in plants for the liquefaction, vaporisation, filling and storage of LPG and LNG, for example horizontal single or multistage refinery pumps of process type design or variants of these pump series in the form of vertical pumps (Figs. 1 and 2). In addition multistage mainly vertical can-type pumps (refinery pump) are used for the higher heads. The shaft passage through the casing on these l.g.p.'s is by means of mechanical shaft seals. Depending on the operating conditions, either single-acting metal bellows mechanical seals or double-acting balanced mechanical seals in back-to-back arrangement, with a sealing fluid feed, are adopted.

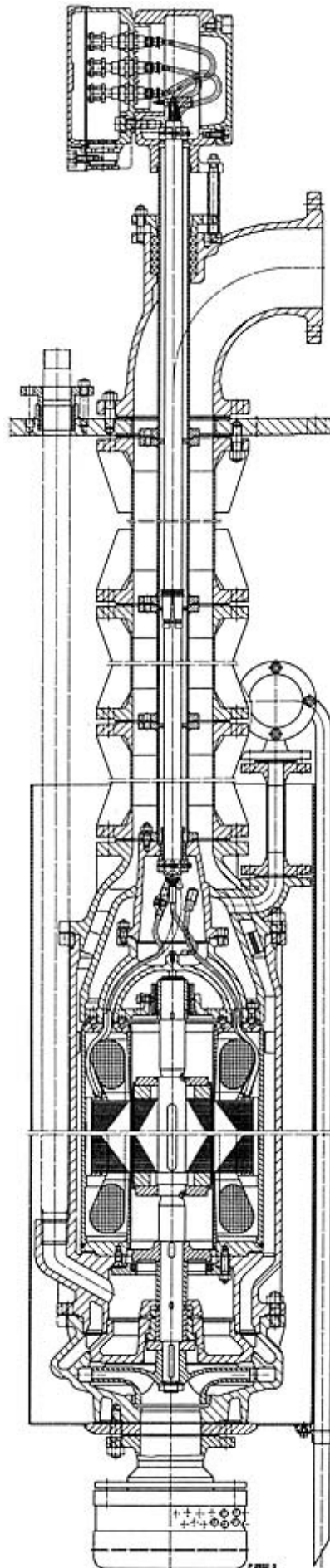


Fig. 2: Chemical submersible motor pump (liquefied gas pump)

As liquefied gases are usually pumped at temperatures close to their boiling point (vapour pressure), the NPSH (net positive suction head, suction behaviour) of the l.g.p. assumes great importance. On conventional centrifugal pumps, this NPSH is usually insufficient to enable the tanks to be emptied down to a level of a few centimetres only, so that it becomes necessary to use inducers in many cases for the emptying of tanks.

All the metal parts of l.g.p.'s in contact with the fluid are made of appropriate materials which are ductile at low temperatures (tough at subzero); apart from chrome-nickel steels, aluminium alloys are also used.

Liquefied Natural Gas

Liquefied natural gas
Gaz naturel liquéfié

LNG for short, is liquefied gas from natural sources. LNG consists mainly of methane (-161 °C boiling point at atmospheric pressure), and nearly always contains a certain amount of CO₂ and N₂ (liquefied gas pump).

Liquefied Petroleum Gas

Liquefied petroleum gas
Gaz de pétrole liquéfié

LPG for short, is a liquefied form of petroleum gas. It is a by-product in the refining process. Under pressure it can be liquefied, and when depressurized it returns to its natural gaseous state. Usually this gas is known as propane and butane, and their gas olefins, propylene and butylene, and combinations thereof. The boiling point at atmospheric pressure is between -0.5 °C (n-butane) and - 61 °C (n-butylene). Propane and butane are growing in importance as combustible gases in house-hold, commercial and industrial applications, whereas the olefins are used almost exclusively in the petrochemical industry. Storage and transportation (liquefied gas pump) is accomplished in pressurized bottles, for small quantities, and in pressure tank car or trucks, for large quantities.

Liquid Metals Pump

Flüssigmetallpumpe
Pompe à métaux liquides

The l.m.p. is a centrifugal pump which because of its design and materials is capable of handling liquid metals. A simple l.m.p. is the mercury pump; most l.m.p.'s however, have to be adapted to the high temperatures of liquid metals by means of thermal barriers and special shaft seals as well as a suitable selection of materials.

Liquid Ring Pump

Flüssigkeitsringpumpe
Pompe à anneau liquide

see Water Ring Pump

Long-Turn Y-Branch

Bogen-Abzweigstück
Raccord courbé de dérivation

see Fittings

Loop

Loop
Boucle

see [Closed Loop Test](#)

Loss Coefficient

Verlustbeiwert
Coefficient des pertes

L.c., symbol ζ , is the nondimensional characteristic number for the calculation of the head loss H_v (pressure loss) using

$$H_v = \zeta \frac{v^2}{2g}$$

with

v characteristic flow velocity in the relevant component exposed to flow, usually the flow velocity in the cross-section of the connection immediately downstream of the component,
 g gravitational constant.

Low Pressure Pump

Niederdruckpumpe
Pompe à basse pression

L.p.p. is a centrifugal pump with a head not exceeding 80 m (e.g. chemical pump). In contrast, we have medium pressure, high pressure and super pressure pumps.

Lubricating Oil Pump

Schmierölpumpe
Pompe d'huile de graissage

L.o.p.'s are used to supply oil to lubricating points (particularly for plain bearings).

In the case of circulation system lubrication, a flow of oil is aspirated from a reservoir, forced through the lubricating points and then led back to the reservoir. Gear type oil pumps are preferably used for this purpose (positive displacement pump); they are self-priming and generate high static pressures in the lubricating oil system.

In the case of the fresh oil or economy lubrication system, each lubricating point receives a metered oil supply adequate for its needs. For this purpose, special plunger pumps are usually required (positive displacement pump).

The lubricating pump used for grease-lubricated plain bearings is also a plunger pump; in this case the grease reservoir is situated above the pump, to ensure that grease is always fed positively to the lubricating pump with the aid of a mechanically driven paddle stirrer and a slight overpressure (also called grease gun).

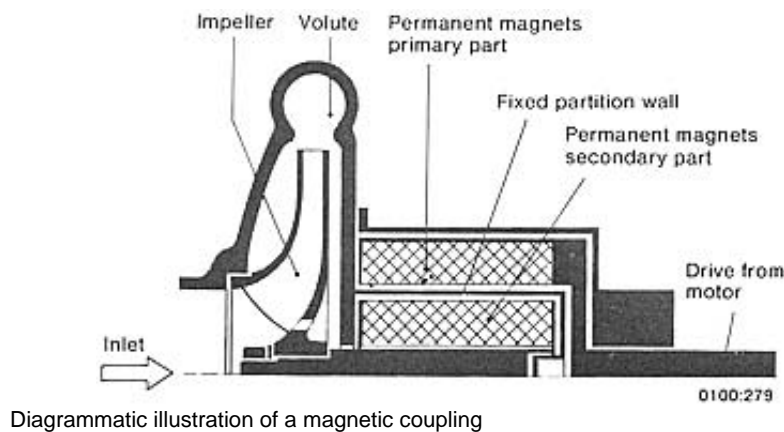
M

Magnetic Coupling

Magnetkupplung
Coupleur magnétique

The m.c. used for the drive of centrifugal pumps (see illustration) consists of a primary part (rigidly connected to the motor shaft) and of a secondary part arranged on a shaft (pump shaft) with the impeller. The primary part and secondary part are equipped with permanent magnets. Poles of opposite polarity face one another in each case, and they attract each other mutually as a result of the magnetic flux. When the primary part is driven, the secondary part will be entrained at the same rotational speed. Centrifugal pumps equipped with m.c.'s are also called magnetic pumps and have been built with two-pole drives (number of poles) up to powers of 200 kW.

When the torque transmitted increases, the revolving field angle between the axes of the magnets of the primary and secondary part will also increase. When the torque transmitted exceeds a given maximum value, the so-called pull-out torque or stalling torque, the coupling will fall out of step and the secondary part will come to a standstill. The driver (drive) must then be switched off, because a run-up to full operating speed is only possible synchronously



Diagrammatic illustration of a magnetic coupling

This disadvantage, which incidentally is usually of no importance in the case of the drive of centrifugal pumps, can be avoided if the secondary part is designed in the form of a squirrel-cage rotor of an asynchronous motor, thus making it possible to have an asynchronous run-up to full speed and asynchronous operation.

By means of the provision of a partition wall (which is often double for safety reasons) in the gap between the primary and the secondary part of the m.c., the pump can be sealed off absolutely leak-tight. It follows that centrifugal pumps fitted with a m.c. (Fig. 3 under chemical pump) transmission of the drive have similar applications to canned motor pumps and other pumps with wet rotor motor.

The m.c. must not be confused with the electromagnetic coupling used to control or protect against loads.

Magnetic Pump

Magnetpumpe
Pompe étanche à entraînement magnétique

see Magnetic Coupling

Maintenance

Wartung
Entretien

The various checks and m. work duties which have to be carried out at regular intervals during operation come under the designation of m. The time intervals and scope of m. are laid down in detail in the operating instructions. For specific products, m. contracts, also known as "inspection contracts" are offered and entered into. The regular m. of pumps and their drives prolongs their service life and saves costs.

External care of pumps and drives. The pumping sets should be cleaned externally at regular intervals, depending on the amount of dirt accumulation. On no account should the pumps and drives be simply hosed down with water jets; rust patches should be removed and suitably treated; bright parts should be coated with rust preventive grease.

M. of bearings. The bearings should be maintained at regular intervals. As the lubricants are used up at differing time intervals, they must be either topped up or renewed (oil change. topping up of oil, complete repacking with grease). The operating instructions give details of the relevant time intervals, lubricants and lubricant qualities.

M. of shaft seals. The m. of shaft seals depends on the type of seals fitted to the pump:

Soft packed stuffing boxes should always drip slightly during operation. If they fail to drip (i.e. seal completely leak-tight), damage will ensue to the sealing elements (shaft, shaft protecting sleeve). Therefore the nuts on the gland studs should be slackened in this case until the stuffing box (shaft seals) starts to drip slightly. The same applies to metallic packings, sleeve packings and plastic packings.

In the case of mechanical seals (shaft seals), any leakage is usually not visible from outside. No special m. is required. If there is excessive leakage, the sealing elements should be replaced by new ones. Further details are given in the operating instructions.

Other m. work.

Checking the quietness of the centrifugal pump and of its drive; if applicable, re-tightening of V-belts (belt drive);

checking the motor output (current consumption);

checking the coupling alignment (shaft coupling);

inspecting the flexible transmission elements for signs of wear (coupling face plate, coupling bolts, flexible coupling membrane packs);

examining the flushing and sealing water supply systems;

checking the balancing device (balance disc) if applicable;

checking the functioning of the automatic grease pumps (lubricating oil pump) and grease lines if applicable.

All necessary cleaning media and lubricants, and small spare parts required as a result of the inspection must be provided by the customer.

It is recommended to allow the operating personnel to participate in the inspection for the purposes of additional information.

M. is usually carried out according to a so-called check list. An example is given below:

1. Check smooth and quiet running of pump and driver.
2. Check performance data, including current consumption.
3. Check oil and grease feeds (bearings).
4. Check shaft seal (soft packed stuffing box, mechanical seal, radial sealing rings).
5. Repack soft packed stuffing box with new packing if necessary, fit new carbon ring and rotating seal ring on mechanical seal if necessary.
6. Check coupling alignment.
7. Examine flexible transmission elements for signs of wear (coupling face plate, coupling bolts, flexible coupling

membrane packs).

8. Check functioning of isolating and nonreturn valves.
 9. Inspect plant for signs of corrosion (external surfaces) and inspect pump internals for signs of cavitation and erosion.
 10. Carry out a trial run after m.
 11. Determine what spare parts are required for stock, and make the appropriate suggestions to the customer.
 12. Inspect flushing and sealing water supply system if applicable.
 13. Clean solenoid valves in the installation if applicable.
 14. Check balancing device if applicable.
 15. Check electropneumatic control system (only applies to sewage pumps).
 16. Check functioning of automatic grease pumps and grease lines if applicable.
 17. Check wear by inspection through handhold cover (only on sewage pumps).
 18. Measure rotor play.
 19. Check throttling gaps.
-

Mammoth Pump

Mammutpumpe

Pompe système Mammouth

see Pump Types

Manifestation of Corrosion

Korrosionserscheinungen

Indices de corrosion

see Corrosion

Manometer

Manometer

Manomètre

see Measuring Technique

Manometric Pressure

Manometerdruck

Pression manométrique

see Pressure

Manufacturing Tolerance

Bautoleranz

Tolérance de construction

The m.t. of a centrifugal pump is the permissible deviation of the value achieved in the acceptance test from the guaranteed value (guarantee, which results from inaccuracies in the manufacturing process. According to DIN 1944, a permissible m.t. is quoted for the capacity Q of centrifugal pumps with a so-called "non-flat throttling curve" (characteristic curve), whilst it is quoted for the head H of centrifugal pumps with a "flat throttling curve". In this context, a throttling curve (characteristic curve) is considered "flat" at a design point (design duty point) defined by Q_L and H_L specified in the supply contract at a constant

speed n_L , if

$$\frac{Q_L}{H_L} \cdot \left| \frac{dH}{dQ} \right| \leq 0.2$$

where

Q_L is the capacity specified in the supply contract,

H_L is the head specified in the supply contract.

The slope dH/dQ of the throttling curve should be inserted as an absolute value.

No permissible m.t. is quoted for the efficiency η ; the guaranteed efficiency is the minimum efficiency attainable at the duty point.

The magnitude of the m.t. is a distinguishing characteristic for the standardized degrees of accuracy (guarantee) listed in the Table.

Table: Manufacturing tolerances in accordance with DIN 1944 (October 1968)

Degree of accuracy		
III	II	I
Manufacturing tolerance on capacity Q for centrifugal pumps with a non-flat throttling curve		
from -5 to +15%	from -5 to +10%	from -5 to +5%
of the capacity Q agreed in the supply contract		
Manufacturing tolerance on head H for centrifugal pumps with a flat throttling curve		
from -1 to +3 %	from -1 to +2 %	from -1 to +1 %
of the head H agreed in the supply contract		

In the ISO acceptance test code (acceptance test code for centrifugal pumps), the m.t. is explicitly defined but is also implicit in the overall tolerance laid down (designation in the ISO acceptance test code: acceptance tolerance); thus according to the ISO acceptance test code, a permissible m.t. is again only quoted for the capacity and on the head.

Marine Pump

Schiffspumpe
Pompe marine

The concept m.p. is a collective concept for all pumps complying with the relevant regulations issued by the Shipbuilding Classification Societies. The m.p. as centrifugal pump on board ship performs a great variety of duties (application fields for pumps under point 4. shipbuilding): in the engine room as boiler feed pump, condensate pump, centrifugal pump for cooling water (seawater or freshwater), in the bow as bow-thruster, in special pump rooms as cargo oil pump. Butterworth pump, ballast pumps, for special duties as antirolling pump, antiheel pump, trimming pump; as drainage pump, pump -out pump, bilge pump, fire-fighting pump, and as service pumps for a variety of services. The dock pump also belongs to m.p.'s.

Cooling water pumps using either seawater or freshwater (seawater pump) and fire-fighting pumps take their fluid supply from the sea chests via suction pipelines. These sea chests are tanks arranged beneath the water line on the inner side of the ship's side, with apertures facing seawards, covered over by intake trash racks.

Due to the restricted installation space on board, a special type of construction has evolved for most m.p.'s, viz. the radially split (radially split casing) vertical pump with the motor directly on top of the pump; sometimes one also comes across axially split casing pumps. Seawater as the pumped medium requires suitable pump materials, gunmetal (red bronze) or bronze (e.g. multi-alloy aluminium bronze) for the pump casing and the impeller, chrome nickel steel for the pump shaft.

Large quantities of air have often to be evacuated from the bilge and ballast water lines; therefore many of the m.p.'s to be found on board ship are self-priming pumps. These are water ring pumps which rotate continuously with the main pump. In recent times separate venting devices (venting), such as ejectors, central vacuum systems etc-. have tended to become more commonly adopted.

Fig. 1 illustrates a vertical radially split m.p. (radially split casing) for a wide variety of applications. The range of capacities stretches from 10 to 300 l/s at heads from 15 to 125 m. The impeller is of single suction type for the low capacities, and of double suction type (multisuction pump) for the higher capacities. The pump shaft is guided in greaselubricated anti-friction bearings arranged outside the pump casing. For inspection and repair purposes, the rotating assembly can be easily lifted out in the upward direction after removal of the coupling spacer sleeve (shaft coupling). The motor need not be dismantled, and the pump casing can remain in situ on the pump foundation.

Fig. 2 illustrates an axially split m.p. (axially split casing) for capacities ranging from 300 to 1200 l/s at heads from 15 to 85 m. The impeller is of double suction type (multisuction pump). The pump shaft is guided in a grease-lubricated anti-friction bearing located beneath the shaft coupling, and in a plain bearing lubricated by the fluid pumped (rubber bearing) situated beneath the impeller. In order to dismantle the rotating assembly of the pump, it is necessary first to remove the front part of the casing (pump casing) to one side.

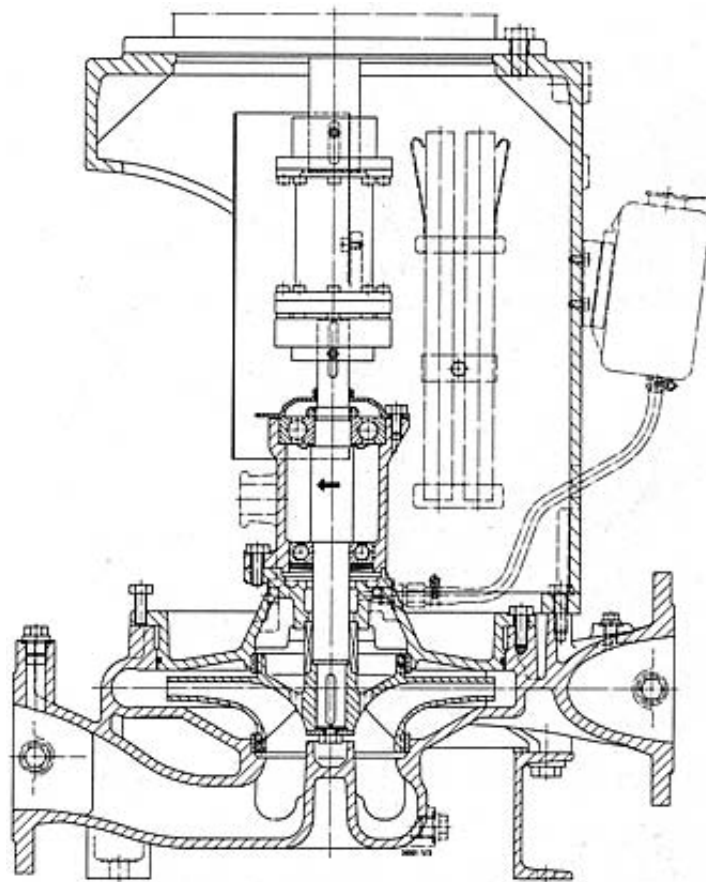


Fig. 1: Radially split vertical marine pump

On turbine ships equipped with large boiler plants and correspondingly large condensers, double suction centrifugal pumps (multisuction pump) are used for cooling water supply duties. Because of the low heads involved, propeller pumps with an axially split casing are sometimes adopted (Fig. 3). The capacities range up to 3750 l/s in this case, at heads from 10 to 2 m.

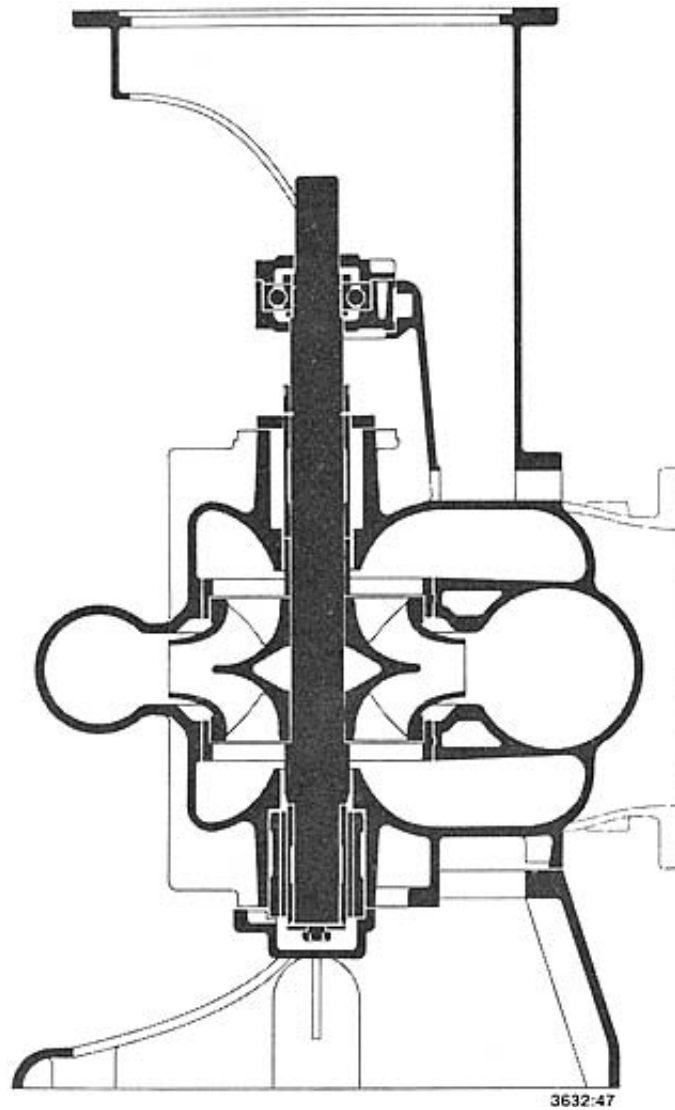


Fig. 2: Axially split vertical marine pump

Marine condensate pumps have to operate on very low positive suction heads (suction behaviour). Fig. 4 illustrates a two stage condensate pump. The first stage impeller (suction impeller) is arranged at the bottom and the approach flow is from beneath. The left-hand side of the figure illustrates a design with an inducer installed to improve the NPSH value (net positive suction head). The bearings are arranged outside the medium pumped. Such pumps often operate with cavitation-conditioned self regulation (condensate pump). Consequently their first stage impellers are made of cavitation-resistant materials. The performance data of such pumps are: 2 to 70 l/s capacity, 140 to 30 m head.

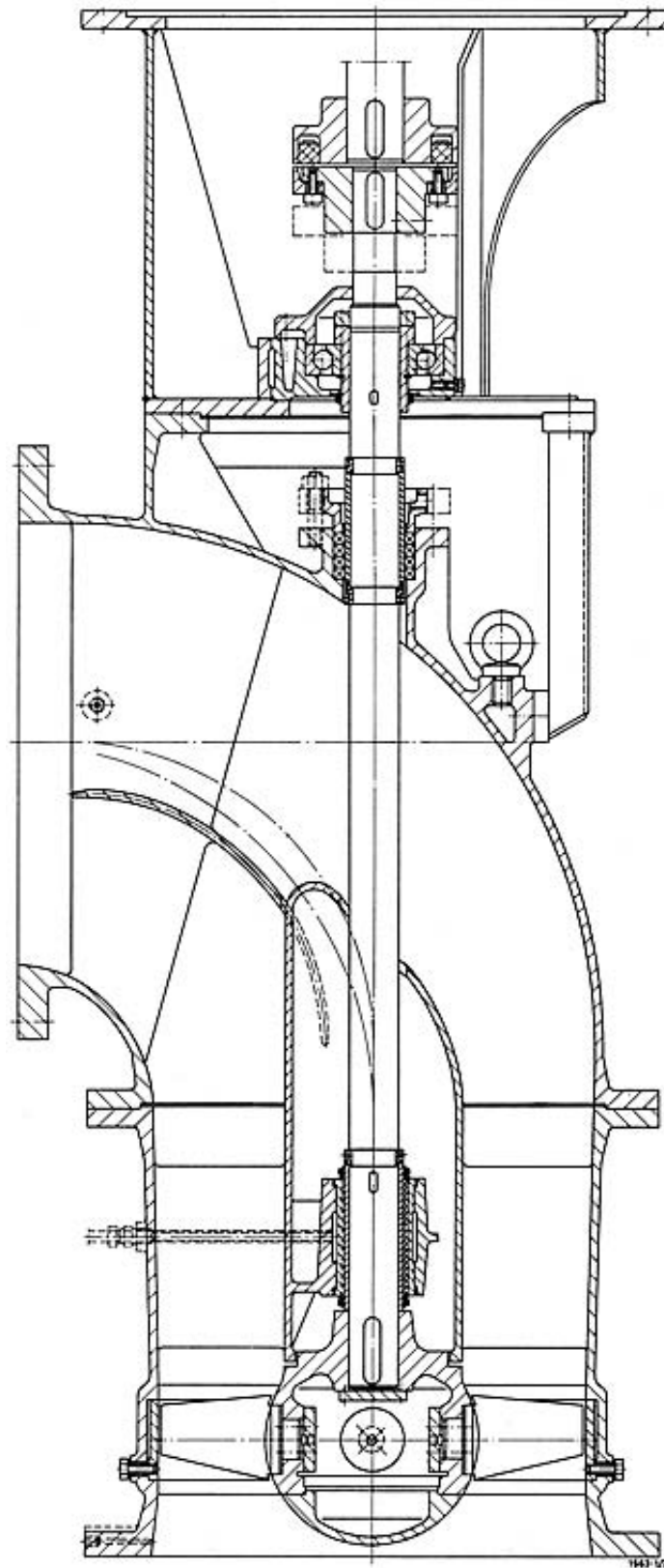


Fig. 3: Axially split propeller pump (marine pump)

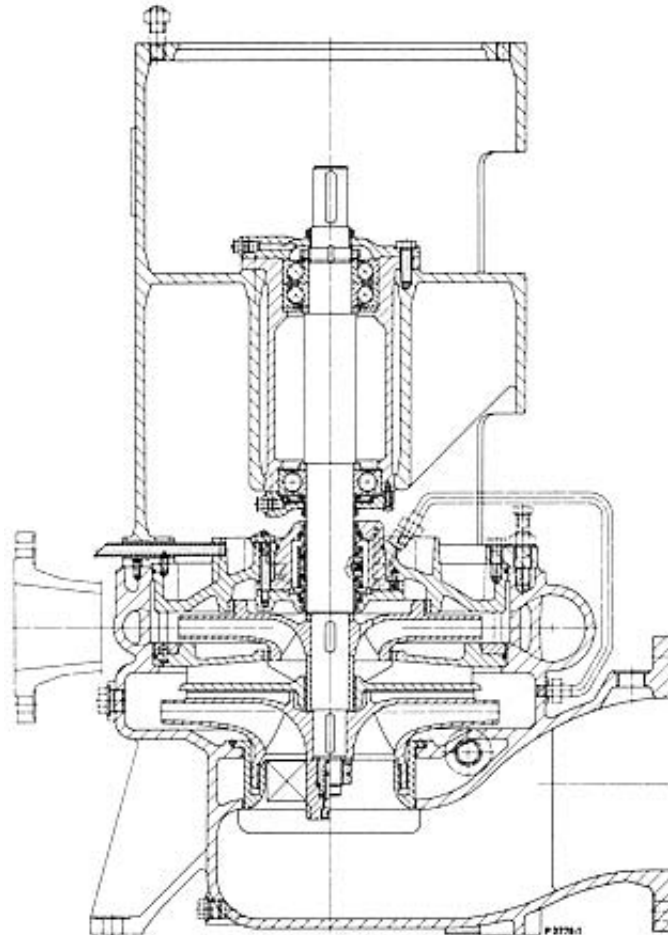


Fig. 4: Volute casing pump (two stage marine condensate pump) with suction impeller (on the right) and inducer (on the left)

Mass

Masse
Masse

see [Unit](#)

Mass Flow

Massestrom
Débit-masse

The m.f. \dot{m} is the mass of fluid pumped through the outlet cross-section and through any extraction cross-sections of the pump during a unit of time.

$$\dot{m} = \rho \cdot Q$$

with

ρ density of pumped medium,
Q capacity.

The SI unit of m.f. is kg/s, but in centrifugal pump technology the units kg/h and t/h are also used.

Materials

Werkstoffe

Matériaux

Today centrifugal pumps are constructed of metallic and nonmetallic m. (plastics and ceramics).

Metallic m. can be cast, forged, rolled or formed by other methods. Cast m. are still the most popular for use in constructing centrifugal pumps, (pump casings and impellers) due to their possibility for free design and their large variety in alloys. The most important cast m. for centrifugal pumps, their composition and mechanical properties are shown in Table 1.

Plastics are divided into four groups (according to DIN 7724), due to the temperature dependence of their mechanical properties:

1. Thermoplastics: highly polymers m. which can repeatedly be deformed under temperature; e.g. polystyrene (PS), polyethylene (PE) and polyvinyl chloride (PVC).
2. Elastomers: wide-meshed high-polymere m.; with elastic properties in all temperature ranges, e.g. natural rubber, polyisobutylene.
3. Thermoplastics: wide-meshed high-polymere m., with elastic properties above 20 °C; e.g. natural rubber with more than 10% sulfur, meshed polyethylene. high-molecular polymethacrylate.
4. Duroplastics: highly meshed hardening highpolymere m.; e.g. polyester resins, epoxy resins, phenol-formaldehyde resins.

Of these, thermoplastics are the most important for constructing impellers and diffusers, whereas elastomers are used for seals and protective liners. Some typical thermoplastics and their basic properties are given in Table 2.

Ceramic m. are defined as nonmetallic, inorganic and of more than 30 % crystalline structure. They are used nowadays as modern for seals and bearing parts (plain bearings), but also increasingly for impeller rings, casing wear rings (for suction side clearance gaps) and also for impellers. Characteristic for this group is the high mechanical, thermal and chemical resistance. Extreme brittleness, sensitivity to tensile and bending stresses restrictions for difficult geometries and design and cost intensive production require special design. An overview of the most important ceramics and their characteristics can be found in Table 3.

Table 2: Stantard values of important plastic materials used in centrifugal pumps.

Chemical designation	Tradename	Codes acc. to DIN 7728	Density	Tensile strength	Elongation at fracture	Modulus of elasticity	Impact strength	Continuous temperature		Coefficient at linear expansion for 20 °C	Water absorption ⁶⁾	Special characteristic features
			$\frac{\text{g}}{\text{cm}^3}$	$\frac{\text{N}}{\text{mm}^2}$	%	$\frac{\text{N}}{\text{mm}^2}$	$\frac{\text{kJ}}{\text{m}^2}$	water °C	dry °C	$\frac{1}{\text{K}}$	%	
Polyamide	Ultramid B3 WG 7	PA 6 - GF 35 ¹⁾	1.40	130 ³⁾	6 ³⁾	7000 ³⁾	50	90	125	$20 \cdot 10^{-6}$	6.2	wear resist, very tough
Polyphenylene oxide	Noryl GFN 2	PPO - GF 20	1.21	90	2-3	6500	30	110	110	$40 \cdot 10^{-6}$	0.14	good hot water resistance, low water absorption
	Noryl GPN 3	PPO - GF 30	1.27	120	2-3	9000	30	120	120	$40 \cdot 10^{-6}$	0.12	high stiffness
Polyether sulfone	Victrex 4 01 GL 30	PES - GF 30	1.60	140	3	8000	27	140	180	$23 \cdot 10^{-6}$	0.15	very good hot water resistance, low water absorption, high dimensional stability in the heat
Polybutylenterephthalat	Pocan 3235	PBTP - GF 30	1.55	130	2.7	10500	45	60	120	$30 \cdot 10^{-6}$	0.3	high stiffness, good sliding and abrasion behaviour
Polycarbonate	Makrolon 8030	PC - GF 30	1.44	90	3.5	5500	30	60	120	$27 \cdot 10^{-6}$	0.3	wear resistance, low water absorption low distortion
Acrylonitrile/styrol/acrylate	Luran 776 S	ASA	1.07	47	20	2300	no fracture	60	85	$90 \cdot 10^{-6}$	0.45	excellent weathering resistance, high impact strength
Polypropylene	Hostalen PPNVP 7180 TV/20	PP - TV 20 ²⁾	1.04	33	20	2300 ⁴⁾	38	60	100	$80 \cdot 10^{-6}$	0.2	good resistance to chemicals, good surface luster, mar resistant surface
	Hostalen PPNVP 7780 GV/20	PP - GF 20	1.05	32	50	2400 ⁴⁾	50	60	100	$60 \cdot 10^{-6}$	0.2	
	Hostalen PPNVP 7799 GV 2/30	PP - GF 30	1.114	71	5	5500 ⁴⁾	15	70	110	$70 \cdot 10^{-6}$	0.2	good resistance to chemicals, high stiffness
Polyethylene soft	Lupolen 1810 H	PE-LD	0.92	9	>400	230	no fracture	60	80	$230 \cdot 10^{-6}$	0.2	high flexibility, good resistance to chemicals

- 1) GF 35: 35 weight % glass fiber
 2) TV 14: 20 weight % talcum
 3) exposed to standard atmosphere (up to saturation of approx. 1% water)
 4) modulus of flexional elasticity
 5) in case of storage in water of 23 °C

Table 1: Typical metallic cast materials for centrifugal pumps

Quality/ Tradename	Designation due to DIN 17 006	Material No.	German No.	Comparable ASTM standard	Heat treatment 4)	Structure 5)	Weldability 6)	Remarks	
Cast irons and cast steel maximum weight of castings: 1700 kg									
Grey cast iron	GG-25	0.6025	DIN 1691	A 48-40 B	-	P	(+)	max. weight 4000 kg	
Nodular cast iron	GGG-40	0.7040	DIN 1693	A 536, Cl. 60-40-18	-	(F)	(+)		
Nodular cast iron	GGG-40.3	0.7043	DIN 1693	A 536, Cl. 60-40-18	G	F	(+)	1) U-notch sample	
Ni-Resist D-2W®	GGG-NiCr Nb 20 2	0.7659	DIN 1694	A 436, Type 2	-	A+C	+		
ERN	GGL-NiMO 77	-	KSB-WSZ 1930		-	B	-		
NORIHARD® NH 15 3	G-X 250 CrMo 15 3	-	KSB-WSZ 1941		V	M+C	-	max.weight 1000 kg	
NORILOY® NL 25 2	G-X 170 CrMo 25 2	-	KSB-WSZ 2878		V	F+C	-		
Cast steel	GS-C 25 N	1.0619.01	DIN 17245	A 216 WCB	N	F+P	+		
Cast stainless and special steels maximum wight of castings: 1700 kg									
Cast ferritic stainless steel	G-X 8 Cr 14	1.4008	DIN 17445	A 743 A 217-79	CA 15	V	M	+	
Cast martensitic stainless steel	G-X 5 CrNi 13 4	1.4313	DIN 17445	A 743 A 487	CA 6 NM	V	M	+	
Cast austenitic stainless steel	G-X 6 CrNi 18 9	1.4308	DIN 17445	A 743-79 A 351-79	CF 8	L	A	+	KSB standard C ≈ 0.04
	G-X 6 CrNiMo 18 10	1.4408	DIN 17445	A 743-79 A 351-79	CF 8M	L	A	+	KSB standard C ≈ 0.04
	G-X 6 CrNiMo 17 13	1.4448	KSB-WSZ 1641	A 743-79 A 351-79	CG 8M	L	A	+	SEW 410-70
	G-X 5 CrNiNb 18 9	1.4552	DIN 17445	A 743-79 A 351-79	CF 8M	L	A	+	KSB standard C ≈ 0.04
	G-X 5 CrNiMoNb 18 10	1,4581	DIN 17445	A 743-79 A 351-79		L	A	+	
	G-X 7 CrNiMo CuNb 18 18	1.4585	KSB-WSZ 2770			L	A	+	SEW 410-70

Cast special austenitic stainless steel	G-X 7 NcrMo CuNb 25 20	1.4500	KSB-WSZ 2830	A 743-79 A 351-79	CN 7M	L	A	+	KSB standard C \approx 0.04
NORICID®	G-X 3 CrNisiN 20 13	9.4306	KSB-WSZ 2872			L	A(+F)	+	
NORIDUR®	G-X 3 CrNiMoCu 24 6	9.4460	KSB-WSZ 2745	A 743-79 A 351-79 CD4MCu		L	F+A	+	
NORICLOR® NC 24 6	G-X 3 CrNiMo CuN 24 6 5	-	KSB-WSZ 2747			L	F+A	+	
Cast copper and nickel base alloys maximum wight of castings: 300 kg									
Tin bronze	G-CuSn 10	2.1050.01	DIN 1705	B 584, C 90500	-			+	2) hardness: HB10
NORICOR®	G-CuAl 10 Ni	2.0975.01	DIN 1714	B 148, C 95500	-			+	3) Al \leq 8.5 + 0.5 Ni
MONEL®	G-NiCu 30 Nb	-		A 494, M 35	-			+	

4) Heat treatment:
 G = annealed
 V = hardened and tempered
 N = normalized
 L = solution annealed and water quenched

5) Structure:
 P = perlite
 F = ferrite
 A = austenite
 M = martensite
 B = bainite
 C = carbides

Table 1: continued

Designation due to DIN 17 006	Chemical composition								Mechanical properties				
	C	Si	Mn	Cr	Ni	Mo	Cu	others	Hardness HV50	0.2-Yield strength N/mm ²	Tensile strength N/mm ²	Elongation %	Impact energy J(ISO-V)
GG-25										-	\geq 245		-
GGG-40										\geq 250	\geq 400	\geq 15	
GGG-40.3										\geq 250	\geq 400	\geq 18	\geq 141)
GGG-NiCr Nb 20 2	<3.0	1.9	1.0	1.8	20.0			Nb: 0.2		\geq 210	\geq 370	\geq 7	
GGL-NiMO 77	3.2	1.8	0.7		1.8	0.7			300				
G-X 250 CrMo 15 3	2.6	0.6	0.7	15.0		2.6			750-1000				
G-X 170 CrMo 25 2	1.7	1.0	1.0	25.0		2.0			>500		\geq 400		
GS-C 25 N	0.2	0.5	0.7	\leq 0.3						\geq 245	\geq 440	\geq 22	\geq 24

G-X 8 Cr 14	0.09	≧ 1.0	≧ 1.0	13.0	1.5	≧ 0.5			≧ 170	≧ 440	≧ 590	≧ 15	≧ 27
G-X 5 CrNi 13 4	≧ 0.07	≧ 1.0	≧ 1.5	13.0	4.0	≧ 0.7			≧ 240	≧ 550	≧ 760	≧ 15	≧ 50
G-X 6 CrNi 18 9	≧ 0.07	≧ 2.0	≧ 1.5	18.0	10.0				≧ 130	≧ 175	≧ 440	≧ 20	≧ 60
G-X 6 CrNiMo 18 10	≧ 0.07	≧ 2.0	≧ 1.5	18.0	11.0	2.3			≧ 130	≧ 185	≧ 440	≧ 20	≧ 60
G-X 6 CrNiMo 17 13	≧ 0.07	≧ 1.0	≧ 2.0	17.0	13.5	4.5			≧ 130	≧ 185	≧ 400	≧ 15	
G-X 5 CrNiNb 18 9	≧ 0.06	≧ 1.5	≧ 1.5	19.0	10.0			Nb ≧ 8x%C	≧ 130	≧ 175	≧ 440	≧ 20	≧ 35
G-X 5 CrNiMoNb 18 10	≧ 0.06	≧ 1.5	≧ 1.5	19.0	11.5	2.3		Nb ≧ 8x%C	≧ 130	≧ 185	≧ 440	≧ 20	≧ 35
G-X 7 CrNiMo CuNb 18 18	≧ 0.08	≧ 1.5	≧ 2.0	17.5	20.0	2.3	2.1	Nb ≧ 8x%C	≧ 130	≧ 185	≧ 440	≧ 20	
G-X 7 NiCrMo CuNb 25 20	≧ 0.08	≧ 1.5	≧ 2.0	20.0	25.0	3.0	2.0	Nb ≧ 8x%C	≧ 130	≧ 185	≧ 440	≧ 20	
G-X 3 CrNiSiN 20 13	≧ 0.04	4.5	4.5	20.0	13.0	≧ 0.05		N	≧ 200	≧ 250	≧ 600	≧ 30	≧ 80
G-X 3 CrNiMoCu 24 6	≧ 0.04	≧ 1.5	≧ 1.5	25.0	6.0	2.4	3.1	N	≧ 240	≧ 450	≧ 700	≧ 16	≧ 85
G-X 3 CrNiMo CuN 24 6 5	≧ 0.04	≧ 1.0	≧ 1.0	24.0	6.0	5.0	2.0	N	≧ 240	≧ 485	≧ 485	≧ 16	≧ 85
G-CuSn 10	89.0	≧ 2.0		10.0	≧ 0.2			Pb ≧ 1.0; Zn ≧ 0.5 Pb ≧ 0.2; Sb ≧ 0.2	≧ 70 2)	≧ 130	≧ 270	≧ 18	
G-CuAl 10 Ni	≧ 76.0	5.2			4.5		3.0	≧ 0.8	≧ 140 2)	≧ 270	≧ 600	≧ 12	
G-NiCu 30 Nb	30.0	64.5			≧ 4.0	≧ 2.0	≧ 2.0	Nb: 1.3	≧ 120 2)	≧ 220	≧ 450	≧ 25	

6) Weldability:
+ ... good
(+) ... possible
- ... impossible

Table 3: Ceramic materials for pumps

Material	Chemical composition	Special characteristics
SiC, sintered	SiC: > 96 %; Si: < 0.5 %	high thermal conductivity
SiSiC, reaction bounded, silicide infiltrated	SiSiC: 80 - 90 %; Si: 8 - 15 %; C: 0 - 5 %	high thermal conductivity
Si ₃ N ₄ , hot-pressed	Si ₃ N ₄ : > 98 %; MgO, Y ₂ O ₃ , CeO ₂ : 0 - 2 %	high strength
Al ₂ O ₃	Al ₂ O ₃ : > 94 %; SiO ₂ , MgO, ZrO ₂ , TiC: 0 - 6 %	low thermal spalling resistance
ZrO ₂ , stabilized	ZrO ₂ : > 85 %; MgO, CaO, Y ₂ O ₃ : 0 - 15 %	high strength, high thermal expansion

Measuring Device

Meßgerät

Appareil de mesure

see [Measuring Technique](#)

Measuring of Speeds

Drehzahlmessung

Mesure de vitesse

see [Measuring Technique](#)

Measuring Orifice Plate

Meßblend

Diaphragme de mesure

see [Standard Orifice](#)

Measuring Technique

Meßtechnik

Technique de mesure

Just as varied as are the problems which arise during the development, manufacture and operation of centrifugal pumps, are the problems of measurement associated with them. Ever increasing technical and economic requirements demand an ever greater quantification of technical-physical states and magnitudes, with the result that the variety and range of m.t.'s become progressively greater. The same applies to the use of measuring devices for monitoring and control purposes. Conventional measurement tasks can be solved with the aid of improved methods, and new measurement tasks have been added. The methods of electrical and electronic m.t.'s are being adopted increasingly, to the point where they have become routine and irreplaceable in many fields. Present-day technical developments are characterized by the adoption of electrical (digital) transmission of measured values, of multipoint measuring devices and data processing equipment, which involve heavy capital expenditure. In the following paragraphs, we have attempted to describe some of the more important and successful measuring devices and processes used in centrifugal pump technology.

1. *Strain measurement technique (WSG technique = wire strain gauge technique).* Measurement with wire strain gauges (WSG) has become one of the most important and adaptable measuring methods used today. Use is made of the physical property which metal conductors (also semiconductors in certain special applications) exhibit of changing their electrical resistance when strained longitudinally. The WSG is a passive receiver and must be fed from an electrical potential source (d.c. or a.c. voltage).

In centrifugal pump technology, the WSG used almost exclusively consists of a thin electrical conductor (metal foil) embedded in a plastic foil. The plastic foil act: as an insulant against the object to be measured, and must be suitably and properly attached to it (e.g. by gluing or welding). Connecting lugs for the connection to the measuring cables are provided at the ends of the so-called measuring grid. A whole series of different WSG types, together with reliable equipment (measuring bridges) for the assembly of measuring devices are available for the most varied applications. Basically, very accurate measurements are made possible in this way. The temperature, surrounding medium, acceleration (e.g. in the event of vibrations) and the material of the object to be measured do not present any fundamental obstacles to the use of WSG's. WSG'S are used successfully not only to measure elongation (and hence stresses by subsequent calculation) at the surface of components under investigation, but also for a whole series of further static and dynamic measurement magnitudes e.g. force (force measurement pressure cells (dynamometers), axial thrustmeasuring rings), torque, pressure, travel.

The main applications of WSG technology to centrifugal pumps are concerned with strength investigations on pump casings and impellers, the measurement of axial thrusts and torques (see point 4., power measurement) and of torsional stresses on pump shafts.

2. *Pressure measurement.* The following measuring devices are important practical devices used in the measurement of pressure of liquids and gases: vertical tube manometers (Figs. 1 and 2); inclined tube manometers (Fig. 3); piston manometers (or pressure balances, ring balance manometers, Fig. 4); spring pressure gauges (Figs. 5 and 6); Barton cells and various other electrical measuring devices consisting of a pressure transmitter, an intermediate circuit (e.g. an amplifier) and an indicating or recording device. The measuring systems illustrated in Figs. 1 to 4 inclusive are characterized by the fact that they measure the pressure directly and do not require any special calibration. They are even used to calibrate other measuring instruments. In order to measure pressures which fluctuate with time, more complicated devices are needed. They must possess the characteristic of reproducing without error the fluctuation with time of the pressure within their frequency range (frequency); the recording device must also meet these requirements.

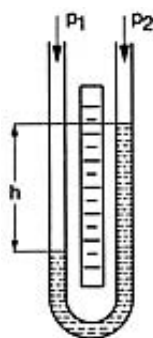


Fig. 1: U-tube manometer

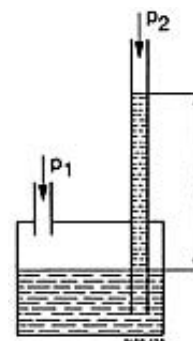


Fig. 2: Cistern manometer

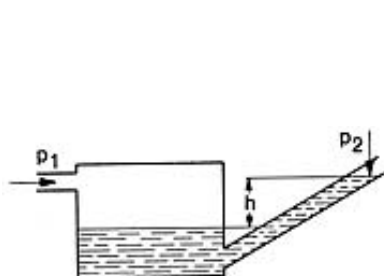


Fig. 3: Inclined tube manometer

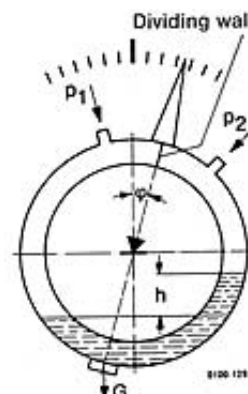


Fig. 4: Ring balance manometer

Pressure measurements on centrifugal pumps are needed to determine the head and NPSH value of the pump (net positive suction head). The liquid manometers most frequently used in centrifugal pump technology comprise the U-tube manometer, the cistern manometer and the inclined tube manometer (as examples of single leg liquid manometers) and the ring balance manometer (Figs. 1 to 4); with the aid of these devices, and using mercury as the sealing liquid, over-pressures up to 1.5 bar (even up to 3 bar with special instruments) can be measured, as well as all the underpressures or partial vacua which occur in practice in centrifugal pump technology (see manometric pressure under pressure). For the measurement of higher pressures, spring and piston manometers (pressure gauges) are used in centrifugal pump technology. The best-known types of spring pressure gauges are the Bourdon tube pressure gauge, and the plate spring or diaphragm pressure gauge (Figs. 5 and 6). They operate on the principle of the deformation of the spring by the action of the pressure, this deformation resulting in the rotation of the indicator needle. DIN 1944, October 1968 edition, contains further details about pressure measurement methods.

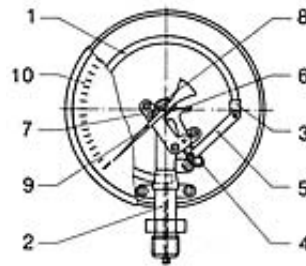


Fig. 5: Bourdon tube spring pressure gauge

1 tube spring; 2 spring holder; 3 spring end piece; 4 segment; 5 pull rod; 6 toothed segment; 7 indicator needle spindle; 8 helical spring; 9 indicator needle; 10 graduated dial

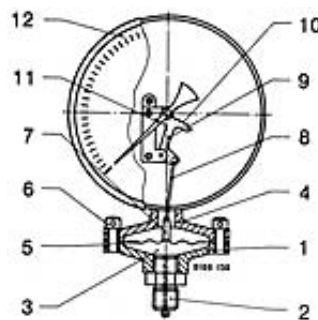


Fig. 6: Diaphragm spring pressure gauge

1 bottom measuring flange; 2 connection piece; 3 pressure compartment; 4 top measuring flange; 5 diaphragm spring; 6 lock screw; 7 ball joint; 8 push rod; 9 segment; 10 toothed segment; 11 indicator needle; 12 graduated dial

As an example, we shall discuss an NPSH (= net positive suction head) measurement. The respective NPSH value of a pump under any conditions can be determined as follows: the flow velocity v_s at point s (Fig. 1 under net positive suction head) is determined from the dimensions of the pump inlet part and from the capacity Q . The vapour pressure P_D for the existing temperature of the fluid pumped and the barometric pressure p_b are assumed to be known. The pressure p_s at the inlet crosssection A_s (net positive suction head) is best measured with the aid of single leg manometers (Figs. 1 to 3) with a checking vessel arranged upstream. For the measuring device illustrated in Fig. 7, we have:

$$p_s = \rho_M \cdot g \cdot h \quad \text{or} \quad \frac{p_s}{\rho \cdot g} = \frac{\rho_M}{\rho} \cdot h$$

with

ρ density of pumped medium,

ρ_M density of sealing liquid in the single leg, manometer (Figs. 1 to 3),

h level of liquid column in accordance with measuring arrangement illustrated in Fig. 7,

g gravitational constant.

The NPSH value of the installation is therefore:

$$\text{NPSH}_{\text{vorh}} = \frac{\rho_M \cdot g \cdot h + p_b - p_D}{\rho \cdot g} + \frac{v_s^2}{2g} + z_s.$$

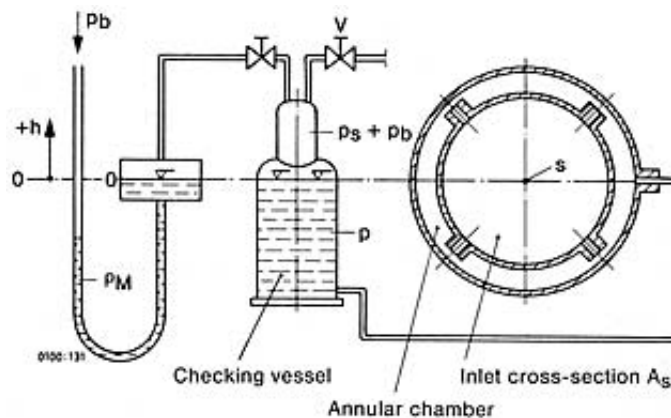


Fig. 7: Measuring device for determination of net positive suction head

In order to determine the NPSH value of the pump (at $v_s = \text{constant}$), the magnitude $(p_s + p_b - p_D)/(\rho \cdot g)$ is reduced e.g. by connecting a vacuum pump, or by throttling the suction side of the pumping plant, or by other suitable means, in order to reduce the NPSH_{av} of the installation down to the point where $\text{NPSH}_{\text{av}} = \text{NPSH}_{\text{req}}$ in accordance with one of the cavitation criteria listed under NPSH_{req} (net positive suction head). In the case of closed loop tests a characteristic curve of NPSH_{av} head at constant capacity Q is run and plotted (Fig. 8). From the course of this characteristic curve, the NPSH value of the pump relating to any one cavitation criterion can be determined.

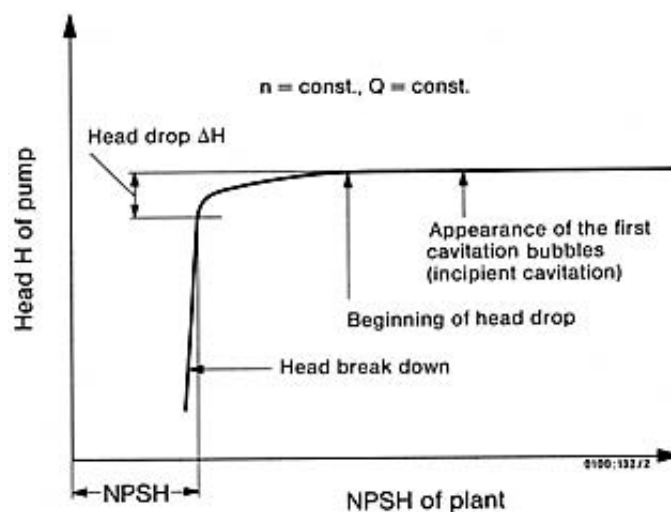


Fig. 8: Characteristic curve of net positive suction head for a given operating point defined by a capacity Q , a head H and a pump rotational speed n (diagrammatic representation)

3. Flow and velocity measurements. The measurement of the capacity (volume flow) is one of the most important measurements in centrifugal pump technology. Of the technically important measurement methods, those using a throttling device for the flow measurement (standard orifice, standard nozzle, standard venturi nozzle) take first place (differential pressure method). The magnitude actually measured is the differential pressure arising at the throttling device. The volume flow measurement is correlated to a differential pressure measurement. Further details relating to flow measurement with the aid of throttling devices in accordance with the differential pressure method are given in the various sets of rules, including DIN 1952, DIN 1944, VDI guideline 2040, ISO 5167. These rules also describe numerous theoretical and practical sources of error arising in flow measurement. In particular, the error caused by an insufficient undisturbed length of piping both upstream and downstream of the throttling device should be mentioned. Quantitative data (standard orifice) are given in the publications referred to. Difficulties arise when measurements have to be made of quantities outside the range of values covered by the data in rules publications.

The differential pressure is usually measured by means of an U-tube (see point 2., pressure measurement), filled with a measuring fluid (sealing liquid) of density ρ_M . The measuring lines (tubing) must be completely filled with a one-phase medium (e.g. the fluid pumped without any gas inclusions). The volume flow Q (capacity) is determined by the equation

$$Q = \alpha \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}$$

If the U-tube is not calibrated in pressure units but in length units, the equation takes the form

$$Q = \alpha \cdot A \cdot \sqrt{\frac{2g \cdot (\rho_M - \rho) \cdot h}{\rho}}$$

with

- $A = \pi d^2/4$, smallest flow cross-section of the throttling device, of diameter d (see Fig. 2 under standard orifice),
- h difference in levels of the sealing liquid in the two legs of the U-tube (see Fig. 1 under standard orifice),
- Δp differential pressure,
- g gravitational constant,
- α flow coefficient (standard orifice),
- ρ density of pumped medium,
- ρ_M density of sealing liquid.

Mercury is mainly used as the sealing liquid in the U-tube, with $\rho_M = 13545 \text{ kg/m}^3$; for smaller differential pressure heads, ethane tetrabromide with $\rho_M = 2967 \text{ kg/m}^3$ or methane tetrachloride with $\rho_M = 1585 \text{ kg/m}^3$ are used. These densities relate to 20 °C.

Measuring device according to Fig. 7.

Example of capacity measurement with a standard orifice:

- $\rho = 1000 \text{ kg/m}^3$ (water),
- $d = 75 \text{ mm} = 0.075 \text{ m} = 7.5 \cdot 10^{-2} \text{ m}$,
- $\beta = 0.75$ (diameter ratio),
- $Re = 5 \cdot 10^4$ (standard orifices; from the above, according to DIN 1952: $\alpha = 0.7352$,
- $\Delta p = 0.77 \text{ bar} = 0.77 \cdot 10^5 \text{ Pa} = 0.77 \cdot 10^5 \text{ N/m}^2$ (measured).

If the U-tube is calibrated in pressure units, we obtain:

$$Q = 0.7352 \cdot 7.5^2 \cdot 10^{-4} \cdot \frac{\pi}{4} \sqrt{\frac{2 \cdot 0.77 \cdot 10^5}{1000}} = 0.0403 \text{ m}^3/\text{s} = 145 \text{ m}^3/\text{h}.$$

If the U-tube is calibrated in length units, we have the measured value $h = 626 \text{ mm} = 0.626 \text{ m}$ column of mercury and

$$\begin{aligned} Q &= 0.7352 \cdot 7.5^2 \cdot 10^{-4} \cdot \frac{\pi}{4} \times \\ &\times \sqrt{\frac{2 \cdot 9.81 (13545 - 1000) 0.626}{1000}} = \\ &= 0.0403 \text{ m}^3/\text{s} = 145 \text{ m}^3/\text{h}. \end{aligned}$$

In the case of flow measurement by filling a tank, the mass of liquid accumulated in the tank over an adequately long measurement period is determined from the difference in liquid levels or from the difference in liquid levels at a given tank cross section or from in weight force before and after the measurement.

Turbine meters, Woltmann meters (hydrometric vanes) and a large variety of water meters for small capacities operate in the measuring cross section with a calibrated turbine wheel whose revolutions are counted and thus allow to measure the volume flow.

The Inductive Flow Measurement technique (IFM) is a system according to FARADAY's induction law and does not incorporate any restricting or throttling elements or any moving parts (Fig. 9). An alternating magnetic field is traversed by a flow of electrically conducting fluid (minimum conductivity: approx. 20 $\mu\text{S}/\text{cm}$ - normal tap water has a conductance level of 400 $\mu\text{S}/\text{cm}$). 1 S (Siemens) is the electrical conductance of a conductor with an electrical resistance value of 1 Ω (Ohm).

This induces at the measuring electrodes a voltage E , which is directly proportional to the velocity of flow v and, for a known cross section A , to the rate of flow Q .

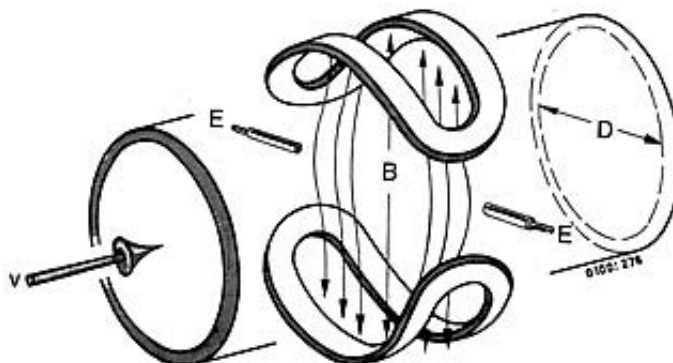


Fig. 9: Measuring principle of inductive flowmeter (IFM)

During the last few years, this measuring instrument has been steadily improved, and it can now be installed, in a special construction version, for applications which require Board of Weights and Measures certification.

Due to the fact that this measuring principle is virtually independent of the flow profile, it becomes possible to adopt appreciably shorter approach flow and wake flow lengths of piping than those required for throttling instruments (standard orifice, standard nozzle, and similar). for the same system accuracy (see VDI 2641).

Velocity head (stagnation pressure) probes (PRANDTL tubes) are mainly used to measure local velocities in aflow. With these velocity head probes and total pressure probes (PITOT tubes), reference is made to the relationship between static and dynamic pressure in a steady flow given by the BERNOLLI equation (fluid dynamics). The directional field of a flow is measured by cylindrical, tapered or wedge-shaped flow probes which are sensitive to direction, or by means of thread probes (if the accuracy requirement is not so great); the latter are the simplest and clearest direction indicating device.

Of the many other measuring methods used, we should mention the hot wire anemometer, which plays an important part in centrifugal pump model tests on air, and more recently also laser anemometers.

4. Power measurement. The mechanical power of rotating machines is usually determined by measuring the torque and the rotational speed (see point 5., measurement of rotational speed) with the aid of two separate measuring devices. There are however also measuring systems with an output variable which is proportional to the product of torque and rotational speed ("electrical product formation").

The torque is measured via its actions of torsion (twist) and moment of reaction. For this purpose, the measuring shaft (torsion dynamometer) or the pump shaft, fitted with wire strain gauges, is used. The direct measurement magnitudes are the angle of twist or the linear strain 45° to the longitudinal axis. The measuring shaft suffers from the disadvantage of influencing the vibration behaviour of the measured object. In any event it represents an additional yielding element in the shafting, which lowers the natural torsional frequency (vibration). To avoid this disadvantage, or in cases where there is no physical possibility of fitting a measuring shaft, four WSG's are fitted directly onto the shaft to be measured. The scientific instrumentation industry has developed several methods of solving the problem of transmitting a feed voltage signal and a measuring signal between the stationary and rotating parts of a machine. We shall only mention slip ring transmission, transformation, capacitive and telemetry transmission among them. A particularly attractive method appears to be that by which the change in resistance of the WSG triggers a frequency change in an electrical oscillating circuit. The frequency signal so obtained can readily be transmitted by transformation.

On torque test beds, the moment of reaction of a pivot mounted driving motor (pendulum type electric motor), which is loaded by a work producing machine or by a dynamometrical brake, is measured. The motor is supported on a force measuring device via a lever of given length.

The most important devices used for measuring torque are described below in greater detail. The Prony brake (Fig. 10) with friction brake is the simplest torque measuring device. The braking lever of known length acts on a balance. Brake shoes are adequate for low powers, but for higher powers brake bands or fluid brakes (e.g. water eddy brakes) have to be used because of the local heat generated. These devices are not applied in m.t. for centrifugal pumps, since they only measure the torque emitted by a driving machine, but not the torque absorbed by a centrifugal pump.

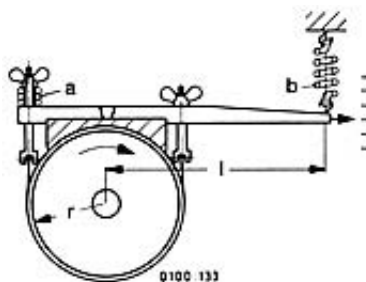


Fig. 10: Prony friction brake a spring; b spring balance

Torsion dynamometers are mechanical elements usually fitted between a motor and a work producing machine or between a generator and a prime mover, which transmit the power and at the same time measure the torque. The torque measurement on torsion dynamometers consists of measuring the twist angle of a given length of shaft, the so-called torsion rod.

The torsion rod (Fig. 11) is a length of shaft capable of twisting (calibrated, value of the spring rate stamped on it), which is fitted at a suitable spot, e.g. between motor and centrifugal pump. There is a disc at either end of this length of shaft. Under the influence of the torque T , the two discs twist in relation to each other. The twist angle is a measure of the torque T . Reading of this angle is effected with the aid of stroboscopic lighting (light pulse stroboscope, perforated disc or similar).

The torque metering hub (Fig. 12) represents a further development of the torsion rod. Various (mainly electronic) systems are used to transmit the signal. The difficulties reside mainly in the transmission of the signal from the rotating to the stationary system.

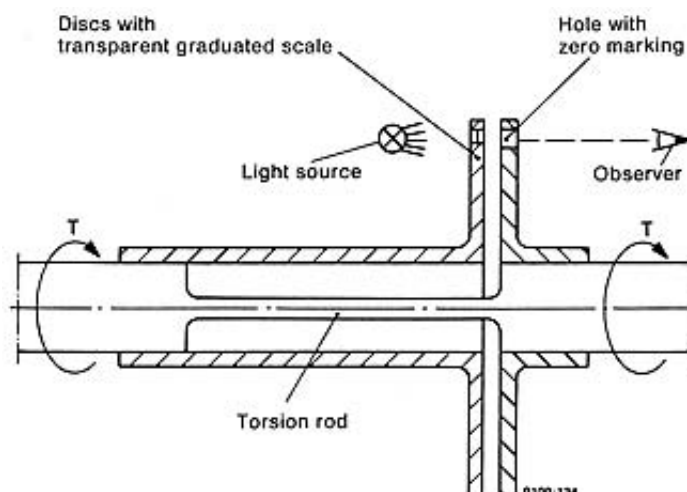


Fig. 11: Torsion rod (diagrammatic)

Slip-rings were used at first, but they were frequently liable to failure, and are now replaced in high accuracy measuring systems by non-contact transmission systems. Fig 12 illustrates a torque metering hub in perspective representation. Coil S_1 is displaced in relation to coil S_2 by the twisting of measuring rod T_d and this creates a change of coupling. The sinusoidal feed voltage is fed to coil S_1 via transformer T_1/T_p ; the measurement signal created by induction at coil S_2 is led to the measuring instrument via transformer T_2/T_s . The measuring

arrangement is illustrated in Fig. 13.

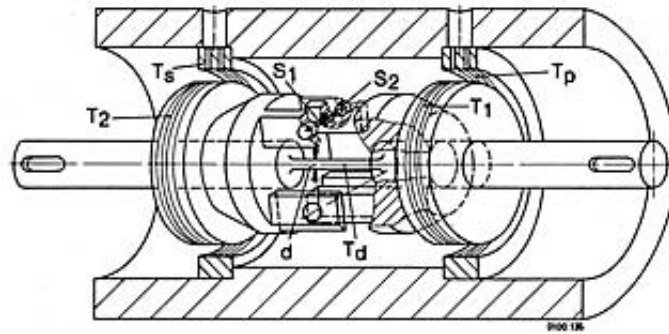


Fig. 12: Torque metering hub rotating/stationary

Other systems use the change of resistance of one or more WSG's glued onto the measuring shaft as measurement signal (Fig. 14). Depending on the system, amplitude or frequency modulation is used to transmit and display the measurement signal.

The hydrostatic torque metering coupling consists of two components fitted with paddle blades (paddle blade housing and paddle wheel) pivoted inside one another. The two form a multicell positive displacement system. A flow of oil fed into the metering coupling is influenced by the torque which passes through the coupling in such a way that an oil pressure proportional to the instantaneous torque at any moment is built up and measured.

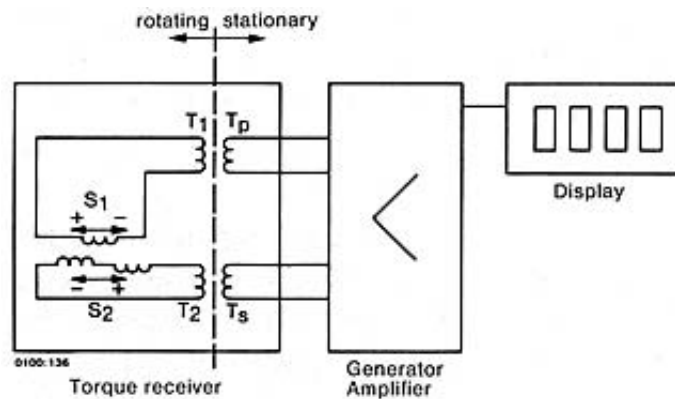


Fig. 13: Twist angle measurement via inductive displacement measurement

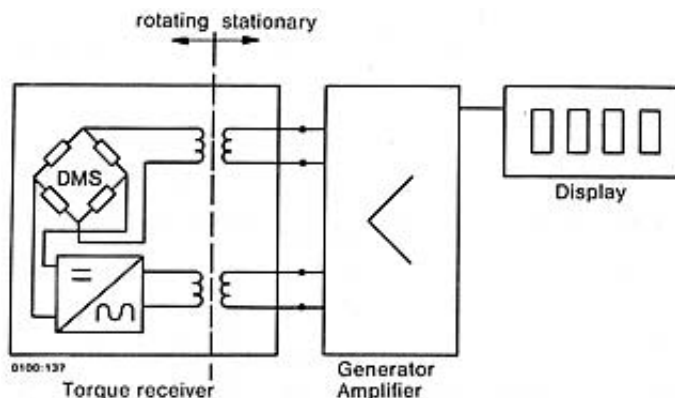


Fig. 14: Twist angle measurement via wire strain gauges

As regards the measurement of electrical power, the following points should be mentioned (see Table):

- in the case of d.c.: measurement of voltage U and current J with a voltmeter and an ammeter;
- in the case of single-phase a.c.: measurement of electrical power P_w with a wattmeter;
- in the case of three-phase current: measurement of two partial powers P_{w1} and P_{w2} with two wattmeters as illustrated in Fig. 15 a; the total electric power $P_w = P_{w1} + P_{w2}$.

This two wattmeter method can always be used (even if the neutral conductor is inaccessible, if the phases are loaded unequally, or if the power fluctuates). If the power remains constant, P_{w1} and P_{w2} can also be measured in succession with the same wattmeter (change-over switch).

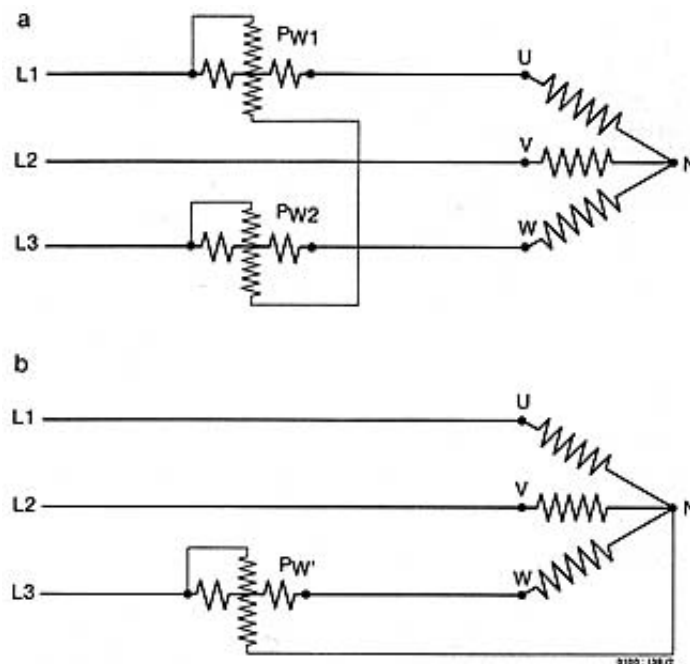


Fig. 15: Power measurement in a three-phase current system

- a) two wattmeter method (power user: motor, for example);
- b) simplified method when the neutral point is accessible or an artificial neutral point is used

If the power factor $\cos \varphi$ is less than 0.5, the indicator needle of one of the wattmeters swings to the negative side; in that case, the voltage circuit of the wattmeter in question should be reversed, and the reading subtracted so that $P_w = P_{w1} - P_{w2}$.

Rule: for $P_{w1} = 0$ is $\cos \varphi = 0.5$;
 for $P_{w1} = 0.5 \cdot P_{w2}$ is $\cos \varphi = 0.866$;
 for $P_{w1} = P_{w2}$ is $\cos \varphi = 1.0$.

If the neutral conductor is accessible (motors with star connection), and the phase loading is uniform, it will suffice to measure one partial power P_w' only (see bottom diagram of Fig. 15 b); the total electric power in this case is $P_w = 3 \cdot P_w'$.

Instead of using individual wattmeters, one can also use wattmeters with double or treble measuring elements. They indicate the total power in a single reading, and have the added advantage of compensating power oscillations (hunting), which, in the case of individual wattmeters, often lead to phase-displaced fluctuations of the individual instrument needle deflections. The equations for the calculation of the electric power in kW from the voltage U in V, the current J in A, the motor efficiency η_M and the power factor $\cos \varphi$ are given in the Table.

Table: Powers of electric motors: U in V J in A

Nature of current	Absorbed electric power kW	Power output in kW
direct current	$P_w = \frac{U \cdot J}{1000}$	$P_M = \frac{U \cdot J \cdot \eta_M}{1000}$
alternating current	$P_w = \frac{U \cdot J \cdot \cos \varphi}{1000}$	$P_M = \frac{U \cdot J \cdot \cos \varphi \cdot \eta_M}{1000}$
three-phase current	$P_w = \frac{\sqrt{3} \cdot U \cdot J \cdot \cos \varphi}{1000}$	$P_M = \frac{\sqrt{3} \cdot U \cdot J \cdot \cos \varphi \cdot \eta_M}{1000}$

5. Measurement of rotational speed of pumps. The measurement of the rotational speed is usually effected with the aid of:

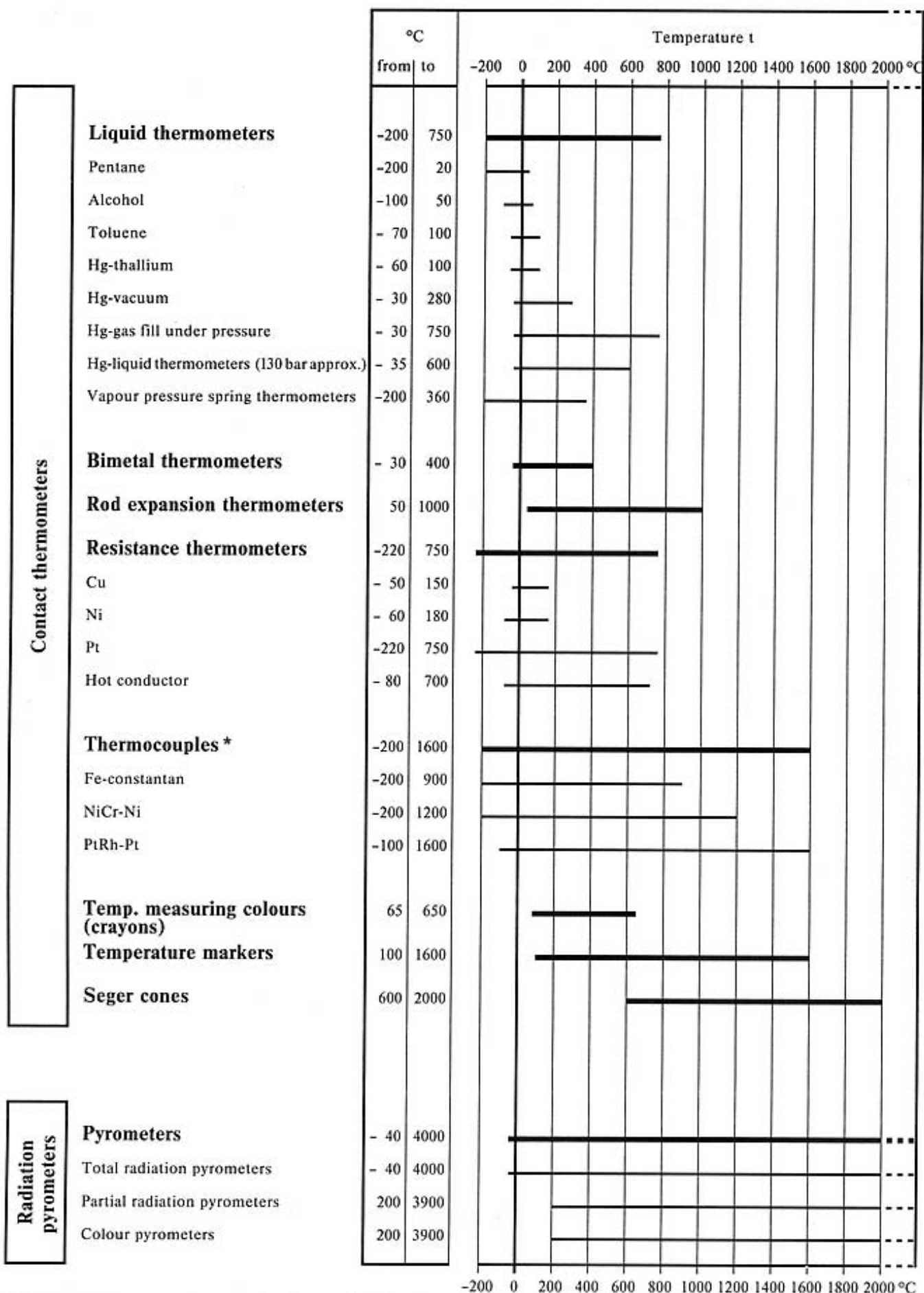
- a) electromagnetic induction rotational speed transmitters, which generate an electric voltage proportional to the rotational speed;
- b) hand tachometers operating on the eddy current principle (hand tachometers applied to the end face of the pump or motor stub shafts;
- c) electronic counters (when a high degree of accuracy is required), in which the pulses proportional to the pump speed are usually transmitted without physical contact (e.g. optically or magnetically) from the shaft to the counter;
- d) vibrating reed frequency meters (tachometers), placed on the casing of the machine or on the pipings;
- e) slip meters (slip coils or slip magnetic needles) arranged at a given axial distance from the motor.

Electrical rotational speed measurement methods mainly make use of a.c. measuring generators (analogue signal) or of revolution impulse transmitters (inductive or optical pulses) as rotational speed transmitters. In the case of the analogue method, the indication is effected by means of a voltmeter with a rectifier. These measuring devices are equally suitable for the detection of non-constant rotational speeds. In the case of the impulse method, the number of revolution impulses (or rotating "occurrences") per unit of time, or the number of units of time per impulse period is counted by an electronic counter. This method is extremely accurate, because of the digital counting and the very short cycle of the counters (quartz generators). The great advantage of this method is the ability to operate with non-contact and non-interacting receivers.

If the electric driving motor is supplied together with the pump by the pump manufacturer, e.g. in the case of close-coupled pumping sets, the supply contract may be drawn up to specify the frequency instead of the rotational speed. In such cases, the frequency must be measured in lieu of the rotational speed.

DIN 1944, October 1968 edition, gives further details of rotational speed measurement procedures.

6. Temperature measurement. So-called contact thermometers, in which the temperature sensor is in direct contact with the medium to be measured, are mainly used in centrifugal pump technology. The intensive heat exchange between the medium and the temperature sensor is brought about by the arrangement, e.g. in the flow of medium itself, also by a suitable material selection and by design measures on the sensor. Steps must be taken to prevent the flow of heat to the outside through the temperature sensor. Conventional contact thermometers include liquid thermometers, resistance thermometers and thermocouples. Books of rules about these include: VDE/VDI 3511, 3512, DIN 16 160 and many others. The temperature measurement method selected is governed on the one hand by the temperature range concerned (Fig. 16) and on the other hand by the special requirements of a given measuring task (e.g. installation conditions, accuracy, dynamics of the measurement magnitudes, signal transmission).



* additionally thermocouples see under thermoelectric series

7. Noise measurement. Noise measurements on machines are carried out in accordance with DIN 45635. This standard consists of a general set of rules for measurement (Part 1), and of a number of addendum sheets for special types of machines (e.g. for centrifugal pumps - Part 24) which contain additional stipulations. DIN 45635 establishes the precondition for the determination of noises radiated to the surrounding air by centrifugal pumps (noise emission) according to uniform procedures, which ensure comparable results. DIN 45 635, Part 24 encompasses all "liquid pumps" (pump types, centrifugal pumps, positive displacement pumps, jet pumps see under deep well suction device, gas mixture siphons, hydraulic rams, elevators). The following concepts are frequently used in sound measurement technology.

a) A-sound pressure level L_{pA} and A-impulse sound pressure level L_{pAI} . As the acoustic measurement magnitude of the noise generated by the machine, either the A-evaluated sound pressure level L_{pA} in dB, referred to in the standards as A-sound pressure level, or the A-impulse sound pressure level L_{pAI} are used, which are measured with a measuring device in accordance with DIN IEC 651.

b) Measuring surface sound pressure level \bar{L}_{pA} . This is the A-sound pressure level averaged over the measuring surface S , and if necessary cleared of the influence of any extraneous noises, and interactions from the room; it is given in dB.

c) A-acoustic power level L_{wA} . The measure of the overall noise radiated by the machine to the surrounding air under clearly defined installation conditions is given by the A-evaluated acoustic power level L_{wA} in dB, and is designated A-acoustic power level in the DIN standard.

d) Measuring surface. The measuring surface is an imagined surface surrounding the machine, on which the measuring points are situated.

e) Measuring surface dimensions L_S . As the measuring surface S is only used in conjunction with logarithmated ratio magnitudes, it is not given direct in m^2 , but in the form of a measuring surface dimension:

$$L_S = 10 \cdot \lg \frac{S}{S_0}$$

with

$S_0 = 1 \text{ m}^2$ reference surface area,

L_S in dB.

f) Sound spectrum. The sound spectrum describes the level distribution within the interested frequency range. For this purpose, the sound pressure levels or acoustic power levels are determined in consecutive frequency bands, e.g. of octave or tertial width.

g) Sound level meter. The A-sound pressure level L_{pA} shall be measured by means of a precision sound level meter, or by means of a precision impulse sound level meter in accordance with DIN IEC 651.

h) Filters. The octave filters (analyzers) shall be basic element filters and shall comply with DIN 45 651; the tertial filters shall comply with DIN 45652.

i) Testing of the measuring device. Before each series of measurements, the sound level meter shall be checked in respect of correct indication by means of the accompanying or built-in testing device. This checking procedure shall be repeated at intervals in the case of protracted measurements. The precision sound level meters used for noise acceptance tests shall be officially tested at least once every two years.

The measurement conditions shall specifically lay down very precisely which accessories (e.g. lubricating oil systems, valves and fittings, minimum flow devices, nameplates, piping, vessels, coolers) form part of the pump to be tested, and also whether the driver, gearbox or other plant components are to be included in the measurement. The operating condition during the measurements shall enable the characteristic noises which arise in continuous service at the design point conditions to be detected.

If the measurement is not carried out on site, the pump shall be installed under conditions as similar as possible to those existing on site.

If the installation under site service conditions is not known, or if there are various possible different site arrangements, then the mode of installation used for the measurements shall be stated in the measurement report. When measuring on the test bed, the pump shall be installed in such a way that no additional sound radiation from structural elements, drive, loading or other test bed devices occurs. The measuring surface shall be a surface parallel to the pump surface at a distance of one metre, and shall approximate the shape of a simple parallelepiped, ignoring and projecting components which do not play an important part in radiating sound. The measuring surface can either be enclosed in itself or it can terminate at sound-reflecting limiting surfaces of the installation site (e.g. at the floor or at walls). If the measuring surface sound pressure level at 1 m distance cannot be detected with absolute certainty, other distances of the measuring surface are permissible. Any distance differing from 1 m shall be mentioned in the measurement report.

When carrying out the measurement, a distinction must be made between measurement of the A-sound pressure level and measurement of the sound pressure spectrum:

a) Measurement of A-sound pressure level L_{pA} . For steady noises, the sound pressure level at the individual measuring points is measured on the "slow" position of the sound level meter. The small level fluctuations of less than 5 dB which are frequently present in such cases can be averaged by reading the arithmetic average of the indication direct. Noises containing impulses are measured with an impulse sound level meter. It is recommended to measure noises containing impulses and noises with a level which fluctuates with time both in the "impulse" and in the "slow" positions; the difference in indications shall be noted in the measurement report.

b) Measurement of sound level spectrum. The sound level spectrum is determined in the partial frequency ranges, usually unevaluated. Measurement in frequency bands of one octave width is sufficient, and should preferably be adopted in cases where no distinctly audible individual tones (notes) are apparent; otherwise tertial or narrow band spectra are indicated.

The following applies to the calculation of the acoustic magnitudes: the measuring surface S in m^2 shall be calculated from

$$S = 4 (ab + ac + bc)$$

with

- a length of right parallelepiped in m (length of machine \times 0.5 + measuring distance),
- b width of right parallelepiped in m (width of machine \times 0.5 + measuring distance),
- c height of right parallelepiped in m (height of machine + measuring distance).

The measuring surface dimension L_S in dB is calculated from the measuring surface content:

$$S = \frac{4 (ab + ac + bc)}{S_0}$$

with $S_0 = 1 m^2$, reference surface area

The following applies to the evaluation:

a) Calculation of measuring surface sound pressure level \bar{L}_{pA} . The measuring surface sound pressure level is calculated from the measurement values obtained on the measuring surface S , taking into account the influences of extraneous noises and interactions from the room.

b) Calculation of A-acoustic power level L_{wA} . The A-acoustic power level is calculated as the sum of the measuring surface sound pressure level \bar{L}_{pA} plus the measuring surface dimension L_S :

$$L_{wA} \cong \bar{L}_{pA} + L_S$$

Because of the tolerances in the characteristics of the measuring instruments, the possibilities of disturbances during the measurement and the uncertainty relating to the corrections given above, the uncertainty in measurement of the A-sound pressure level or of the measuring surface sound pressure level amounts to ± 2 dB even under favourable conditions.

Measuring Tolerance

Meßspiel
Jeu de mesure

see [Uncertainty of Measurement](#)

Mechanical Drive

Mechanischer Antrieb
Commande mécanique

see [Drive](#)

Mechanical Efficiency

Mechanisecher Wirkungsgrad
Rendement mécanique

The m.e. η_m (efficiency) is the ratio of the [shaft power](#) P minus the mechanical power losses P_m and the shaft power P:

$$\eta_m = \frac{P - P_m}{P}$$

The power loss P_m consists mainly of the frictional losses in the pump bearings ([plain bearings](#), [anti-friction bearings](#)) and in the [shaft seals](#), and is usually of the order of 0.5 to 2 % of the [shaft power](#) P (the lower loss figures usually apply to the more powerful pumps).

Mechanical Seal

Gleitringdichtung
Garniture mécanique

see [Shaft Seals](#)

Medium Pressure Pump

Mitteldruckpumpe
Pompe à moyenne pression

M.p.p. is a [centrifugal pump](#) with a [head](#) ranging between 80 and 200 m (e.g. [irrigation pump](#)).

In contrast, we have [low pressure](#), [high pressure](#) and [super pressure pumps](#).

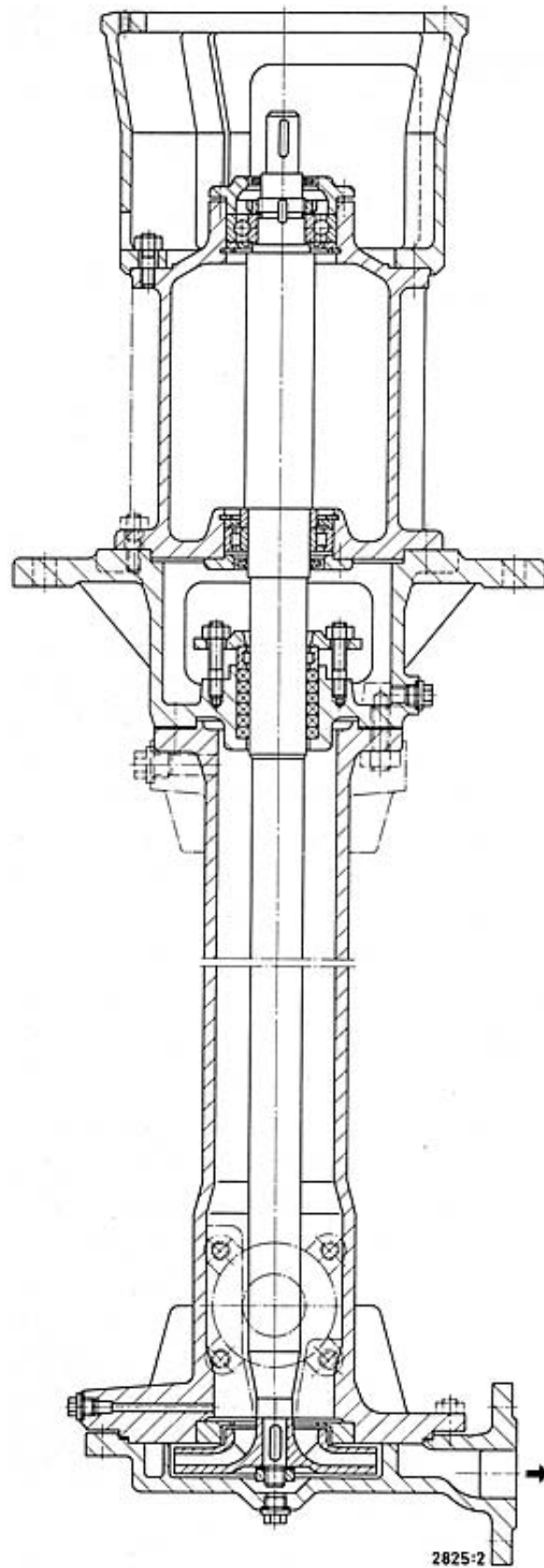
Mercury Pump

Quecksilberpumpe
Pompe à mercure

The m.p. is used to pump mercury from the chloralkali electrolysis according to the mercury process. A stream of mercury flowing over the cell bottom of the electrolysis serves as a cathode. Amalgam from the mercury is formed on this cathode, and this is re-converted to mercury in a secondary cell, the so-called decomposer or electrolyses under simultaneous formation of alkaline solution and hydrogen; the mercury is returned to the electrolysis by the

m.p.

M.p.'s are usually vertical, dry installation centrifugal pumps, mounted on the side of the electrolysis cell or on the decomposer. The pump shaft with an overhung-impeller is supported in grease-lubricated anti-friction bearings and is designed for subcritical running (see illustration). The pump is arranged in such a way that the elevation of the mercury level surface can be kept as low as possible above the impeller inlet, which is usually arranged facing upwards. The shaft seal is a soft packed stuffing box arranged inside the suspension pipe of the pump at an adequate height above the surface of the mercury level, and it is not in contact with the liquid pumped. In most cases it is fed with sealing water, and its purpose is to prevent the leakage of water contaminated with mercury and simultaneously to prevent the ingress of dirt. By using double acting mechanical seals (shaft seals) fed with sealing water, any leakage of mercury at the shaft seal can be completely avoided. In certain cases, the use of completely grandness canned motor pumps has been considered for this duty.



Vertical single suction single stage double volute casing pump with radial pump suction and discharge branch (mercury pump)

Meridional Section Plane

Meridianschnittebene
Plan de coup méridienne

see [Flux Line](#)

Mine Drainage Pump

Wasserhaltungspumpe
Pompe d'exhaure

Drainage is always necessary when a construction project is carried out below the groundwater level. In *underground working* or mining operations, the water which penetrates into the galleries must be pumped up to the surface; in *opencast mining operations*, the groundwater level is lowered through drainage well trenches until the pit is dry; the same applies to excavations on building sites.

Overcoming the high heads in *underground workings* (1000 m or more) requires the use of multistage centrifugal pumps with "single suction, with all impellers facing the same way", "single suction, with back-to-back or opposed impellers" or "double suction" impellers (Figs. 16, 17 and 18 under impeller).

As the water involved often contains solids and matter in suspension (erosion) and is often chemically corrosive (selection of materials), impellers with wide channels, shaft protecting sleeves and wear-resistant or corrosionresistant materials (corrosion, table of corrosion resistance) are adopted. M.d.p.'s installed as horizontal dry installation pumps usually have no positive suction head (suction behaviours); they are driven by flameproof electric motors. Vertical pumps in the form of underwater motor pumps have many advantages for automatically operated mine drainage systems. The submersible motors are water-filled and arranged beneath the pumps. They are flooded and thus adequately cooled. Flameproofing is only required for the cable connections.

In *opencast mining* (strip mining), drainage well trenches equipped with underwater motor pumps are arranged at the edge of the workings and also in the various levels in staggered arrangement. The submersible motors of m.d.p.'s can be built for high tensions of 3 or 6 kV (in some cases up to 10 kV) as well as for low tensions. The capacities Q and heads H of m.d.p.'s in underground mining can attain up to $Q = 900 \text{ m}^3/\text{h}$ and $H = 1050 \text{ m}$, and in opencast mining up to $Q = 1800 \text{ m}^3/\text{h}$ and $H = 400 \text{ m}$. It is technically feasible to go beyond these limits.

Drainage or the lowering of the groundwater level is also necessary in the case of excavations (underground railways, bridges, building and construction work). Large building sites are protected by drainage well trenches equipped with underwater motor pumps (lowering of the groundwater level), similar to the arrangements made in opencast mining. Heavily contaminated leakage water (seepage water) is often pumped out by portable submersible motor pumps. The motors of the latter are arranged above the pump. Their impellers are specially designed to handle dirty effluent (sewage pump). A mechanical seal (shaft seals) protects the motor against the ingress of water. The motor cooling is adequately sized to enable the pumping set to run non-submerged if necessary.

Minimum Flow

Mindestförderstrom
Débit minimum admissible

see [Boiler Feed Pump](#), [Head](#), [Valves and Fittings](#)

Minimum Positive Suction Head

Mindestzulaufhöhe
Hauteur minimum d'amenée

see [Suction Behaviour](#)

Mixed Flow Impeller

Halbaxialrad

Roue semi-axiale

see [Impeller](#)

Mixed Flow Pump

Schraubenradpumpe

Pompe Hélicocentrifuge

M.f.p.'s are [centrifugal pumps](#) with mixed flow [impellers](#). Their [specific speeds](#) n_q lie between 35 and 80 min^{-1} for the slow specific speed m.f.p.'s, and between 80 and 160 min^{-1} for the high specific speed m.f.p.'s (in special cases even higher), and they therefore represent the transition range between radial and axial pumps ([propeller pump](#)). The impellers of m.f.p.'s are combined with a volute casing (Fig. 1) or annular casing ([pump casing](#)) in the case of the lower specific speed pumps, and with a tubular casing ([pump casing](#), [tubular casing pump](#)) together with a [diffuser](#) in the case of the higher specific speed pumps (Fig. 2). The optimal n_q range for tubular casing m.f.p.'s is not sharply defined with regard to construction costs and efficiency. However, from $n_q = 130 \text{ min}^{-1}$ upwards (approx.), the peripheral component of the [absolute velocity](#) at the impeller outlet becomes so small in comparison with the [flow velocity](#) at the [pump suction branch](#) that the end cross-sections of the volute or annular casing required to convey the flow ([capacity](#)) would become unmanageably large. In such cases, the m.f.p. would have to be provided with a radial casing of very inflated dimensions, which might conceivably be made of concrete (Fig. 6 under cooling water pump), but would be very expensive if it were made of cast iron or steel. For this reason, m.f.p.'s of higher [specific speeds](#) are usually provided with an axial tubular casing and an "onion-shaped" or axial [diffuser](#) through which the fluid flows ([capacity](#)) towards the [pump discharge branch](#) (Fig. 2).

The range of heads of the m.f.p. with tubular casing complements that of the [propeller pump](#) at the top end of the [head](#) range. As the [circumferential velocity](#) of the helical impeller is limited to 25 to 30 m/s because of the danger of [cavitation](#) ([suction behaviour](#)), the m.f.p. can only exceed its max. [head](#) limit of $H = 60 \text{ m}$ by multistage design ([multistage pump](#) in practice with two to three stages only).

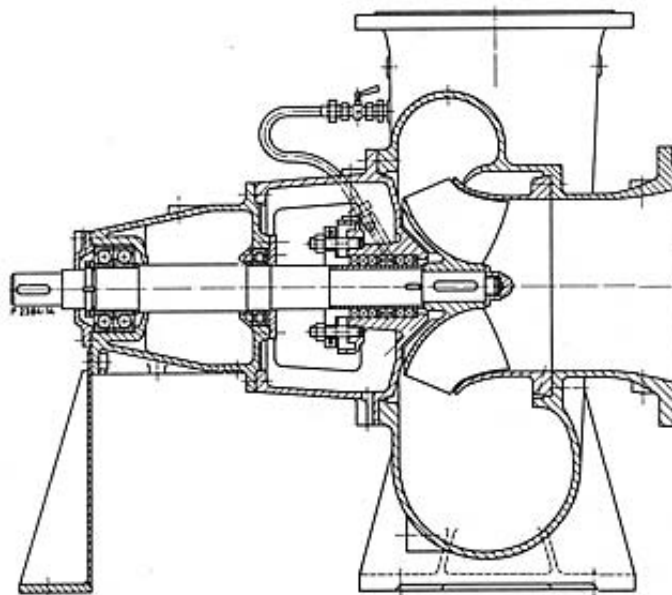


Fig. 1: Volute casing pump with open helical impeller and free vortex volute

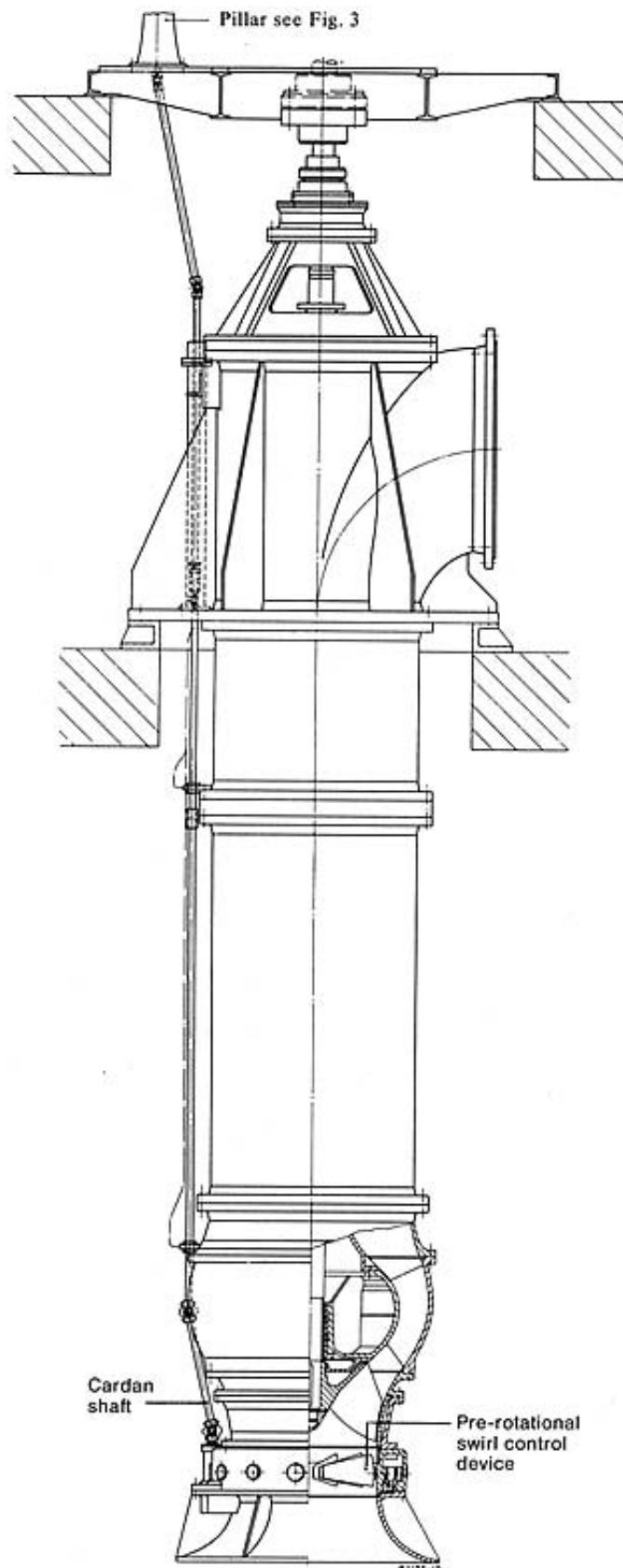


Fig. 2: Mixed flow pump with pre-rotational swirl control device

Because of the impeller geometry, control by blade pitch adjustment (impeller blade pitch adjustment) is not practicable on m.f.p.'s, as opposed to propeller pumps, but pre-rotatio | swirl adjustment is very much recommended. Such a pre-rotational swirl control device (by adjustable inlet guide vanes) is actuated by a linkage system consisting of cardan shafts outside the tubular casing, led above floor level (Figs. 2 and 3). In the case of multistage m.f.p.'s with tubular casing, such a pre-rotational swirl control system can be and has been successfully fitted upstream of the impeller of each stage. In many cases, however, the overall control effect of only one

pre-rotational swirl control device fitted upstream of the first stage has proved quite adequate.

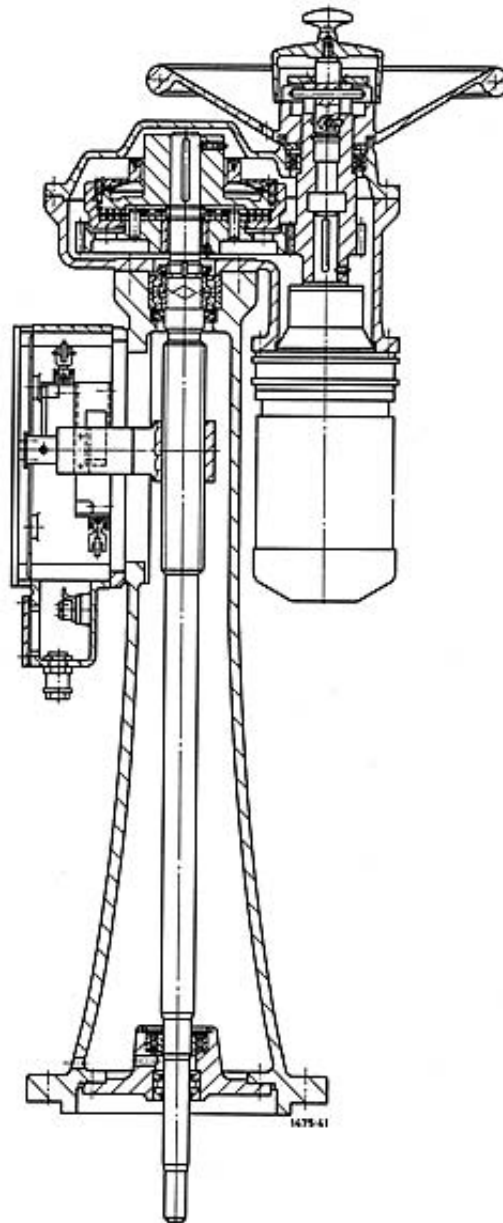


Fig. 3: Pillar with hand wheel and adjusting motor, fitted with device for direct mechanical display and remote electrical display of the guide vane pitch angle of the pre-rotational swirl control device. This illustration complements Fig. 2 (mixed flow pump)

If the fluid pumped is clean and the head is low (up to 15 m), the front coverplate on the helical impellers can be dispensed with, as illustrated in Fig. 1 (open impellers). On the other hand, a closed impeller is to be preferred for high heads, to reduce the clearance gap losses (clearance gap width).

Model Laws

Modellgesetze

Lois de similitude

When investigating flow phenomena, it is often necessary to carry out the investigations on geometrically similar models instead of on the original (similarity conditions). Physical similarity is also required, as well as geometric similarity, i.e. the basic physical laws (differential equations including boundary conditions) must have an invariant formulation in respect of similarity transformations. In general this is achieved by dividing all the physical magnitudes which arise by exponential products of magnitudes (characteristic number) which are characteristic of

the phenomenon, so that one is left with ratio magnitudes of the unit 1.

Physical similarity is achieved if the ratio magnitudes or so-called characteristic numbers of the original and of the model are the same. The relationships obtained between the physical magnitudes in the original and in the model by means of the characteristic numbers are called m.l.

Example: the following characteristic numbers can be formed for a centrifugal pump, assuming a frictionless, incompressible, non-cavitating flow, with the aid of the characteristic magnitudes D (impeller diameter) n rotational speed, g gravitational constant and ρ density of pumped medium:

- of flow velocity $v/(n \cdot D)$,
- of pressure $p/(\rho \cdot n^2 \cdot D^2)$,
- of specific energy $Y/(n^2 \cdot D^2)$,
- of head $g \cdot H/(n^2 \cdot D^2)$,
- of capacity $Q/(n \cdot D^3)$,
- of shaft power $P/(\rho \cdot n^3 \cdot D^5)$.

Therefore the following m.l. apply to two geometrically similar pumps operating under physically similar conditions:

- for the flow velocities $\frac{v_1}{v_2} = \frac{n_1 \cdot D_1}{n_2 \cdot D_2}$,
- for the pressures $\frac{p_1}{p_2} = \frac{\rho_1 \cdot n_1^2 \cdot D_1^2}{\rho_2 \cdot n_2^2 \cdot D_2^2}$,
- for the specific energies $\frac{Y_1}{Y_2} = \frac{n_1^2 \cdot D_1^2}{n_2^2 \cdot D_2^2}$,
- for the heads $\frac{H_1}{H_2} = \frac{n_1^2 \cdot D_1^2}{n_2^2 \cdot D_2^2}$,
- for the capacities $\frac{Q_1}{Q_2} = \frac{n_1 \cdot D_1^3}{n_2 \cdot D_2^3}$,
- for the values of shaft power (assuming identical pump efficiencies) $\frac{P_1}{P_2} = \frac{\rho_1 \cdot n_1^3 \cdot D_1^5}{\rho_2 \cdot n_2^3 \cdot D_2^5}$.

Since the pump efficiencies are more or less dependent on the conditions of friction, they are also subject to other conversion laws (efficiency re-evaluation).

The selection of characteristic magnitudes for the determination of the characteristic numbers is largely arbitrary, and for instance when studying the theory of flow in radial impellers, the circumferential velocity u of the impeller, its outlet diameter D and the outlet width b of the vane passage are selected as characteristic magnitudes, leading to the formation of the two characteristic numbers:

- flow coefficient $\Phi = Q/(\pi \cdot D \cdot b \cdot u)$,
- pressure coefficient $\Psi = \Delta h_s/(u^2/2)$

where Δh_s is the isentropic increase (entropy) of the generalized specifics enthalpy of the fluid pumped. If we insert $\Delta h_s = Y$ (specific energy) we obtain the pressure coefficient in its known form:

$$\Psi = \frac{2 \cdot Y}{u^2} = \frac{2g \cdot H}{u^2}.$$

As a general rule, a length l and a velocity v are selected as characteristic magnitudes for flow investigations.

Flows subject to friction are characterized by the kinematic viscosity ν (viscosity); the reciprocal value of the characteristic number of kinematic viscosity is the

$$\text{REYNOLDS number } Re = \frac{v \cdot l}{\nu}$$

which also gives the ratio of inertia force to frictional force.

If the force of gravity has to be taken into account as an external force, the characteristic number of acceleration due to gravity (gravitational constant) is $g \cdot l / v^2$; its reciprocal value is the

$$\text{FROUDE number } Fr = \frac{v^2}{g \cdot l}$$

Fr can also be interpreted as the ratio of inertia force to force of gravity.

If further physical phenomena such as compressibility, heat transport, surface tension etc. have to be taken into account, further characteristic numbers must be introduced.

As characteristic numbers are not independent of one another, it becomes possible that physical similarity can no longer be achieved when several characteristic numbers have to be taken into account.

Model tests are widely used not only to investigate flow mechanics problems, but also design strength and heat transport problems.

Model Test

Modellversuch

Essai de maquette

see [Model Laws](#)

Moment of Gyration

Schwungmoment

Moment de giration

The m.o.g. represents a characteristic magnitude (unit kg m^2) for the "inertia in respect of rotations" (starting process) of a rigid rotating body (e.g. the rotor of a centrifugal pump and drive). The m.o.g. is designated and calculated by means of mD^2 where m is the mass (weight) of the rotating body and D the so-called *diameter of gyration*, with

$$D^2 = \frac{\sum (m_i \cdot d_i^2)}{\sum m_i}$$

with

m_i individual mass on rotation circle of diameter d_i and

$\sum m_i = m$ sum of the individual masses.

The m.o.g. mD^2 is often replaced by the *mass moment of inertia* J, with

$$mD^2 = 4 J$$

(same units used for both mD^2 and J).

Moment of Inertia

Massenträgheitsmoment Trägheitsmoment
Moment d'inertie de masse, moment d'inertie

see [Moment of Gyration](#)

Monitoring Device

Kontrollgerät
Instrument de contrôle

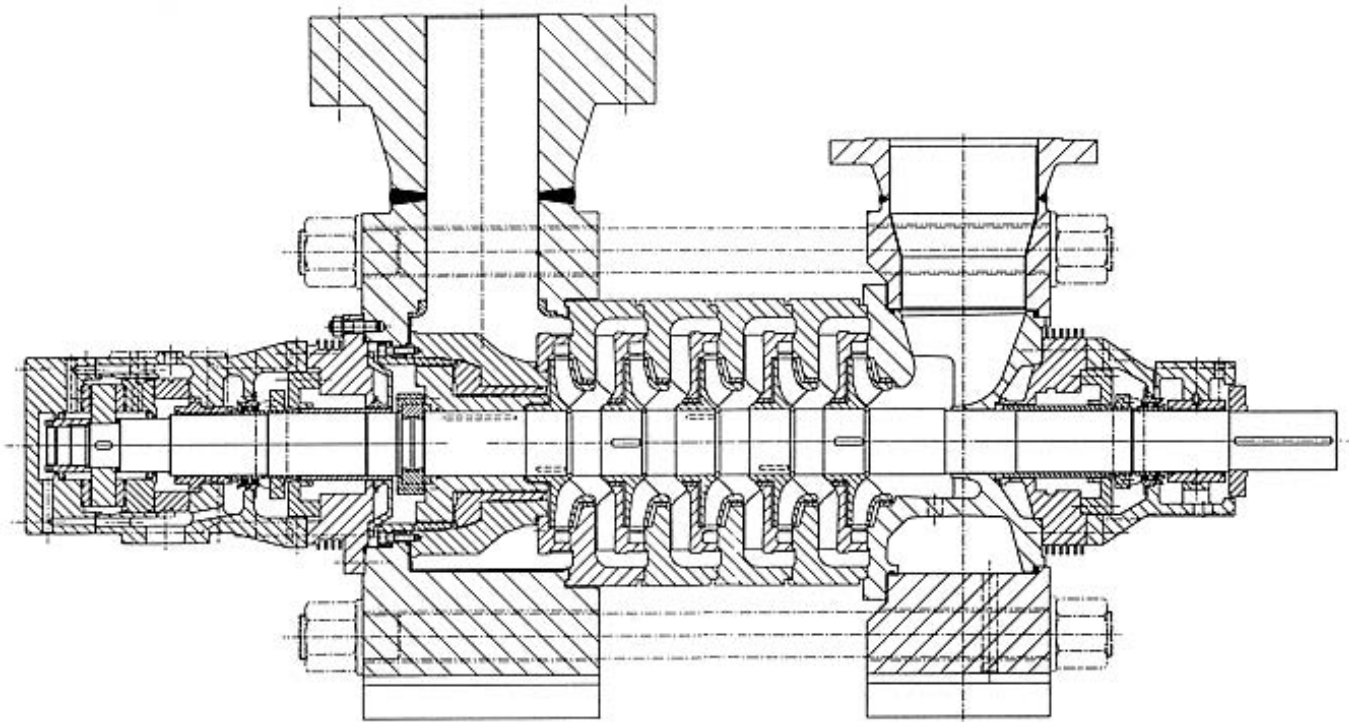
see [Measuring Technique](#)

Multistage Pump

Mehrstufige Pumpe
Pompe multicellulaire

The head of a single stage centrifugal pump is governed by the circumferential velocity and by the type of impeller. There are certain limits set to the rotational speed of the pump by the inlet conditions, and also to the increase in size of the impeller diameter by the increase in impeller side friction, which results in a steady decrease in pump efficiency; when these limits are exceeded, it becomes necessary to fit several impellers in sequence on the same pump, as stages. If the dimensions and speeds remain constant, the increase in number of stages of a m.p. will not alter the capacity, but the head and shaft power will increase proportionally to the number of stages.

Pumps fitted with several identical stage casings arranged in tandem behind each other are called ring section pumps; this type of construction is frequently adopted in barrel pumps and in boiler feed pumps (see illustration). Each stage consists of an impeller, a diffuser and sometimes combined with the diffuser) a row of return guide vanes, which are all arranged inside each stage casing. A suction casing with a radial, or sometimes an axial (end) suction branch (e.g. on condensate pumps) is arranged upstream of the first stage, whereas the final stage is arranged in the discharge casing (pump casing), which also contains the balancing device (axial thrust) and one of the shaft seals. The design of these pump components is independent of the number of stages, and only the common pump shaft, the tiebolts and the baseplate (pump foundation) must be adapted to the relevant number of stages in each case.



Multistage ring section type boiler feed pump

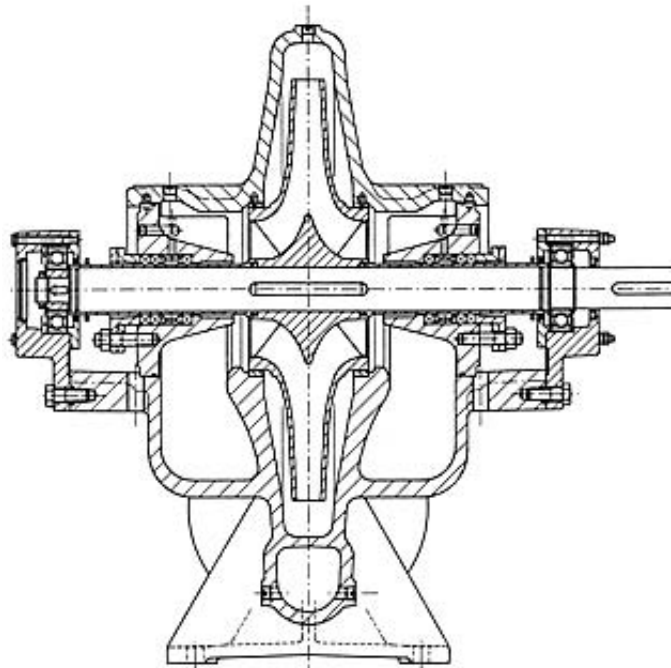
The individual stages of the m.p. do not have to be arranged in tandem, i.e. facing the same way; they can be arranged back-to-back in pairs or groups, to facilitate the balancing of the axial thrust (back-to-back impeller pump). Typical examples of such arrangements are to be found in pipeline pumps.

M.p.'s fill in the higher pressure regions of the grids (characteristic curve) of pump series in an economic fashion. Other advantages of m.p. construction comprise the possibility of a simple extraction after a given stage, and the provision of empty stages for subsequent increases in system pressures. A disadvantage in the case of a very large number of stages is the increasing sensitivity of the pump rotor to vibrations, e.g. during operation under cavitation or part load (operating behaviour).

Multisuction Pump

Mehrströmige Pumpe
Pompe à plusieurs flux

M.p., is usually a double suction pump in practice. In multistage centrifugal pumps, the head can be increased at constant capacity by arranging several impellers in sequence, whereas in m.p.'s the impellers are arranged in parallel, and the capacity can therefore be increased at constant head. The m.p. principle is adopted when the capacity of a centrifugal pump becomes too large for the inlet cross-sections of an impeller, or when the flow velocity in the inlet cross-section of the first stage impeller has to be reduced for reasons of suction behaviour. In the case of the double suction pump illustrated, the pair of impellers is arranged back-to-back on the pump shaft, to compensate the axial thrust at the same time; this advantage, together with the common pump suction branch and the common volute casing (pump casing) make this double suction arrangement an attractive one, e.g. for cargo oil pumps. In principle, four and sixfold suction pumps can be built (Fig. 18 c under impeller), but the branches and connecting pieces which their design entails make them very expensive, and they are seldom used. Multisuction arrangements can of course also be combined with multistage arrangements, e.g. in pipeline pumps and large water supply pumps (Fig. 1 under back-to-back impeller pump).



Double suction volute casing pump

N

NAVIER-STOKES-Equation

NAVIER-STOKES-Gleichung
Équation de NAVIER-STOKES

see Fluid Dynamics

Net Positive Suction Head (NPSH)

Haltedruckhöhe, NPSH
Charge nette absolue d'aspiration, NPSH

The NPSH value is an important concept for judging the suction behaviour of a centrifugal pump, i.e. it allows a prediction to be made of the margin of safety against the effects of cavitation during operation of the pump.

The brochure entitled "NPSH in Centrifugal Pumps - Significance, Calculation, Measurement" of EUROPUMP (European Committee of Pump Manufacturers) contains a detailed explanation of the NPSH value concept. This brochure was published in 1974 by Maschinenbau-Verlag GmbH in Frankfurt/M.

Additional explanations will be found in the standards DIN 24260 (Centrifugal Pumps and Centrifugal Pumping Plants, Concepts, Symbols, Units; edition 1986) and ISO 2548 and 3555 (Centrifugal, mixed flow and axial pumps - Code for acceptance tests).

In the DIN 24260 standard, edition 1971, the concept of "Haltedruckhöhe" (retaining pressure head) is used; physically its significance is the same as that of the NPSH value. There may be a difference in the numerical values of z_s of the "retaining pressure head" and of the NPSH of one and the same pump, because different datum levels (reference levels) are laid down for their calculation (see Fig. 1). In practice, only the NPSH value is used today.

In line with the publications referred to above, the main points relating to the NPSH value (and to the "retaining pressure head") are discussed below.

During the progress of the flow across the impeller of a centrifugal pump, there occurs first of all a decrease in the static pressure (pressure) in comparison with the value upstream of the impeller, particularly in the region of the inlet to the vane passages. The magnitude of this pressure decrease will depend on the rotational speed, the operating point, the geometry of the impeller inlet, the velocity profile of the approach flow, the density and the viscosity of the pumped medium.

In order to avoid cavitation in the impeller of the centrifugal pump, or at least to reduce cavitation to an acceptable level, it is therefore necessary for the pressure upstream of the impeller to be situated at a given level above the vapour pressure of the pumped liquid.

A distinction is made between the NPSH value **required** by the centrifugal pump ($NPSH_{req}$, where the suffix req. = required), which corresponds to the "retaining pressure head" (H_H) of the pump, and the existing NPSH value of the installation (plant) ($NPSH_{av}$, where the suffix av. = available), which corresponds to the "retaining pressure head" (H_{HA}) of the installation (plant).

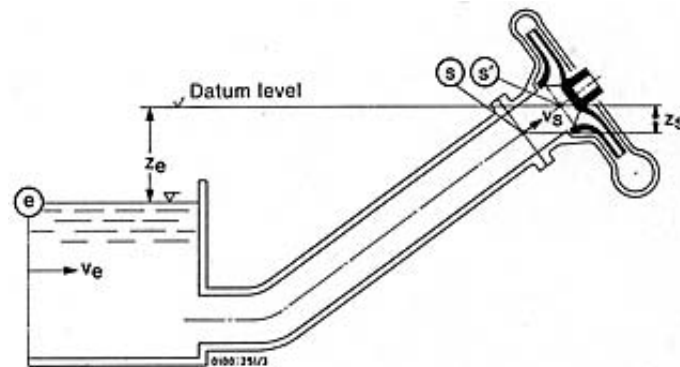


Fig. 1: Position of datum points (reference points) s' for the NPSH value and s for the "retaining pressure heads" (in this example the impeller is approached from below)

NPSH available. The NPSH value of the installation (pumping plant), symbol NPSH_{av}, is defined as follows:

$$\text{NPSH}_{av} = \frac{p_{\text{tot.s}} - p_D}{\rho \cdot g}$$

where

p_D vapour pressure of fluid pumped at point s (Fig. 1),
 ρ density of fluid pumped at point s,
 g gravitational constant
 $p_{\text{tot.s}}$ total pressure at pump suction branch
 (subscript s, see Fig. 1).

Point s relates to the centre of the suction branch in the above equation (Fig. 1). If there is no pump suction branch, e.g. on a inline pump with welded-in pipe connections (welded-in pump) or on a submersible pump with entry nozzle, a location s which corresponds clearly to the pump suction branch must be defined and accurately specified when stating the NPSH value.

The total pressure at point s can also be expressed as

$$p_{\text{tot.s}} = p_s + p_b + \frac{\rho \cdot v_s^2}{2} \pm z_s \cdot \rho \cdot g$$

with

$p_s + p_b$ absolute static pressure at point s, (p_b barometric pressure, p_s static over or underpressure in relation to p_b),
 v_s flow velocity at cross-section through point s,
 z_s geodetic head at point s, related to the datum level (reference level) (see Fig. 1).

The datum point (reference point) s' for the NPSH value is the centre point of the impeller, i.e. the intersection point of the axis of the pump shaft with the plane situated at right angles to the pump shaft and passing through the outer points of the vane inlet edge (see Fig. 2). In the equations above and below the negative sign of z_s applies when the impeller is approached from below as illustrated in Fig. 1; otherwise, i.e. if point s is situated above the reference level s', the positive sign is to be applied.

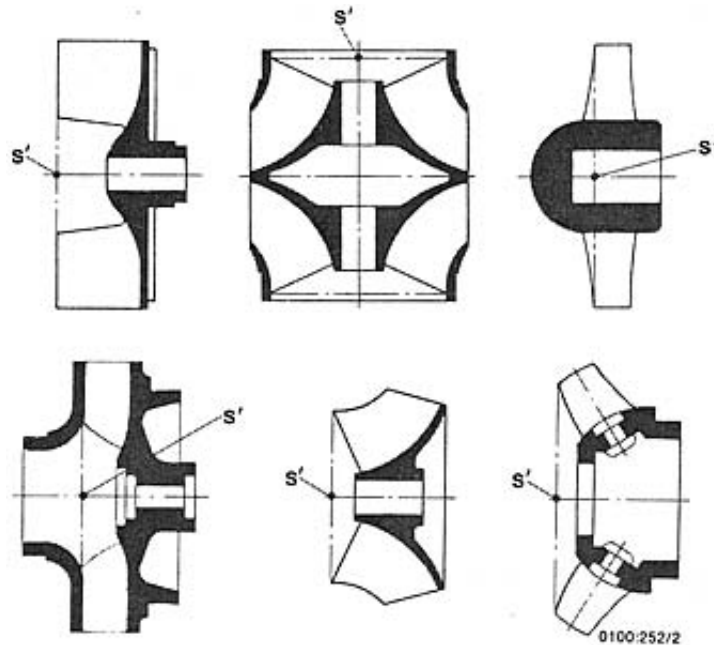


Fig. 2: Position of reference point s' in various types of pump impellers

Thus the NPSH value of the plant becomes:

$$NPSH_{av} = \frac{p_s + p_b - p_D}{\rho \cdot g} + \frac{v_s^2}{2g} \pm z_s.$$

If we insert the values at the inlet cross-section of the plant (head of plant):

$$NPSH_{av} = \frac{p_e + p_b - p_D}{\rho \cdot g} + \frac{v_e^2}{2g} \pm z_e - H_{v,e,s}$$

with

p_e pressure at inlet cross-section of plant,

v_e flow velocity at inlet cross-section of plant,

z_e difference in geodetic heads between the inlet cross-section of the plant and the reference level s'. If this reference level is situated above the liquid level, as shown in Fig. 1, the negative sign for z_e has to be applied; if it is situated below the liquid level, the positive sign applies. For a definition of the reference level please refer to Fig. 2,

$H_{v,e,s}$ head loss (pressure loss) in the suction part of the plant (Fig. 2 under head), from inlet cross-section A_e of the plant to inlet cross-section A_s of pump, including any entry losses and pressure drops across valves, appliances, fittings etc.

NPSH of pump. The NPSH value of the pump, subscript req means required, is defined in similar fashion to the NPSH value of the plant.

The NPSH value of the pump is:

$$NPSH_{req} = \left(\frac{p_{tot,s} - p_D}{\rho \cdot g} \right)_{min} = \left(\frac{p_s + p_b - p_D}{\rho \cdot g} + \frac{v_s^2}{2g} \pm z_s \right)_{min}.$$

The symbols in the brackets have the same meaning as those for the NPSH value of the installation. A significant difference, however, is that the magnitudes defined by the terms in brackets *must neither be equaled nor less than a given minimum value* (subscript min), specific to the pump and the application; otherwise one of the following "cavitation criteria for centrifugal pumps" would be disregarded.

1. Visible incipient cavitation at $NPSH_i$ (bubbles) at entry edge of blade up to a max. bubble length LB_1 (by definition) of 5 mm for example (only measurable by visual observation).
2. Drop head ΔH , caused by cavitation, of a max. amount by definition, of, say, $\Delta H = 0.03 \cdot H$, or, as is often applied in the case of high specific speed pumps, of $\Delta H = 0.00 \cdot H$, i.e. right where the total head starts to be affected by cavitation. The corresponding NPSH values are indexed $NPSH_3$ and $NPSH_0$.
3. Removal of material in the centrifugal pump due to cavitation, up to a max. mass per time (by definition).
4. As point 2. above, but related to the pump efficiency (e.g. $\Delta \eta = 0.03 \cdot \eta$).
5. Increase in noise level due to cavitation up to a maximum sound pressure level (by definition) (measuring technique).
6. Drop head caused by cavitation, up to the complete collapse of the head.

When giving the NPSH value of the pump, it is indispensable to also specify the related cavitation criteria; however, criteria 4. to 6. are hardly used, and measuring criteria 1. runs up large experimental costs. Due to this it is usually agreed that $NPSH_{req} = NPSH_3$.

All listed cavitation criteria and their derived NPSH values are dependent on the operating point (characteristic curve), as is illustrated in Fig.3. Here one sees the curves of the $NPSH_{req}$ of a specific impeller versus flow. Parameterized according to the type of cavitation, such as, for example, the length L_{B1} , of the resulting bubble trail in relation to the blade spacing t , (cascade). If one were to depict on this graph the $NPSH_{av}$ then one could determine the type of cavitation expected as a function of capacity.

The upper curve ($NPSH_i$) depict incipient cavitation. If $NPSH_{av}$ is larger than $NPSH_i$, then no cavitation will occur, and the impeller will run without formation of bubbles. The lower the $NPSH_{av}$ value gets, the longer the bubble trail will get. From a minimum value the bubble length will increase at partial load and runout flow if $NPSH_{av}$ remains constant. The capacity at the minimum cavitation bubble length, corresponds to the flow direction of the shockfree entry, and represents the condition of smallest increases in fluid velocity on the pressure side and suction side of the blade. It is thus labeled as a shock-free capacity, $Q_{shock-free}$. When $Q < Q_{shock-free}$, cavitation on the suction side increases, and when $Q > Q_{shock-free}$ cavitation on the pressure side increases.

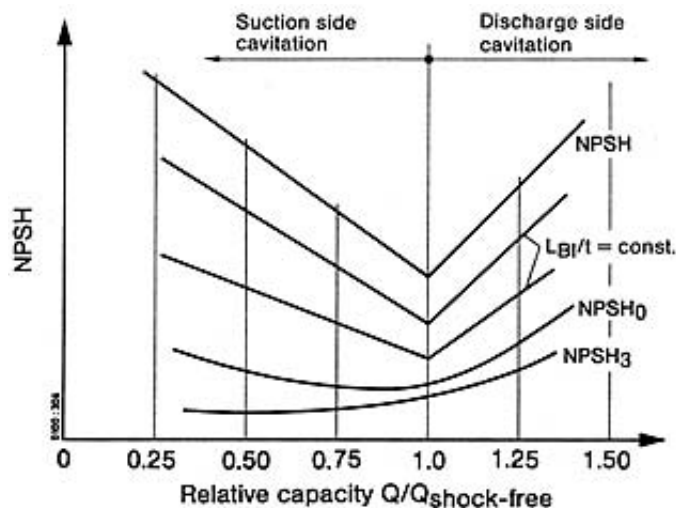


Fig. 3: $NPSH_{req}$ for various criterions as a function of the relative capacity $Q/Q_{shock-free}$

When specifying the NPSH required, it is essential therefore to specify also the relevant cavitation criterion, bearing in mind that criteria 4. to 6. are less frequently used.

The determination of the NPSH required is largely a matter of experience involving NPSH tests, particularly when converting the $NPSH_{req}$ of the pump from one rotational speed to another or on similar pumps from one pump size to another and from one medium to another (particularly if there is any dissolved or undissolved gas in the medium pumped, gas content of pumped medium).

The operating point of a centrifugal pump can only be taken as a point of continuous operation if

$$\text{NPSH}_{\text{av}} \geq \text{NPSH}_{\text{req}}$$

at this point.

In connection with the NPSH value the following magnitudes are sometimes introduced: suction number S,

$$\text{cavitation coefficient } \sigma_u = \frac{\text{NPSH}}{u_{1,a}^2/2g}$$

with

$u_{1,a}$ maximum circumferential velocity of the impeller entry edge.

Neural Conductor

Nulleiter

Fil neutre

see Center Conductor

NEWTONian Liquid

NEWTONsche Flüssigkeit

Liquide NEWTONien

N.1., according to DIN 1342 also called isotropic pure viscous fluid (viscosity) is a fluid which behaves according to the so-called NEWTON's law of viscous shear.

$$\tau = \eta \frac{\partial v_x}{\partial y}$$

with

τ shear stress on a fluid particle of a viscous fluid (τ arises when two particles are displaced in opposite direction to each other, and is the force per unit of area opposed to the displacement action),

η proportionality factor related to temperature and pressure, also called dynamic viscosity,

$\frac{\partial v_x}{\partial y}$ local change in absolute velocity v in the direction perpendicular to the flow line; unit: 1 metre per second and metre.

A fluid is a N.1. if its dynamic viscosity depends only on temperature and pressure, but not on the velocity v or the velocity gradient $\partial v_x / \partial y$.

Water and mineral oils from distillates are N.I.'s, in so far as they can be pumped by conventional centrifugal pumps. Non-N.I.'s are e.g. oils at temperatures close to their pour point temperature. The pumping of non-N.I.'s by centrifugal pumps is generally only possible or economic after heating of the fluid.

Water and mineral oils obtained from distillates are N.I.'s, as long as they are transferred with normal centrifugal pumps. Liquids that are non-NEWTONian are sewage sludge, finely powdered mineral sludge (hydrotransport of sand, coal or ore), but also oils, if their temperature is nearing the solidification point. To pump a non-N.I. through a centrifugal pump is sometimes possible only if the medium has been heated. The flow characteristics of structurally viscose, dilatant, thixotropic or rheopectic fluids affects not only the pressure losses in pipings, but also the layout of a centrifugal pump (pulp pumping).

Noise Abatement Measures

Schallschutzmaßnahmen
Mesures insonorisantes

see Noise in Pumps and Pumping Installations

Noise in Pumps and Pumping Installations

Geräusch bei Pumpen und Pumpenanlagen
Bruit de pompes et d'installations de pompage

In centrifugal pumps, energy is transmitted via the impeller vanes (blade) to the fluid pumped. Because the number of blades is finite, periodic pressure fluctuations arise with more or less pronounced amplitudes. Because the flow in the centrifugal pump runs against a rising static pressure, the boundary layers are in constant danger of flow separation. Flow around the blades and flow separation lead to a unsteady state of flow in a centrifugal pump. The pressure fluctuations arising from this unsteady flow excite vibrations in to the pump casing and attached piping, which in turn transmit them to the surrounding air in motion; thus generating an audible noise.

In single stage volute casing pumps, which consist of only a few simple components, the unsteady flow separations are the main source of noise, if we disregard the noise sources arising from the driver (electric motor, steam turbine, internal-combustion engine) and assume the centrifugal pump to be in perfect mechanical working order.

In the case of multistage centrifugal pumps with balancing devices (axial thrust) the conditions surrounding noise sources are more complex. In such pumps, apart from the noises already mentioned, there are also considerable turbulence noises (fluid dynamics) as a result of the usually very high stage heads. In addition, considerable noises can arise when high pressures are relieved (e.g. in the balancing devices).

The trend towards higher rotational speeds has an unfavourable effect on the noise developed by centrifugal pumps, because the overall dimensions of the machine are reduced and the energy conversion takes place in a smaller volume (increased power density). In addition, the better use made of materials (thinner walls) also has an unfavourable effect on noise development.

Because of all these many interdependent causes, it is not possible to formulate generally valid rules about noise sources and noise development in centrifugal pumps. It seems clear however that the shaft power represents the main magnitude which influences noise.

According to VDI 3743, Part 1, the following formulae can be used for the approximate calculation of the noise power level:

for volute casing or annular casing pumps

$$L_{WA} = 71 + 13.5 \lg \frac{P}{P_0}$$

valid for $4 \text{ kW} \leq P \leq 2000 \text{ kW}$,

for pumps with diffuser vanes and return guide vanes

$$L_{WA} = 83.5 + 8.5 \lg \frac{P}{P_0}$$

valid for $4 \text{ kW} \leq P \leq 20000 \text{ kW}$,

for tubular casing pumps with axial impeller

$$L_{WA} = 21.5 + 10 \lg \frac{P}{P_0} + 57 \frac{Q}{Q_{opt}}$$

valid for $10 \text{ kW} \leq P \leq 1300 \text{ kW}$ and $0.77 \leq Q/Q_{opt} \leq 1.25$

where

L_{WA} A-rated noise power level in dB,
 P shaft power of the centrifugal pump in kW,
 P_0 = 1 kW,
 Q/Q_{opt} ratio of capacity at measuring point
to optimum capacity.

As a general rule, these formulae err on the generous side (values slightly too high).

Statistical research in another direction has demonstrated that a portion varying between 10^{-9} and 10^{-6} of the shaft power of the centrifugal pump is converted into noise, depending on the pump type. This rough evaluation amounts to a scatter of 30 dB.

It is very difficult to predict what the noise value in pumping installations will actually be, because of the lack of sufficient experimental data. A special difficulty resides in the prediction of influences by the foundations (pump foundation), buildings (enclosed workshops or hangars) as well as neighbouring machines and equipment.

Measurement of noise in pumps and pumping plants. DIN standard 45635 is used as guideline for noise measurement. Another internationally used regulation, which agrees with DIN 45 635 in all main points is ISO recommendation No. 3744 (measuring technique).

Both of these contain definitions of the most important concepts in acoustics, and also useful information on measuring instruments and measuring conditions, and on the method and evaluation of sound measurements, and also on the uncertainties in measurement. They also provide the basis for determining the noises radiated by centrifugal pumps to the surrounding air using a unified method, thus ensuring comparability of results. The values determined in this way are, amongst other things, suitable for the comparison of similar and dissimilar machines, for the evaluation of noise emission (immission protection act, environmental protection) and for the planning of soundproofing measures.

Sound-proofing measures for pumps and pumping plants. The most important active sound-proofing measure is the correct pump selection. This applies both to the pump size and pump type, which should be matched to the given application. Centrifugal pumps develop differing noise intensities, depending on the position of the operating point (operating behaviour) on their throttling curve (characteristic curve). Pumps should wherever possible operate at the operating point of optimum pump efficiency, both for energy and acoustic reasons; at their operating speed (rotational speed) the noise emission is usually lowest in the region of the optimal point.

If very stringent requirements are set in connection with the noise emitted by centrifugal pumps, certain insulation measures (passive soundproofing) cannot be avoided (incorporation of expansion joints in the pipings; installation of the centrifugal pump on rubber-metal shockabsorbing elements or spring elements; encasing of the pump or pump and drive in special sound-absorbing and sound attenuating wall elements).

Noise Measurement

Geräuschmessung
Mesure de bruit

see Measuring Technique

Nominal Diameter

Nennweite

Diamètre nominal

N.d. (diamètre nominal, international symbol DN in accordance with ISO 6708, e.g. DN 500), is a standardized characteristic magnitude, without any mention of a unit, applying to pipings, valves and fittings and benches of pump casings. The n.d. closely approximates the internal (or clear) diameter in mm of the two components connected to each other. The Table below gives a synopsis of the n.d.'s standardized in DIN; ISO 6708 does not state numerical value.

Table: Nominal diameters according to DIN 2402

	10	100	1000
	12 *)	125	1200 1400
	15 **) 16 *)	150	1600 1800
	20	200	2000 2200 2400
	25	250	2600 2800
3	32	300 350	3000 3200 3400 3600 3800
4	40	400 450	
5	50	500	
6	65	600 700	
8	80	800 900	

*) these nominal diameters are used in cases where a finer graduation becomes necessary e.g. for screwed pipe connections

**) this nominal diameter is used in cases where a coarser graduation is adequate, e.g. for flanges

Nominal Pressure

Nenndruck

Pression nominale

N.p. (pression nominale, internationally abbreviated PN, according to ISO 7268, e.g. PN 16), is the designation for a selected pressure-temperature relationship to which resort is made for the standardization of structural components. Parts (such as flanges) with the same n.p. have the same nominal dimension at equal nominal diameter. The n.p. is without units, and DIN 2401 (Table 1) gives the allowable pressure for parts and building materials in units of bars at 20 °C.

Table 2 shows the approximate values of the n.p. according to ISO 7268.

Table 1: Nominal pressure ratings in accordance with DIN 2401

	1	10	100	1000
		12.5	125	1250
	1.6	16	160	1600
	2	20	200	2000
	2.5	25	250	2500
	3.2	32	315	
	4	40	400	4000
0.5	5	50	500	
	6	63	630	6300
			700	
	8	80	800	

Table 2: Nominal pressure ratings in accordance with ISO 7268

2.5	6				25	40					
		10	16	20			50	100	150	250	420

Nominal Speed of Rotation

Nennzahl

Vitesse de rotation nominale

N.s.o.r., abbreviated nN, is the rounded value of the speed in the speed range (nominal value).

According to DIN 40200 the n.s.o.r. is not, as it was earlier, the speed agreed upon in the contract, but rather a suitable approximate quantity value used to designate or identify a speed such as e.g. 2900, 1450 and 980 min⁻¹ at a frequency of 50 Hz, and 3500, 1750 and 1180 min⁻¹ at a frequency of 60 Hz.

Nominal Value

Nennwert

Valeur nominale

According to DIN 40200 the n.v. is a suitable approximate quantity value used to designate or identify a component, device or equipment. The n.v. of a size helps in the classification of a useful range such as:

nominal diameter (e.g. DN 100),

nominal pressure (e.g. PN 20),

nominal speed of rotation (e.g. nN = 2900 min⁻¹),

The actual values need not be exactly equal to the n.v., but need merely lie in the vicinity.

In this case the nominal speed of rotation is not the actual speed, but rather a classifying speed (e.g. 1450 or 2900 min⁻¹).

Non-Clogging Impeller

Kanalrad
Roue à canaux

see [Impeller](#)

Nozzle

Düse
Tyère

see [Entry Nozzle](#), [Standard Nozzle](#)

Number of Poles

Polzahl
Nombre de pôles

The number of poles or of pole pairs p of a threephase motor ([asynchronous motor](#)) determines the so-called [synchronous speed](#) in

$$n_{\text{synchron}} = \frac{60 \cdot f}{p}$$

with

f [frequency](#) of three-phase supply in Hz,

p number of pole pairs.

The Table below lists the most usual n.o.p. and of pole pairs on centrifugal pump drivers ([asynchronous motor](#) in the form of squirrel-cage motors and of slip-ring motors, and synchronous motors), together with their related synchronous speeds, depending on the frequency of the mains supply.

Table: Numbers of poles and synchronous speeds of three-phase motors

Number of poles	2	4	6	8	10	12
Number of pole pairs p	1	2	3	4	5	6
Mains supply frequency in Hz	synchronous speed n_{synchron} in min^{-1}					
50	3000	1500	1000	750	600	500
60	3600	1800	1200	900	720	600

The full load speeds of [asynchronous motor](#) range from about 1 % below the synchronous speed in the case of large (high power) motors, and down to 6% below in the case of small (low power) motors, whereas under no-load conditions, the synchronous speed is almost attained.

Variable pole motors are [asynchronous motor](#) that allow for several speeds (Fig. 3 under control). Several pole configurations of the stator winding can be selected by combining several independent winding coils or switching between parts of a single stator winding. Both methods were designed to handle up to four different speeds. A pole switch is necessary in both cases.

Squirrel-cage motors are generally used ([asynchronous motor](#)); with pole modulated slip-ring motors the rotor winding must also be switched to adjust for pole numbers.

Number of Working Hours

Betriebsstundenzahl

Nombre d'heures de fonctionnement

see Economics

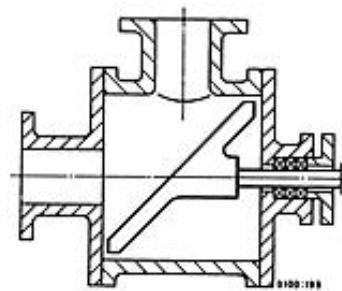
O

Oblique Plate Pump

Schrägscheibenpumpe

Pompe à disque oblique

O.p.p.'s (also known as slanting rotor pumps) are pumps with a rotor consisting of a disc (see illustration) arranged at an inclined angle on the drive shaft and rotating in a cylindrical casing with an axial suction branch (pump suction branch) and a radial or tangential discharge branch (pump discharge branch). This generates a centrifugal force field inside the liquid filled casing which sustains a pulsating flow. O.p.p.'s are suitable for pumping thick stocks (pulpes), mashes and slurries (pulp pumping); if the rotor and casing are suitably shaped, the o.p.p. is also capable of comminuting (pulverizing) the medium pumped.



Oblique plate pump

Oil Pump

Ölpumpe

Pompe à huile

see Lubricating Oil Pump

Operating Behaviour of Centrifugal Pumps

Betriebsverhalten von Kreiselpumpen

Comportement en marche de pompes centrifuges

The o.b.o.c.p. is a collective concept for all the pump characteristics at the operating point concerned, e.g. hydraulic, mechanical, acoustic characteristics.

The position of the operating point in relation to the design duty point has a marked effect on the o.b.o.c.p. If the capacity (rate of flow) at the actual-operating point happens to be much lower than that at the design duty point, the pump will run in the *part load region of operation*, and this may lead to flow separation (boundary layer) as a result of the unfavourable flow incidence to the impeller vanes and/or guide vanes (blade), and consequently lead to mechanical vibrations, noises and cavitation. Low specific speed centrifugal pumps (specific speed), such as radial pumps have a relatively flat throttling curve (characteristic curve), depending on the diffuser device and on the discharge angle of the impeller vanes. In some instances the throttling curve intersects the system characteristic curve at a flat angle, and this can result in pump pulsations due to operation in an unstable region (unstable throttling curve).

High specific speed centrifugal pumps (e.g. axial pumps, mixed flow pumps) are characterized by a more or less pronounced saddle in their throttling curve in the part load region (Fig. 3) which should be passed through very rapidly on startup and shutdown to avoid pronounced vibration, noise, and possible cavitation. Continuous operation (see QH diagram, Fig. 3) should take place only below the so-called operating limit ("rumour limit").

Two effects are mainly responsible for the unfavourable behaviour of high specific speed centrifugal pumps at part load (continuous) operation: firstly there is a flow separation in the impeller vane cascade at the suction end (Fig. 1), and secondly instability is induced by the part load eddies at the outer end of the impeller inlet and at the inner end of the impeller outlet (Fig. 2).

In Fig. 1, v_{1m} is the meridian component of absolute velocity (velocity triangle) at the design duty point. This component decreases to the value v_{1mT} at part load operation representing as it does a measure of the magnitude of the capacity; correspondingly, the relative velocity w_1 will change to w_{1T} . This results in a very unfavourable flow incidence to the vane cascade, due to the orientation of w_{1T} , and the relative flow can no longer follow the blade contour along the suction side and a flow separation takes place (boundary layer). Similar conditions arise in the overload region (see Fig. 1), but in this case the flow separates on the discharge side of the blades. Any flow separation represents an unsteady flow phenomenon (unsteady flow); such phenomena disturb the flow deflection at the cascade profile (flow profile) quite appreciably (deflection required to generate the head), and lead to pulsations in the medium pumped and vibrations in the pump components guiding the flow.

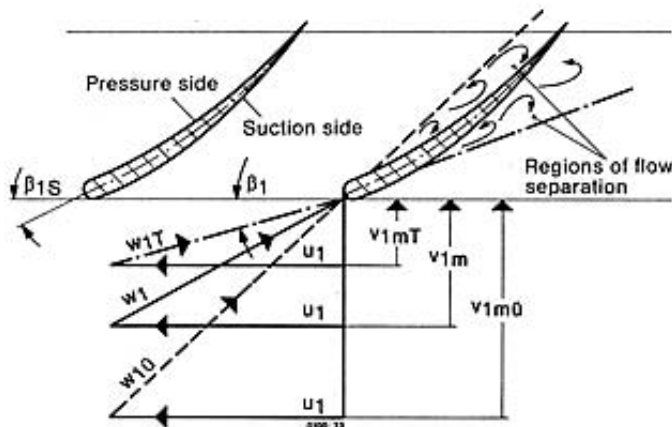


Fig. 1: Flow separation in an axial impeller vane cascade at part load (index T, chain-dotted line) and overload (index ü, dotted line) operation

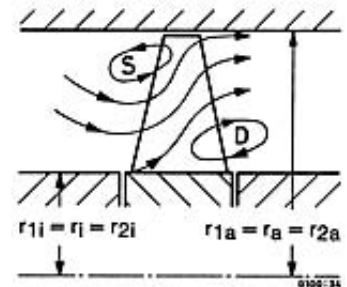


Fig. 2: Part load eddies S and D in an axial impeller

The positions of the part load eddies S and D are illustrated in Fig. 2. Part load eddies arise mainly as a result of flow separation; the fluid pumped can no longer follow the appreciable rise in pressure in the impeller at part load operation (caused by the high heads at part load) and consequently separates, under the effect of a lower pressure rise along the wall. The part load eddy S on the suction side can be traced or detected for quite a distance (equal to several suction pipe diameters) in the direction opposite to that of incoming flow, even in the case of low specific speed pumps. The stretching out of this part load eddy in the axial direction can however be limited by incorporating ribs, sharp changes in cross-section or abrupt pipe elbows.

The so-called "rumour-limit" of high specific speed pumps is due to the fact that under certain operating conditions only the part load eddy S or the part load eddy D is present; there is a constant jumping to and for between S and D, resulting in great disturbance to the flow, and the excitation of excessive vibrations in the pump components guiding the flow. When the capacity is further reduced, both part load eddies S and D are present simultaneously, the flow lines are pushed more and more in the radial direction until finally the flow in the impeller is mainly radial (condition at pump shut-off point).

In the overload operation region, i.e. when the capacity is much higher than at design duty point, there is also a general limit which should not be exceeded if a trouble-free operation of low or high specific speed pumps is desired. This limit is set by the suction behaviour of the pump and by flow separation on the discharge side (Fig. 1).

The unfavourable o.b.o.c.p. at part loads can be improved by increasing the capacity (e.g. by providing a by-pass) in order to obtain trouble-free operation. As illustrated in Fig. 3, the performance chart of a propeller pump with adjustable pitch blades, the impeller blade pitch adjustment represents a suitable means of avoiding the unpleasant consequences of part load operation, within certain limits of the capacity Q. The steps described under control of centrifugal pumps to alter the operating point are the main means of influencing the o.b.o.c.p.

When centrifugal pumps with a high head are operated at very low capacities (i.e. at the extreme lower limit of part load operation), or even against a closed discharge valve (valves and fittings), it can happen in certain cases that the pumped liquid is heated to such an extent that it evaporates, particularly during expansion in the balancing device. For a description of this phenomenon, and for suitable precautions to be adopted in this connection (e.g.

nonreturn valves), see under valves and fittings.

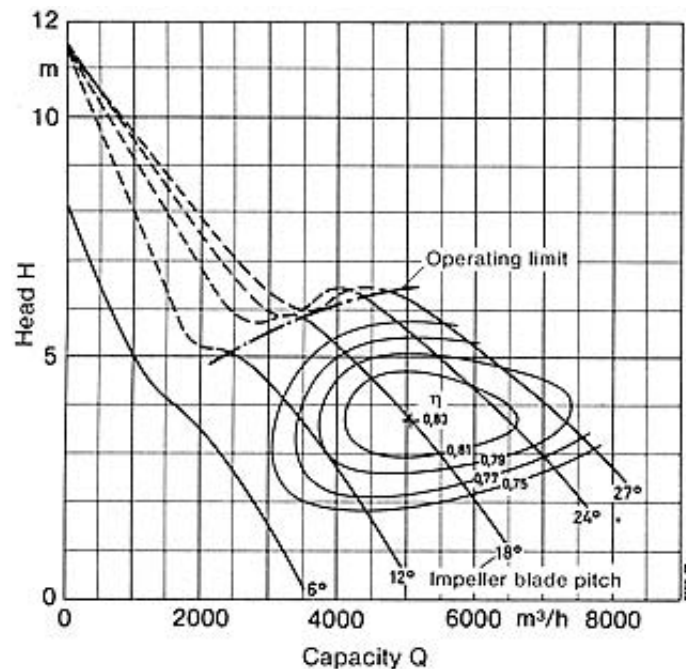


Fig. 3: Performance chart of an adjustable pitch blade propeller pump showing the operating limit

Operating Characteristics

Betriebseigenschaften

Caractéristiques de fonctionnement

see [Operating Behaviour of Centrifugal Pumps](#)

Operating Conditions of Centrifugal Pumps

Betriebsbedingungen bei Kreiselpumpen

Conditions de fonctionnement de la pompe centrifuge

O.c.o.c.p. are the requirements laid down by the purchaser who places the contract in respect of certain operating characteristics of the pump. The o.c.o.c.p. have a determining influence on the selection of the pump in respect of type, size, drive etc.

The operating conditions primarily encompass the supply contract data on the medium pumped (density, temperature, viscosity etc.), on the magnitude of the capacity (rate of flow) and of the head, on the suction behavior, and occasionally also on the rotational speed of the centrifugal pump. Further data include the size and power supply details of the driver, the running conditions, switching frequency or controllability of the set (control), and plant or environmental factors effecting the pump, e.g. max. permissible noises (environmental protection), piping forces (piping), shaft vibrations (vibration), danger of explosion (explosion protection).

Quite apart from the operating points proper, many of these operating conditions apply generally to one or more application fields for pumps. Thus the o.c.o.c.p. are often laid down in requirements or directives which apply to entire industries (e.g. refineries, power stations), or within the framework of environmental protection (immission protection act).

Operating Costs

Betriebskosten

Frais d'exploitation

see [Economics](#)

Operating Point

Betriebspunkt

Point de fonctionnement

The o.p. of a centrifugal pump is the intersection of the throttling curve $H(Q)$ (characteristic curve, head, capacity) with the plant characteristic curve $H_A(Q)$. $H(Q)$ is the head value *generated by the pump*, $H_A(Q)$ is the head value required by the installation or plant (centrifugal pump, pumping plant).

Fig. 1 illustrates the o.p. as intersection B of a stable throttling curve $H(Q)$ at constant rotational speed n with the plant characteristic curve $H_A(Q)$; $H_A(Q)$ usually corresponds to a pipeline characteristic with an aspect governed by the pressure losses which increase as the square of Q .

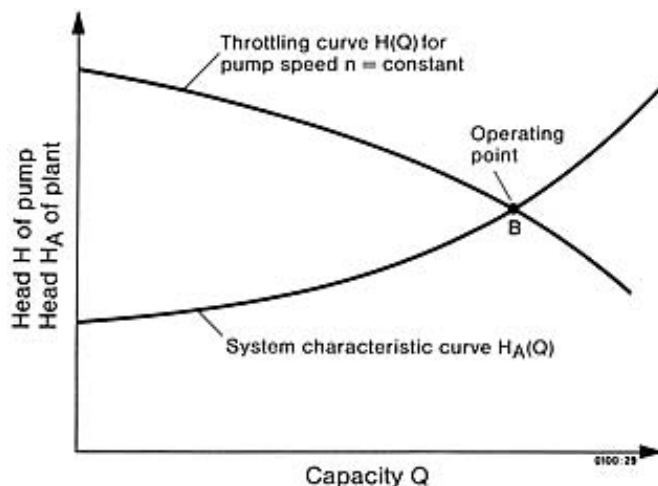


Fig. 1: Definition of operating point of a centrifugal pump

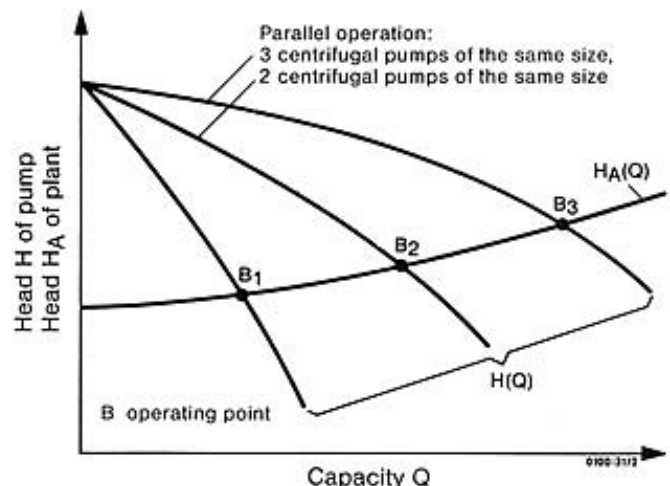


Fig. 2: Displacement of operating point from B_1 to B_3 along the plant characteristic $H_A(Q)$ due to increase in pump speed from n_1 to n_3

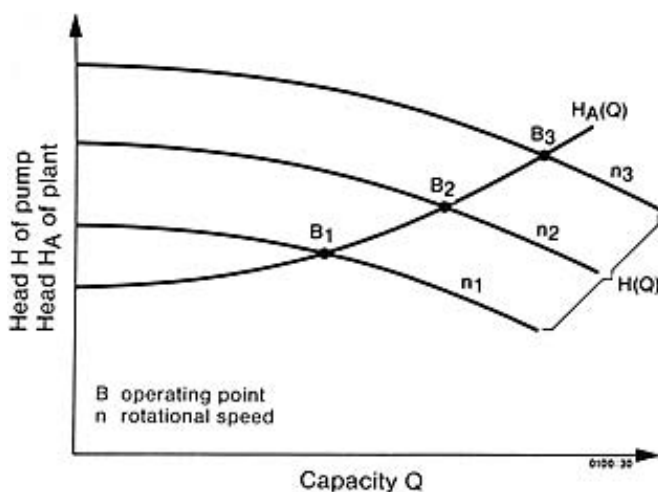


Fig. 3: Displacement of operating point from B_1 to B_3 along the plant characteristic $H_A(Q)$ when a second and third pump of the same size operating in parallel are switched on

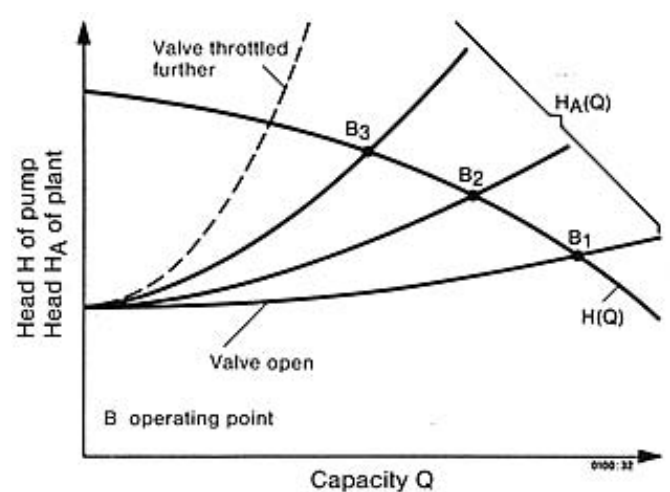


Fig. 4: Displacement of operating point from B_1 to B_3 along the throttling curve $H(Q)$ due to increasing throttling

The position of the o.p. will change if the throttling curve $H(Q)$ and/or the plant characteristic curve $H_A(Q)$ take up a different position or aspect.

Case a: $H(Q)$ changes but $H_A(Q)$ remains constant. This happens e.g. in the case of speed-controlled pumps (control). Fig. 2 illustrates how the o.p. moves from B_1 to B_3 along the plant characteristic curve as the rotational speed increases, towards higher capacities and heads.

In Fig. 3, the displacement of point B from B_1 to B_3 along the plant characteristic curve is illustrated for the case of a centrifugal pump operating in parallel with other pumps of the same size, when a second and a third pump are switched on (parallel operation).

Case b: $H_A(Q)$ changes, $H(Q)$ remains constant. This happens e.g. in the case of control by throttling (control). Fig. 4 illustrates how point B moves from B_1 to B_3 as the throttling increases, i.e. as the pipe resistance increases, towards lower capacities and higher heads.

In most cases design duty points and so-called inquiry points do not exactly correspond to the o.p.

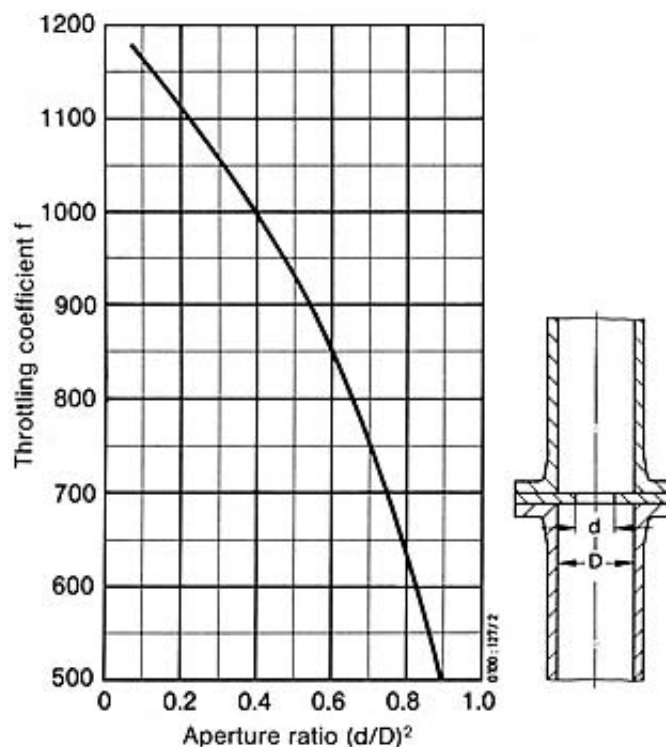
Orifice Plate

Blende, Lochblende
Diaphragme sténopé

In order to achieve a given ratio of the volume flow (capacity) in pipings branches, o.p.'s (illustration) are built into the individual piping branches for throttling purposes. The required hole diameter d of the o.p. will depend on the desired volume flow Q , on the static pressure differential upstream and downstream of the o.p. (standard orifice), and on a nondimensionalized throttling coefficient f determined by the aperture ratio $(d/D)^2$ (see illustration).

We have:

$$d = f \cdot \sqrt{\frac{Q}{\sqrt{g \cdot h}}}$$



Orifice plate and throttling coefficients of same

with

- d inside diameter of o.p. in m,
- g gravitational constant = 9.81 m/s^2 ,
- h effective (differential) pressure head in m (standard orifice),
- Q volume flow in m^3/s .

Because the aperture ratio is an unknown quantity when first determining d, the value of the throttling coefficient f is guessed in the first instance and then corrected by repeating the calculation.

If it is desired to subdivide the flow very accurately, the bore of the o.p. should be selected slightly larger, and the fine adjustment affectedly means of a control valve.

Outlet Branch

Druckstutzen

Tubulure de refoulement

see Pumping Discharge Branch

Outlet Cross-section

Austrittsquerschnitt

Section de sortie

One must differentiate between the o.c.s. of the pump and the o.c.s. of the plant (pumping plant).

The o.c.s. of the pump is the same as the crosssection of the pump discharge branch (cross-section A_d , see Fig. 2 under head). If there is no outlet branch, the o.c.s. of the pump must be defined, e.g. as being the cross-section at the outlet end of the discharge elbow.

The o.c.s. of the plant (cross-section A_a , see Fig. 2 under head) is a cross-section in the discharge piping to be mutually agreed, or a cross-section in another space on the discharge side with known geometric and flow data.

Outlet Width

Austrittsbreite

Largeur de sortie

see Impeller

Overall Efficiency

Gesamtwirkungsgrad

Rendement total

see Pump Efficiency

Overall Measuring Tolerance

Gesamtmeßspiel

Jeu de mesure total

see Acceptance Test Codes for Centrifugal Pumps

Overall Tolerance

Gesamttoleranz

Tolérance totale

O.t. is the tolerance formed by the manufacturing tolerance and the uncertainty of measurement when proving a guarantee.

The o.t. is not specified in the form of a value in the DIN 1944 acceptance test code (October 1968 edition), because the uncertainties of measurement in relation to the measuring points and the manufacturing tolerance in relation to the performance data must be taken into account separately as criteria for the fulfilment of the guarantee when evaluating the acceptance test.

In the ISO acceptance test code, however, the o.t. (which is designated "acceptance tolerance" in this code) is the square root of the sum of the squares of uncertainty of measurement and of manufacturing tolerance. When assessing the acceptance test the o.t. serves to check compliance with the guarantee. For this purpose, the throttling curve $H(Q)$ (characteristic curve) through the measuring points, and the guarantee point (Q_L/H_L specified in the supply contract) are plotted; then the segments ΔH and ΔQ are plotted in the graph (see illustration). The guarantee for the performance data is deemed to be fulfilled if

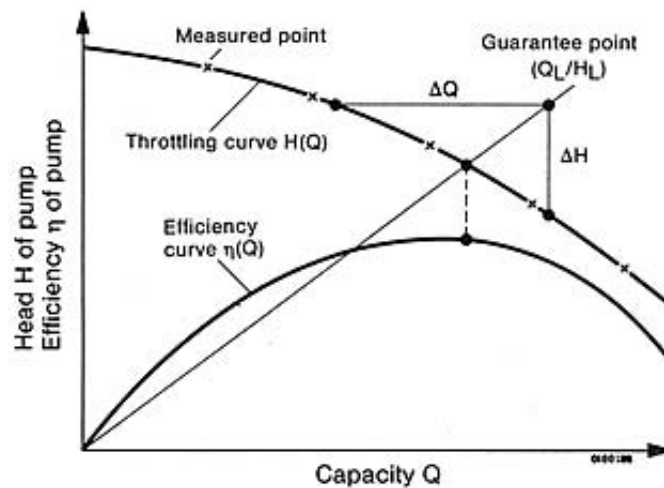
$$\left(\frac{H_L \cdot x_H}{\Delta H}\right)^2 + \left(\frac{Q_L \cdot x_Q}{\Delta Q}\right)^2 \geq 1$$

where x_H and x_Q are the o.t.'s (acceptance tolerances) for the head H_L and the capacity Q_L specified in the supply contract (see numerical example below). The guarantee for the efficiency is deemed to be fulfilled if it does not deviate from the guaranteed value by more than the uncertainty of measurement, at the point given by the intersection of a straight line running through $Q = 0$, $H = 0$ and Q_L , H_L with the throttling curve $H(Q)$ obtained from the measured points (see illustration).

Example: Acceptance test in accordance with Class C, ISO 2548 (guarantee), according to which (acceptance tolerances) $x_Q = 0.07$ and $x_H = 0.04$. $Q_L = 80 \text{ m}^3/\text{h}$, $H_L = 50 \text{ m}$; $\Delta Q = 2.8 \text{ m}^3/\text{h}$, $\Delta H = 2 \text{ m}$;

$$\begin{aligned} &\left(\frac{H_L \cdot x_H}{\Delta H}\right)^2 + \left(\frac{Q_L \cdot x_Q}{\Delta Q}\right)^2 = \\ &= \left(\frac{50 \cdot 0.04}{2}\right)^2 + \left(\frac{80 \cdot 0.07}{2.8}\right)^2 = \\ &= 1 + 4 > 1. \end{aligned}$$

The guarantee for the performance data is deemed to be fulfilled.



Guarantee of performance data H , Q , η as specified in the ISO acceptance test code

Overload Operation

Überlastbetrieb

Marche en surcharge

see [Operating Behaviour](#) of [Centrifugal Pumps](#)

Overvoltage Suppressor

Überspannungsableiter

Éclateur déchargeur

O.s., an electrical protective surge suppressor against atmospheric and internal overvoltages. Electrical machines are not only endangered by atmospheric overvoltages but also by internal overvoltages caused by switching or shorting to earth. In medium voltage distribution installations, voltage peaks amounting to 5 to 6 times the value of the operating voltage can arise. In order to protect the machine windings against such overvoltages, special surge suppressors for machines have been developed, with a blowing alternating voltage preset at 1.8 times to twice the rated voltage, i.e. below the value of the test alternating voltage.

In the case of overhead power line cathode fall arrestors, the blowing alternating voltage is considerably higher, and the protection they give to electrical machines is therefore lesser.

P

Packing

Packung
Garniture à tresse

see [Shaft Seals](#)

Parallel Operation

Parallelbetrieb
Marche en parallèle

If two centrifugal pumps I and II are connected in parallel, the capacity Q_{I+II} is the sum of the capacities of the individual pumps

$$Q_{I+II} = Q_I + Q_{II}$$

at the same head

$$H_{I+II} = H_I = H_{II}.$$

This relationship is illustrated in Fig. 1. Each of the pumps must be equipped with its own nonreturn valve (valves and fittings) for safety reasons. Centrifugal pumps in p.o. (operating point) present no running problems if their throttling curves are stable (stable throttling curve), and both have the same, or almost the same shut-off head (head). This does not prevent them from exhibiting differing characteristic curves $H(Q)$. Fig. 1 shows that when Q_{I+II} decreases to Q'_{I+II} , the individual capacities Q_I and Q_{II} also decrease to Q'_I and Q'_{II} . If the shut-off heads H_0 of pumps I and II are different, however, as illustrated in Fig. 2, pump I is rapidly pushed back towards the shut-off point, whilst II still continues pumping.

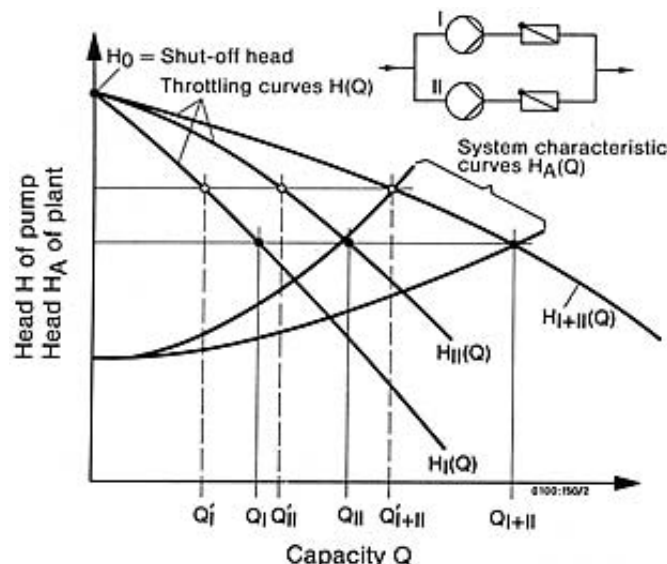


Fig. 1: Parallel operation of two centrifugal pumps I and II with stable throttling curves

Fig. 3 illustrates two unstable centrifugal pump characteristic curves $H_I(Q)$ and $H_{II}(Q)$ with identical peak heads and shut-off heads H_0 . Pumps I and II can be operated in parallel in the capacity range 4 to 5, and any further similar pump can be switched on without causing any problems. At point 4, only one other similar pump can be switched on, because of the shut-off head H_0 ; between points 4, 3, 2, 1 down to almost 0, this is no longer possible. In this operating range the pump, having a shut-off head H_0 smaller than H_{I+II} , would not be able to open

the nonreturn valve (valves and fittings) on account of the pressure exerted on it by the other pumps.

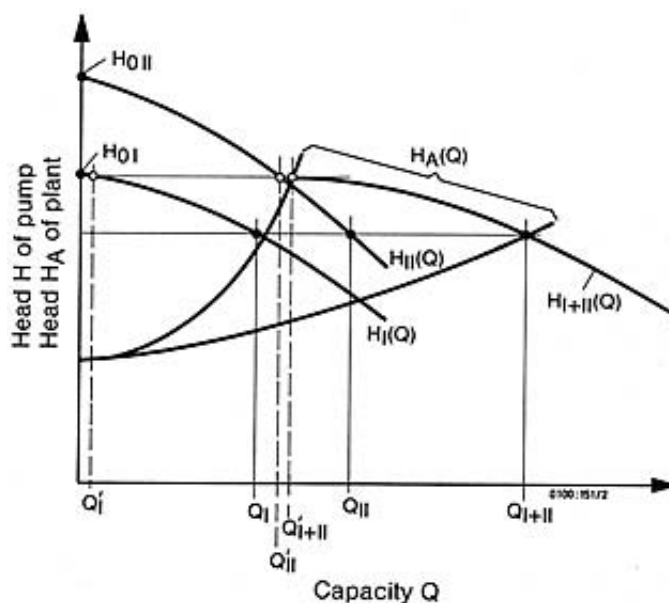


Fig. 2: Parallel operation of two centrifugal pumps I and II with differing shut-off heads

Fig. 4 shows a case of p.p. of two pumps, I and II, with unstable characteristic curves. The head at the crest, H_{Sch} (head) of pump I is greater than that of pump II.

As soon as the head at the operating point, H_{I+II} , is greater than the smallest value of the head at the crest, H_{SchII} , p.o. will lead to a very complex unsteady flow pattern.

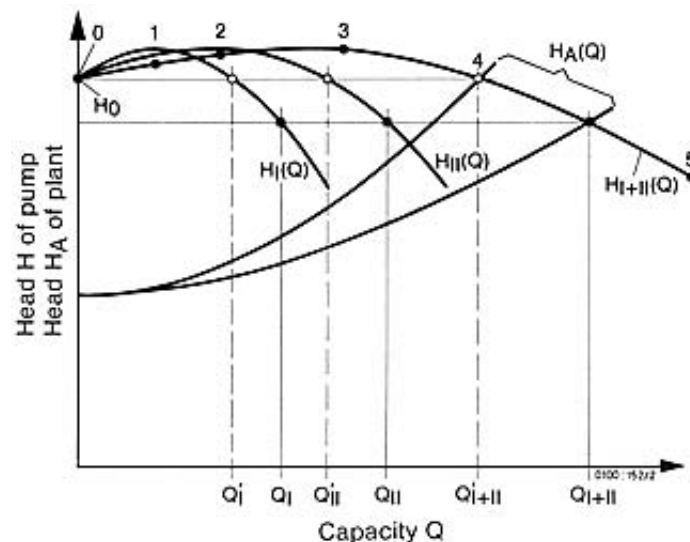


Fig. 3: Parallel operation of two centrifugal pumps I and II with unstable characteristics and the same peak heads

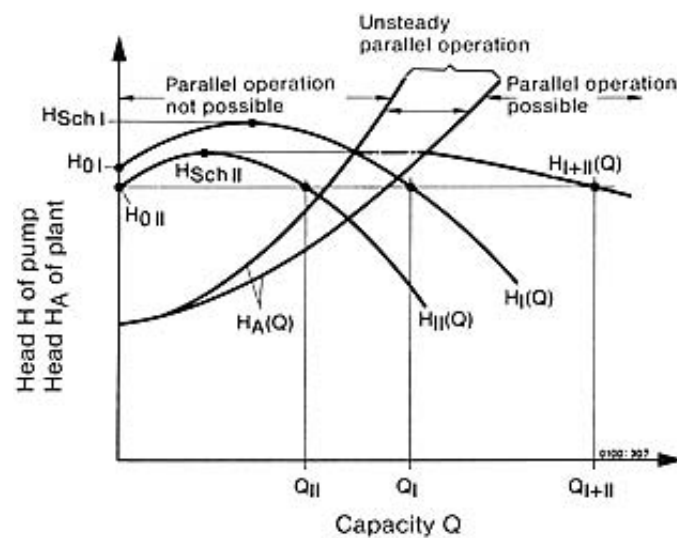


Fig. 4: Parallel operation of two centrifugal pumps I and II with unstable characteristics and different peak heads

When starting parallel centrifugal pumps with unstable characteristic curves and whose shut-off head H_0 , are not equal (Fig. 4), the pump with the smaller shut-off head H_{0II} , will not be able to open its check valve against the pressure of the other pump ($H_{0I} > H_{0II}$), and must therefore be started first.

This type of restriction can lead to interruptions in the operation of the pumps and can be eliminated by meticulous planning before construction of the system.

If a stepwise control, that is a system where one of the two parallel pumps must be switched on and off, is not used, it is less costly and simpler to design the system as two impellers in a back-to-back impeller pump rather than as two separate centrifugal pumps.

In the case of p.o. of a plunger pump (positive displacement pump) with a centrifugal pump, the capacities are again added to one another, viz. the practically constant capacity (in relation to head) of the plunger pump Q_K is added to the capacity of the centrifugal pump Q_I as illustrated in Fig. 5.

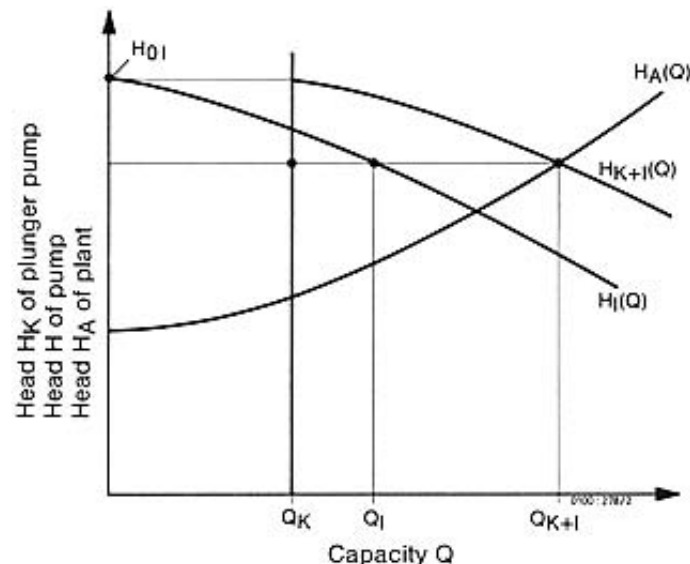


Fig. 5: Parallel operation of a plunger pump and of a centrifugal pump

Part Load Operation

Teillastbetrieb

Opération à charge partielle

see [Operating Behaviour of Centrifugal Pumps](#)

Pattern Maker's Section

Schreinerschnitt

Coupe pour le modelage

In the graphical representation of a double curvature blade in plan projection, the p.m.s. is a contour line, i.e. a line of constant geometric elevation above an arbitrarily selected datum level, perpendicular to the pump shaft.

The name p.m.s. derives from the fact that the pattern maker cuts out a number of little wooden boards of 5 to 10 mm thickness (or plastic boards, or boards of another suitable material) in the shape of the contour lines shown on the drawings, and glues all the boards together in accordance with the contour line drawing (p.m.s. representation) to obtain the pattern of the blade surface after filling out or abrading the step-like shoulders (projections).

The plan projection representation of an impeller in p.m.s. form is very useful for checking the smooth and continuous pattern of a blade surface. The p.m.s. still plays an important part even in the more recently developed blade surface manufacturing methods, e.g. by numerically controlled milling (N.C. milling).

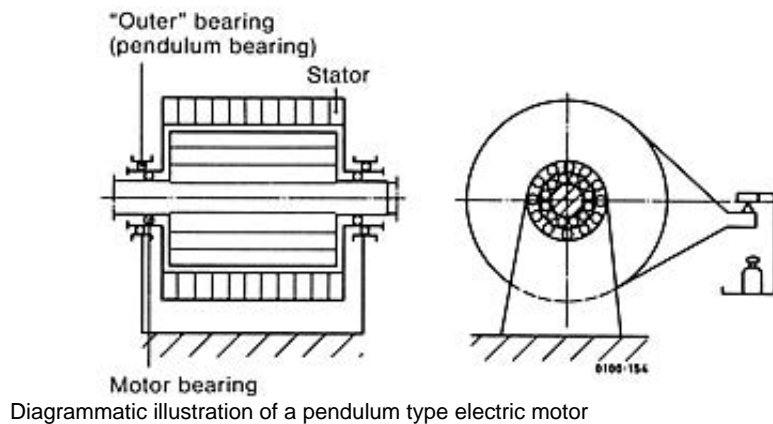
Pendulum Type Electric Motor

Pendelmotor

Moteur balance

The p.t.e.m. of an energy-producing machine (e.g. centrifugal pump) or the pendulum type generator of a prime mover (e.g. a hydraulic turbine) offers a very accurate torque measuring facility (measuring technique) if it is carefully designed and constructed. The torque is determined by the measurement of the moment of reaction (countermoment) on the stator of the motor or generator, generally by weighing at a known lever arm length (see illustration). The stator of the p.t.e.m. must be pivotally about the axis of the machine. This is usually accomplished by a so-called double bearing arrangement (see illustrations using ball or roller bearings (anti-friction bearing)). If however a very high degree of measuring accuracy is desired, the "outer" bearings should be designed in the form of hydrostatic or aerostatic bearings.

With this arrangement, only the torque actually delivered to the work-producing machine (e.g. centrifugal pump) or actually delivered by the prime mover (e.g. hydraulic turbine) is measured. Any losses arising within the outside bearings do not influence the measurement. When the accuracy requirements are exceptionally high therefore, every effort is made to connect all sources of losses which are of no interest to the measurement concerned (e.g. pump bearings) to the pivoting component (stator).



Percentage by Weight

Gewichtsprozent

Pourcentage en poids

see [Pulp Pumping](#)

Performance Chart

Kennfeld

Réseau de courbes caractéristique

A p.c. is a graphic presentation of several characteristic curves of a machine, in one picture, to illustrate the effect of changing various parameters. Examples of such parameters for a centrifugal pump are impeller diameter (cutdown of impellers), the rotational speed (control), blade angles of impeller vanes (impeller blade pitch adjustment) or pre-rotational swirl controller (cooling water pump) or the size of the pump.

Fig. 1 shows the p.c. of a multisuction volute casing pump, whose characteristic curves of the head H and the shaft power P are displayed versus various impeller diameters. The NPSH value (net positive suction head) usually does not change due to the cutdown of the impeller. In a p.c. the pump efficiencies, referred to as "constant efficiency curves" are entered as an additional parameter.

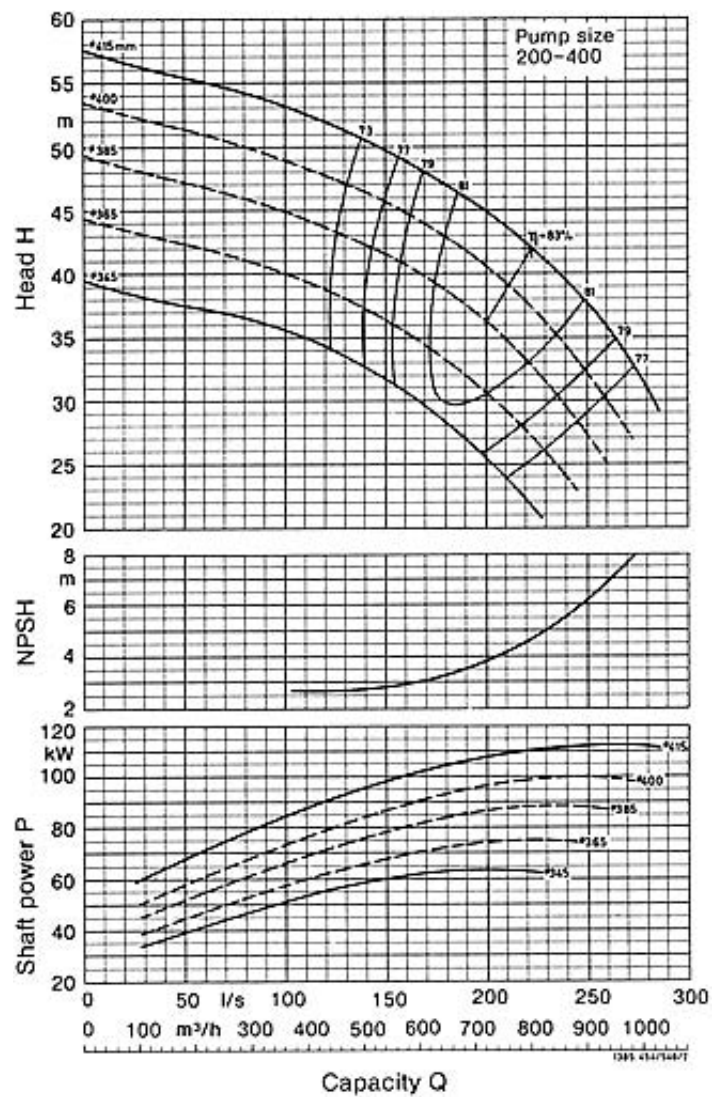


Fig. 1: Performance chart of a double suction volute casing pump with the characteristic curves for five different impeller diameters, at the rotational speed of $n = 1450 \text{ min}^{-1}$

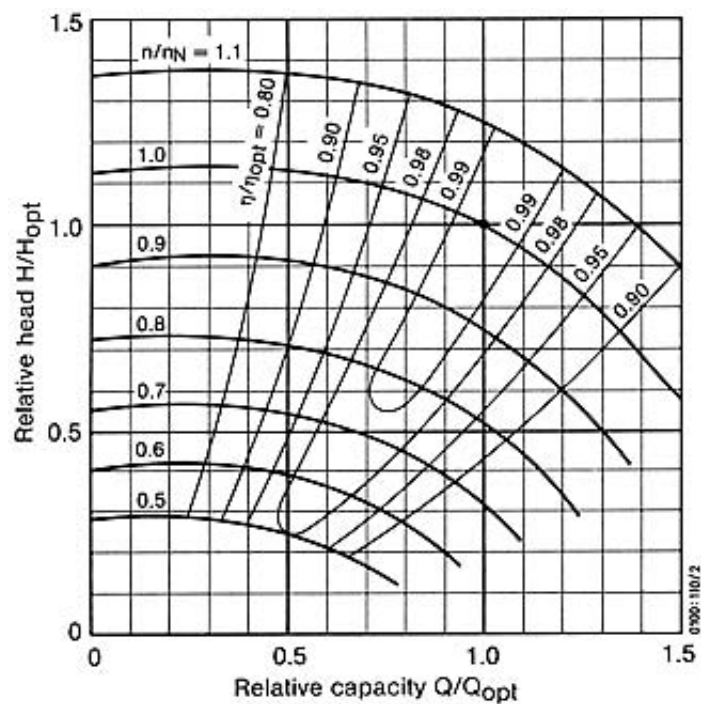


Fig. 2: Performance chart of a speed-controlled centrifugal pump, specific speed $n_q \approx 20 \text{ min}^{-1}$

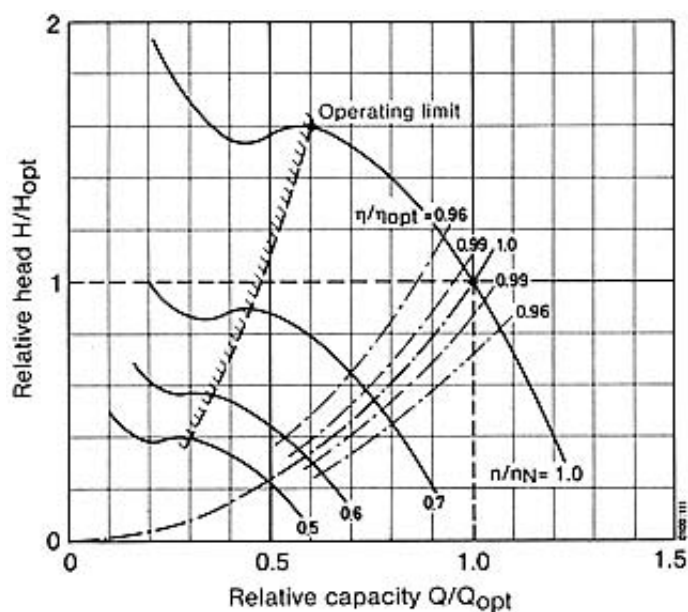


Fig. 3: Performance chart of a speed-controlled centrifugal pump, specific speed $n_q \approx 200 \text{ min}^{-1}$

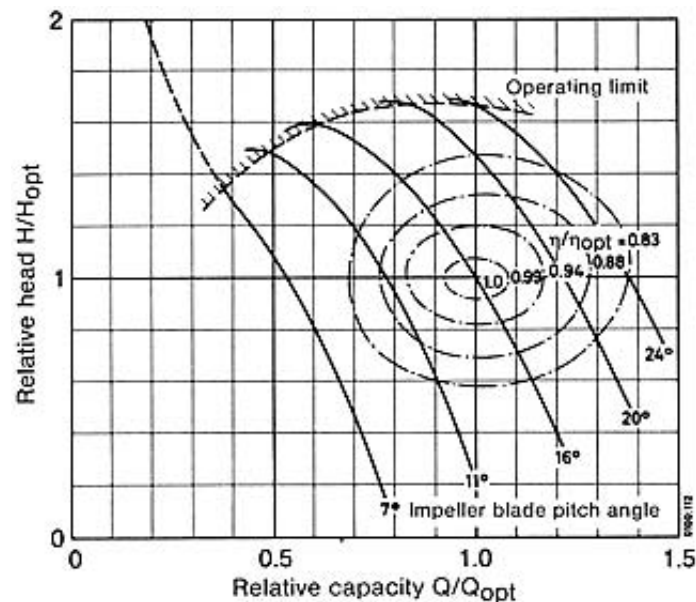


Fig. 4: Performance chart of a centrifugal pump with impeller blade pitch adjustment, specific speed $n_q \approx 200 \text{ min}^{-1}$

Figs. 2 and 3 show the p.c. of the head H of centrifugal pumps with a radial and an axial impeller at various rotational speeds n . The "constant efficiency curves" are also entered in this graph. All the numerical values are related to the optimal values of the best efficiency point and are represented dimensionlessly.

In a similar manner, Fig. 4 shows the p.c. of the head of a propeller pump with impeller blade pitch adjustment. Fig. 5 shows the p.c. of a mixed flow pump with an pre-rotational swirl controller (control). In both cases the blade angle of the blades is given as a parameter.

The complete list of the p.c. of various pump sizes or even of various pump series, in a QHdiagram, is given in a coverage chart. An example of this is Fig. 6, in which is shown the coverage chart of a pump series and its various sizes. The information for Fig. 1 was taken from this graph.

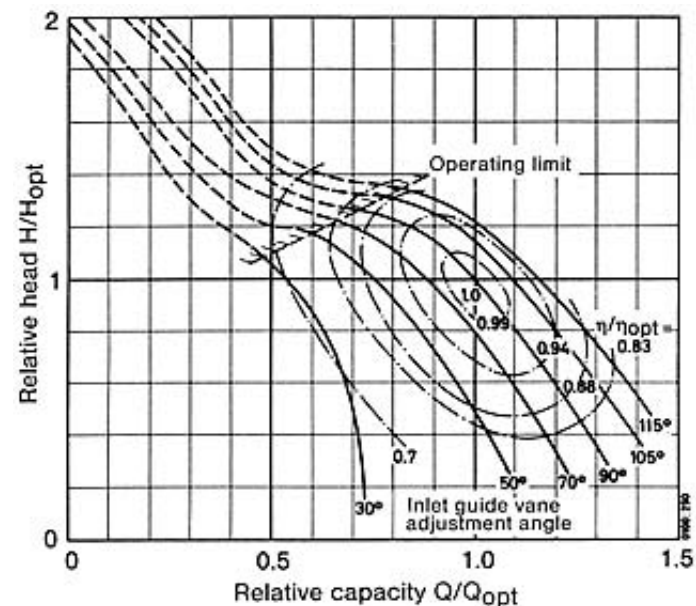


Fig. 5: Performance chart of a centrifugal pump with pre-rotational swirl controller, specific speed $n_q \approx 160 \text{ min}^{-1}$

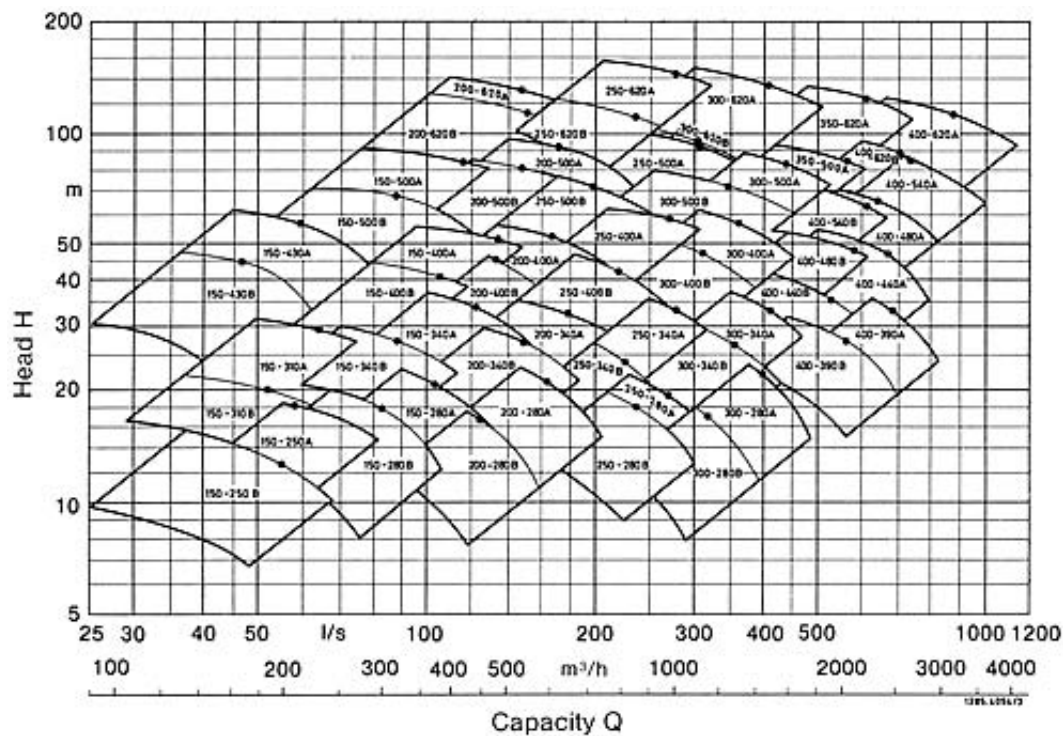


Fig. 6: Coverage chart of a series of double suction volute casing pumps as a family of curves of the various sizes of pumps in a QH-diagram, where $n = 1450 \text{ min}^{-1}$

Performance Coefficient

Leistungszahl

Nombre de puissance

see Characteristic Number

Peripheral Impeller

Peripheralrad

Roue à cellules périphériques radiales

see Impeller, Peripheral Pump

Peripheral Pump

Peripheralpumpe

Pompe à entraînement de liquide

The p.p. (see Fig. 1), also called vortex pump (torque-flow pump), is a centrifugal pump with an impeller equipped with a large number of small radial blades on either side of its periphery (approach flow from both sides of the impeller). The impeller rotates in a virtually concentric casing channel with an inlet and an outlet aperture, and the repeated flow of the pumped medium between impeller and casing channel and vice-versa ensures that a large amount of energy is imparted to the medium (Fig. 2). Because of this concentrated increase in energy, p.p.'s are relatively small and often built in the form of close-coupled pumping sets. The pressure coefficients (characteristic number) of p.p.'s even exceed those of side channel pumps, and their characteristic curves (Fig. 4) are steeper than those of side channel pumps. The shaft power of a p.p. decreases with increasing capacity.

A p.p. can be converted to a multistage pump by adding as many as three blade rings (Fig. 3) at different diametric stages. Running at a normal rotational speed of 2900 min^{-1} , such pumps can develop heads peaking at 1200 m. In the low capacity range, they therefore cover part of the service range of geared high-pressure pumps (geared pump) and even positive displacement pumps.

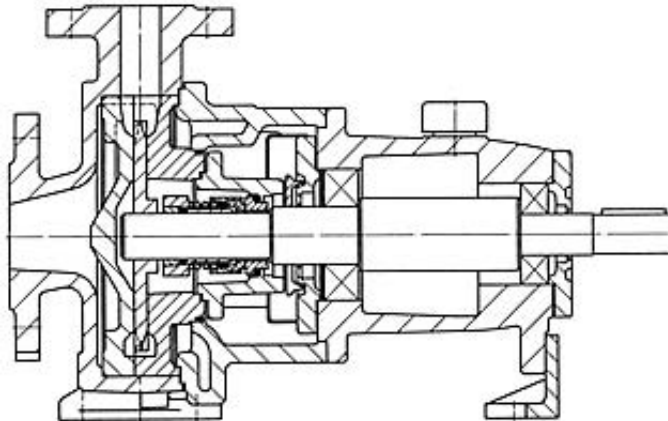


Fig 1: Meridional section through a peripheral pump

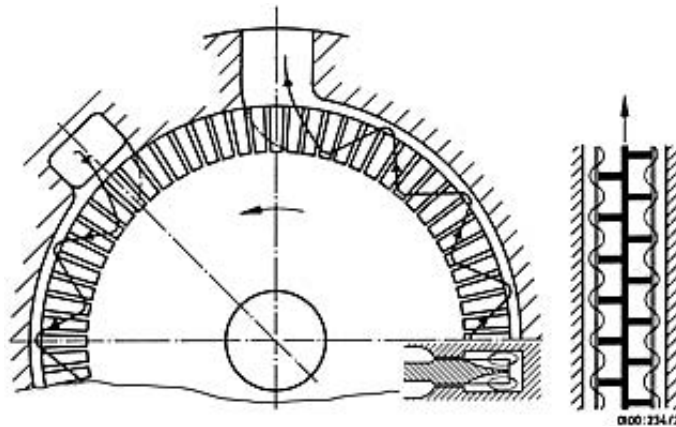


Fig. 2: Principle of energy supply in the impeller of a peripheral pump

Another characteristic feature of the p.p. is that it is able to pump fluids containing relatively large gas fractions (two-phase flow), in contrast to other centrifugal pumps. P.p.'s. are also able to continue operating under conditions of considerable vapour bubble formation (cavitation) without any flow separation and without any noticeable disturbance in their quietness (Fig. 4). Under these conditions, the throttling curve (characteristic curve) moves away only very gradually from the throttling curve measured under cavitation-free conditions (no sudden separation). Another advantage is the symmetrical approach flow (multisuction pump) to the impeller (very low axial thrust).

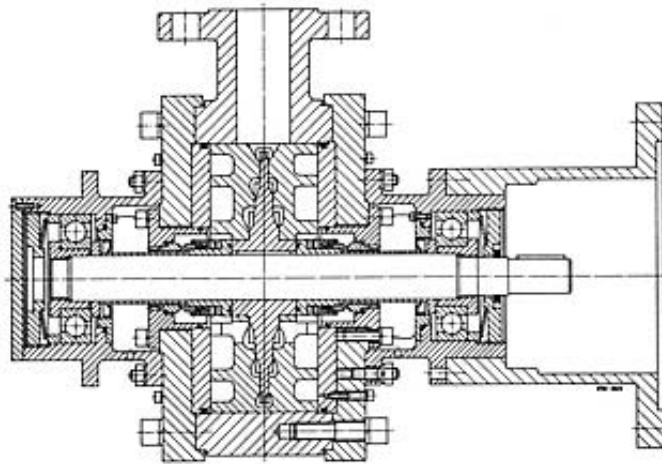
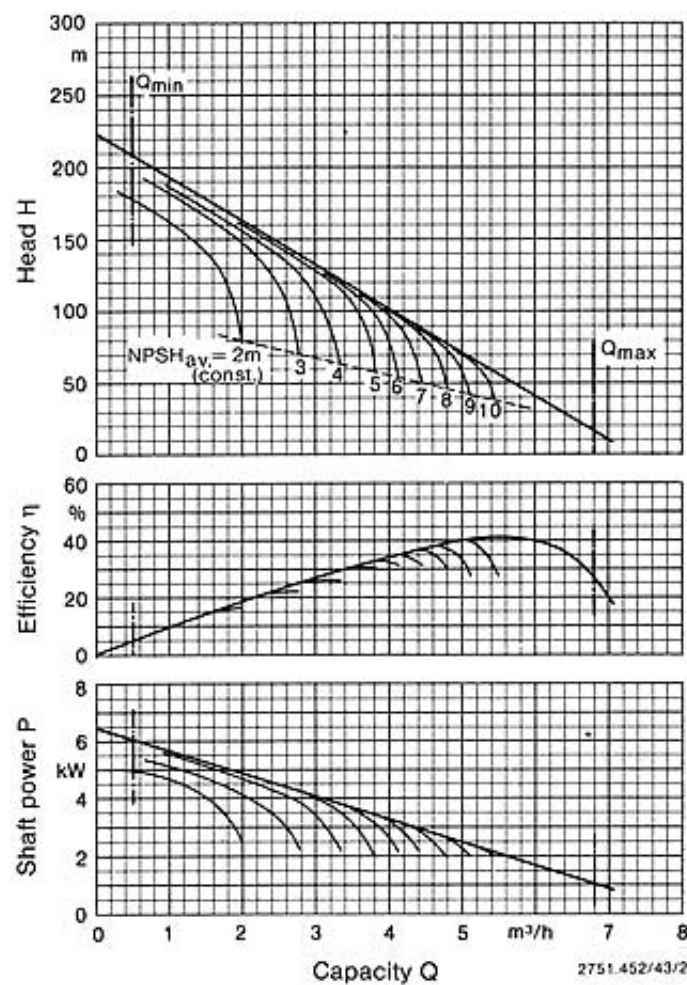


Fig 3: Two-bearing, multistage peripheral pump

Fig. 4: Characteristic curves of a peripheral pump (nominal diameter DN 25; impeller diameter 125 mm; rotational speed $n = 2900 \text{ min}^{-1}$)

Compared to radial flow pumps (impeller), p.p.'s have lower pump efficiencies.

P.p.'s are suitable for the pumping of clean (uncontaminated) liquids, e.g. as boiler feed pumps for small boilers, as pressure boosting pumps, as car washing pumps and as pumps for the chemical industry, and for all applications where low capacities are required in conjunction with high heads.

Peristaltic Pump

Schlauchpumpe
Pompe péristaltique

see [Positive Displacement Pump](#)

Phase Displacement

Phasenversetzung
Déphasage

see [Three-phase Current](#)

Phase Voltage


Phasenspannung
Tension de phase

see [Three-phase Current](#)

pH-Value

pH-Wert
Valeur pH

The pH-v. is the negative common logarithm of the effective hydrogen-ion concentration or hydrogen-ion activity, expressed in mole/l (DIN 38 404, Part 5, 1984 edition). The pH-scale extends from 0 to 14:

pH-Value	chemical reaction (temperature-dependent)	 decreasing H ⁺ -ion concentration
0 - 3	strongly acidic	
4 - 6	acidic	
7	neutral	
8 - 10	alkaline (basic)	
11 - 14	strongly alkaline	

The pH-v. can be determined by various methods. The calorimetric analysis method uses indicators which show colour changes at certain pH-v. - universal indicators are also available (paper, strips etc. with appropriate colour comparison scales). pH-v. can be determined with a high degree of accuracy by the electrometric method which uses electrodes (e.g. glass electrodes) and comparison solutions (buffer solutions).

DIN 19261 offers details on the measurement of pH-v.'s.

Pipe Friction Loss

Rohrreibungsverlust
Perte de charge par frottement dans la conduite

see [Pressure Loss](#)

Pipeline Pump

Pipelinepumpe
Pompe d'oléoduc

The petroleum industry uses centrifugal pumps and pipelines to transport crude oil from the oil fields or tank farms to the refineries (crude oil pipelines) and product pipelines to transfer the refined products such as petrol (gasoline), kerosene, heating oil or Diesel fuel to the user regions. The p.p.'s (Figs. 1 and 2) overcome differences in altitude (geodetic altitude) and pipe friction losses (pressure loss). P.p.'s can also be used as lye solution pressure pumps in gas washing plants or as underground cavern pumps for the flushing out and filling in of salt stock; these are usually application fields for which pumps without a balancing device are required.

P.p.'s have to meet the following requirements:

1. Safety and reliability of operation, i.e. simple and sturdy construction. This means a small number of stages and the balancing of the axial forces by a back-to-back arrangement of the impellers, or the use of double suction impellers (back-to-back impeller pump, multisuction pump).
2. Easy dismantling and reassembly, i.e. good accessibility and interchangeability of wear parts. This is achieved by an axially split casing at shaft centralize height, and by the arrangement of the branches on the lower half of the casing (Fig. 11 under pump casing).
3. High efficiency over as wide a capacity range as possible. This requirement can be met by selecting multistage pump back-to-back impeller pumps, or in the case of large capacities, single stage pumps with a double suction impeller. If necessary the pumps can be connected in parallel or in series (parallel operation, series operation). The viscosity of the medium pumped must be taken into account at the design stage of the p.p.
4. Adaptability in respect of changing performance requirements at later stages of exploitation, by fitting different internal components. This adaptability is achieved by changing the impeller width or the impeller diameter. - On the pump casing of multistage pumps, the interstage crossover ducts are shaped to provide favourable hydrodynamic flow characteristics (Fig. 2). Renewable sleeves and bushes protect all running surfaces exposed to wear.

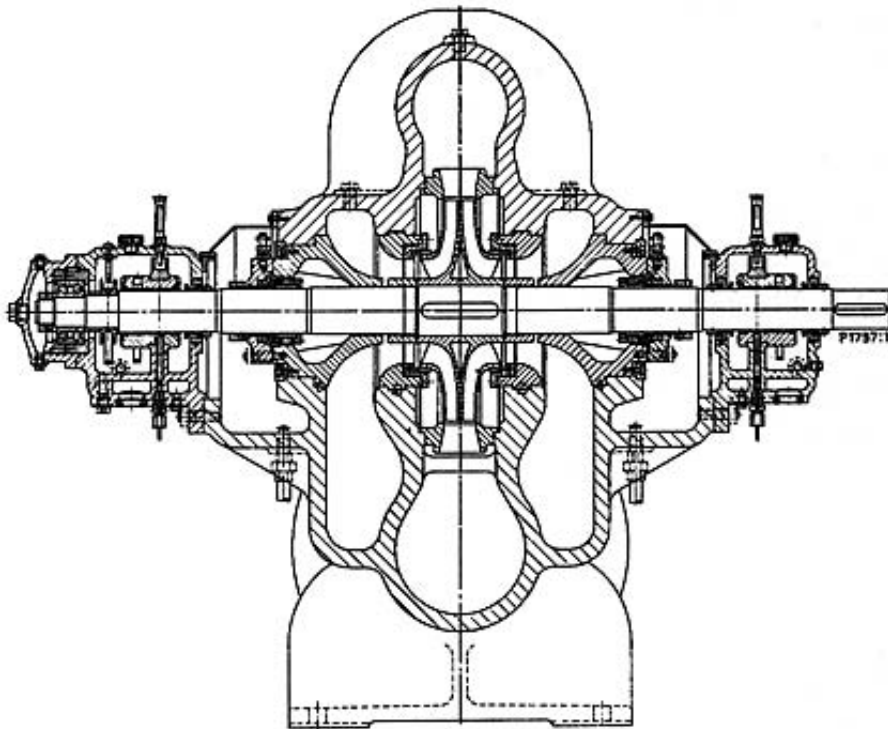


Fig. 1: Pipeline pump with axially split casing, single stage, double suction design

Mechanical seals are used almost exclusively as shaft seals. Soft packed stuffing boxes can however be fitted if desired. Both anti-friction and plain bearings are fitted as bearings, and segmental thrust bearings can also be fitted to absorb the axial thrust. By providing a flexible spacer-type shaft coupling, the mechanical seal or the bearing at the drive end can be renewed without having to dismantle the pump rotor; neither is it necessary to move the driver

or to remove it from the baseplate (pump foundation).

A booster pump is almost always installed, so that in general there is no requirement for an exceptionally low NPSH value (net positive suction head, suction behaviour).

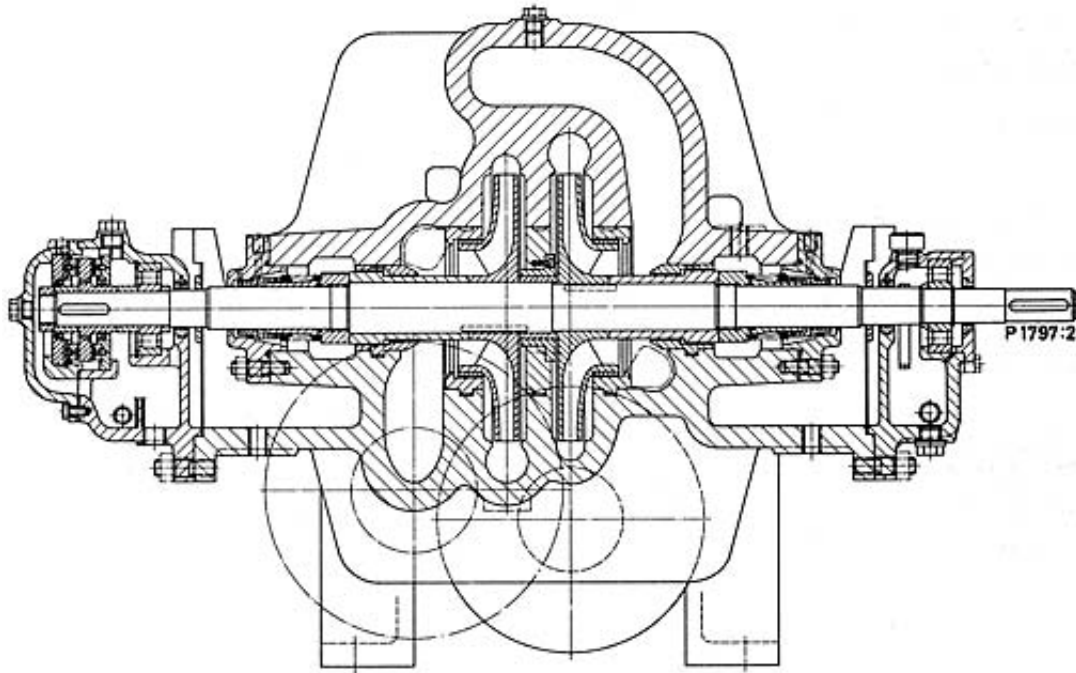


Fig. 2: Pipeline pump with axially split casing, two stage, single suction design

Because p.p.'s are usually operated fully automatically from a remote control centre it becomes necessary to monitor the following values: 1. inlet pressure, 2. bearing temperature, 3. shaft seal pressure, leakage, 4. temperature of fluid pumped, 5. oil pressure upstream of the segmental thrust bearing, 6. vibration amplitude, to keep a check on the bearings and on the suction behaviour (cavitation).

During start-up, steps must be taken to ensure, by means of sequential control, that the suction valve, discharge valve, and main valve of the pumping station are all opened in the correct sequence and at the correct time intervals, and that the lubricating oil pump and booster pump are also correctly switched on.

Piping

Rohrleitung
Conduite

P. is used to transport substances, fluids in particular. The inside diameters of p. are graded according to nominal diameters (DN), and the loading capacity by the internal pressure is graded according to nominal pressures (PN). For short lengths of p. and the transport of water, the following guideline values are usual for the flow velocity v and the capacity Q respectively:

DN	mm	25	40	65	100	150	200	300	500
v	m/s	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.9
Q	m ³ /h	2.5	7	21	56	140	270	660	2050

The velocity values listed in the Table may not be reached in the case of long pipelines and/or extended periods of operating. The economics of the installation should be borne in mind.

For suction pipes, the stated velocity values should be multiplied by 0.8, but the inside diameter should never be smaller than that of the pump suction branch (due attention must be paid to net positive suction head and interference from components).

Expansion joints (bellows type) are built into the p. system to absorb movements in the p., whatever their cause may be. Expansion joints are able to absorb the movements illustrated below (Fig. 1).

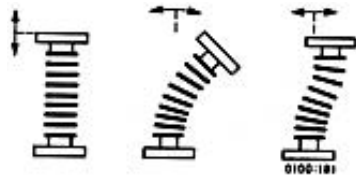


Fig. 1: Compensating movements of expansion joints

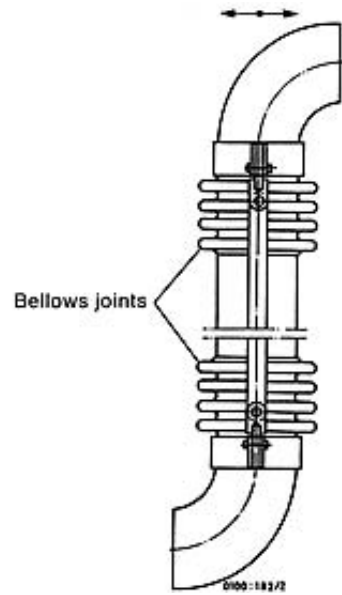


Fig. 2: Articulated expansion joint

P. can be installed in closed or open systems.

1. *The closed system* features friction type connections for the transmission of the axial forces, e.g. flanges and rigid fittings. The axial force arising from the internal pressure is absorbed as an internal force by the pipe wall and the pipe connections. The supports and fastenings of a closed p. system are only loaded by the weight and the dynamic forces. Thermal expansion is absorbed by flexible pipe supports or by articulated expansion joints (Fig. 2). The span l_s in m between adjoining supports of water-filled steel p. should be selected approximately according to the following rule:

$$l_s = 16 \cdot \sqrt{d}$$

with

d pipe diameter in m.

The wall thickness b of steel pipes loaded by internal pressure is calculated in accordance with DIN 2413. For pipes subjected mainly to static loading, the following rule applies:

$$b = b_0 + b_1 + b_2$$

with

$b_0 = d_a \cdot p \cdot K_1 / (20 \cdot K_2 \cdot K)$ and

b_1 allowance for short-fall in plate thickness in mm,

b_2 corrosion and wear allowance (e.g. 1 mm) in mm,

d_a outside diameter in mm,

K characteristic strength factor in N/mm^2 (for St 00, $K \leq 190 \text{ N/mm}^2$),

K_1 safety factor ≈ 1.75 ,

value coefficient of welding seam:

K_2 1.0 for seamless pipe,

0.5 to 1.0 depending on the quality and approval test of the weld seam,

p max. operating pressure in bar.

In the case of appreciable temperature variations, the change in length of a straight pipe is calculated according to the following formula:

$$\Delta l = \alpha \cdot l \cdot \Delta T$$

with

α coefficient of expansion in m/(m·K)

$10 \cdot 10^{-6} \text{ m/(m·K)}$ for cast iron } for 270
 $12 \cdot 10^{-6} \text{ m/(m·K)}$ for steel } to 370 K,

l pipe length in m,

ΔT temperature difference in K.

2. *The open system* has socket joints (couplings), loose fittings and axial expansion joints (Fig. 3) that provide compensation for thermal expansion. The axial force arising from the internal pressure must be absorbed as an external force by fixed points at the beginning and the end of the pipe run and at any change of direction or cross-section. Adequate guidance, e.g. in the form of floating clips, roller bearings, etc., must be provided to keep the pipe from buckling.

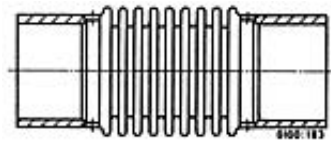


Fig. 3 : Axial expansion joint

Forces on the fixed points:

End securing (Fig. 4)

$$F = A \cdot p_{\max}$$

with

$A = d_a^2 \pi / 4$ pipe cross-section (or, in the case of an expansion joint, the effective bellows cross-section),

p_{\max} max. pressure,

d_a outer diameter of pipe.

Branch securing (Fig. 5)

$$F = A_{ab} \cdot p_{\max}$$

with

A_{ab} cross-section of branching pipe.

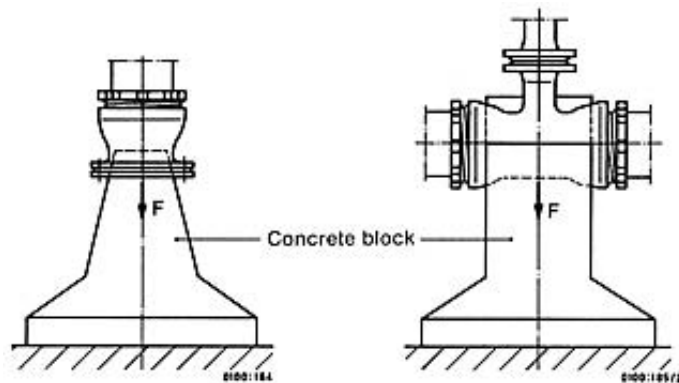


Fig. 4: End securing

Fig. 5: Branch securing

Elbow securing (Fig. 6)

$$F = 2 \cdot A \cdot p_{\max} \cdot \sin \frac{\alpha}{2}$$

with

α deflection angle.

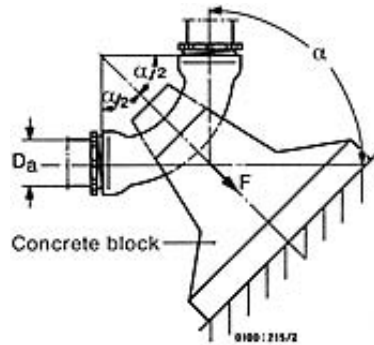


Fig. 6: Elbow securing

Piping Characteristic Curve

Rohrleitungskennlinie

Courbe caractéristique de réseau

see System Characteristic Curve

Piston Pump

Kolbenpumpe

Pompe à piston

see Positive Displacement Pump

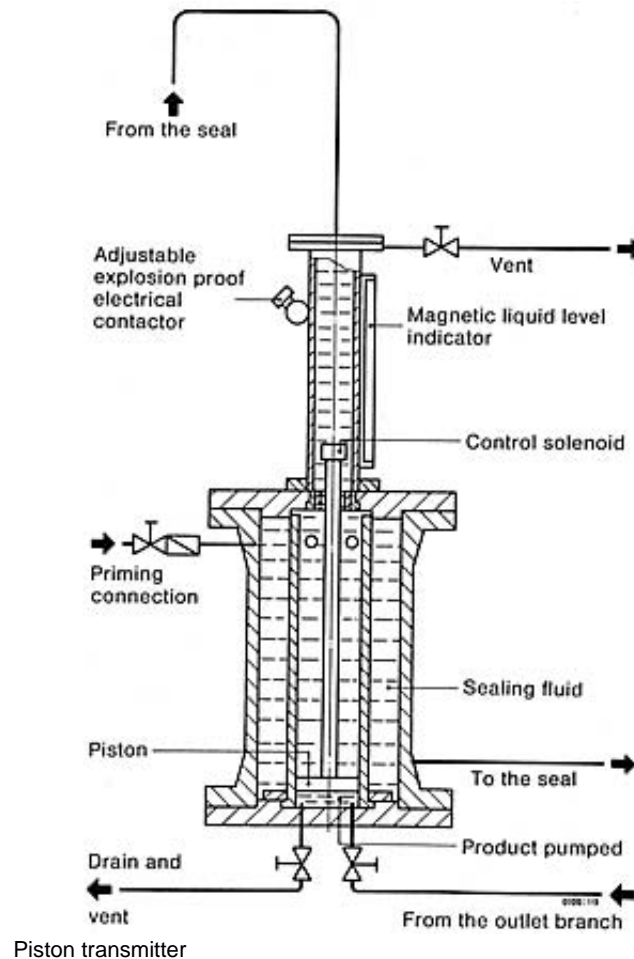
Piston Transmitter

Kolbentransmitter

Transmetteur à piston

The p.t. is a glandless pressure transmitting device and for the protection of the mechanical seal (shaft seals) produces a sealing liquid pressure depending on the pressure of the pump or medium pumped. Consisting mainly of a double wall reservoir in which a piston effects the separation of the sealing product from the product pumped (see illustration).

In order to generate the required sealing pressure, the bottom part of the cylinder is connected to the pump discharge branch, or to a product line at the required pressure. The top part of the cylinder is filled with sealing fluid and is connected to the compartment situated between the two sealing surfaces of the doubleacting mechanical seal (Fig. 4 under shaft seals).



PITOT Tube

PITOT-Rohr
Tube de PITOT

see [Measuring Technique](#)

Pitting

Anfressung
Attaque

see [Erosion](#)

Plain Bearing

Gleitlager
Palier lisse

The p.b. is frequently used as an element in the construction of centrifugal pumps (Fig. 1). A moving component slides inside a stationary component. A distinction is made between radial p.b.'s for radial forces (transverse forces) and axial (or thrust) p.b.'s for axial forces (longitudinal forces).

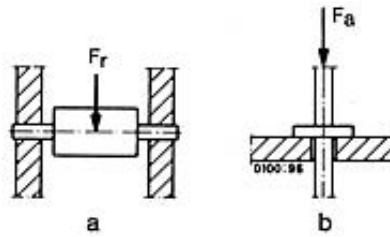


Fig. 1: a radial plain bearing; b axial plain bearing (diagrammatic sketch)

a) *Radial p.b.'s*. The moving part is the pin or journal of the axle or shaft. The stationary part, in various forms, consists of: cylindrical bearing shell (or brass) (Fig. 2 a), two-face bearing shell with "lemon clearance" (Fig. 2 b), two-face bearing shell offset radially (Fig. 2 c), three-face bearing shell (Fig. 2 d), three-face and multiple-face bearing shell with lubrication grooves or pockets (Fig. 2 e), rubber bearing (Fig. 2 f), multiple-face bearing with tilting radial segments (pads) (Fig. 2 g).

The variety of bearing designs is related to the control of the dynamic response of centrifugal pump rotors. The general vibration performance of p.b. rotors depends on the rotor mass, the mass distribution, and the stiffness of the shaft and the stiffness and damping characteristics of the bearings under a given load. Through an appropriate design of the rotor bearings, both types of lateral vibrations (namely the forced and self-induced vibrations) of the rotor can be eliminated or reduced to an acceptable level for the machine; the dynamic bearing coefficients can be optimized. The choice of bearings is a key aspect in this optimization, since each type of bearing offers different performance characteristics.

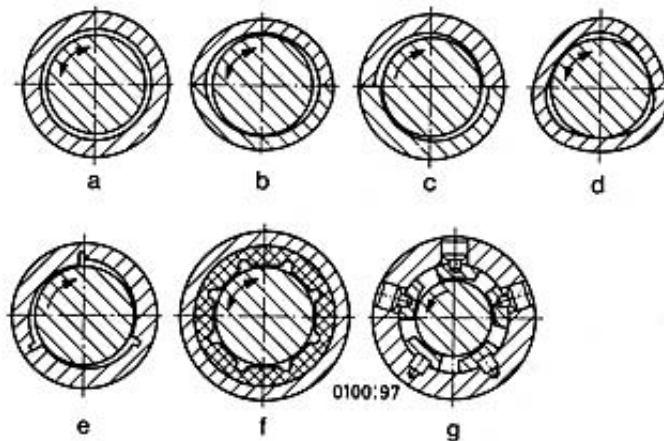


Fig. 2: Various types of bearing shells

By the choice of suitable manufacturing dimensions of both functional components, or of their mutual adjustment in operation, an operating bearing clearance is created which in most cases is fed with a liquid or solid (grease type) lubricant, in order to reduce the sliding friction to a minimum. The operating bearing clearance provides space for a load-bearing lubrication wedge to be formed by the lubricant when the circumferential velocity of the bearing journal is high enough. Such a lubrication wedge separates the sliding faces from one another, i.e. the bearing operates on full lubrication. This process is the typical one for *hydrodynamic p.b.'s*.

Hydrostatic p.b.'s represent a different type, in which the lubricant is fed under high pressure to the individual pressure chambers of the running surface. The bearing force is absorbed as a result of the difference in the following pressures: static pressure in the chambers on the loaded side of the running surface (very small clearance during operation, therefore very small decrease in pressure in the lubricant layer) and static pressure in the chambers on the unloaded side of the running surface (large clearance during operation, therefore considerable pressure drop in the lubricant layer).

Advantages and disadvantages of hydrodynamic and hydrostatic p.b.'s:

Type of bearing	Advantages	Disadvantages
hydrodynamic	Simple manufacture; the lubricant is fed either pressureless or at a very low feed pressure to the bearing during operation (a very low, or negligible energy requirement for the oil supply system).	During the starting and stopping phases of the machine, there is no pure fluid friction present (increased wear on the sliding faces).
hydrostatic	Pure fluid friction static is present at all times, including the starting and stopping phases (no danger of excessive wear). In comparison with hydrodynamic bearings of equal load-bearing capacity, smaller dimensions and smaller frictional losses.	More expensive to manufacture (increased number of manufacturing operations). More expensive to operate, because a pressure-boosting plant is required for the lubricant (increased capital investment and running costs, energy costs).

Both types can be combined. Furthermore, in the case of p.b.'s which operate hydrodynamically in the steady state, the increased friction during starting and stopping can be reduced and excessive wear avoided by providing an auxiliary hydrostatic lubrication at a high lubricant pressure via longitudinal grooves which terminate inboard of the extremities of the bearing shells. The auxiliary lubricant feed is shut off during normal running, so as to ensure that the hydrodynamic pressure is maintained in the lubricating clearance gap.

Various *friction conditions* in a p.b. are illustrated diagrammatically in Fig. 3, starting with

- dry friction without any separating lubricant layer between the stationary and moving components, through
- mixed friction, which is part dry, part fluidfriction, onto
- which is fluid friction with a separating layer of lubricant.

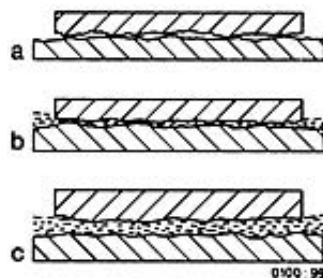


Fig. 3: Various friction conditions in a plain bearing

All three types of friction can occur in a hydrodynamic p.b. during the three phases of operation: during start-up, during steady running and during shut-down to a standstill. Start-up is the operating phase from a standing start to the attainment of full operating speed. As the sliding speed steadily increases, the zone of mixed friction is passed through, in the case of hydrodynamic p.b.'s, with the amount of dry friction diminishing and that of fluid friction increasing. Finally the transition point is reached where the sliding faces separate from one another and pure fluid friction with a minimum of friction losses prevails. As the sliding speed increases further, the thickness of the lubricating film also increases, but the friction losses rise again slightly. This frictional behaviour has been investigated experimentally by STRIBECK (see Fig. 4 for diagrammatic explanation).

When a machine runs down to a standstill, the reverse of the sequence described above for the starting process takes place for the p.b.

A p.b. should as a general rule have its steady state operating point in the region of fluid friction. If mixed friction prevails during continuous operation, there will be excessive wear at the sliding faces, and particular care should be taken in selecting and matching the two materials in sliding contact correctly (wear, conduction of heat).

Many pumps are equipped with guide bearings that are **lubricated by the pumped fluid**. In cases such as this the choice of bearing materials is especially important, since each fluid has its own characteristics as a lubricant. When using clean water as lubricant one has several bearing materials available for use that have good tautological characteristics: metallic alloys, elastomers, hard rubber, electro-graphite with or without resin binders, hard graphite with resin binders or antimony impregnation, etc. If the pumped medium as a lubricant is dirty or contains such solids as sand, it is recommended to use hard metals or ceramics (such as siliconcarbide) materials as bearing materials: bushings and shaft sleeves from the same material result in a maintenancefree bearing.

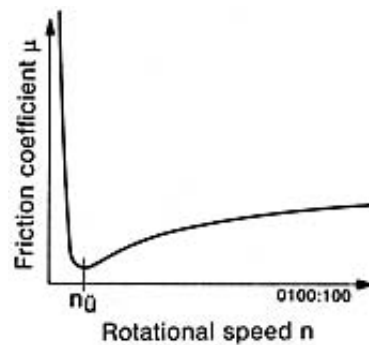


Fig. 4: Friction coefficient as function of speed (STRIECK curve). Subscripts indicates the transition point

Fig. 5 illustrates diagrammatically the position taken up by the journal at various rotational speeds, and also the pressure distribution in the wedge-shaped clearance of a cylindrical radial p.b.

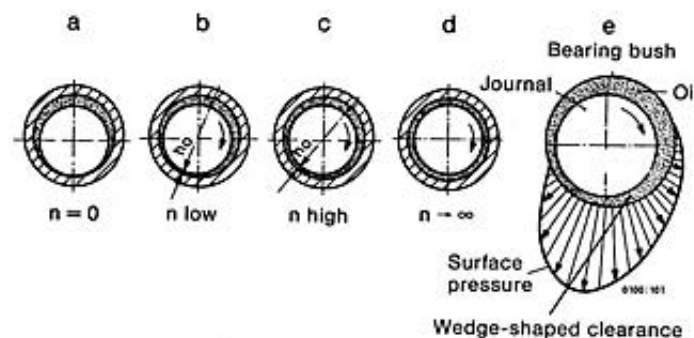


Fig. 5: Position of journal at various speeds n , h_0 thickness of lubricating film at the narrowest point of the clearance

The friction losses (friction power) are converted into heat, which is partly led away to the surrounding air by the bearing housing, and partly transmitted through the shaft. The p.b. should therefore not exceed a given max. bearing operating temperature. If necessary a cooling system must be provided for the bearing or the lubricant (usually water cooling).

The *design of hydrodynamic p.b.'s* involves the solution of a complex problem which takes into account a number of factors such as the geometry and size of the bearing, the bearing load, the viscosity of the lubricant, the sliding speed, the nature of the flow in the bearing (laminar or turbulent fluid dynamics), and the mutual influence of all these factors on one another. The object of the design of p.b.'s is to be reasonably sure to attain pure fluid friction during operation. The entire design procedure is based partly on theoretical considerations, partly on experimental data, which incorporate a number of interrelated characteristic factors in a suitable manner, e.g. in the case of a radial p.b.:

bearing ratio	B/d ,
relative bearing clearance	$\Psi = (D - d)/d$,
relative thickness of lubricating film	$\delta = h_0(S/2)$,
SOMMERFELD number	$So = p\Psi^2/(\eta \cdot \omega)$

Where

- B effective bearing width,
- D diameter of bearing bore,
- d diameter of bearing journal,
- h_0 absolute thickness of lubricant film,
- S absolute bearing clearance = $D - d$,
- p mean surface pressure of bearing = $F/(B \cdot d)$,
- F bearing force (radial force),
- η dynamic viscosity of lubricating oil,
- ω angular velocity (rotational speed).

Fig. 6 illustrates the relationship between δ , S_o and B/d for radial p.b.'s.

P.b.'s which do not exhibit purely laminar flow characteristics (this happens frequently in bearings with very high sliding speeds and simultaneously very low viscosities of the lubricant) tend to have a higher load-bearing capability and also higher frictional losses than purely laminar p.b.'s. Apart from the characteristic factors for p.b.'s already mentioned, the qualitative difference between a laminar and a turbulent flow plays a decisive part in this case. The design of turbulent p.b.'s is much more complicated than that of laminar p.b.'s, because of the lack of comprehensive data, and the designer must rely to a greater extent on practical experience.

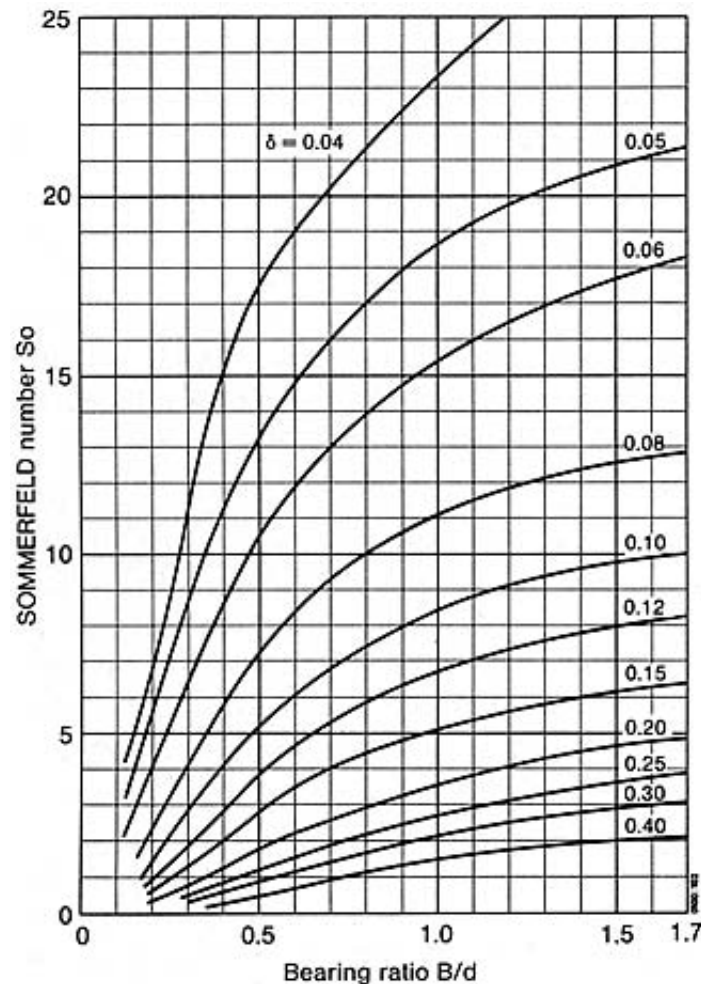


Fig. 6: Relative thickness of lubricating film δ in function of SOMMERFELD number S_o and of bearing ratio B/d for a radial laminar flow plain bearing (according to DECKER)

b) *Axial (thrust) p.b.'s*. The moving part is the thrust bearing disc or plate. The stationary part has various forms such as: thrust bearing ring (Fig. 7 a), thrust bearing ring with machined wedge faces (Fig. 7 b), thrust bearing ring with stepped damming gap (Fig. 7 c), eccentrically supported tilting pads or segments (Fig. 7 d); often centrally supported tilting pads are also used, e.g. where a cooling water pump has to rotate in reverse because of a reflux from the piping system.

Depending on the type, axial p.b.'s are also subdivided into hydrodynamic, hydrostatic, and combined hydrostatic-hydrodynamic p.b.'s for special applications. Both types must ensure by suitable design measures that the supported shaft can move sufficiently in the axial direction to accommodate the thickness of the lubricant film, which varies according to load, viscosity of the lubricant, and sliding speed. The same arguments as for radial p.b.'s apply to the advantages and disadvantages of hydrodynamic versus hydrostatic axial p.b.'s.

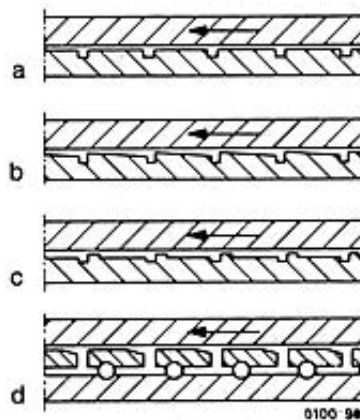


Fig. 7: Various types of stationary element of a thrust bearing

Fig. 8 illustrates an example of a water-lubricated turbulent p.b. in the bearing arrangement of a centrifugal pump shaft.

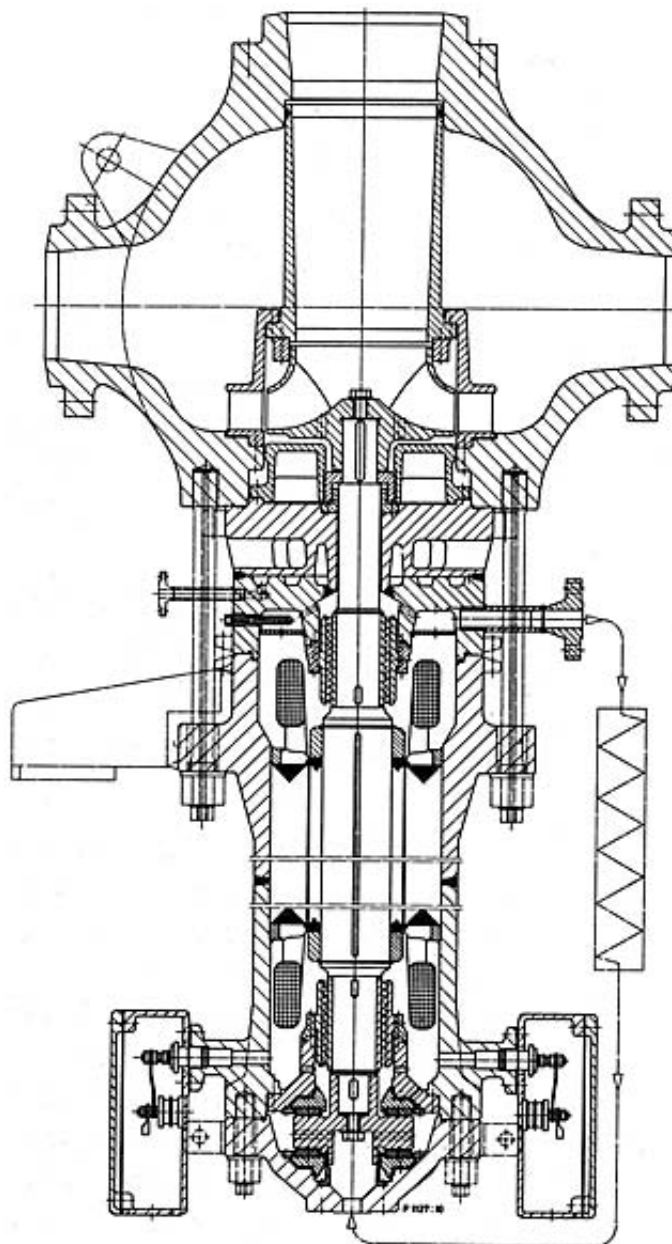


Fig. 8: Plain pump shaft bearings of a glandless boiler circulating pump

Plastic Pump

Kunststoffpumpe
Pompe en matière plastique

P.p. is a centrifugal pump with components in contact with the fluid pumped largely made of plastic materials (Table 2 under materials).

Plug-In Pump

Einsteckpumpe
Pompe accessoire incorporée

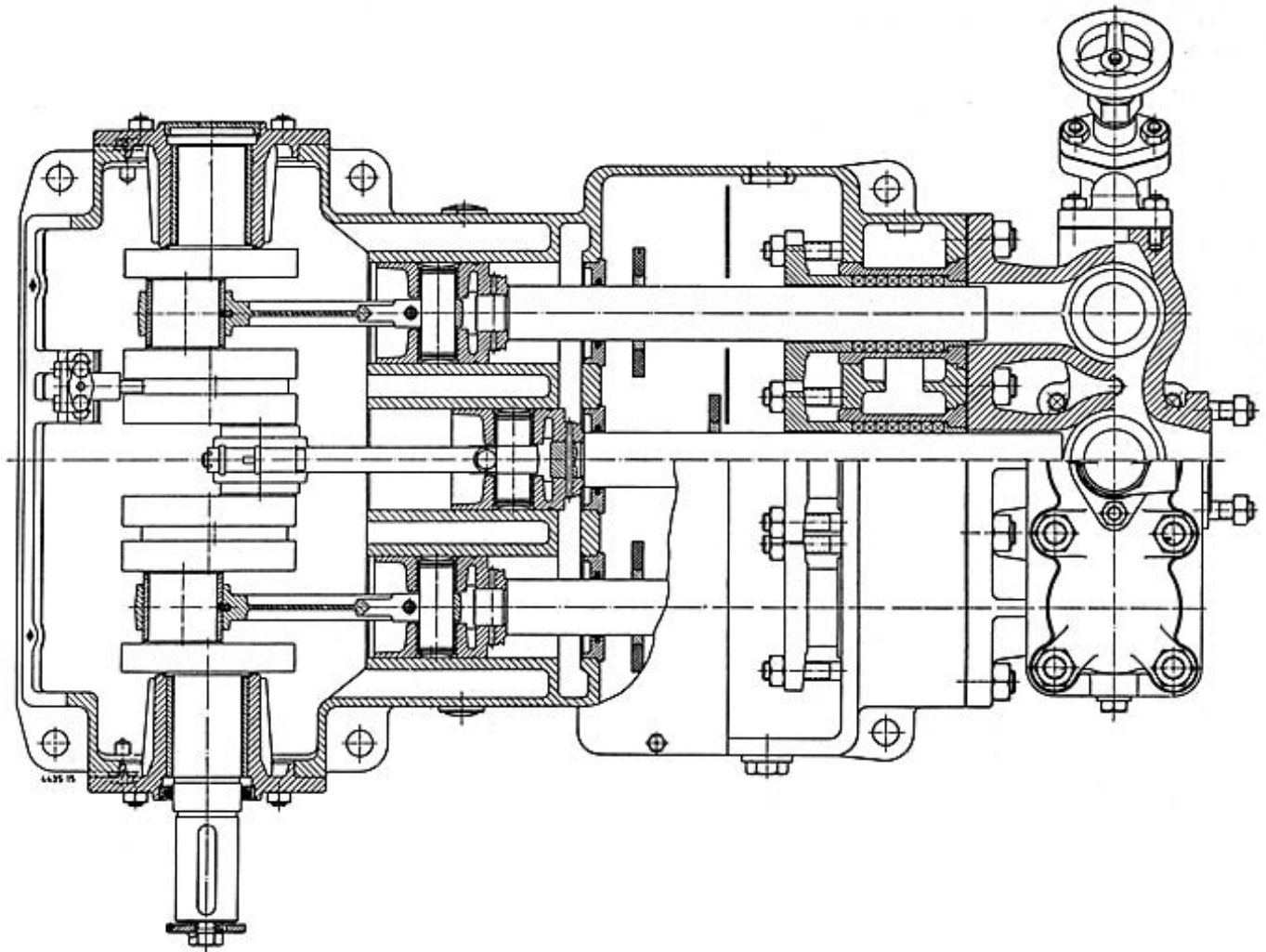
see Reactor Pump

Positive Displacement Pump

Verdrängerpumpe
Pompe volumétrique

P.d.p., also known as volumetric pump is the collective concept for all pumps operating according to the positive displacement principle, i.e. pumps which transmit energy to the fluid pumped by means of displacement bodies which periodically increase and decrease the working volumes, which are separated from the feed and the discharge by separating elements.

Depending on the motion, a distinction is made between reciprocating (*oscillating*) and rotary displacement bodies; the oscillating motion can be a straight-line motion (e.g. in the case of piston or plunger pumps or diaphragm pumps) or it can be a motion along the arc of a circle (e.g. in the case of semirotary wing pumps). Examples of *rotary* p.d.p.'s are the gear pump (where the meshing teeth of a pair of gearwheels form the displacement element and the separating element at one and the same time) the helical rotor pump (where the displacement element- and simultaneously the separating element- is the helical rotating spindle running eccentrically in a volute, double-pitch casing), the liquid ring pump (water ring pump) for the pumping of gases (the displacement element is an eccentrically rotating liquid ring), the roller vane pump (the displacement element is an eccentric hollow cylinder which bounds the radially mobile cell walls) and the peristaltic pump (where the displacement element and simultaneously the separating element is formed by the places on the tube which are periodically squeezed in sequence).



Three-plunger pump

The best-known p.d.p.'s are the *piston or plunger* pumps with straight-line stroke motion. Their displacement bodies are either disc-shaped pistons sliding to and fro within a hollow cylinder and sealing at their periphery, or plungers which slide to and fro within a stuffing box and increase and decrease the working volume (see illustration). The separating bodies are nonreturn valves or check valves (valves and fittings) which are automatically controlled by pressure differences, or, in the case of higher rotational speeds, slide valves actuated by the crank mechanism. These valves are designed to control the admission into and discharge from the working volume of the fluid pumped at the right times, so as to generate a flow of fluid by the alternating suction and discharge action of the displacement body, in a manner governed by the design of the pump. By arranging two working volumes opposite one another, using the same displacement body, one obtains a double-acting piston or plunger pump instead of a single-acting one, with a more even torque pattern and a steadier flow pattern. The pulsating discharge flow is further smoothed by air receivers acting as shorttime accumulators (pressure vessel) arranged in the suction and discharge pipes, or by having several cylinders in parallel operated off the same crankshaft (duplex pump, triplex pump, see illustration).

The cylinders may be arranged horizontally or vertically, and the crankshaft is usually horizontal.

Piston and plunger pumps are either provided with a very simple by-pass adjustment which returns a greater or lesser portion of the flow from the discharge to the suction as required (the losses are quite acceptable on the smaller units), or with a more economical speed adjustment (control), which is complicated and expensive in the shape of a stepless electricmotor control but is an acceptable compromise in the form of a pole-changing motor combined with by-pass adjustment. The number of strokes of steam-driven piston pumps on the other hand is easily and economically controllable.

Piston and plunger pumps are mainly used for handling very low capacities at very high heads (pressured); they are therefore suitable for very low specific speeds. Outside of this region, they are being superseded to an increasing degree by centrifugal pumps in their varied specialized construction types (multistage pump, peripheral pump).

The advantages of the piston and plunger pump include its self-priming capability (self-priming pump) and its high efficiency; its disadvantages include its pulsating delivery, and, in the case of large units, its large space requirements, heavy weight and high capital investment costs related to the unit of pump output.

The main application fields for p.d.p.'s are plants which require water or oil under pressure (pressure pumps), particularly in the chemical industry, also on board ship and in small and medium-sized industrial plants. P.d.p.'s are also widely used in drive technology applications (in the form of radial and axial piston or plunger pumps for stepless energy transmission).

Positive Suction Head

Zulaufhöhe

Hauteur de charge

see Suction Behaviour

Potential Flow

Potentialströmung

Écoulement potentiel

P.f. is a vortex-free and source-free flow (apart from isolated singularities). Its velocity field \vec{v} fulfils the condition of freedom from rotation ($\text{rot } \vec{v} = 0$), and can be derived from a velocity potential Φ in accordance with $\vec{v} = \text{grad } \Phi$. In the case of incompressible media, the velocity potential Φ satisfies the potential equation $\Delta\Phi = 0$. Δ is the LAPLACE operator. The potential equation is capable of verifying the condition of tangential flow direction at stationary walls, but not the no-slip conditions. A closed body surrounded by flow on all sides does not experience drag in a p.f. but only lift.

Simple examples of p.f. are parallel flow, source or sink flow and the potential vortex. In a source or sink flow, the radial component v_r (the only one present), and in a potential vortex, the circumferential component v_u (the only one present) of the velocity varies in inverse ratio to the radial distance r from the centre. In both instances there is a singularity at the centre, where $r=0$, as the velocity there increases without limit.

Strictly speaking, p.f.'s can only be flows of frictionless media. However, the flows of real media subjected to friction can be assumed as approaching p.f.'s at large enough distances away from stationary walls, and the friction forces need only be taken into account in a thin boundary layer close to the wall. The flow around bodies, in particular e.g. the flow and pressure distribution in a cascade can be calculated as a p.f. with a fairly good approximation, for high REYNOLDS numbers (model laws), i.e. for thin boundary layers, if the body contour is increased in the calculation by the displacement thickness of the boundary layer.

Important methods of calculating a p.f. (aerofoil theory) include the conformal representation method, by means of which a flow field can be transposed with angular fidelity from one complex plane into another, and the singularity method, by means of which the flow field is represented by superimposing separate or continuous singularities (sources, sinks, vortices) on the bodies surrounded by the flow. When handling the p.f. in a rotating cascade, it must be borne in mind that, although the absolute flow \vec{v} is rotation-free, the relative flow \vec{v}' does not fulfil the condition of freedom from rotation, because of the rotating reference system (rigid body), therefore $\text{rot } \vec{v}' \neq 0$. As a result, a so-called relative vortex rotating in the opposite direction to the rotation of the impeller is formed in the vane channel of an impeller through which a frictionless flow passes.

Power

Leistung
Puissance

P. is work accomplished in a given time. In centrifugal pump technology, the concepts of mechanical p., electrical p. and thermal p. are often used.

Mechanical p. is the quotient of mechanical work A over time t

$$P = \frac{A}{t} = \frac{F \cdot s}{t}$$

A is accomplished when a force F, acting in the direction of displacement, displaces its point of impact over a distance s. When a body is rotated, the mechanical p. is the product of the torque T and the angular velocity ω (rotational speed)

$$P = T \cdot \omega$$

The SI unit of mechanical p. is the Watt.

Electrical p. is the product of current (active current or energy component) and the voltage (power factor $\cos \varphi$). The SI unit of electrical p. is the Watt. Electrical p. is 1 W (Watt) when a current of 1 A flows in an electrical circuit at a voltage of 1 V. This is only true if the current and voltage are in phase. If there is a phase displacement between current and voltage by the angle φ in an alternating current circuit, only the active current $J_w = J \cdot \cos \varphi$ should be entered in the p. calculation equation. This active current is in phase with the voltage. The current component which is displaced 90° in relation to the voltage is the reactive or wattless current $J_q = J \cdot \sin \varphi$. The product of voltage U by current (apparent current) J is the apparent power P_s .

Active p. (true p.) $P_w = U \cdot J \cdot \cos \varphi$,
reactive (wattless) p. $P_q = U \cdot J \cdot \sin \varphi$,
apparent p. $P_s = U \cdot J$.

The unit of apparent p. is generally referred to as "Voltampere" (VA) instead of Watt (W), and the unit of reactive p. as "Voltampere reactive" (var).

Thermal p. (or refrigerating p.) is the heat flow, i.e. the quotient of quantity of heat (SI unit: 1 Joule) and time. The SI unit of thermal p. is 1 Watt = 1 Joule/second.

Power Control

Leistungsregelung
Réglage de puissance

see Control

Power Factor $\cos \varphi$

Leistungsfaktor $\cos \varphi$
Facteur de puissance $\cos \varphi$

Inductive and capacitive reactances in alternating current circuits cause a phase displacement in time of the current flow in relation to the voltage. One period corresponds to 360° . The angle φ of phase displacement between voltage and current is $+90^\circ$ for capacitance and -90° for inductance. For Ohmic resistance, $\varphi = 0^\circ$. Mixed resistances are termed impedances. Their phase displacement angles lie between 0° and $+90^\circ$ or 0° and -90° respectively.

The cosine of the phase displacement angle φ is termed p.f., and it is used to calculate the active current J_w or the active (or true) power P_w :

$$J_w = J \cdot \cos \varphi ,$$

$$P_w = U \cdot J \cdot \cos \varphi .$$

Power Measurement

Leistungsmessung
Mesure de puissance

see Measuring Technique

PRANDTL Tube

PRANDTL-Rohr
Tube de PRANDTL

see Measuring Technique

Pressure

Druck
Pression

P., in the sense of a force per unit area can be interpreted on the one hand as mechanical stress or strength (resistance per unit area of a material against shear, rupture etc.), and on the other hand as static p. in a fluid. Preferred units of p. are 1N/mm^2 in the first instance and 1 bar or 1 mbar in the second instance.

The *static* p. in a fluid is the p. which a probe entrained with the fluid would measure (pressure measurement, measuring technique). Apart from the static p., symbol p, the concepts dynamic p. P_{dyn} and total p. plot are used in fluid dynamics.

The dynamic p. is defined as

$$p_{\text{dyn}} = \frac{\rho \cdot v^2}{2}$$

where

ρ density of fluid, and
 v absolute velocity.

p_{dyn} at a stagnation point corresponds precisely to the increase in static p. along the flow line through the stagnation point (flow line) as a result of the fluid being brought there to rest; p_{dyn} is therefore also designated as "stagnation pressure".

The *total* p. is defined as

$$p_{\text{tot}} = p + p_{\text{dyn}} + \rho \cdot g \cdot z$$

where

z geodetic altitude above an arbitrarily selected datum level, and
 g gravitational constant.

In centrifugal pump technology, "pressure" usually refers to a *static p*. In this connection, German Standard DIN 24260 (Liquid pumps; centrifugal pumps and installations; terms, letter symbols, units) lays down the following rules: the barometric pressure p_b and the vapour pressure p_D of the fluid pumped must be specified as absolute p., whereas all other static p.'s must be specified as gauge p. (in relation to barometric pressure). In the case of p.'s below atmospheric, the values will be negative ones.

The most widely used static p.'s in centrifugal pump technology are (see Fig. 2 under head):

1. p. at inlet cross-section of the pump, abbreviation symbol p_s , which is the gauge p. at the inlet cross-section of the pump at the elevation z_s ;
2. p. at outlet cross-section of the pump, symbol p_d , which is the gauge p. at the outlet cross-section of the pump, at the elevation z_d ;
3. manometer (or p. gauge) reading at inlet cross-section of the pump, symbol p_{sM} , which is the gauge p. at the manometer or p. gauge; the p. p_s is related to p_{sM} as follows:

if the measuring line to the gauge is filled with liquid

$$p_s = p_{sM} + \rho \cdot g \cdot Z_{s,M}$$

where

$Z_{s,M}$ difference in elevations of p. gauge centralize and measuring point elevation at inlet cross-section of the pump,

ρ density of liquid in measuring line,

g gravitational constant;

if the measuring line to the gauge is filled with air:

$$p_s = p_{sM}$$

The same reasoning applies to the manometer reading at the outlet cross-section of the pump, symbol p_{dM} .

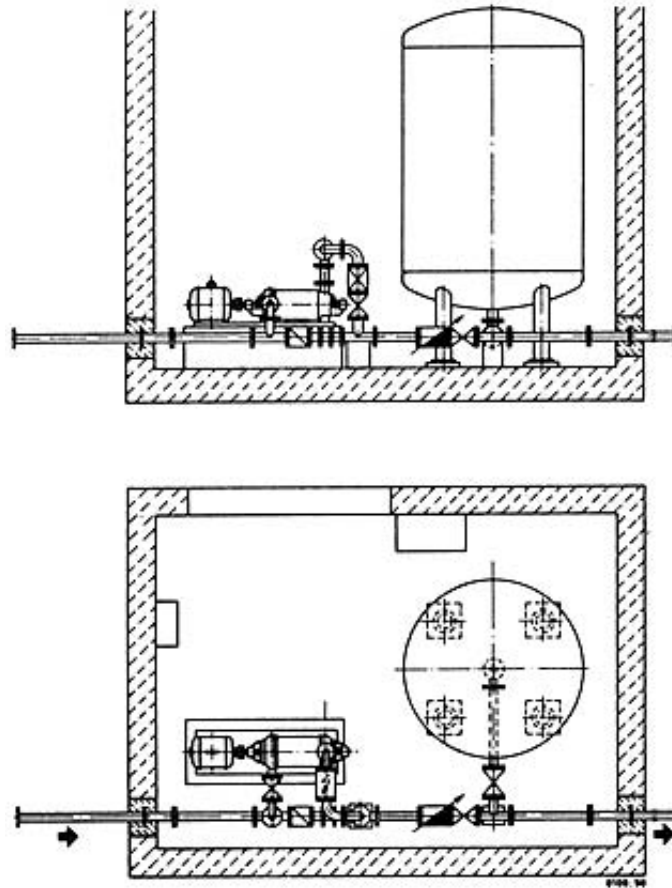
4. p. at inlet cross-section of the plant, symbol p_e , which is the gauge p. at the inlet cross-section A_e at elevation z_e i.e. if there is a liquid level present, the gauge p. on this liquid level;
5. p. at outlet cross-section of the plant, symbol p_a , which is the gauge p. at the outlet cross-section A_a at elevation z_a , i.e. if there is a liquid level present, the gauge p. on this liquid level;
6. barometric pressure at pump installation site, symbol p_b (atmospheric pressure);
7. vapour pressure (vaporization p.) of the fluid pumped, symbol p_D , which is the absolute p. at which the fluid pumped vaporizes at the temperature reigning at the inlet cross-section of the pump.

Pressure Booster Plant

Druckerhöhungsanlage
Installation de suppression

In cases where the normal mains pressure of an existing water supply installation is inadequate to ensure the supply to consumers located high up, e.g. in high rise apartment blocks or in the hills surrounding a city, it becomes necessary to boost the mains pressure. This is accomplished by means of p.b.p.'s, which consist of one or more pressure boosting pump arranged in a bypass line to the mains (see illustration). In order to prevent any pumping in a closed circuit, a nonreturn valve (valves and fittings) is arranged in the mains between the suction and discharge connections of the booster pumps. If the pressure upstream of the pump is subject to violent fluctuations,

a pressure reducing valve, preset to the minimum upstream pressure, is installed upstream of the pumps to protect the plant. The pumps are controlled by switching on and off in function of the pressure, via a pressure vessel of adequate capacity; in some cases the booster pump is also switched off if the temperature rise of the water becomes excessive (circulation through a by-pass) or if the flow velocity decreases in the mains. A pressure switch is often incorporated on the suction side, which switches off the p.b.p. when the pressure upstream drops below the minimum preset value. Parallel operation of two or more pumps (in most cases multistage pumps) may increase the economics of a p.b.p.; the number of pumps started up automatically is determined by the capacity required. The relatively high expenditure for control mechanisms justifies the application of microprocessors. Standardized compact p.b.p.'s incorporate all the components necessary for automatic operation and can be connected up with a minimum of installation work.



Pressure booster plant

Pressure Coefficient

Druckzahl

Coefficient de hauteur

see Characteristic Number

Pressure Drop

Druckabfall

Chute de pression

see Pressure Loss

Pressure Fluctuation

Druckschwankung
Fluctuation de pression

see Surge Pressure

Pressure Head

Druckhöhe
Hauter piézométrique

Terminology for

$$\frac{p}{\rho \cdot g}$$

where

p static pressure
 ρ density of pumped medium,
 g gravitational constant.

According to DIN 24260, p.h. is the energy of pressure being exerted by the fluid handled under the static pressure, p. as referred to its weight.

In BERNOULLI's equation (fluid dynamics) it stands alongside of dynamic head and geodetic altitude.

Pressure Loss

Druckverlust
Perte de pression

The p.l. p_v is the pressure differential arising as a result of wall friction and internal friction in piping runs, fittings, valves and fittings etc.

The generally valid formula for the p.l. of a flow in a straight length of pipe is:

$$P_v = \frac{\lambda \cdot U \cdot L}{4 A} \cdot \frac{\rho \cdot v^2}{2}$$

where

p_v pipe friction loss,
 λ pipe friction coefficient,
 U wetted periphery of section A through which the fluid flows,
 L length of pipe,
 ρ density of pumped medium,
 v flow velocity across a section A which is characteristic for the p.l.

Straight lengths of circular cross-section piping are defined by the following equation:

$$P_v = \frac{\lambda \cdot L}{D} \cdot \frac{\rho \cdot v^2}{2}$$

where

D bore of pipe.

The pipe friction coefficient λ varies with the state of flow of the medium and the internal surface finish of the pipeline through which the medium is flowing. The state of flow is determined by the REYNOLDS number (model laws):

$$Re = \frac{v \cdot D}{\nu}$$

for noncircular sections

$$Re = \frac{v \cdot 4 A}{\nu \cdot U}$$

where

ν kinematic viscosity.

λ can be calculated for smooth bore pipes (new rolled steel pipes):

In the region of laminar flow in the pipe ($Re < 2320$) the friction coefficient is:

$$\lambda = \frac{64}{Re}$$

In the region of turbulent flow in the pipe ($Re > 2320$) the test results can be represented by an empirical equation by ECK:

$$\lambda = \frac{0.309}{\left(\lg \frac{Re}{7}\right)^2}$$

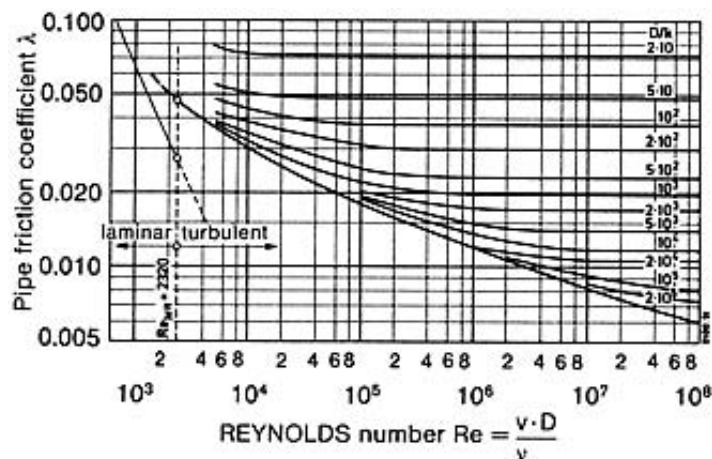


Fig. 1: Pipe friction coefficient λ in function of REYNOLDS number and of relative wall roughness D/k (see Large-scale diagram)

In the region of $2320 < Re < 108$ the deviations are less than 1%.

Fig. 1 shows, that λ is solely dependent on the parameter D/k at relatively high REYNOLDS numbers; k/D is the "relative roughness", obtained from the "absolute roughness" k and the pipe bore diameter D, where k is defined as the mean depth of the wall surface roughness (coarseness). According to MOODY the following applies:

$$\lambda = 0.0055 + \frac{0.15}{\sqrt[3]{\frac{D}{k}}}$$

Table 1 gives rough approximations of k.

Fig. 2 gives the losses of head H_v per 100 m of straight pipe run for practical usage. The head losses H_v in this context are calculated according to

$$H_v = \zeta \cdot \frac{v^2}{2g}$$

where

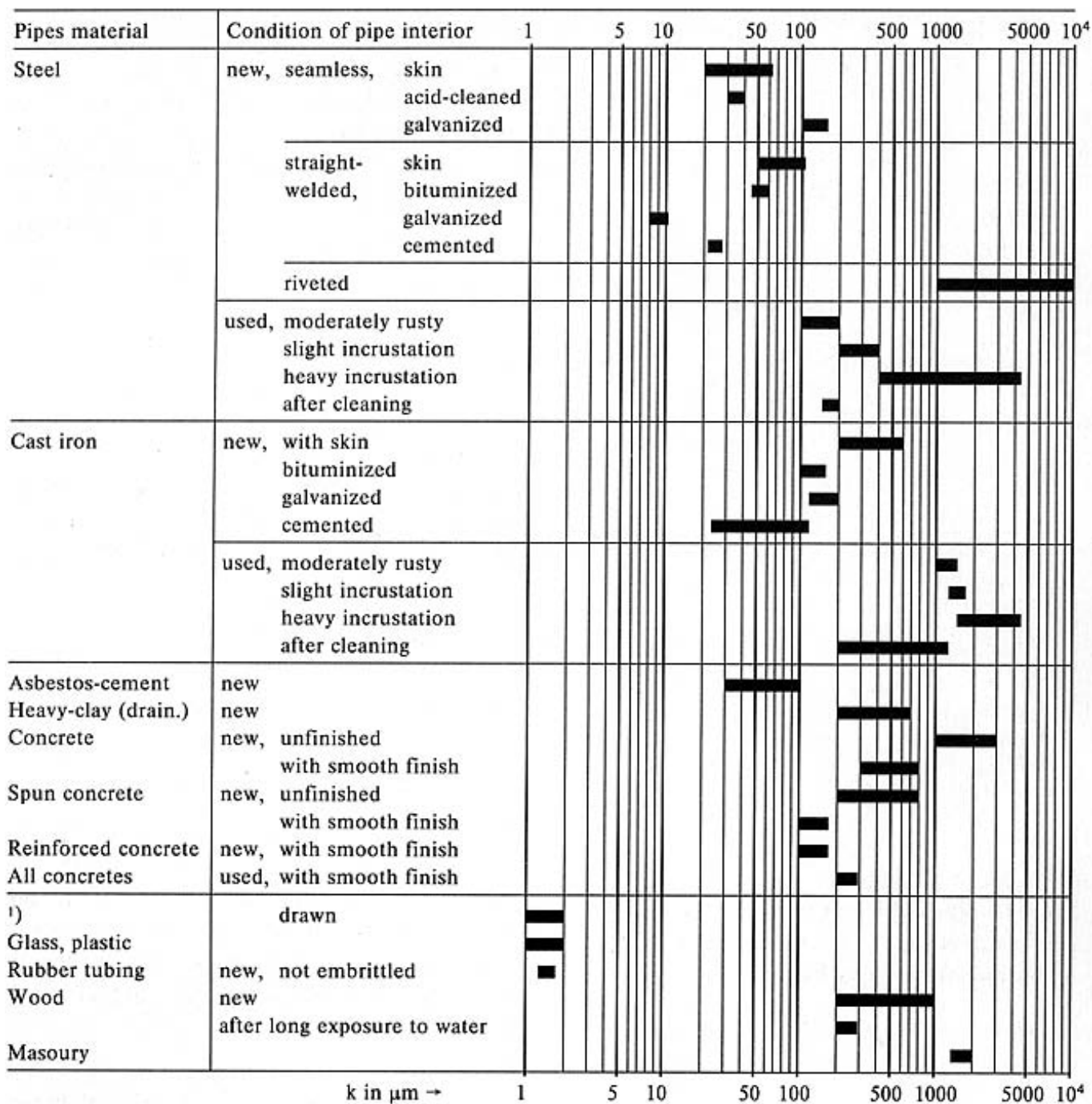
- η loss coefficient,
- v flow velocity,
- g gravitational constant.

The values in Fig. 2 apply to clean water at 20° C and to fluids of equal kinematic viscosity, assuming the pipng is completely filled, and consists of new cast iron pipes, with an internal bitumen coating ($k = 0.1$ mm). The head losses H_v of Fig. 2 should be multiplied by;

- 0.8 for new rolled steel pipes,
- 1.7 for pipes with incrustations (the reduced pipe cross-section due to the incrustations is the determining factor),
- 1.25 for old slightly rusty steel pipes.

In the case of pipes with very heavy incrustations, the actual head loss can only be determined by experiments. Deviations from the nominal diameter have a profound effect on the head loss, e.g. an actual bore of 0.95 times the nominal bore (i.e. only a slight bore reduction) pushes up the head loss H_v to 1.3 times the "as new" loss. New rubber-hoses and rubber-lined canvas hoses have H_v values approximately equal to those indicated in Fig. 2.

Table 1: Mean peak-to-valley heights k (absolute roughness)



¹⁾ Nonferrous metal, light alloy

Example of use of Fig. 2:

Assuming a capacity $Q = 140 \text{ m}^3/\text{h}$ and a new cast iron pipe of 150 mm inside diameter, we obtain: head loss $H_v \approx 3.25 \text{ m}/100 \text{ m}$ pipe length, flow velocity $v \approx 2.2 \text{ m/s}$.

Head losses in plastic pipes H_{vK} . The p.l.'s of polyvinyl chloride (PVC) and polyethylene "hard" and "soft" (drawn) plastic pipes are approximately equal. For the practical calculation of H_{vK} , the respective p.l.'s for cast iron pipes H_{vG} (Fig. 2) should be multiplied by the correction coefficients μ of Fig. 3, which are dependent on the flow velocity v . The p.l.'s evaluated in this way apply to water at a temperature of 10 °C. If the water temperature is other than 10 °C these p.l.'s must in addition be multiplied by a temperature factor Φ (Fig. 4). Thus

$$H_{vK} = H_{vG} \cdot \mu \cdot \Phi$$

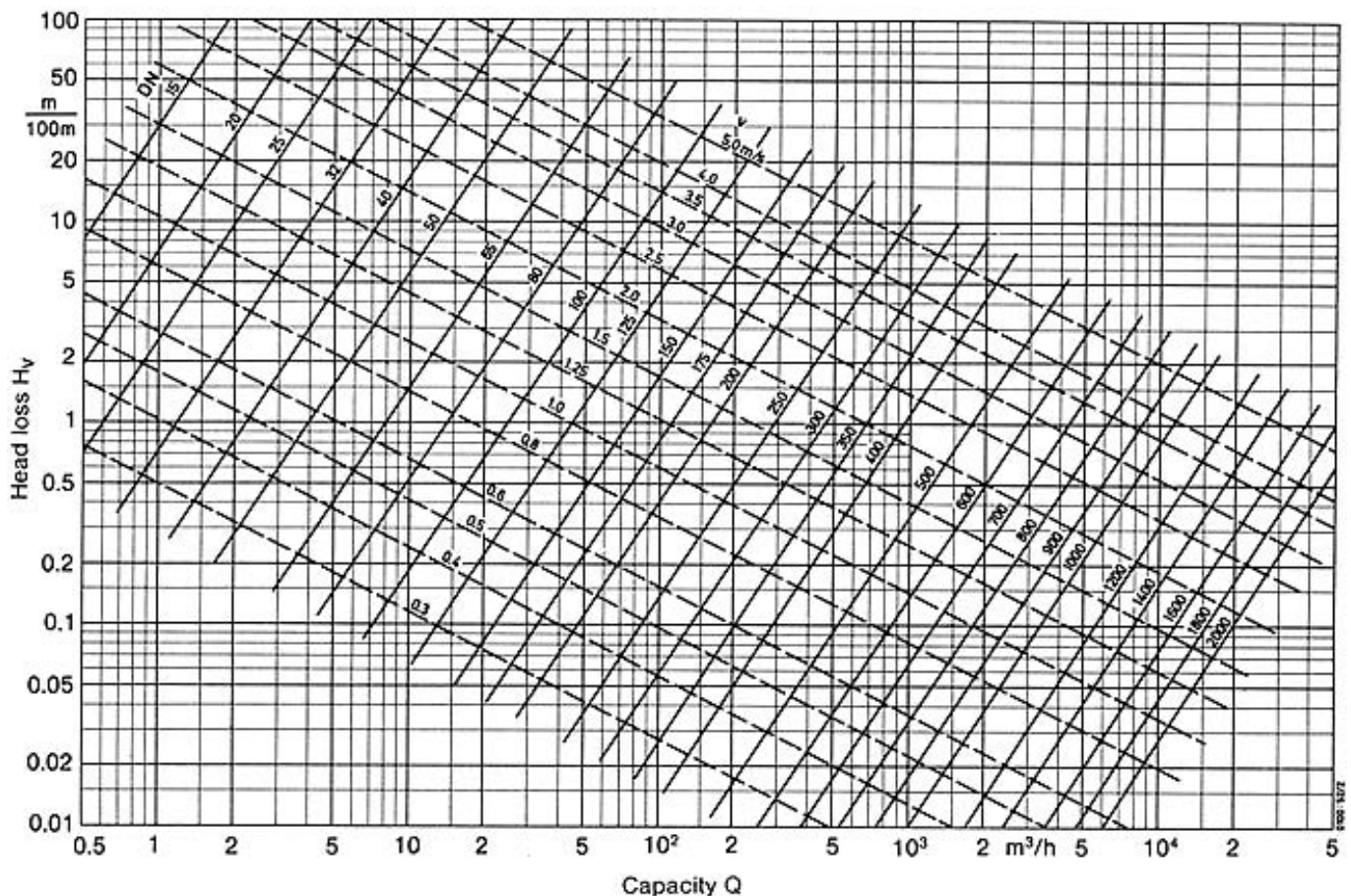


Fig. 2: Head losses in straight pipes (cast iron pipe, new condition) from DN 15 to 2000 mm and for capacities Q from 0.5 to 50000 m³/h (flow velocity v in m/s, water at 20 °C) (see Large-scale diagram)

where

H_{vK} p.l. in plastic pipes,

H_{vG} p.l. in cast iron pipes act. to Fig. 2,

μ correction coefficient acc. to Fig. 3,

Φ temperature factor acc. to Fig. 4.

Increments of 20 to 30% should be added for sewage or untreated water.

Head losses for viscous fluids in straight pipes. The head loss of a viscous fluid (subscript FI) can be ascertained for practical purposes with the aid of Fig. 5, after having obtained the headless for cold water (20 °C, $\nu = 10^{-6}$ m²/s) (subscript W) from Fig. 2:

$$H_{vFI} = \frac{\lambda_{FI} \cdot H_{vW}}{\lambda_W}$$

See [viscosity](#) for conversion of viscosity values. Example of usage of Fig. 5: Assuming a capacity Q = 100 m³/h and a new cast iron pipe of 250 mm inside diameter, a kinematic viscosity $\nu = 2 \cdot 10^{-4}$ m²/s, we obtain from Fig. 2:

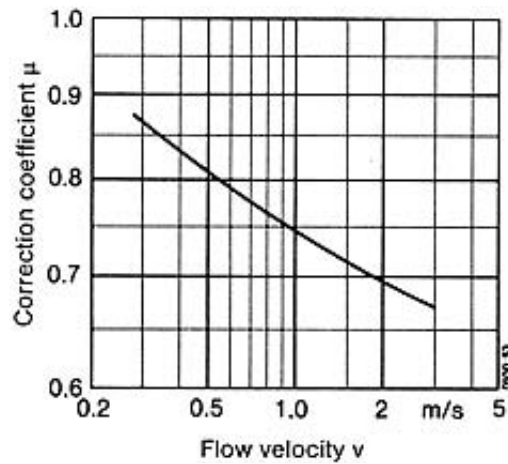


Fig. 3: Correction coefficient μ for conversion or pressure losses in a cast iron pipe at 20 °C water temperature to values in a plastic pipe at 10 °C water temperature; plotted in function of flow velocity v

$H_{vW} = 0.14 \text{ m/100 m}$, from Fig. 5: $\lambda_{FI} = 0.08$, $\lambda_W = 0.021$,

$$\text{thus } H_{vFI} = \frac{0.08 \cdot 0.14 \text{ m}}{0.021 \cdot 100 \text{ m}} = 0.53 \text{ m/100m.}$$

One quite common viscous fluid is cellulose (pulp pumping), the viscosity of which depends on the flow velocity, since the material in question is "non-NEWTONian". Figs. 6 a through 6 g offer reference values for the head loss H_v per 100 m length of straight pipe run plotted against capacity Q ($H_v = f(Q)$; nominal bore: 100, 125, 150, 200, 250, 300 and 350 mm) for conveying unbleached sulfite cellulose at 15 °C 26 °SR (grinding state, °SR-Schopper-Riegler degree of freeness) and with a pulp density (pulp pumping) of 1.5 to 7 % bone dry.

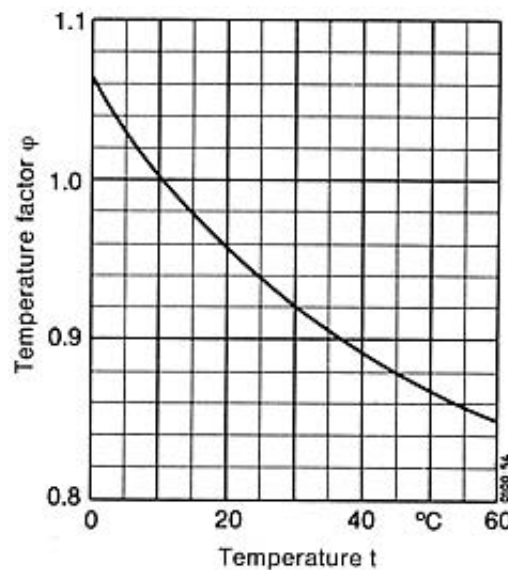


Fig. 4: Temperature factor ϕ for calculation of head losses in plastic pipes at water temperatures between 0 and 60 °C

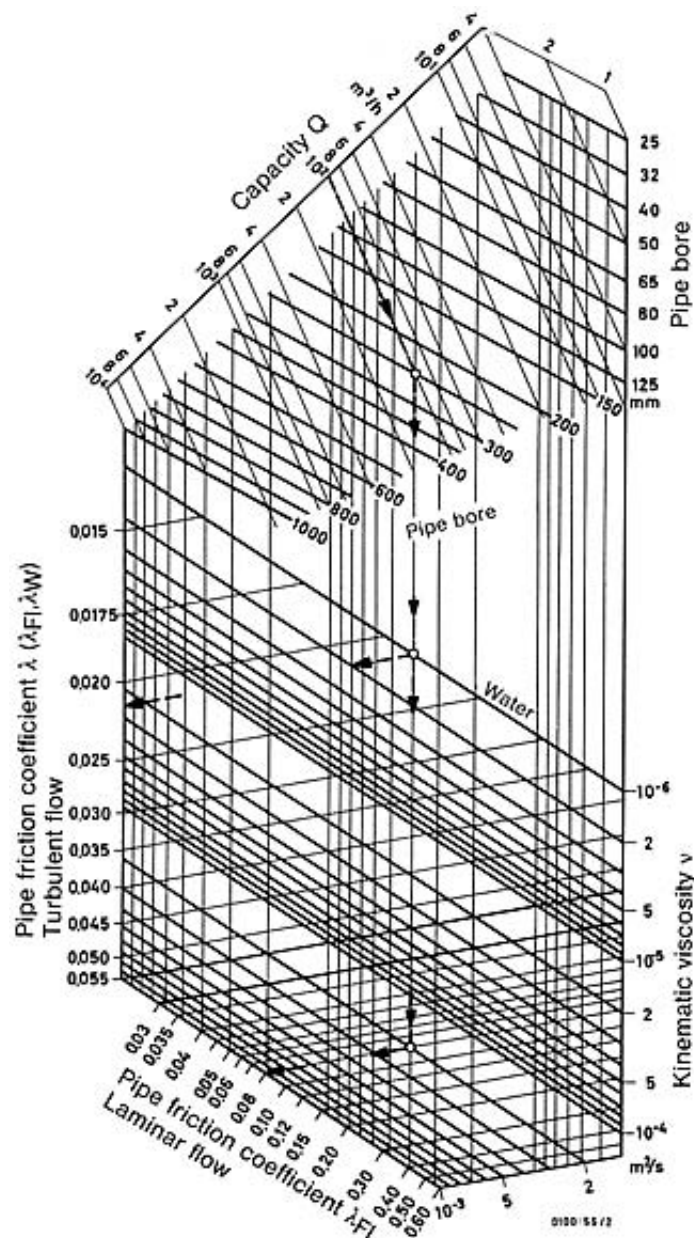


Fig. 5: Resistance coefficients λ for flow of viscous fluids in straight pipes (see [Large-scale diagram](#))

If the pump slurry concerned differs from that used for the purpose of plotting the curves of Fig. 6. then the values obtained from Fig. 6 should be multiplied by the following factors:

- K = 0.9 for bleached sulfite - sulfate cellulose, waste paper pulp,
- K = 1.0 for boiled (digested) wood pulp,
- K = 1.4 for white and brown raw wood pulp.

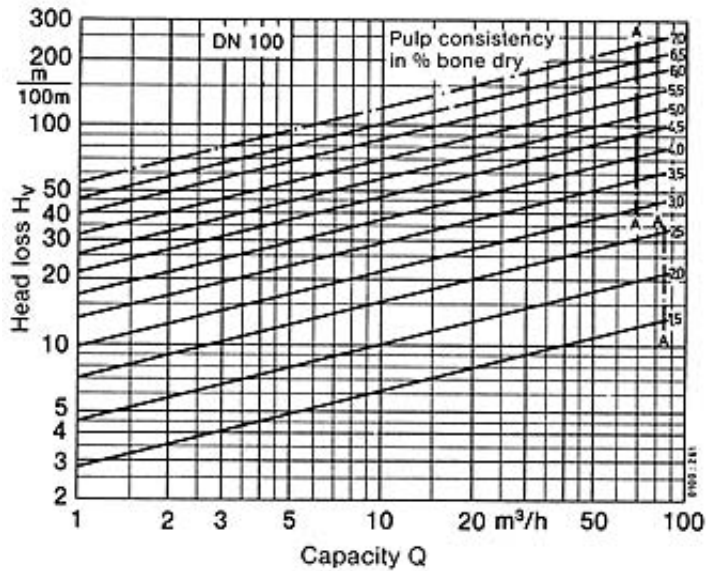


Fig. 6a

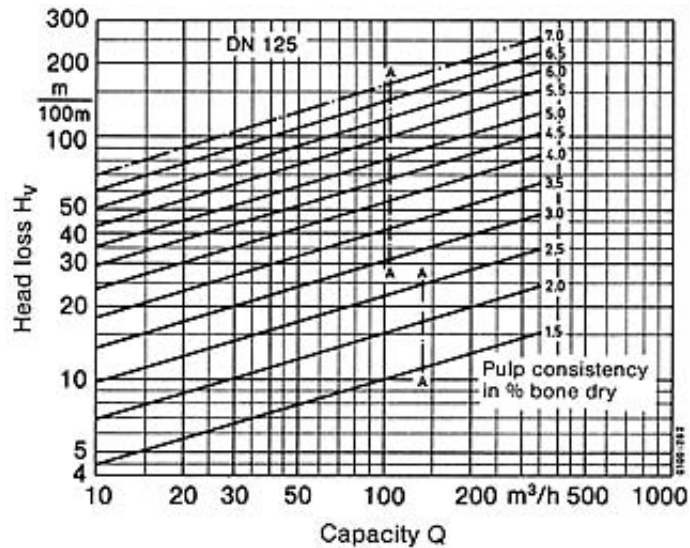


Fig. 6b

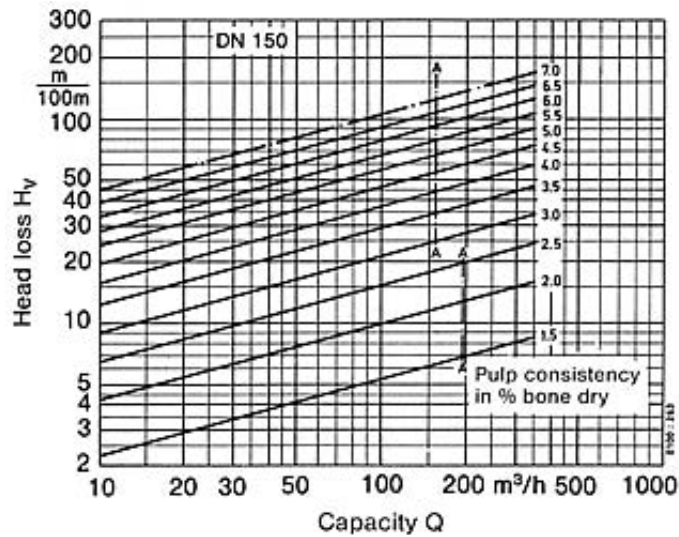


Fig. 6c

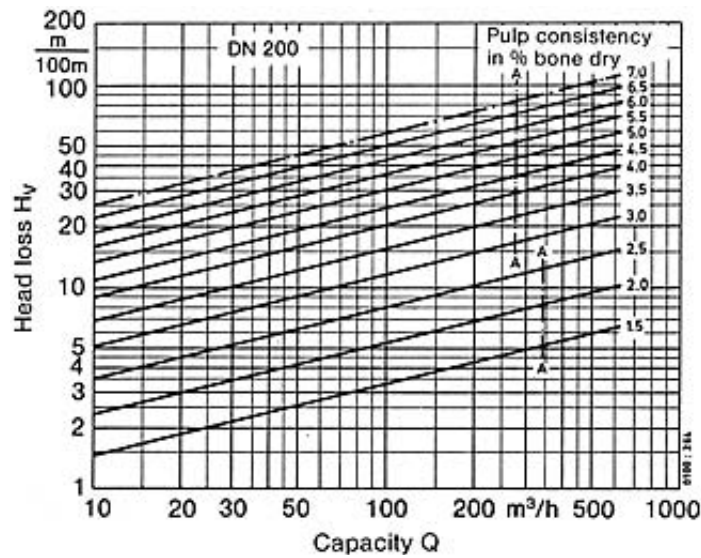


Fig. 6d

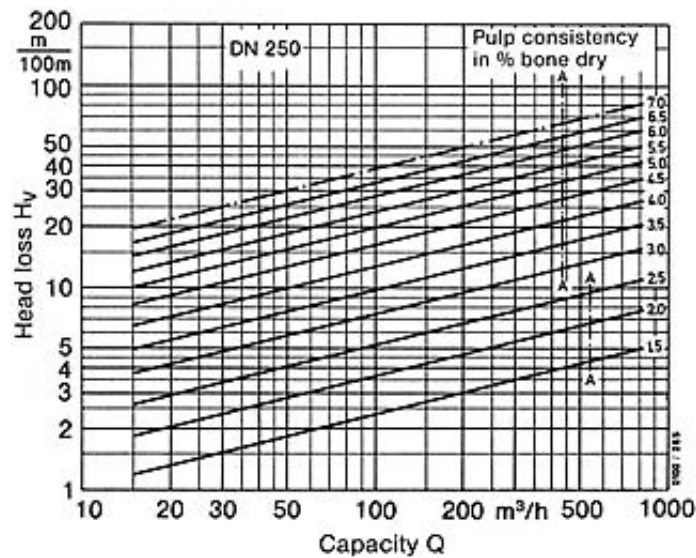


Fig. 6e

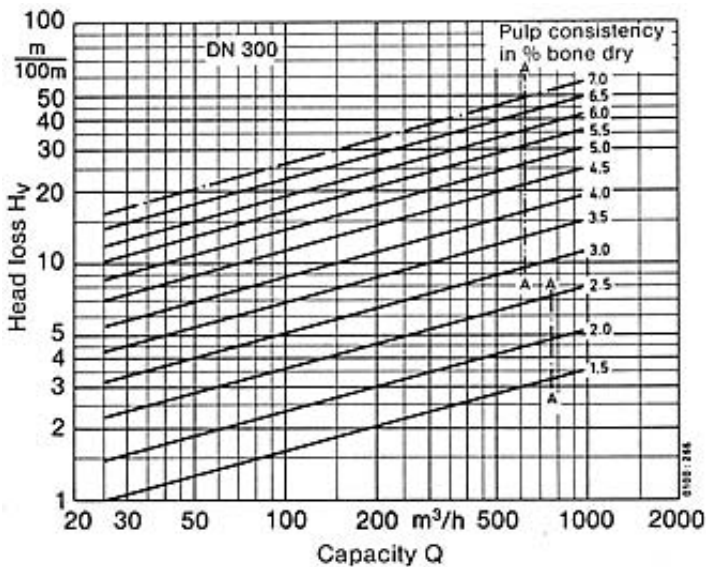


Fig. 6f

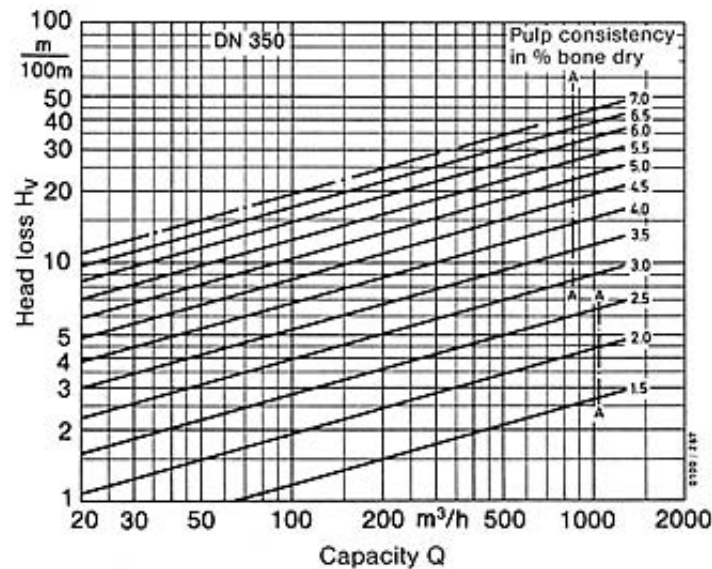


Fig. 6g

Figs. 6a - g shows a plot of the head losses H_v for conveying sulphite cellulose of various pulp densities at a temperature of 15 °C and a grinding grade of 26 °SR (pipe diameters DN 100 to 350) A-A = maximum velocity (2.44 or 3.05 m/s) in the delivery pipe for economical operation (see Large-scale diagrams from 6a to 6g)

Furthermore, the head loss obtained from Fig. 6, and if necessary corrected by one of the factors listed above, should be corrected additionally if the pulp slurry concerned is at a temperature higher than 15 °C. In this case, 1% of the head loss value which applies to 15 °C should be deducted for every 2° of temperature difference. In the case of plastic pipes, the H_vK value is obtained by multiplying the H_v value for steel pipes by 0.9.

The head loss value is reduced even further if fillers such as kaolin (China clay) are contained in the pulp slurry concerned. For an 18 % kaolin content, the head loss value will decrease by 12 %, and for a 26.5% kaolin content, it will decrease by 16 %.

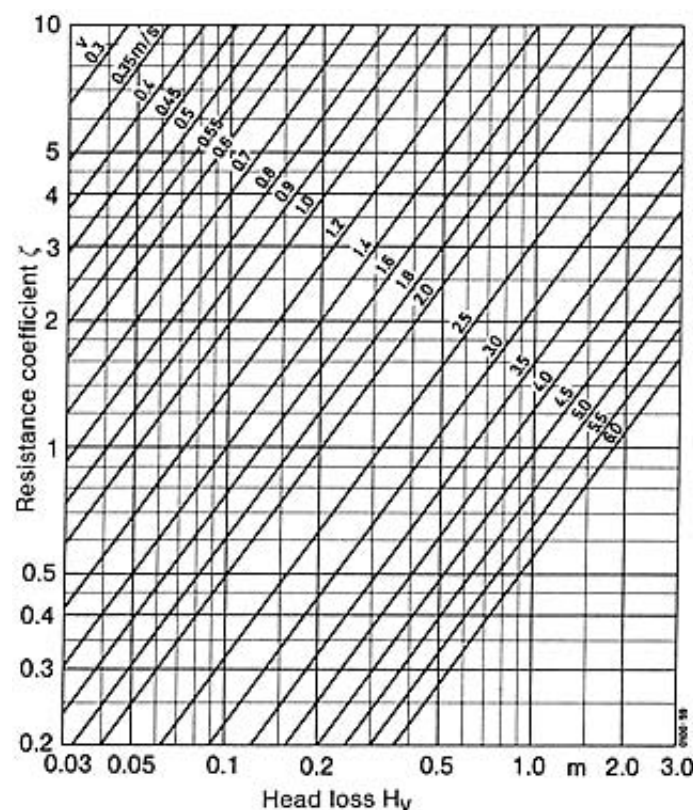


Fig. 7: Determination of head losses H_v in valves and fittings; flow velocity v relating to the actual cross-sectional area through which the fluid flows (see Large-scale diagram)

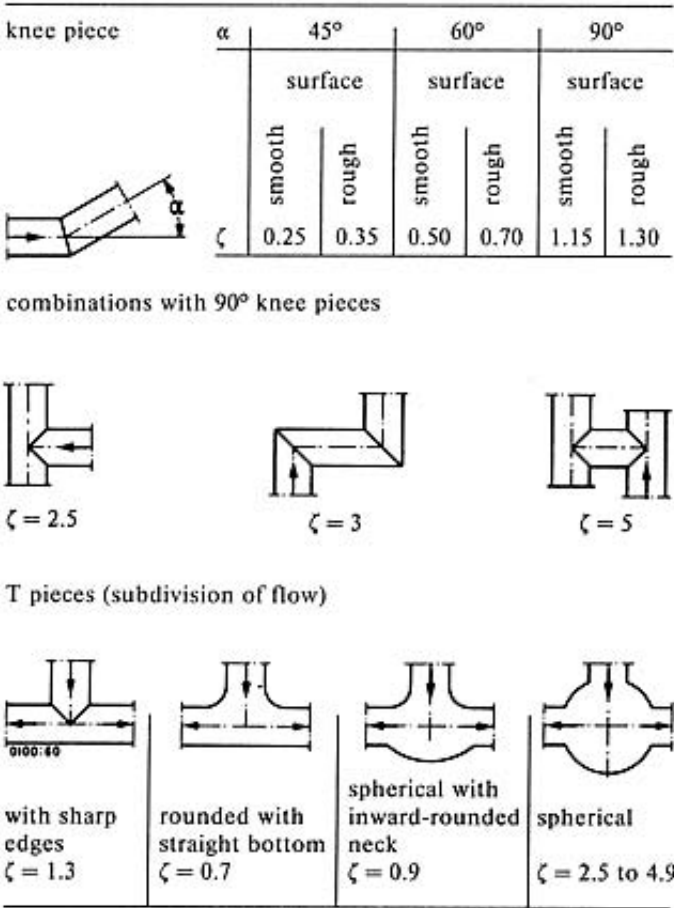


Fig. 8: Illustration of fittings with related loss coefficients ζ

For p.l.'s in valves and fittings the following equation applies:

$$P_v = \zeta \cdot \frac{\rho \cdot v^2}{2}$$

where

- ζ loss coefficient,
- ρ density of pumped medium,
- v flow velocity across a section A which is characteristics for the p.l.

Tables 2 to 4 and Figs. 7 to 10 give details of individual loss coefficients ζ and head losses H_v in valves and fittings for operation with water.

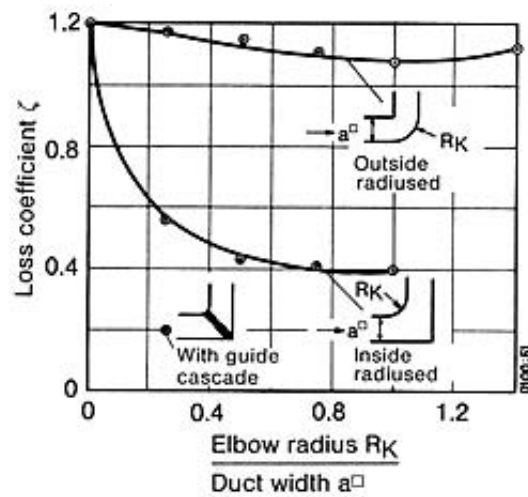


Fig. 9: Influence of rounding off of concave and convex side on the loss coefficient of elbows with quadratic cross-section

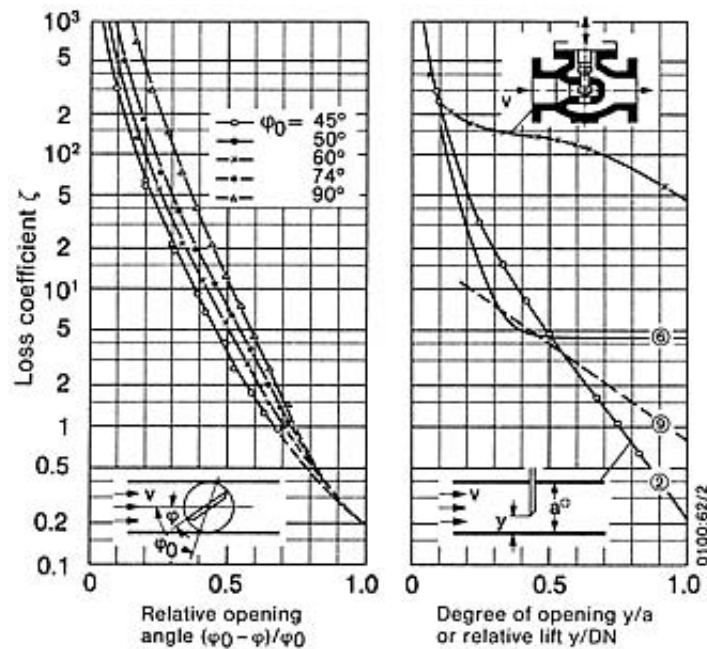


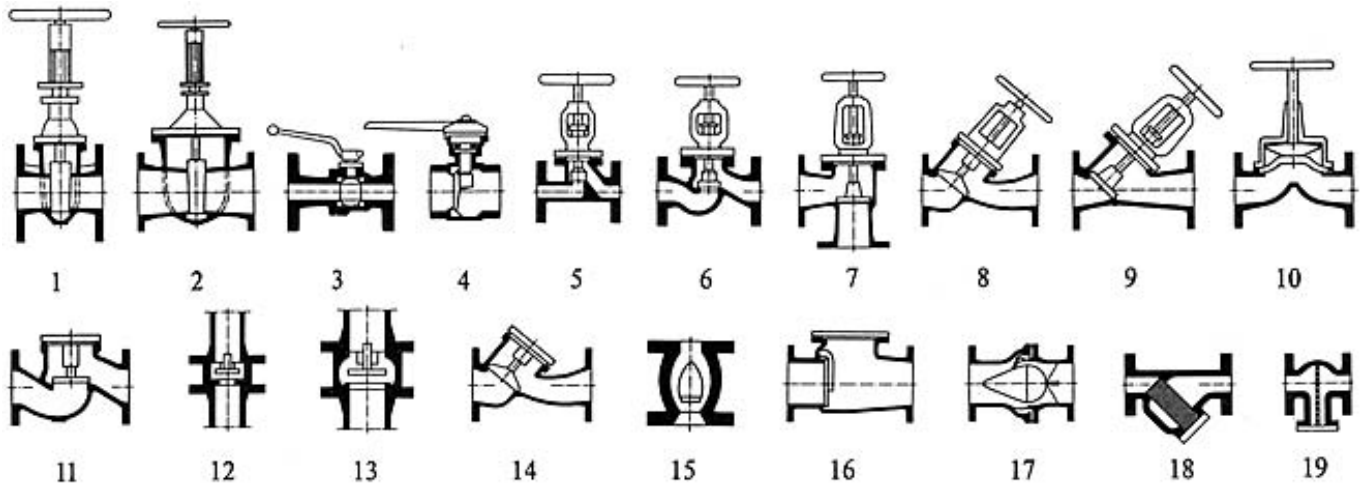
Fig. 10: Loss coefficients of butterfly valves, globe and gate valves in function of opening angle or degree of opening (position numbers according to Tabled design configuration) (see Large-scale diagram)

	Type of valve/fitting	Des. conf. ^{a)}	Loss coefficient ζ for DN =																	Remark				
			15	20	25	32	40	50	65	80	100	125	150	200	250	300	400	500	600		800	1000		
Shut-off valves	flat slide gate valves ($d_E = DN$)	min max	1	0.1 0.65	←	0.6	0.55	0.5	0.5	0.45	0.4	0.35	0.3	←							→	0.1 0.3	for $d_E < DN$ cf. footnote ¹⁾	
	round-body slide gate valves ($d_E = DN$)	min max	2						0.25 0.32	0.24 0.31	0.23 0.30	0.22 0.28	0.21 0.26	0.19 0.25	0.18 0.23	0.17 0.22	0.16 0.20	0.15 0.19	0.13 0.18	0.12 0.16	0.11 0.15	0.11 0.14		
	cocks ($d_E = DN$)	min max	3	0.10 0.15	0.10 ←	0.09	0.09	0.08	0.08	0.07	0.07	0.06	0.05	0.05	0.04	0.03	0.03 →	0.02 0.15						for $d_E < DN$ $\zeta = 0.4$ to 1.1
	butterfly valves PN ≥ 2.5 PN ≤ 40		4					0.90	0.76	0.60	0.50	0.42	0.36	0.30 1.50	0.25 1.20	0.20 1.00	0.16 0.92	0.13 0.83	0.10 0.76	0.08 0.71	0.06 0.67	0.05 0.63		
	globe valves, forged	min max	5			6.0 6.8	←	←	←	6.0 6.8														
	globe valves, cast	min max	6	3.0 6.0	←													3.0 6.0						$\zeta = 2$ to 3 possible for optimized valve
	angle valves	min max	7	2.0 3.1	←														2.0 6.6					
	slanted-seat valves	min max	8	1.5 2.6	←														1.5 2.6					
	full-bore valves	min max	9	0.6 1.6	←														0.6 1.6					
	diaphragm valves	min max	10	0.8 2.2	←										0.8 2.2									
Nonreturn valves	nonreturn valves, straight-seat	min max	11	3.0 6.0	←										3.0 6.0									
	nonreturn valves, axial	min max	12	3.2 3.4	←	3.4	3.5	3.6	3.8	3.2 4.2	3.7 5.0	5.0 6.4	7.3 8.2											
	nonreturn valves, axially expanded	min max	13										4.3 4.6	←	←	←	4.3 4.6							
	nonreturn valves, inclined-seat	min max	14	2.5 3.0	←	2.4	2.2	2.1	2.0	1.9	1.7	1.6	1.5	←				1.5 3.0						
	foot valves	min max	15						1.0 3.0	0.9 3.0	0.8 3.0	0.7 3.0	0.6 3.0	0.5 3.0	0.4 3.0	0.4 3.0	0.4 3.0	(7.0)	(6.1)	(5.5)	(4.5)	(4.0)	() in groups	
	swing check valves	min max	16	0.5 2.4	←	2.3	2.3	2.2	2.1	0.4 2.0	1.9	1.8	1.8	1.7	1.6	1.5	1.5	0.4 1.4	0.3 1.3	1.2	1.2	1.1	0.3 1.0	swing-tape valves with- out levers and weights ²⁾
	hydrostops $v = 4$ m/s $v = 3$ m/s $v = 2$ m/s		17						0.9 1.8 5.0				3.0 4.0 6.0		3.0 4.5 8.0	2.5 4.0 7.5	2.5 4.0 6.5	1.2 1.8 6.0	2.2 3.4 7.0					
	filters		18						2.8	←														
screens		19						1.0	←									1.0						in clean condition

1) If the narrowest shut-off diameter d_E is smaller than the nominal diameter DN, the loss coefficient ζ must be increased by $(DN/d_E)^x$, with x = 5 to 6

2) In the case of partial opening, i.e. low flow velocities, the loss coefficients increase

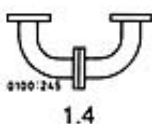
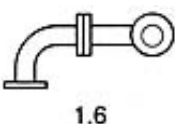
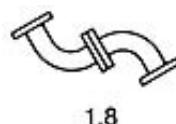






3) Design configurations



Design configurations (1 to 10) see Table 2

The minimum and maximum values listed in Table 2 include figures taken from the most pertinent trade literature and apply to fully open valves and fittings under uniform conditions of flow. The losses attributable to flow disturbances in a length of pipe equalling ca. 12 X DN downstream of the valve or fitting are also included in those values (cf. VDI/VDE guideline 2173). Nonetheless, the actual values are subject to wide variance, depending on the conditions of inflow and outflow, the model in question, and the design objectives.

Table 3: Loss coefficients for fittings

Elbows:					
cast elbows 90°, R = D + 100 mm,					
all nominal diameters $\zeta \approx 0.5$					
pipe bends 90°, R = 2 to 4 x D					
nominal diameter DN	50	100	200	300	500
ζ	≈ 0.26	≈ 0.23	≈ 0.21	≈ 0.19	≈ 0.18
if the deflection angle amounts					
	60°	45°	30°	15°	
to the above ζ values should be multiplied	0.85	0.7	0.45	0.3	
Knee pieces:					
deflection angle	90°	60°	45°	30°	15°
ζ	≈ 1.3	≈ 0.7	≈ 0.35	≈ 0.2	≈ 0.1
Combinations of elbows and pipe bends:					
The ζ value of the single 90° elbow should not be doubled, but only be multiplied by the factors indicated to obtain the pressure loss of the combination elbows illustrated:					
  					
Expansion joints:					
bellows expansion joint with / without guide pipe $\zeta \approx 0.3/2.0$					
smooth bore pipe harp bend $\zeta \approx 0.6$ to 0.8					
creased pipe harp bend $\zeta \approx 1.3$ to 1.6					
corrugated pipe harp bend $\zeta \approx 3.2$ to 4					
Inlet pipe fittings:					
     					
inlet edge	$\zeta \approx 0.5$	3	0.55	0.20	0.05
sharp chamfered	$\zeta \approx 0.25$				
for $\delta = 75^\circ$ $\zeta = 0.6$ 60° $\zeta = 0.7$ 45° $\zeta = 0.8$					
Discharge pieces:					

- $\zeta \approx 1$ downstream of an adequate length of straight pipe with an approximately uniform velocity distribution in the outlet cross-section.
 $\zeta \approx 2$ in the case of very unequal velocity distribution, e.g. immediately downstream of an elbow, a valve etc.

Loss coefficients of flow meters:

short venturi tube $\alpha = 30^\circ$ standard orifice



ζ is related to the velocity v at diameter D.

diameter ratio d/D	=	0.30	0.40	0.50	0.60	0.70	0.80
aperture ratio $m = (d/D)^2$	=	0.09	0.16	0.25	0.36	0.49	0.64
short venturi tube	$\zeta \approx$	21	6	2	0.7	0.3	0.2
standard orifice	$\zeta \approx$	300	85	30	12	4.5	2

water meters (volumetric meters) $\zeta \approx 10$.

In the case of domestic water meters a max. pressure drop of 1 bar is prescribed for the rated load, and in practice the actual pressure loss is seldom below this figure.

Branch pieces (branch of equal bore):

Note:

the resistance coefficients ζ_a for the diverted flow Q_a or ζ_d respectively for the main flow $Q_d = Q - Q_a$ relate to the velocity of the total flow Q in the branch. On the basis of that definition ζ or ζ_d may take on negative values in which case they are indicative of pressure **gain** instead of pressure **loss**. Not to be confused with reversible pressure **changes** according to BERNOULLI's equation (cf. annotation to Table 4).

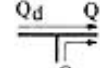
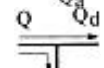
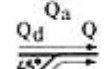
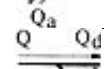
$Q_a/Q =$		0.2	0.4	0.6	0.8	1.0
	$\zeta_a \approx$	-0.4	0.08	0.47	0.72	0.91
	$\zeta_d \approx$	0.17	0.3	0.41	0.51	-
	$\zeta_a \approx$	0.88	0.89	0.95	1.10	1.28
	$\zeta_d \approx$	-0.08	-0.05	0.07	0.21	-
	$\zeta_a \approx$	-0.38	0	0.22	0.37	0.37
	$\zeta_d \approx$	0.17	0.19	0.09	-0.17	-
	$\zeta_a \approx$	0.68	0.50	0.38	0.35	0.48
	$\zeta_d \approx$	-0.06	-0.04	0.07	0.20	-

Table 4: Pressure change coefficients in transition piece Expansion Reduction

Expansion				Reduction				
	form		d/D =	0.5	0.6	0.7	0.8	0.9
	I		ζ	≈ 0.56	0.41	0.26	0.13	0.04
	II for	$\alpha = 8^\circ$	ζ	≈ 0.07	0.05	0.03	0.02	0.01
		$\alpha = 15^\circ$	ζ	≈ 0.15	0.11	0.07	0.03	0.01
		$\alpha = 20^\circ$	ζ	≈ 0.23	0.17	0.11	0.05	0.02
	III		ζ	≈ 4.80	2.01	0.88	0.34	0.11
	IV	for $20^\circ < \alpha < 40^\circ$	ζ	≈ 0.21	0.10	0.05	0.02	0.01

Note:
in the case of branch pieces as per Table 3 and transition pieces as per Table 4, differentiation is made between irreversible pressure loss (= pressure reduction)

$$p_v = \zeta \cdot \frac{\rho \cdot v_1^2}{2}$$

on the one hand and reversible pressure changes involving frictionless flow as per BERNOULLI's equation ([fluid dynamics](#))

$$p_2 - p_1 = \frac{\rho}{2} (v_1^2 - v_2^2)$$

on the other. In the case of accelerated flow e.g. through a pipe reduction, $p_2 - p_1$ is negative. Conversely, it is positive in pipe expansions. By contrast the pressure loss ascertained by way of the loss coefficients ζ are always negatives if the overall pressure change is calculated as the arithmetic sum of p_v and $p_2 - p_1$.

In the case of water transport through valves and fittings, the loss coefficient, ζ , are occasionally neglected in favour of the so-called k_v -value:

$$p_v = \left(\frac{Q}{k_v} \right)^2 \cdot \frac{\rho}{1000}$$

where

Q volume flow in m^3/h ,

ρ density of water in kg/m^3 (effective temperature vapour pressure, Table 1).

p_v p.l in bar.

The k_v -value (m^3/h) represents the volume flow of cold water ($\rho = 1000 \text{ kg}/\text{m}^3$) at $p_v = 1 \text{ bar}$ through a valve or fitting; it therefore gives the relationship between the p.1. p_v in bar and the volume flow Q in m^3/h .

Conversation:

$$\zeta \approx 16 \cdot \frac{d^4}{k_v^2}$$

where

d reference diameter (nominal diameter) of the valve or fitting in cm.

Pressure Measurement

Druckmessung

Mesure de pression

see Measuring Technique

Pressure Ratings

Druckstufen

Étages de pression

P.r. are the nominal pressure ratings graduated on the lines of the preferred number series in accordance with DIN 2401; they represent the basis for the elaboration of standards for piping components.

Pressure Vessel

Druckbehälter, Druckkessel

Réservoir sous pression

P.v. 's for the attenuation of pressure surges (surge pressure) are energy stores, the function of which is to prolong the switching interval. They should be sized and equipped with valves and fittings, compressed air supply and instrumentation on the basis of a pressure surge calculation.

P.v. 's for the automatic pressure switching of water supply plants (pressure booster plant) are often arranged vertically (Fig. 1) and only in special circumstances horizontally. The size of vessel is determined by the number of permissible hourly switchings Z on and off, in view of motor overheating and contact wear. The value of Z should not exceed 12 to 15. These limit values are however not absolutely binding. The general rule is: the bigger the motor output, the lower the number of switchings per hour should be. The lowest water level at the switching on pressure P'_e must be selected high enough to flood the compressed air check valve located in the connection nozzle of the vessel. This compressed air check valve is intended to prevent the undesirable ingress of compressed air into the discharge line when the pump is stopped. Therefore the volume V of the vessel must be selected 25 % larger than the effective vessel volume J (Fig. 1). If horizontal vessels are used, the depth of cover must be checked; if necessary, the connection must be placed at a lower level (dome). This safety margin is included in the formula given below for the sizing of the vessel.

$$V = K \cdot 0.312 \cdot \frac{Q_m}{Z} \cdot \frac{p'_a + p_b}{p'_a - p'_e}$$

where

V volume of vessel in m³,

Q_m mean capacity in m³/h = (Q'_e + Q'_a)/2,

p_b barometric pressure in bar,

p'_a switching off pressure in bar,

p'_e switching on pressure in bar,

Z switching frequency in h⁻¹,

Q'_e capacity at switching on pressure in m³/h,

Q'_a capacity at switching off pressure in m³/h,

K correction factor as per Fig. 2, if Q'_a / Q'_e < 0.5.

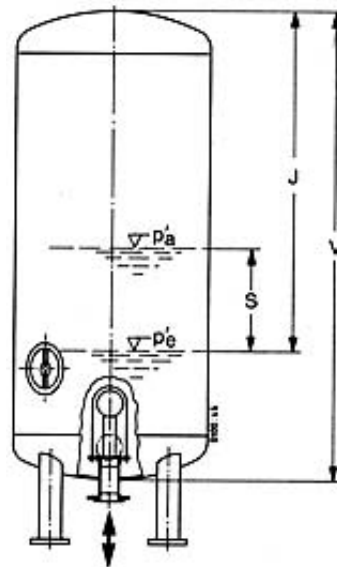


Fig. 1: Pressure vessel for the automatic pressure switching of water supply plants

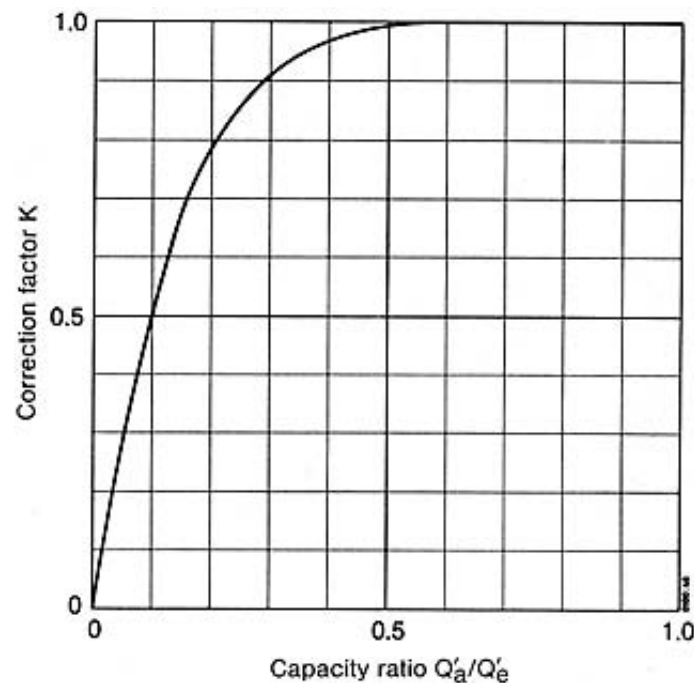


Fig. 2: Correction factor K

An example of the calculation of required vessel volume V is given below:

$$\begin{aligned}
 p_b &= 1.0 \text{ bar,} \\
 p'_a &= 6.5 \text{ bar,} \\
 p'_e &= 4.5 \text{ bar,} \\
 Q'_a &= 80 \text{ m}^3/\text{h,} \\
 Q'_e &= 148 \text{ m}^3/\text{h,} \\
 Q_m &= 114 \text{ m}^3/\text{h,} \\
 Q'_a / Q'_e &= 0.54, \\
 K &= 1.0, \\
 Z &= 13,
 \end{aligned}$$

$$V = 1.0 \cdot 0.312 \cdot \frac{114}{13} \cdot \frac{6.5 + 1.0}{6.5 - 4.5} = 10.26 \text{ m}^3.$$

A vessel of $V = 10 \text{ m}^3$ should be selected.

The proportion of useful water volume S in m^3/h in relation to the total volume V depends solely on the switching on and switching off-pressures and can be calculated from

$$S = V \cdot 0.8 \cdot \frac{p'_a - p'_e}{p'_a + p_b}$$

The container size can be decreased by meeting the following requirements:

1. With set-ups having more than one of the same pump, cyclical exchanges can be used to increase the start frequency Z .
2. Through a delayed shut-off the start frequency can be set independently. Since the pump is run here for a certain amount of time at shut-off, it is necessary to examine the conditions carefully.
3. The capacity Q'_a at switch off pressure and therefore the mean capacity Q_m can be decreased through the use of an orifice in the connecting line. This also causes the correction factor K to become more suitable.

Especially with smaller units, balloon air cushions are provided for. In this case neither a compressed air check valve nor compressor is required. In that case the mentioned enlargement of the effective volume J by 25% is not required. The number of pumps in a pressure booster plant has no bearing on the calculation of the vessel volume; if there are several pumps of different pump output involved, the mean capacity Q_m of the largest pump should be entered in the equation. In the case of installations in which several pumps are controlled in function of the rate of flow, and only the base load pump is switched on and off in function of the pressure, the vessel size V should be calculated in relation to this base load pump.

A subdivision of the calculated vessel volume V among several p.v.'s is desirable if such smaller vessels can be accommodated more easily in the available space, and the plant costs are thereby reduced.

As a proportion of the air content of the p.v. is gradually absorbed by the water under pressure, the compressed air in the vessel must be topped up from time to time, usually by means of a compressor.

The size of this compressor is governed by the suction capacity Q_k . The calculation of the compressor must start with the filling time T for the complete topping up of the air volume in the p.v. This filling time should preferably not exceed four hours; only in the case of very large installations and high service pressures is a filling time of up to ten hours considered admissible.

$$Q_k = \frac{0.6 \cdot V}{T} \cdot \frac{p'_a}{p_b}$$

where

- Q_k suction capacity in m^3/h ,
 V vessel volume in m^3 ,
 T filling time in h,
 Q'_a switching off pressure in bar,
 p_b barometric pressure in bar.

The above formula is based on the assumption that only $\frac{2}{3}$ approx. of the vessel volume V (corresponding to the water level at switch off pressure P'_a) have to be filled with compressed air.

The operating pressure of the compressor must be selected at least equal to the max. switching off pressure P'_a of the pump. The relief valve on the compressor must be preset so as not to exceed the max. permissible operating pressure of the vessel.

In accordance with the accident prevention regulations for p.v.'s (German Gas and Waterworks Professional Association in D-4000 Düsseldorf, Achenbachstraße 20) it is not mandatory to fit a relief valve on p.v.'s for centrifugal pumps if the throttling curves (characteristic curve) of the pumps are such that they do not exceed 1.1

times the max. permissible operating pressure for the vessel, and steps are taken to prevent overspending of the pumps.

P.v.'s are manufactured in welded, cast, riveted and sometimes in strip-wound construction (the last-named being extensively used in the chemical industry for very high pressures and temperatures). The materials used are steel plate (boiler plate), non-ferrous metal plate, cast steel and plastic. P.v.'s which are used in large numbers are standardized according to design and operating data. The principal standards, recommendations and rules which apply to them are:

American Petroleum Institute: API 610,

American Society of Mechanical Engineers:
ASME Boiler and Pressure Vessel Code, Section I-X,

German Working Committee on Pressure Vessels: AD notices,

Federal Minister for Economy: Protection of Labour Act (Federal Bulletin 4/1980) with among others. Steam Boiler and Pressure Vessel Act,

DIN 3171, 4661, 4664 and 4810,

German Association of Gas and Water Experts:
DVGW worksheet W 314,

Technical Regulations for Steam Boilers: TRDguidelines,

Association of Technical Supervision Societies:
VdTÜV,

Recommendations of Shipbuilding Classification Societies

and others.

Pressure Wave

Druckwelle

Onde de pression

see Surge Pressure

Primary Circulating Pump

Primärumschlepppumpe

Pompe primaire de circulation

see Reactor Pump

Priming Device

Ansaughilfe

Aide au amorçage

see Deep Well Suction Device, Priming Stage

Priming Funnel

Fülltrichter
Entonnoire de remplissage

see [Filling-up of Centrifugal Pumps](#)

Priming Stage

Ansaugstufe
Pompe d'amorçage

P.s., also called self-priming stage or air extraction stage is the device within the pump itself designed to vent the suction pipe of the pump ([venting](#)). The [water ring pump](#) or [side channel pump](#) incorporates a p.s., see also under self-priming pump.

Probe

Sonde
Sonde

see [Measuring Technique](#)

Process Pump

Prozeßpumpe
Pompe process

The p.p. is a [centrifugal pump](#) for use in a chemical process, particularly in oil refineries and the petrochemical industry ([refinery pump](#)). The p.p. differs from the conventional [chemical pump](#) in that it is suitable not only for corrosive chemical fluids, but also for extremely high system pressures and temperatures. In general, p.p.'s comply with the specifications of the American Petroleum Institute (API).

Process Type Design

Prozeßbauweise
Construction à type process

P.t.d. describes a type of [centrifugal pump](#) construction meeting the requirements of refinery technology for rapid dismantling and reassembly, used very often in [process pumps](#). In this context, rapid dismantling and reassembly of the pumps for the purposes of inspection and renewal of the internal components involves: after disconnection of the motor coupling and unscrewing of a connection flange, the entire pump rotor and bearing assembly ([plain bearing](#), [anti-friction bearing](#)) together with the [shaft seal](#) can be pulled out of the [pump casing](#) as one unit, without having to remove the [pump casing](#) from the [piping](#) (Fig. 1). This type of construction, which greatly facilitates maintenance, has also been adopted (under the name of p.t.d.) for other applications than refineries, as it simplifies the task of the operating and maintenance staff.

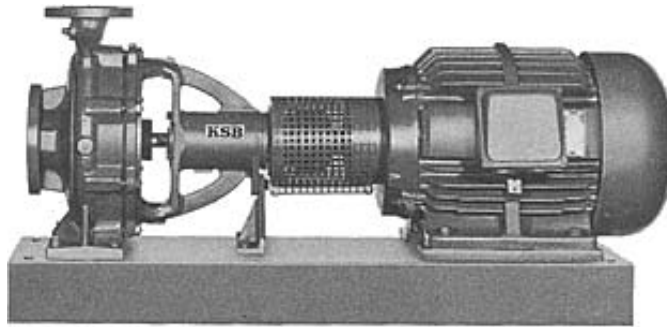


Fig. 1: Low pressure centrifugal pump of process type design with a non-spacer-type flexible coupling

If another shaft coupling with intermediate sleeve (Fig. 2) is used at the same time, the motor may remain on the pump foundation when extracting the rotating assembly.

The electrical leads also remain in situ, because the lateral removal of the coupling spacer gives sufficient axial clearance for the dismantling of the pump rotor; these spacer-type couplings save further erection time.

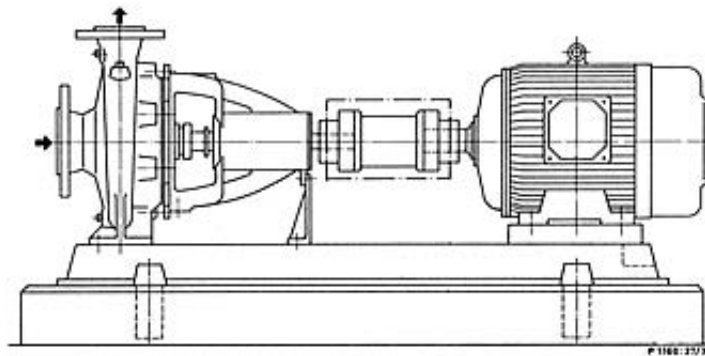


Fig. 2: Low pressure centrifugal pump of process type design with spacer-type flexible coupling

Profile

Profil
Profil

see Flow Profile

Profil Seal

Profildichtung
Joint profilé

see Seals

Propeller

Propeller
Hélice

see Impeller

Propeller Pump

Propellerpumpe

Pompe hélice

The expression p.p. which is commonly used in centrifugal pump technology derives from the ship's propeller, which, like the centrifugal pump, is a hydraulic energy-producing machine, but has another function, i.e. to generate thrust for propulsion. The common feature of the ship's propeller and the p.p. is the axial impeller, which can be either a fixed propeller (with fixed blades (vanes)) or an adjustable propeller (with blades (vanes) which can only be adjusted in respect of pitch when the propeller is dismantled), or again a variable pitch propeller (with pitch adjustment of the blades while the pump is running impeller blade pitch adjustment, control, cooling water pump). Thus pumps with variable pitch blades on a mixed flow type impeller are also designated p.p.'s (see Fig. 4 under impeller and Fig. 2 under impeller blade pitch adjustment).

P.p.'s have the highest specific speeds of any centrifugal pump type ($n_q > 110 \text{ min}^{-1}$, although most p.p.'s have an $n_q > 160 \text{ min}^{-1}$). The higher the specific speed of the p.p., the smaller is (usually) the number of propeller blades, the camber of the profiles (flow profile) and the hub ratio. P.p.'s are meant for high capacities and low heads (usually up to 15 m approx. in a single stage with axial propellers, and 20 m approx. with mixed flow propellers).

The throttling curve (Figs. 3 and 4 under performance chart. Figs. 1 and 4 under characteristic curve) has a steep slope compared with other centrifugal pumps and features a so-called "break away limits", which characterizes the operating behaviour of this type of pump. The shaft power curve (characteristic curve) has its maximum at pump shut-off head (head at capacity $Q = 0$); therefore p.p.'s are usually started (starting process) against an open discharge valve, to avoid overloading the drive. The characteristic aspect of the characteristic curves of this pump type (throttling curve of "saddle" shape, "break away limit" increase in absorbed power with decreasing capacity) makes it worthwhile adopting a by-pass adjustment (control) for these pumps, which would be uneconomic on other pump types: by opening a by-pass, one can reduce the shaft power and head of the p.p. By impeller blade pitch adjustment while the pump is running, either via a mechanical, electrical or hydraulic pitch adjustment gear (see illustrations under impeller blade pitch adjustment), the pitch of the blades, and thereby the head and the "break away limit" can be altered within wide ranges, and this type of control is very widely adopted on p.p.'s. P.p.'s with blades which can be adjusted during mounting (but not when the pump is running) make it possible to perform subsequent adjustment to the head on identical pump components and without having to carry out any subsequent machining of the components. Variable pitch propeller pumps are started up at very small blade incidence angles, i.e. at the lowest shaft power.

Most axial p.p.'s are built in the form of tubular casing pumps (Fig. 1). In the case of mixed flow p.p.'s, we find volute casing pumps as well as tubular casing pumps, and the largest of these are equipped with concrete volute casings. The pump shaft of tubular casing p.p.'s is guided in water or grease-lubricated shaft guide bearings (plain bearings) inside the rising main; the axial thrust is absorbed above the stuffing box (shaft seals) by a sturdy thrust bearing (plain bearing, anti-friction bearing). P.p.'s are built with propeller diameters up to several metres.

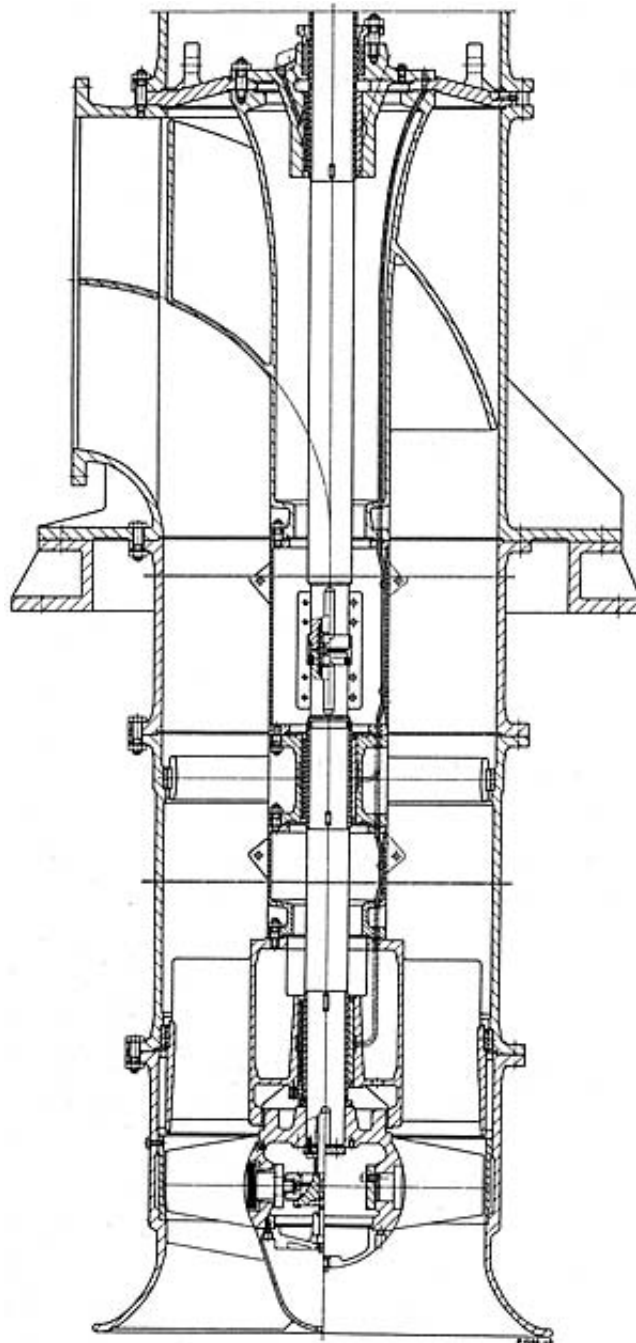


Fig. 1: Propeller pump as tubular casing pump left-hand side: variable pitch blades, shaft guide bearings lubricated by product pumped; right-hand side: adjustable pitch blades, grease-lubricated shaft guide bearings

Because axial blades are very sensitive to disturbances in the approach flow, the intake chambers, entry nozzles or intake elbows must be designed and constructed very carefully (inlet conditions).

Most p.p.'s are of single stage type, but there are a few multistage p.p.'s of axial type. because of their good regulation characteristics by means of impeller blade pitch adjustment. The multistage construction is however much more expensive than the single stage axial construction, particularly in the case of variable pitch propeller pumps or p.p.'s with a withdrawable rotor (back pull out pump, tubular casing pump). Therefore mixed flow pumps or p.p.'s with a mixed flow (semi-axial) impeller are more usually adopted for the higher heads. In the case of the semi-axial propeller (Fig. 4 under impeller), the propeller blades are inclined in relation to the main direction of flow, and the pivoting trunnions of the blade pitch adjustment are inclined from 25° to 45° to the radial direction. Thus p.p.'s with semi-axial propeller blading combine the advantages of the high heads attainable with mixed flow impellers with the advantages of good regulation characteristics achieved by variable pitch propellers (illustration under tubular casing pump).

P.p.'s are generally used as land reclamation pumps for irrigation or drainage, as cooling water pumps (e.g. in power stations, the petrochemical industry or on board ship), as circulating pumps (e.g. in the chemical industry, on boiling water reactors, in central heating plants), as universal elbow casing pumps (Fig. 2, also without diffuser for pumping in either direction of flow, with adjustable blades) in the chemical

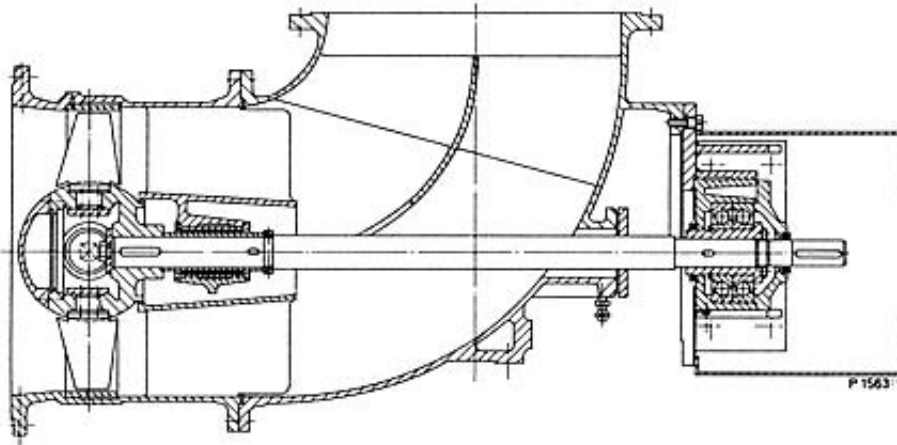


Fig. 2: Propeller pump in the form of an elbow casing pump and foodstuffs industries, and finally in their original role of thrust generation by means of a propeller in bow-thrusters.

Protection against Low Load

Schwachlastschutz

Protection contre marche à débit trop faible

see Boiler Feed Pump

Protective Layer

Schutzschicht

Revêtement de protection

According to DIN 50 902 a layer (also coating) is a cover consisting of one or several layers on the base material. It may either be formed by coating or by a change of the surface material composition (please also refer to DIN 8580), for example as surface layer, passive layer or p.l. formed by lime and rust.

The surface layer (DIN 50900, Part 1) is a layer of solid reaction products formed by corrosion, which covers the surface more or less uniformly. It can have the effect of slowing down corrosion processes. If the surface layer forms unevenly, corrosion cells (see DIN 50900, Part 2) may develop. A surface layer is only regarded as p.l. (for a definition of the term see DIN 50 902), if it is uniformly developed and slows down corrosion substantially (also see DIN 50 905, Part 2).

The passive layer is a p.l., often not detectable by an optical microscope, which occurs in the passive condition of the metal (see DIN 50900, Part 2) (corrosion). The best-known passive layers are those on stainless steels, aluminium and titanium.

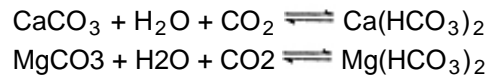
A diffusion layer is a layer formed when a metal or nonmetal diffuses into the base material.

The formation of a lime-rust p.l. depends, among other things, on the following factors: pH-value, oxygen content, salt content, temperature, flow velocity and water hardness.

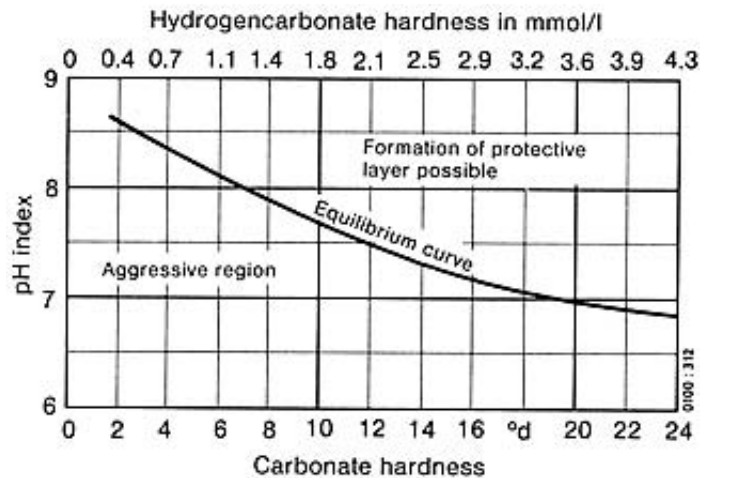
Any natural water has a certain content of free carbonic acid in the form of anions (water hardness).

Bound carbonic acid (CO_3^{2-} , HCO_3^-) is combined as calcium carbonate and/or magnesium carbonate.

Free carbonic acid (CO_2) is dissolved in water in mainly gaseous condition. Part of it the **inherent free carbonic acid** maintains the hydrogen carbonates in solution according to the following equilibrium:



(lime-carbonic acid equilibrium, see illustration). The part of free carbonic acid contained in



Equilibrium curve for the formation of surface layers as a function of the carbonate hardness and the pH value

water in excess of the concentration of inherent free carbonic acid is called excess or aggressive carbonic acid.

All waters which contain inherent free (i.e. no free, excess) carbonic acid only may form surface layers on metals. With regard to unalloyed steel this means that after initial rust attacks corrosion is considerably reduced due to the lime-rust p.l. (e.g. boiler scale).

Pulp Consistency

Stoffdichte

Consistance de pâte

see [Pulp Pumping](#)

Pulp Pump

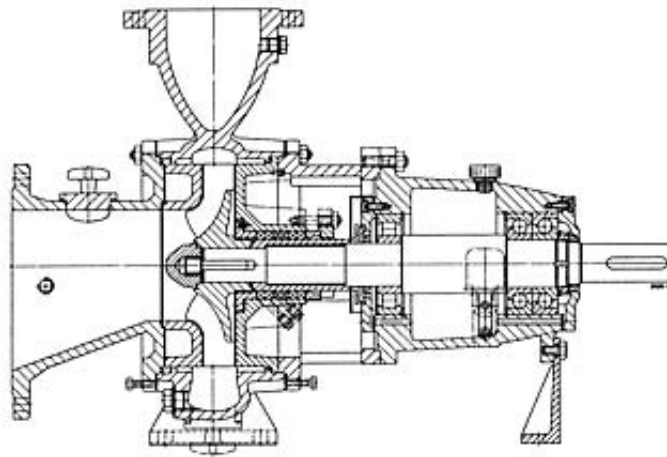
Dickstoffpumpe, Stoffpumpe

Pompe pour liquides épais et chargés, pompe à pâtes

P.p.'s, also known as stuff pumps, are used to pump fibrous material, e.g. for paper and cellulose manufacture.

For designing of p.p.'s see [pulp pumping](#).

P.p.'s are usually horizontal, single flow, single stage centrifugal pumps (see illustration) ([horizontal pump](#), [multisuction pump](#), [multistage pump](#)) of special design in [process type design](#), where the non-NEWTONian flow properties ([NEWTONian fluid](#)) and higher air contents of the pulp slurry ([pulp pumping](#)) are taken into account.



Pulp pump

Large cleaning apertures facilitate the easy and rapid clearing of any blockages. Depending on the type of pulp and on the pump application, various impellers of special design are fitted. These include open impellers, which can be combined with a fan impeller if the pulp suspension culpability is poor, and torqueflow impellers (torque-flow pump) which are fitted in cases where above all pulp suspensions (pulp pumping) containing contaminants (e.g. waste paper) have to be handled, or in cases where the special mixing action of the eddygenerating torque-flow impeller offers advantages for the process. The pump casing is protected against wear by the provision of an easily renewable wear plate on the suction side. The pump shaft is protected against corrosion and abrasive wear by being sealed in such a fashion within the pump that it does not come in contact with the product pumped. Shaft sealing is effected by means of a stuffing box (shaft seals) mechanical seal; in both instances, the seal is fed with sealing water.

Pulp Pumping

Stoffförderung

Transport de pâtes

P.p. is the conveying of pulp suspensions of various pulp densities and of various types of pulp (fibrous material, thick pulp) by means of conventional centrifugal pumps or of pulp pumps in the cellulose and paper industry.

Types of pulp:

Cellulose is an important raw material for the manufacture of paper, artificial silk (rayon), rayon staple (synthetic wool), nitrocellulose etc. Depending on the starting material, cellulose is subdivided into: wood cellulose (wood pulp) (from leaf woods (hardwoods) and pine woofs (soft woods), straw cellulose (straw pulp) and other types of cellulose, which originate from various sources of raw material such as bagasse (sugar cane trash after crushing), reeds bamboo cane etc. In addition, a distinction is made for cellulose according to the chemical treatment process involved, i.e. sulfate cellulose and sulfite cellulose (or pulp).

Wood pulp, also known as mechanical wood pulp, is prepared mechanically by grinding down decorticated hardwood or soft wood. Wood pulp is an important starting material for the manufacture of paper and cardboard (paper pulp or stock).

In contrast to the hydrotransport, where a distinction is possible between the solid matter and the carrier liquid, the pulps of the cellulose and paper industry floating in water form suspensions with a peculiar flow behaviour which must be taken into account when designing (design duty point) the pumps to handle them.

According to their flow characteristics, extremely viscous liquids are classified as follows (Fig. 1):

When the shear stress τ (viscosity) is proportional to the velocity gradient $\partial v_x / \partial y$, a NEWTONian or normal viscosity liquid is at hand for whom all laws of the fluid dynamics are applicable. All other types of liquids are named non-normal viscosity or non-NEWTONian liquids. They are categorized according to their shear characteristics, in which the deformation depends on the magnitude, change or duration of loading, and are usually treated with the subject of p.p.:

If the shear stress τ has a flow limit, one is dealing with a BINGHAM fluid. If the shear stress τ progressively climbs over the velocity gradient the fluid is called dilatant. If it progressively falls, the fluid is called structurally viscous or pseudo plastic. The flow characteristics of all aforementioned abnormal liquids are independent of the time. However, if the shear force is also dependent on times the fluid is known as rheopectic if the viscosity increases overtime with constant shear stress, the liquid is thixotropic if the viscosity lessens with constant shear stress (example: certain paints lessen in viscosity as you stir them). The dependency of the velocity gradient $\partial v_x / \partial y$ on the shear stress τ and on its temporal variation $\dot{\tau}$ is characteristic for visco-elastic fluids, which have both elastic and plastic qualities.

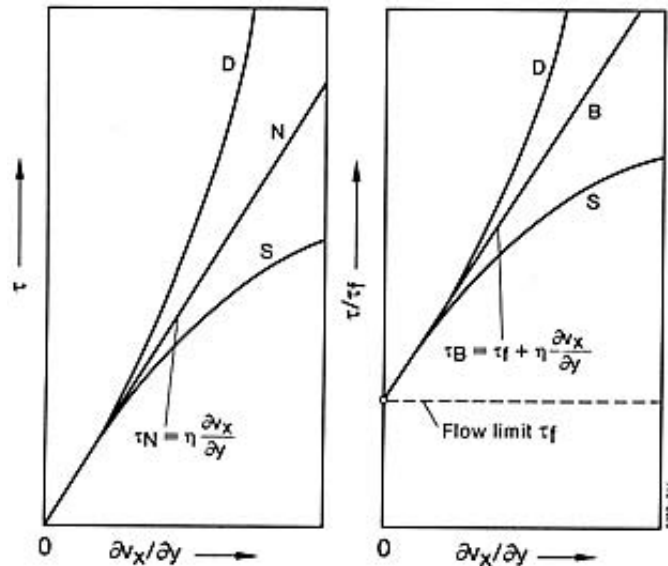


Fig. 1: Graph of the flow characteristic viscosity liquids a without, b with flow limit
N NEWTONian liquid. B BINGHAM fluid, S intrinsic viscous liquid. D dilatant liquid

In view of these influences on the flow characteristics, it is understandable that the following data are valid only for special types of pulp and concentrations. Particularly when calculating the pump capacity and the head of centrifugal pumps that are used for p.p., one can only deal with rough, general correction factors (Fig. 2). With these empirically calculated correction factors f_Q and f_H one can roughly determine the change in the characteristic curves of centrifugal pumps, depending on the operating point on the curve, in the following manner:

$$Q_{St} = \frac{Q_W}{f_Q},$$

$$H_{St} = \frac{H_W}{f_H},$$

$$P_{St} \approx P_W$$

where

Q capacity of the pump,

H head of the pump,

P shaft power of the pump.

Subscript W for the water handling and subscript St for pulp.

When calculating the system characteristic curve one must watch that the pressure losses of the piping and the valves and fittings are dependent on the pulp concentration (Figs. 6 a to 6g under pressure loss). Laboratory analysis, to determine flow curves, is indispensable in many cases.

Apart from the type of pulp (starting material, treatment process (digestion), admixtures etc.) the flow behaviour of pulp suspensions also depends on the pulp density a measure of the pulp density is given by the percentage by weight of dry solids, and a distinction is made between air-dry (% A.D.) and bone-dry (% B.D.).

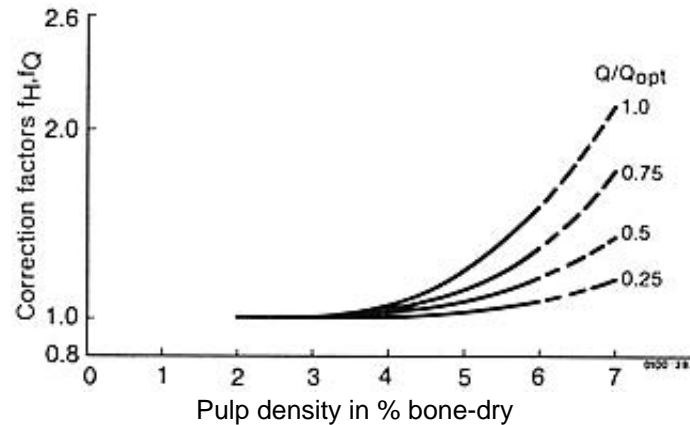


Fig. 2: Empirically calculated correction factors f_H , f_Q for the conversion of capacity and head - for transport of cellulose with centrifugal pumps - to the water values required for the design of pumps; Q/Q_{opt} relates to the characteristic curve for water

The German Wood Pulp Manufacturer's Association has laid down a "standard moisture content" of 12%, which means that an air-dry pulp contains 88 % of dry solids. A pulp density of x % of air-dry pulp therefore corresponds to a pulp density of $0.88 \cdot x$ % bone-dry.

It is also usual for the cellulose industry to specify the pulp content in grammes weight percentage of a 1 kg mixture of pulp and water. For example, a pulp suspension (mixture of pulp and water) of 30 g bone-dry corresponds to a pulp density of 3% bone-dry.

The pulp density should not be confused with the density of pumped media. Centrifugal pumps are used for the pumping of pulp suspensions up to pulp densities of 7 % bone-dry approx., which corresponds to a content of 70 kg of bone-dry pulp (100% dry solids) in one ton of pulp-water mixture.

Pulp suspensions with a pulp density up to 1% bone-dry approx. can be considered as pure water, as far as their flow characteristics are concerned, in respect of the design of the pumps, valves and fittings and piping required to handle them. A very clear picture of the behaviour of higher pulp densities can be obtained from the pouring test in function of the angle of tilt of the container (Fig. 3).

Pulp densities up to 3% bone-dry approx. can be handled by means of conventional centrifugal pumps. Within the region from 3% approx. to 6% approx. bone-dry (maximum), pulp pumps must be used.

Quite apart from the pulp density and air content, the nature of the pulp and the degree of grinding are decisive factors for the pump design. Whereas the pulp density and the degree of grinding will determine the type of pump construction, the size and the impellers the nature of the pulp is a decisive factor in the selection of the most suitable pump material. Depending on the type of treatment and on the pulp suspensions either cast iron, bronze, alloyed cast steel or cast iron with a hard rubber lining will be selected for the pump components in contact with the liquid pumped. Hard rubber lined pumps are adopted in the area of pulp bleaching, because the metallic materials mentioned above tend to corrode in the presence of the chemicals used for bleaching, which have a high free chlorine content.

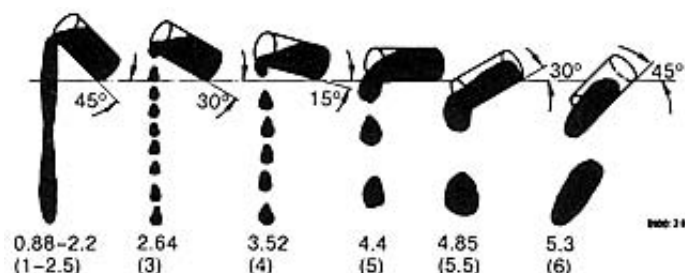


Fig. 3: Pulp density in % bone-dry (in % air-dry)

Pump Casing

Pumpengehäuse

Corps de pompe

P.c. serve to seal oil-the internal pump compartment from the outside in leaktight and pressure-tight fashion. In the case of centrifugal pumps, they surround the pump rotor which transmits energy to the fluid pumped via the rotating shaft; in the case of positive displacement pumps, they surround the rotating or oscillating displacement elements (e.g. the piston(s)).

The inlet and outlet branches (often referred to as suction and discharge branches) serve to feed in and lead out the fluid pumped (capacity); as a general rule, they are attached to lengths of pipings by suitable connections (e.g. flanges, screwed unions, fittings), or, in the case of vertical tubular casing pumps, the inlet branch is immersed in the open liquid level (Fig. 1), or again, in the case of submersible motor pumps, they are immersed together with the complete p.c. in the fluid (Fig. 2).

If the construction of the pump requires the drive shaft to pass through the p.c., shaft seals are provided to prevent any excessive escape of fluid from the p.c. These portions of the p.c. are designated seal housing or stuffing box housing.

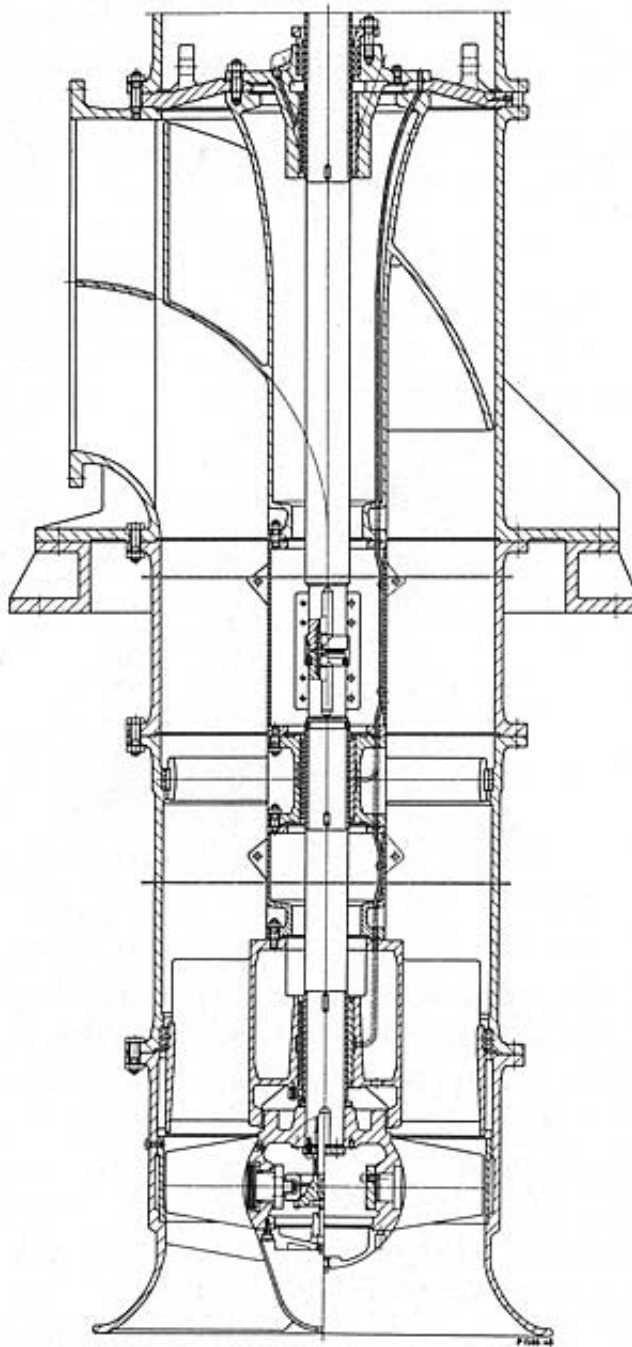


Fig. 1: Propeller pump as tubular casing pump left-hand side: variable pitch blades
plain bearings lubricated by product pumped;
right-hand side: adjustable pitch blades grease-lubricated plain bearings

Nearly every pump type has its own shape of casing, which makes the pump types easily recognizable (in contrast with the impeller shape which is only visible when the pump is dismantled). With increasing specific speeds, the casing shape changes, starting with the volute casing (Fig. 3) (sometimes a double volute, consisting of two spirals offset by 180° to one another to balance the radial thrust, see Fig. 6, under volute casing pump), via the vortex volute (which has a markedly asymmetrical volute cross-section at the vertical centre section, Fig. 4) and the annular or toroidal casing (with a constant cross-section right around the periphery, Figs. 8 and 13) to the tubular casing (Fig. 1) which guides the flow of the pump from the diffuser axially downstream; similarly the elbow casing pump (Fig. 2 under propeller pumps) has a diffuser leading not into a coaxial rising main but into a pipe elbow, the so-called elbow casing.

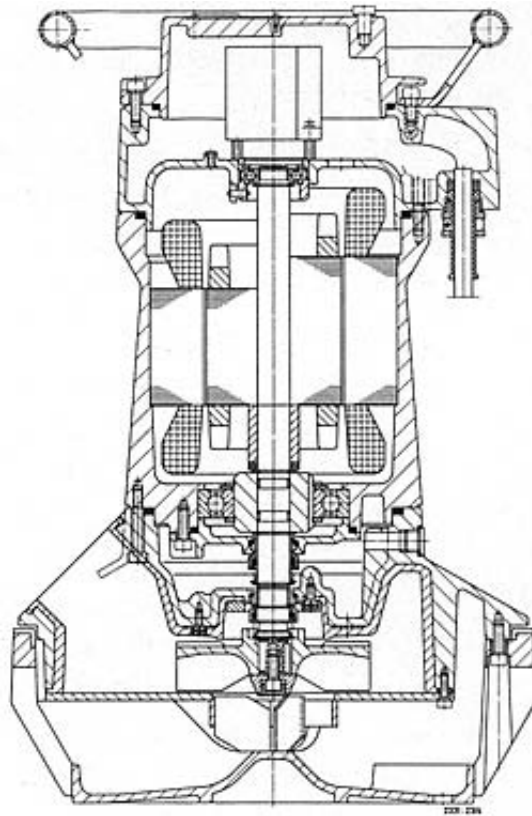


Fig. 2: Submersible motor pump for sewage

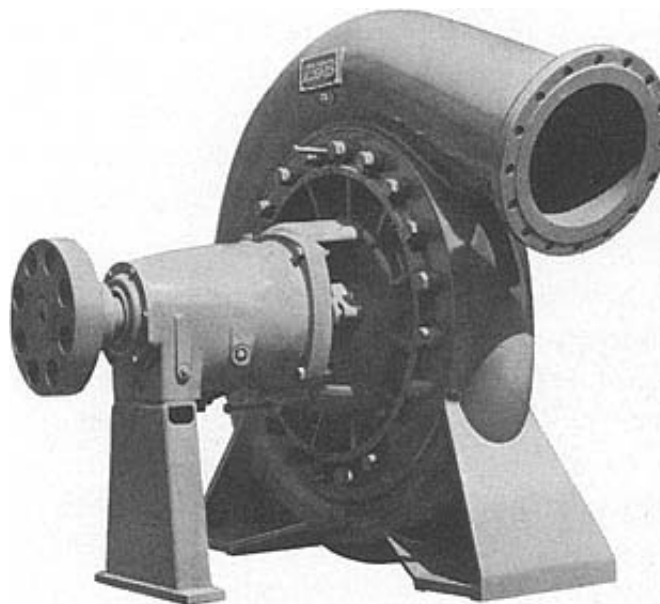


Fig. 3: Volute casing pump

The pressure range also has an influence on casing shapes: low pressure pumps demand different design solutions than those applying to high and super pressure pumps; as the pressure level rises, the geometrical shapes of the outside of volute casing pumps and of multistage pumps become simpler, and approximate cylindrical (barrel pump) (Fig. 5), cortical or spherical (reactor pump) (Fig. 6) configurations. The advantages of these designs, which are favourable from the stress or mounting viewpoints, are counterbalanced by functional disadvantages (e.g. reduced efficiency) or increased casing volume (i.e. increased costs). Vertical can-type pumps, the suction behaviour of which has made them popular for use as condensate pumps (see Figs. 2 and 3 under condensate pump) or refinery pumps (see Fig. 2 under refinery pump), and the can of which encloses its suction end, should not be confused with barrel pumps (or jacket casing pumps).

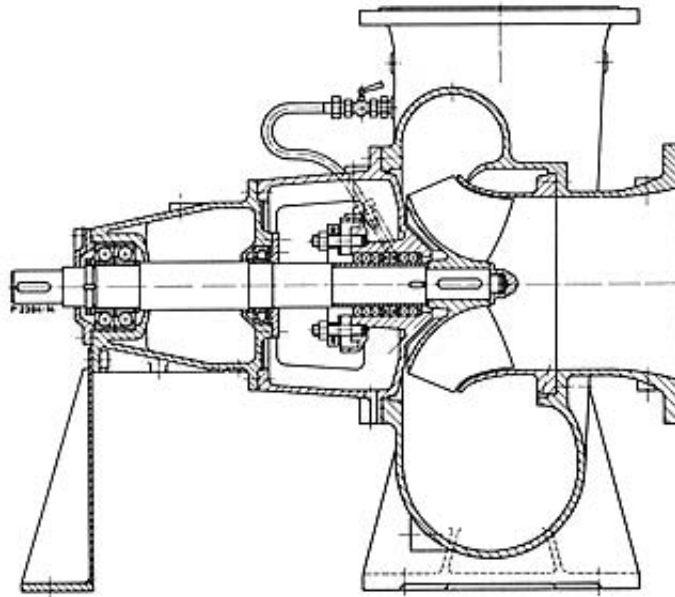


Fig. 4: Volute casing pump with mixed flow impeller and vortex volute

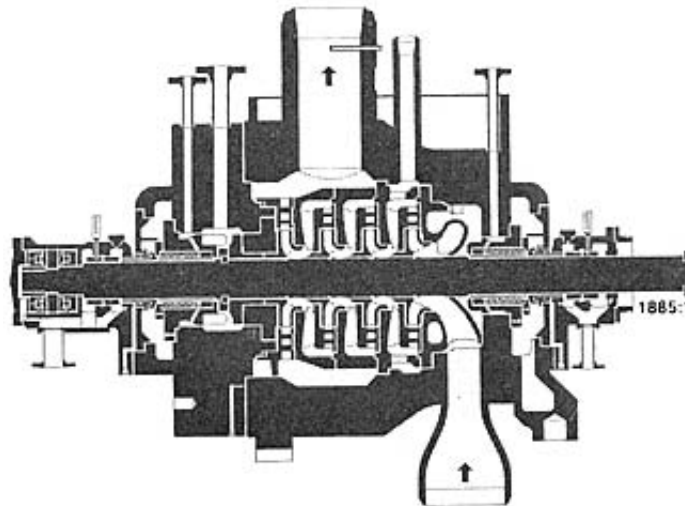


Fig. 5: Barrel type boiler feed pump (diagrammatic)

In addition, the way the casing is split (radially or axially at shaft centralize level, radially split casing, axially split casing), as demanded for mounting reasons, presents further characteristic design differences (Figs. 7 and 8).

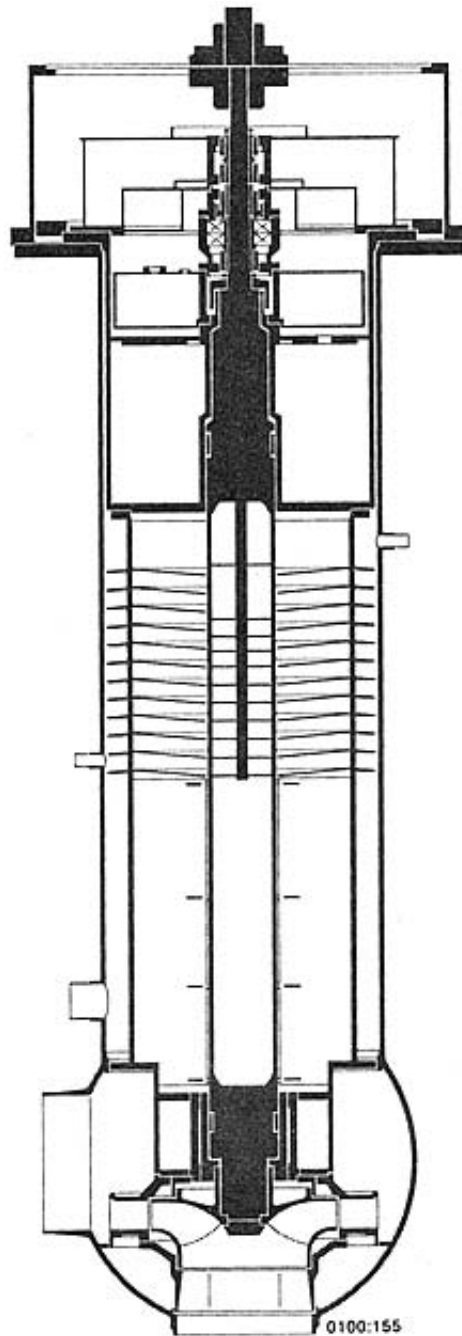


Fig. 6: Liquid metal circulating pump for sodium-cooled breeder reactor (diagrammatic)

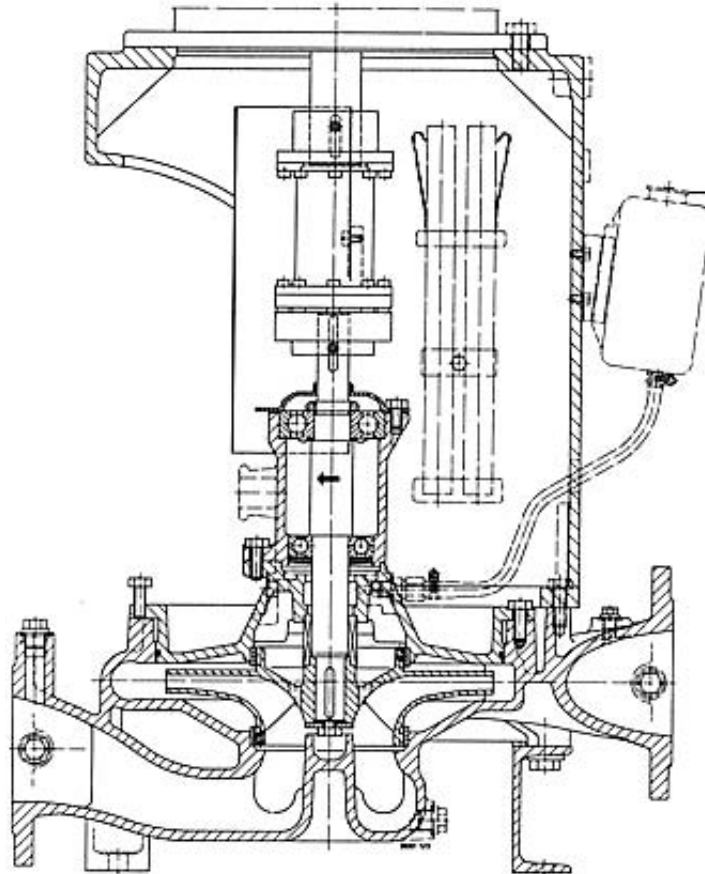


Fig. 7: Radially split vertical volute casing pump with radial impeller

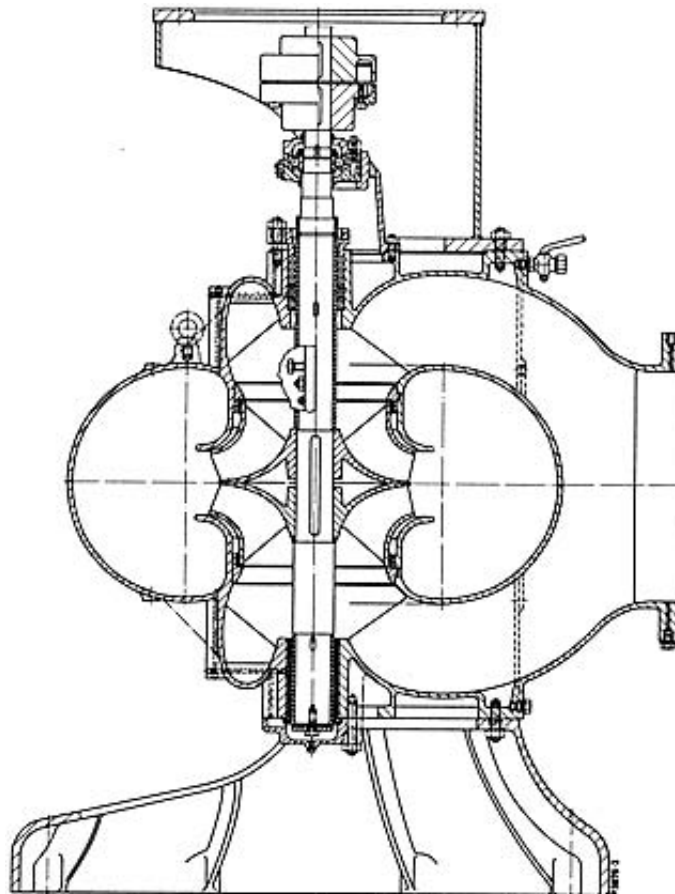


Fig. 8: Axially split vertical volute casing pump with double suction radial impeller

Finally, the location of the pump branches also influences the shape of the p.c.; the axial (or end) suction branch of single stage volute casing pumps is a characteristic feature of this pump type, in contrast to inline pumps which have branches at either side (Fig. 9) or to refinery pumps with "top-top" branches, i.e. both branches pointing vertically upwards (Fig. 10). Even the method of supporting the bearings of overhung pump shafts (Fig. 1 under volute casing pump) in a bearing bracket or in a pedestal bearing (Fig. 5 under volute casing pump) represents a design classification characteristic of the p.c.

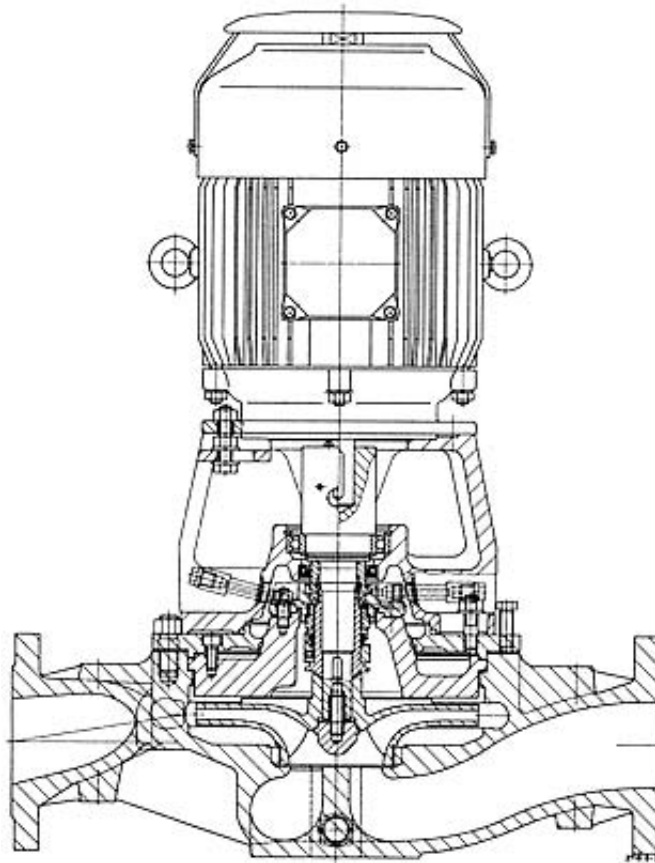


Fig. 9: Inline pump

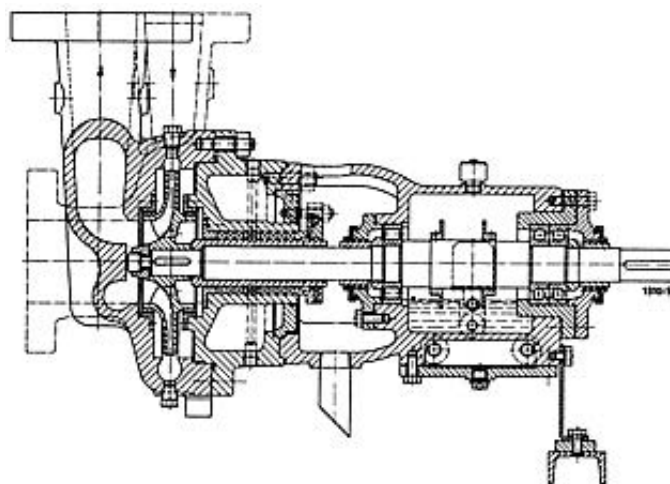


Fig. 10: Refinery process pump

In the case of ring section pumps, the casing components are named according to their function, i.e. suction casing, stage casings (usually several of these are arranged in sequence) and discharge casing. In the fully assembled state, the stage casings and tiebolts (which clamp the suction, stage and discharge casings together in pressure-tight fashion) are often surrounded by a sheet steel cladding, which contains the insulating material (glass fibres, highly polished foil) on hot water pumps, but only serves a decorative purpose on other pumps.

If the casing is split axially at shaft centreline (Fig. 11), it will consist, in the case of a horizontal pump, of a bottom half which accommodates the two branches for connection to the piping and also the pump feet, and of a simply constructed top half. The casing joint consists of a flange (flange construction) on both casing halves, which stretches around the entire p.c. including the two stuffing box housings, and which seals the casing leak- and pressure-tight by means of a number of stud bolts. In the case of axially split casing vertical pumps (Fig. 8), the pump rotor is often guided in a lower bearing lubricated by the product pumped (plain bearing), so that the second shaft seal can be dispensed with; in this case the two casing halves are called rear half (incorporating the piping nozzles and the pump foot) and front half.

Apart from the casing components already described, the p.c. in certain instances also incorporates a thermal barrier and a cooling housing (which is often sealed by a special cooling cover); both these items are designed to reduce the heat flow from the inside of the pump to the pump bearings and to the shaft seal (if there is one), on pumps which handle hot media. Conversely, a pump heating jacket is designed to maintain the contents of the p.c. of a stationary pump at the operating temperature, by an uninterrupted supply of heat, in order to prevent any undesired sedimentation or crystal growth, or even congealing of the pumped medium (Fig. 12).

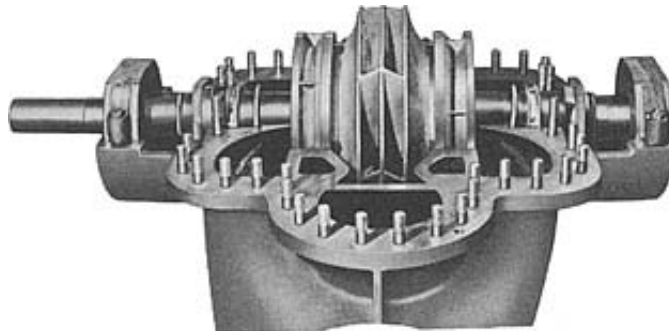


Fig. 11: Bottom casing half and rotor of a double suction radial volute casing pump, axially split at shaft centreline height

P.c. are usually cast, but nowadays also often forged and welded (Figs. 3, 4 and 5 under boiler feed pump), pressed or drawn (Fig. 13). As the operational safety of the machine depends to a large extent on the durability of the p.c., many different regulations relating to specific industries lay down the casing materials to be used, and in some instances the wall thicknesses as well. Metallic casting materials frequently used include cast iron, spheroidal graphite cast iron, ferritic or austenitic chrome steels, austenitic cast iron and cast bronzes (materials).

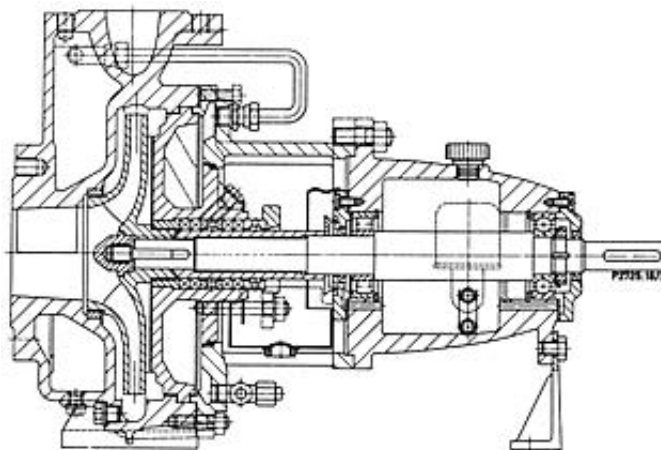


Fig. 12: Standard chemical pump with heating jacket on suction and discharge sides

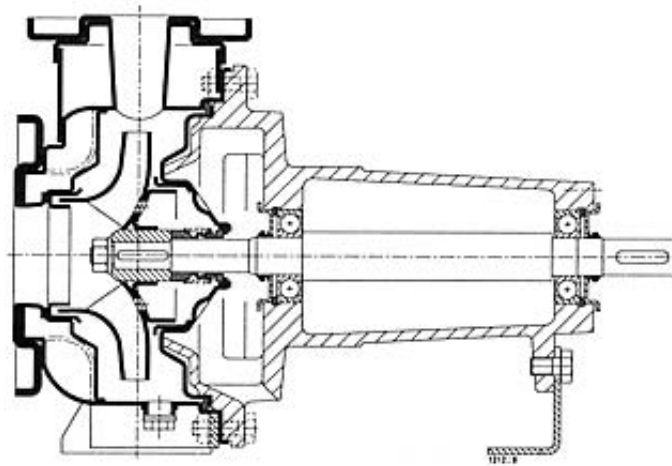


Fig. 13: Centrifugal pump with casing of drawn chrome nickel steel

Recently vertical pumps have increasingly been equipped with pump casings of concrete. This applies above all to tubular casing pumps (Fig. 2 under land reclamation pump), but also to very large volute casing pump (Fig. 6 under cooling water pump).

Pump Discharge Branch

Pumpendruckstutzen
Tubulure de refoulement

P.d.b., a cross-section (circular cross-section) to be defined on the outlet side of the pump casing as the boundary between the discharge part of the pumping plant and the pump (outlet cross-section).

Pumped Medium

Fördermedium
Liquide pompé

P.m., within centrifugal pump technology similar to pumped liquid (density of pumped media).

Pump Efficiency

Pumpenwirkungsgrad
Rendement de la pompe

The p.e. η (efficiency), also known as efficiency at the coupling or overall efficiency, is the ratio of pump output P_Q = $\rho \cdot g \cdot Q \cdot H$ and shaft power P at the operating point considered:

$$\eta = \frac{\rho \cdot g \cdot Q \cdot H}{P}$$

with

ρ density of pumped medium,

g gravitational constant,

Q capacity,

H head, and

P shaft power (at shaft of centrifugal pump), i.e. power transmitted by the drive to the pump shaft at the shaft coupling.

P.e. η is the product of the mechanical efficiency η_m and the internal efficiency η_i :

$$\eta = \eta_m \cdot \eta_i$$

The optimum efficiency η_{opt} is the highest p.e. for the rotational speed and pumped medium specified in the supply contract.

On centrifugal pumps which have no clearly defined separation between the pump shaft and the drive shaft (e.g. on close-coupled pumping sets, submersible motor pumps) it is customary to quote the efficiency of the pumping set η_{Gr} (group consisting of drive and centrifugal pump) instead of the p.e. (see DIN 24260). η_{Gr} is the ratio of pump output P_Q and power absorbed by the driver (drive), which shall be measured at a location to be agreed, e.g. at the motor terminals or at the start of an underwater cable.

The attainable p.e. η depends to a large extent on the specific speed (specific speed of the impeller), the pump size and type, and it increases generally with increasing specific speed. Fig. 1 illustrates very roughly (on the basis of statistical evaluation) the relationship between p.e. η in % and specific speed n_q in min^{-1} for diffuserless volute casing pumps (for various capacities Q); simultaneously, $\Delta\eta$ shows the potential efficiency gain to be obtained by installing a diffusor.

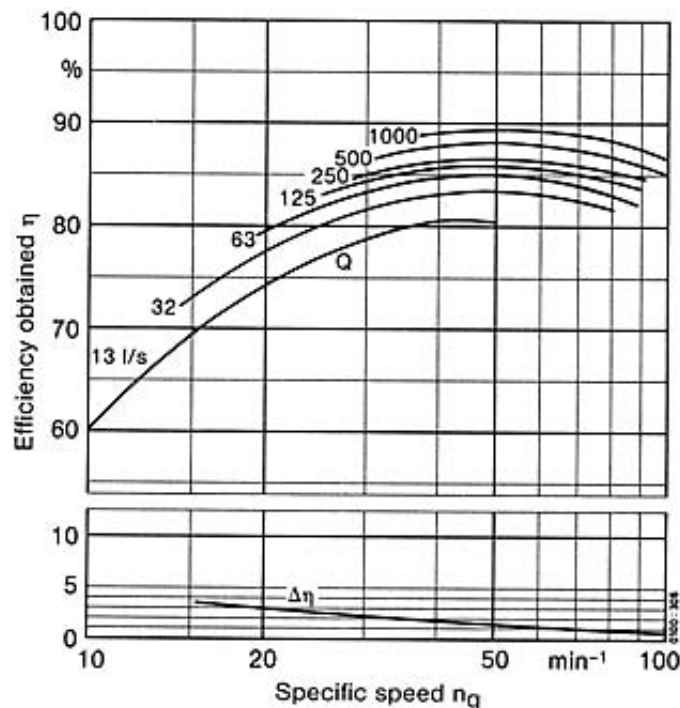


Fig. 1: Feasible efficiencies η of single stage volute casing pumps without diffuser and efficiency gain $\Delta\eta$ due to an additional diffuser as a function of the specific speed n_q

Similarly, Fig. 2 shows the attainable efficiencies of multistage pumps.

The process of efficiency re-evaluation covers the effects of REYNOLDS number (model laws) on the p.e.

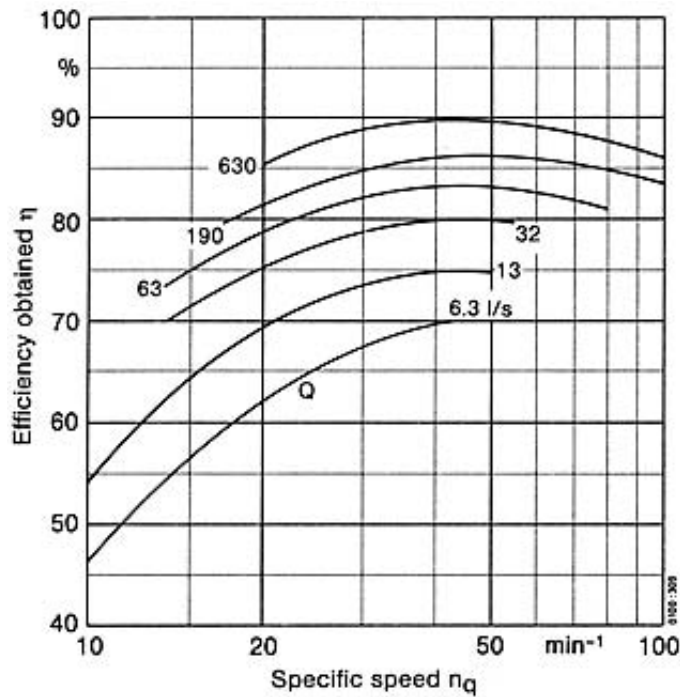


Fig. 2: Attainable pump efficiencies η of multistage high-pressure pumps (after KARASSIK) as a function of specific speed n_q

Pump Foundation

Pumpenfundament

Fondation de pompe

The foundation of a stationary centrifugal pump must be capable of absorbing without any shift of position the forces and torques of the pump placed upon it (quietness of centrifugal pumps), and also in some cases those of the associated drive and the forces of the pipng connected to the pump (branch loading). Of decisive importance to the functional reliability of the p.f. are its strength and its vibration behaviour (high-frequency tuned, low-frequency tuned p.f., see VDI 2056, and under quietness of centrifugal pumps).

The following distinction is usually made between p.f.'s:

Concrete foundation, upon which a combined baseplate for the centrifugal pump and its drive is placed (installation of centrifugal pumps).

Table foundation of concrete or steel, suspended on special vibration damping elements, in order to achieve an optimal vibration insulation against the surroundings (vibration); the individual machines (centrifugal pump, gearbox, drive) are mounted on the table via individual baseplates or base frames (foundation frames).

A special p.f. is not required if the baseplate is e.g. of cast iron, steel or concrete, and is built sufficiently rigid against distortion (warping), so that smaller and medium-sized pump sets with medium loads by the pipng can be installed without a foundation (installation of centrifugal pumps).

Pumping Plant

Pumpenanlage

Installation de pompage

The concept p.p. or simply "plant" arises when determining the head, NPSH (net positive suction head), characteristic curve, operating point, inlet and outlet cross-sections, pressures, and various elevation data.

The definition of the p.p. is illustrated in Fig. 2 under head. It comprises the suction portion of the plant, with the suction vessel and/or the suction piping, including all valves and fittings, and the discharge portion of the plant, with piping, valves and/or discharge vessel. The suction part of the plant is situated between the inlet cross-section A_e of the plant (which has to be specifically defined) and the inlet cross-section A_s of the pump; the discharge portion of the plant is situated between the outlet cross-sections A_d of the pump and the outlet cross-sections A_a of the plant (which has to be specifically defined).

The concept of p.p. as defined above only encompasses the space through which the fluid pumped flows, but no devices in the narrower sense such as auxiliary drives etc.

Apart from the terminal energy receiver (user) in the plant (e.g. pressure vessel, cooler, condenser, overhead tank), the piping with fittings and valves (pressure loss) uses a considerable amount of energy.

Pump Output

Förderleistung

Puissance utile de la pompe

The p.o. P_Q is the useful output transmitted by the centrifugal pump to the fluid pumped:

$$P_Q = \rho \cdot g \cdot Q \cdot H$$

where

ρ density of pumped medium,

g gravitational constant,

Q capacity and

H head.

In the case of appreciable compressibility of the fluid pumped, it is assumed by agreement that the condition prevailing at the pump suction branch shall apply to the density ρ .

For an explanation of the p.o., see Fig. 1 under head.

The SI unit of p.o. is 1 W; in most cases it is expressed in kW.

Pump Power

Pumpenleistung

Puissance de la pompe

see Shaft Power

Pump Shaft

Pumpenwelle

Arbre de ppmpe

The p.s. transmits the driver torque (drive) to the impellers of centrifugal pumps or to the displacement elements of rotating positive displacement pumps. In the case of oscillating positive displacement pumps, the p.s. takes the shape of a crankshaft or a camshaft.

The p.s. is the central component of a centrifugal pump rotor, and it carries the impellers, the shaft sleeves (e.g. spacer sleeves between the impellers or shaft protecting sleeves in the region of the shaft seals), the bearings (anti-friction bearings or plain bearing sleeves and thrust bearing discs), and in some cases balance discs or balance pistons (balancing device) and the coupling, and also other components forming part of the rotor (flinger rings, balancing discs, inducers, shaft nuts); in the case of close-coupled pumping sets there is no coupling (shaft coupling). The hollow shafts of propeller pumps also incorporate the adjustment rod which actuates the blade pitch

adjustment gear (impeller blade pitch adjustment).

The p.s., apart from transmitting energy, has the secondary task of centering the rotating components on the pump rotor within the bores of the pump casing in such a way that the former do not foul the latter when the pump is running, taking the shaft sag into account. It is necessary to keep the radial clearance gaps (so-called sealing gaps) between pump rotor and pump casing as small as possible, in order to reduce the flow losses (clearance gap loss) and the rate of leakage (shaft seals), and fouling may therefore occur at these clearance gaps under certain operating conditions in multistage centrifugal pumps; therefore the materials used in these places must exhibit certain minimum sliding properties.

When sizing a p.s., one must take into account not only the max. torques to be transmitted and the permissible shaft sag, but also any bending and torsional vibrations which may arise. The determination of the critical speed of rotation demands a good deal of practical knowhow, because of the damping influence exercised by the throttling gaps, which cannot be predicted theoretically, and of the special stiffening effects resulting from different modes of installation (installation of centrifugal pumps).

Finally the effects of corrosion on the shaft material must be taken into account, with the exception of "dry" shafts, (shafts not coming into contact with the fluid pumped), i.e. a corrosion-resistant material quality must be used (selection of materials). In all cases, however, steps must be taken to ensure that the shaft material will not suffer permanent deformation through time as a result of temperature fluctuations.

Pump Suction Branch

Pumpensaugstutzen
Tubulure d'aspiration

P.s.b., a cross-section (circular cross-section) to be defined on the inlet side of the pump casing as forming the boundary between the suction part of the pumping plant and the pump itself (inlet cross-section).

Pump Sump

Pumpensumpf
Puisard de pompe

see Pumping Plant

Pump Test Bed

Pumpenprüffeld
Champ d'essai de pompage

A p.t.b. is essential for experimentation and checking purposes during the development stage of pumps, and for the testing of the finished product (amongst other things for acceptance tests designed to prove the guaranteed values).

The test beds for these purposes will differ in design according to the tasks they have to perform, and range from the simple function test bed to complicated test installations designed for the highest accuracy requirements.

There is a basic distinction between "open" and "closed" test beds (Figs. 1 and 2). Fig. 3 illustrates a closed test bed (closed loop test) for research purposes.

In the case of the open test bed, the pump is mounted next to a basin with an open water surface (atmospheric pressure); it draws water out of the basin and discharges it back into the basin via measuring and control devices (measuring technique).

In the case of closed test beds (loops), the most varied operating conditions can be simulated by superposition of suitable system pressures, independently of the installed elevation of the pumps. This is extremely important, particularly for cavitation investigations.

As a general rule, a whole series of characteristic magnitudes has to be measured or calculated on the basis of the measured values in order to measure and evaluate the operating behaviour of pumps. Facilities must therefore exist to determine the following magnitudes (measuring technique) during the test run: capacity, head, torque, shaft power and rotational speed or power absorbed by the drive (drive rating), NPSH (net positive suction head) and if necessary temperatures. Despite the growing tendency to test centrifugal pumps, particularly those with a high shaft power, in situ after installation (acceptance test), it remains necessary for development purposes in centrifugal pump technology to improve, adapt or re-design p.t.b.'s.

The following aspects must be taken into account when designing and planning a p.t.b.:

1. *Object of pump tests.* Demonstration of performance data; check of functional reliability; determination of factors which will help to improve and further develop the product (research and development).
2. *Problems involved.* Measurement of pressures, capacities, power inputs and outputs, rotational speeds, noises, strengths, vibrations, evaluation of pump behaviour under special conditions (operating behaviour).

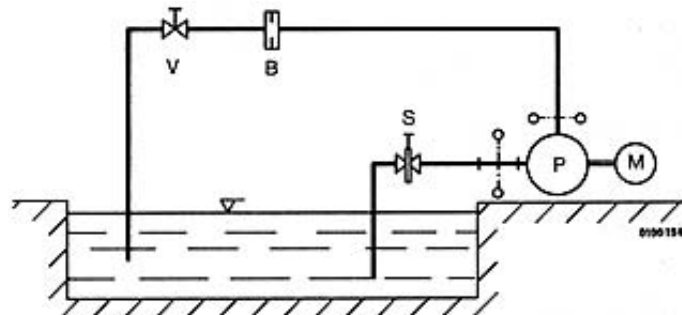


Fig. 1: "Open" test bed (diagrammatic) see Fig. 2 for explanation of symbols

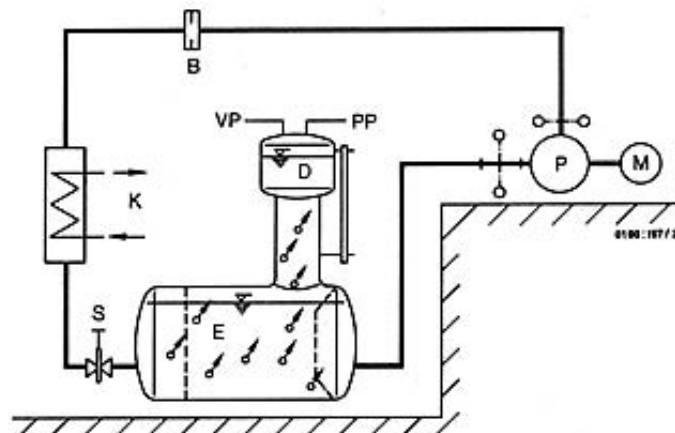




Fig. 2: "Closed" test bed (diagrammatic)

P pump; M motor; B measuring orifice plate; K cooler;
S gate valve; E degassing tank, D dome; V globe valve;
VP vacuum pump; PP compressed air pump;  gas bubbles;  measuring point; - - - strainers

3. *Relevant standards.* For acceptance tests of centrifugal pumps: DIN 1944, ISO 2548 and 3555, EUROPUMP, BS 599, Standards of Hydraulic Institute, ASME Power Test Code and others (acceptance test codes for centrifugal pumps).

4. *Measuring instruments used.* The measuring instruments should be adapted to the particular requirement in each case, as regards class accuracy (C1.), sturdy construction and quality.

Pressure measurements (measuring technique). In general, the following will be adequate on a p.t.b.: spring pressure gauge of C1. 0.6, i.e. max. permissible instrument error = 0.6% of full needle deflection; mercury manometer with hydraulic seal, single leg or twin leg type; in special cases, when nonsteady phenomena have to be investigated, transmitter with amplifier and oscilloscope.

Capacity measurements (measuring technique). In the case of low capacities: tank measurements. More often, standard throttling devices in accordance with DIN 1952 and VDI 2040 (standard orifice, standard nozzle, standard venturi nozzle) are used on the p.t.b. In the case of very large capacities, special methods have to be adopted (hydrometric vane measurements across the cross-section or others).

Power measurements (measuring technique). For very accurate measurements on the p.t.b., pendulum type electric motors are available (very expensive and cumbersome in the case of high powers). In normal cases, the absorbed electric power of the drive is measured by the two wattmeter method (the motor is usually calibrated by determination of the individual losses). Very often, the power is determined via the torque and rotational speed by means of a torsion dynamometer (measuring technique) which are being rapidly superseded by adoption of torque metering hubs.

Rotational speed measurements (measuring technique). Digital counters, which count 60 pulses per revolution, are often used on p.t.b.'s; this gives a direct display in min^{-1} . The display accuracy is very high ($\pm 1 \text{ min}^{-1}$). A simpler method is to use hand tachometers and eddy current tachometers (deviation 0.5% or less).

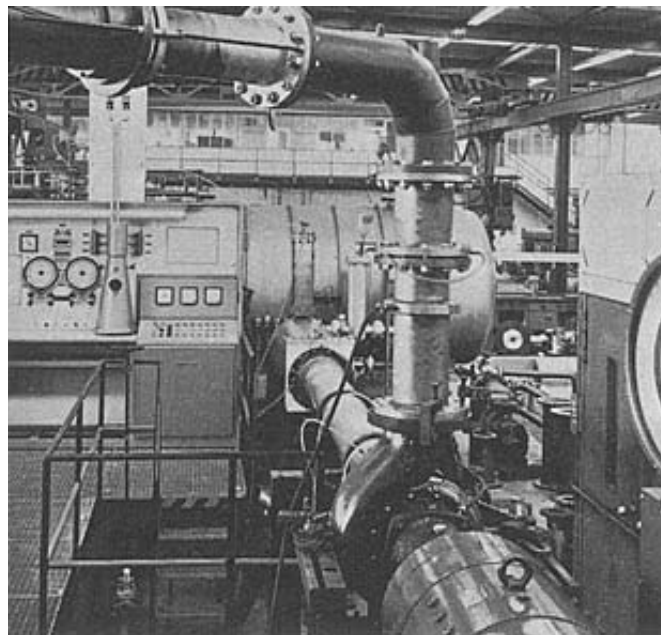


Fig. 3: View of closed test bed with measuring desk

5. *Test bed set-up.* Depending on the p.t.b. facilities, tests are carried out in open or closed circuit. The entire power absorbed by the pump (shaft power) is converted into heat. This makes the provision of a large test bed tank necessary, and of cooling facilities when required.

6. *Special measuring equipment.* When designing a p.t.b., the special requirements relating to (measuring technique), noise measurement, net positive suction head measurements, model tests in air or water, anti-friction or plain bearing trials, fatigue tests, tests on toxic, hot, explosive products must be taken into account (safety regulations).

In the future, test bed measurements which represent the final check on individual products will be simplified still further. By making more intensive use of EDP facilities, it will be possible to automate the pump testing procedure to a great extent. As pump outputs tend to rise more and more, the economic limit of test bed measurements is already foreseeable. There is bound to be a transition to model techniques (model laws) for large units, and to acceptance testing of such units on models, and to improved measurements on the products in situ. The reduced accuracy of an acceptance test in situ will be compensated by an application which is closer to the true conditions. The development of new acceptance test procedures, e.g. the thermodynamic determination of the pump efficiency partly circumvents the problems of determining the shaft power or of measuring the capacity in situ.

Pump Types

Pumpenbauarten

Constructions des pompes

There is no homogeneous pump terminology, because very different criteria are used to designate a pump. The synopsis below only enumerates the main aspects used for classification purposes:

1. operating principle or mode of operation of the pump,
2. design characteristics,
 - 2.1. shape of impeller,
 - 2.2. arrangement of impeller,
 - 2.3. shape of casing,
 - 2.4. mode of installation,
3. application or application field of the pump,
 - 3.1. in respect of mode of operation,
 - 3.2. in conjunction with the pumping plant,
 - 3.3. in respect of the product pumped,
4. pump drive,
5. material of construction of pump.

Item 1.: The operating principle distinguishes between:

Centrifugal pumps. The characteristic of these hydraulic fluid flow machines is the transmission of energy within the impeller blaming on the basis of flow deflection (fundamental equation of fluid flow machines, fluid dynamical; in contrast we have the volume displacement principle of positive displacement pumps. The head of a centrifugal pump is proportional to the square of the pump rotational speed (model laws).

Positive displacement pumps (piston or plunger pumps). Their characteristic is the periodic change in volume of working spaces which are separated from the suction and discharge lines (pumping plant) by separating elements. Their head is independent of the pump speed. A distinction is made between oscillating and rotary positive displacement pumps; examples of oscillating pumps are the piston pump, the diaphragm pump and the semi-rotary wing pump; examples of rotary pumps are the rotary piston pump, the eccentric spiral pump, the geartype pump, the screw pump, the vane pump, the water ring pump and the peristaltic (structure) pump. In addition, the expansion action of vapours or gases can be used for displacement purposes (pulsometer, HUMPHREY pump).

Jet pumps (deep well suction device). The pressure differential in a nozzle through which the driving medium flows is used to convey a fluid (ejector principle). The driving medium can be a liquid gas or vapour. These pumps have no moving parts, and are therefore of very simple construction. Their effectiveness and efficiency is limited.

Airlift pumps (mammoth pumps). Their mode of operation is based on the lift action of a mixture of liquid and gas (two-phase flow); therefore they can only be used in pumping plants with sufficient geodetic head differences.

Hydraulic rams. They make use of the kinetic energy of a flowing liquid column and convert this energy by a sudden braking action into other forms of energy (e.g. pressure energy).

Elevators. They lift liquids to a higher level by using bucket wheels, bucket elevators, Archimedean screws (ARCHIMEDEAN screw pump) and similar devices.

Electromagnetic pumps. Their mode of operation depends on the direct action of a magnetic field on the ferromagnetic medium conveyed, and their application is therefore limited to the pumping of liquid metals.

Item 2.: If design characteristics are called on to distinguish between pumps, they can only reflect some aspect affecting one of the classes listed under 1. above, because the differing operating principles demand basically differing designs from the start. We shall only deal with design differences concerning centrifugal pumps below:

Item 2.1.: Depending on the specific speed of a centrifugal pump, its impeller may be a radial impeller, a mixed flow impeller (also known as a diagonal or helical impeller), or an axial impeller (propeller). Radial and mixed flow impellers can be constructed with open channels (with no front coverplate) or with closed channels (with a front coverplate). The propellers can be cast in one piece with their hubs, or their blades can be attached to the hub in a manner which allows pitch adjustment, in order to improve the characteristic curve, or their pitch can be varied

while the pump is running, to improve control (impeller blade pitch adjustment). The propeller blades need not to be arranged perpendicularly to the pump shaft (propeller pump), particularly if the propeller is intended for higher heads (whilst keeping its design advantages in respect of control) and has to cope with radial velocity components as a result (see under impeller for illustrations of the various types).

In addition, there are special impellers for special fluids pumped:

single vane impeller, two- and three-passage non-clogging impellers, torque-flow impeller (torque-flow pump), peripheral impeller (peripheral pump).

Item 2.2.: The impeller of a centrifugal pump can be supported on bearings at either side, or on one side only (overhung); the overhung arrangement saves one shaft seal, but increases the bending deflection of the shaft all other things being equal. In the case of large capacities, the impeller which is supported on bearings at either side can be designed as a double suction impeller, which also helps to balance the axial thrust (multisuction pump). High heads are catered for by a multistage impeller arrangement (multistage pump); if it is desired to balance the axial thrust in this case, one half of the impeller numbers can be arranged back-to-back with the other half, provided that there is an even number of stages. One can also have combinations of multisuction and multistage arrangements (impeller).

Item 2.3.: The many casing shapes present the greatest variety of design characteristics (pump casing).

Item 2.4.: The various modes of installation of centrifugal pumps present a whole series of different design characteristics. First of all, one can distinguish between horizontal and vertical (shaft) pumps; vertical shaft pumps can be installed dry or wet (dry and wet installation); wet installation centrifugal pumps are also called submersible pumps and are flooded by the pumped medium, such as e.g. the majority of tubular casing pumps. Occasionally one also came across centrifugal pumps installed with an inclined shaft, e.g. very large land reclamation pumps (tubular casing pumps). In addition, the attachment of the centrifugal pump to the foundation (pump foundation) is a distinguishing characteristic; the pump may stand on its own feet (in the case of horizontal pumps for cold media, with feet at the bottom of the casing, and for hot media, with feet arranged at shaft centralize height), or it may be flanged onto its driver in the case of close-coupled pumping sets. There are further differences in the way the pump casing is connected to the driver casing (e.g. flanged motors or drive stools) and also differences in nameplates (combined baseplate for pump and motor, or separate baseplates). Furthermore, a distinction must be made between fixed installation pumps and mobile or portable pumps.

Item 3.: Pump designations in accordance with their application are commonly found (application fields for pumps), and their mode of operation, their application field or the product pumped often play apart in the naming of pumps. These concepts seldom require a more detailed explanation.

Item 3.1.: The following distinctions must be made in respect of the mode or type of operation: besides the service pump, there are standby pumps and spare pumps. In addition to the main pump, there is occasionally a booster pump. Other concepts which designate the operational duty of a pump include: full load or base load pump, partial load (e.g. half-load) or low load pump, peak load pump, auxiliary pump, start-up pump, emergency pump.

Item 3.2.: There are many designations which appear in the relationship between the pump and the plant it serves, and those most frequently used are listed below:

In the field of hydroeconomy (water supply, irrigation and drainage, sewage disposal) we have waterworks pumps, water supply pumps, hydrophor pumps (domestic water supply plant), deep well and borehole pumps, irrigation pumps, sprinkler pumps, land reclamation pumps and drainage pumps (tubular casing pumps), high-water pumps, storm water pumps and many others.

In power stations and central heating installations, we have boiler feed pumps (feedwater pumps), condensate pumps, reactor pumps, storage pumps (which often maybe used in pump and turbine operation), district heating pumps and central heating circulating pumps.

In the chemical and petrochemical industries, we have chemical pumps, pipeline pumps, refinery pumps, process pumps (process type design), inline pumps, charge pumps, blending pumps and recirculating pumps.

In the shipbuilding industry we have marine pumps, cargo oil pumps for the loading and unloading of tankers, ballast pumps, bilge pumps, drainage pumps, and dock pumps for the filling and emptying of dock installations.

Other applications include trench drainage pumps, dredging pumps, fire-fighting pumps, hydrostatic pressure test pumps, scavenging pumps, filling station petrol pumps and vent pumps (siphoning installation).

Item 3.3.: Just as frequently as pumps are designated in relation to the plant they serve, so they are also designated with reference to the product pumped:

Most pumps are used to pump fluids containing a very large proportion of water, viz.: clean water pumps, drinking water pumps, warm and hot water pumps, cooling water pumps, sea water pumps, brine pumps, condensate pumps, feedwater pumps; sewage pumps, faeces pumps, liquid manure pumps, sludge pumps, pulp pumps, solids pumps, cellulose pumps, wood pulp pumps (pulp pumping).

Pumps designated with reference to fluids other than water include oil pumps (heating oil pumps, lubricating oil pumps, hot oil pumps), petrol (gasoline) pumps, heat transfer pumps, refrigerant pumps, liquefied gas pumps, grease pumps, acid pumps, lye solution pumps, beverage pumps (milk, beer, wine pumps), fish pumps, sugar beet pumps, cassette (beet slice) pumps, fruit pumps and concrete pumps (for the pumping of liquid concrete on building sites, not to be confused with concrete casing pumps, which have volute or tubular casings made of concrete, cooling water pump).

Item 4: The designation of a pump according to its type of drive requires little detailed explanation:

Hand pumps, engine-driven pumps, turbine driven pumps, geared pumps, electric motor driven pumps, flanged motor pumps, submersible motor pumps, wet rotor motor pumps (wet rotor motor), canned motor pumps and magnet pumps (the shaft torque on the lastnamed grandness pumps is transmitted by magnetic induction, magnetic coupling).

Item 5.: The designation of centrifugal pumps in accordance with the construction material relates usually to the casing material only, because the various individual components of a pump are made of the material best suited for the particular purpose, and not of the same material (selection of materials). In addition, only the material groups are named in this classification:

Cast iron pumps, spheroidal graphite cast iron pumps, bronze pumps, cast steel pumps, highgrade steel pumps (i.e. stainless steel pumps), plastic pumps, ceramic pumps (stoneware pumps, porcelain pumps), concrete casing pumps (volute casing pump, tubular casing pump).

In many cases, the parts of a pump in contact with the fluid pumped are lined with a protective lining or coating, e.g. rubber-lined pumps, enamelled pumps, plastic-lined pumps and armoured pumps (the surfaces of which in contact with the fluid are provided with a hard facing deposited by welding (build-up) armoured pump)).

The concepts "corrosion-resistant pump" and "wear-resistant pump" are unprecise with regard to pump classification, because of the interrelationship between pump material, fluid pumped, flow velocity and temperature, and they are hardly adequate to convey even a rough definition without more detailed explanations.

Q

Quantity of Heat

Wärmemenge

Quantité de chaleur

see [Energy](#)

Quenching Liquid

Quenchflüssigkeit

Liquide de refroidissement

see [Shaft Seals](#)

Quietness of Centrifugal Pumps

Laufruhe von Kreiselpumpen

Tranquillité de marche

In technical jargon, q.o.c.p. relates more to the evaluation of the mechanical vibrations of the centrifugal pump than to the evaluation of noises. In VDI guideline 2056 "Criteria for assessing mechanical vibrations of machines", the evaluation of the so-called "vibration quality" in function of the vibration speed (swiftness) and of the amplitude of the vibration travel (related to a frequency of 50 Hz) is given for three groups of machines which apply to centrifugal pump technology (extracts given in the Table).

The vibration speed and the amplitude of the vibrations are measured both at the running pump shaft and at various other locations on the pump casing and pump foundation to be mutually agreed between supplier and purchaser, with the aid of electrodynamic vibration speed and vibration travel recorders. Care should be taken to measure not only in the direction of the pump shaft, but also in various pre-selected directions at right angles to the pump shaft. The Table gives approved and reliable guideline values for the evaluation of the q.o.c.p.; the following criteria should also be considered for the final evaluation: permanent presence of people in the vicinity of the centrifugal pump (environmental protection, noise in pumps and pumping installations); smooth and quiet running of machines in close proximity, particularly pump drives; vibration behaviour of pump foundation and building (tuning); vibration behaviour of attached piping; agreed quality and anticipated life duration of the pump, etc.

Table: Quality guidelines relating to the smooth and quiet of small machines(group K), large machines (group G) and turbomachines (group T), in accordance with VDI guideline 2056

Maximum value of vibration speed mm/s	Equivalent travel amplitude for 50Hz μm	Quietness		
		group K	group G	group T
0,63	2	good	good	good
1,0	3,15			
1,6	5	acceptable		
2,5	8			
4,0	12,5	still tolerable	acceptable	acceptable
6,3	20	not tolerable	still tolerable	
10	31,5			
16	50		still tolerable	
25	80			
				not tolerable

The q.o.c.p. can be considered satisfactory if the pump is operated within the operating zone of rotation-symmetrical flow, assuming that the rotating assembly is properly balanced (unbalance of centrifugal pumps). This will generally be the case in the zone of optimal capacity Q_{opt} and disturbance-free approach flow (inlet conditions), assuming the NPSH available is adequate (net positive suction head). If the NPSH available is insufficient, cavitation will occur at the impellers, and this cavitation will in most cases not be rotation-symmetrical, and give rise to more or less important transverse forces (radial thrust).

Centrifugal pumps with radial impellers can occasionally be operated at very reduced loads, but it should be avoided for prolonged periods in the case of large pumps at any rate. Especially in the case of mixed flow and even more so of axial pumps, part load operation is only possible to a limited degree (e.g. for propeller pumps: $Q_{min}/Q_{opt} \approx 0.8$). At the so-called separation limit (operating behaviour), the flow breaks away from the impeller vanes with irregular pulsations, causing the pump to run rough because of the eccentric point of application of the resultant force. This separation limit is particularly pronounced in propeller pumps, but it can be shifted towards lower capacities by impeller blade pitch adjustment to smaller attack angles (flow profile). But Q_{opt} also decreases thereby, and the previously mentioned ratio Q_{min}/Q_{opt} of 0.8 approx. remains the same. Measures which help in curing rough running include:

- elimination of possible unbalanced of centrifugal pumps;
 - improving the inlet conditions;
 -
 - avoidance of unnecessary bends and elbows upstream of the pump;
 - installation of straightening, eddy-reducing, equalizing flow devices in the suction pipe (inlet conditions, pumping plants);
 - increasing the available NPSH value of the plant (net positive suction head) by reducing the pressure drops upstream of the pump (adequately large piping diameter, valves and fittings, elbows and other fittings which exhibit low pressure losses);
 - reducing the required NPSH value of the pump (net positive suction head) by fitting an inducer, increasing the suction mouth of the impeller, providing specially shaped blades, providing a slight rotational swirl in the direction of impeller rotation (vortex flow) and other measures;
 - avoidance of part load operation (operating behaviour), e.g. by control (by-pass, impeller blade pitch adjustment), or by the use of additional smaller pumps, so-called "half load pumps".
-

R

Radial Flow Pump

Radialpumpe
Pompe à roue radiale

see [Centrifugal Pump](#)

Radial Force

Radialkraft
Force radiale

see [Radial Thrust](#)

Radial Impeller

Radialrad
Roue radiale

see [Impeller](#)

Radially Split Casing

Querteilung
Corps à joint radial

The r.s.c. of a [centrifugal pump](#) has a [pump casing](#) joint perpendicular to the shaft axis. In contrast, we have [axially split casings](#).

Radial Thrust

Radialschub
Poussé radiale

R.t., in centrifugal pump technology involves a hydraulic radial force in the plane of the impeller, generated by the interaction between the [impeller](#) and the [pump casing](#) or the [diffuser](#). A distinction is made between steady and unsteady radial forces.

Stationary radial force. The vector of the radial force R , changes its magnitude and direction with the capacity, where $q = Q/Q_{opt}$ ([capacity](#)). The magnitude increases with the [density](#) ρ , the projected impeller discharge area $B \cdot D$ and the [head](#) H , where q is constant, and the direction φ remains unchanged:

$$R = K \cdot \rho \cdot g \cdot H \cdot D \cdot B$$

where

R radial force,
 K radial force factor see Fig. 2,
 ρ density of the pumped medium,
 g gravitational constant,
 H head,
 D impeller outer diameter,
 B impeller discharge width.

For the design point of volute casing pumps ($q = 1$) a minimum is obtained in the graph of R versus q, although the radial force climbs rapidly in the low flow regime ($q < 1$) and the high flow regime ($q > 1$) (operating behaviour).

Even when the impeller is eccentrically set up in the base circle of the volute and $q = 1.0$, decentering forces can occur. At the same time, however, radial forces can be slightly smaller in other duty points.

Fig. 1 shows an example of a volute casing pump, with a specific speed of $n_q = 26 \text{ min}^{-1}$, and its force vector curves, where radial force vectors for six different capacities ($q = Q/Q_{\text{opt}}$) go Out from the center (i.e. the connecting lines between endpoints). It also shows the vectors for one centered and four eccentric positions of the impeller in the volute casing (the direction of A points to the cutwater of the volute).

Vectors $R_{A,B,C,D}$ for the design points ④ are included on all five force vector curves as an example. One can easily see that in the range of $q = 1.0$ the centered rotor position (see curve Z) leads to the smallest radial forces. The minimal forces of the remaining force vector curves for eccentric impeller positions is larger and also lies outside of the $q = 1.0$ range.

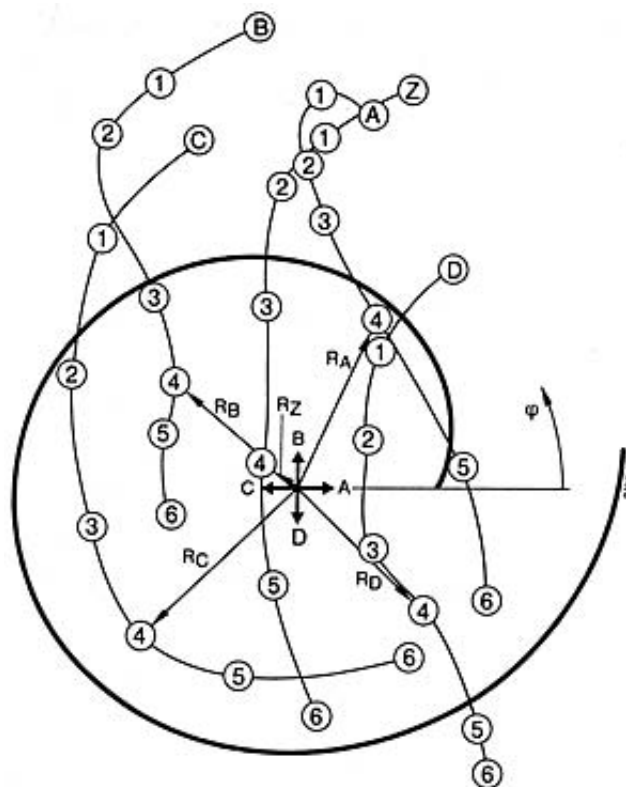


Fig. 1: Radial force vectors R for centered and off-center position of the impeller in the volute casing. The parameters are: ① = the beginning of the force vector curve with $q = 0$ in the centered position of the impeller ②, ③, ④, ⑤ = the beginning of the force vector curves where $q = 0$, and the eccentricity is 2% of the impeller radius in the direction A, B, C and D

① $q = 0.25$ ② $q = 0.5$ ③ $q = 0.75$ ④ $q = 1.0$
 ⑤ $q = 1.25$ ⑥ $q = 1.5$

Steady radial force. The vector of the radial force R changes its magnitude and direction with the capacity ratio $q = Q/Q_{opt}$ (capacity). If $q = \text{constant}$, the magnitude varies with the head, but the direction remains unaltered.

In the case of pumps fitted with diffusers with rotation-symmetrical exit flow, no radial forces are generated if the impeller is concentric with the diffuser. As the eccentricity increases, increasing deciphering radial forces arise over the entire range of q .

The magnitude of the radial forces R in volute casing pumps will depend very largely on the specific speed n_q (Fig. 2). In the case of impellers in annular casings (pump casing), the forces are smallest at part load and rise steadily as the load increases towards overload. In order to reduce these forces, double volute casings (pump casing, Fig. 6 under volute casing pump) are often fitted.

Unsteady radial force. Unsteady, and sometimes also rotating, radial forces may be superimposed on the steady radial forces. They may have different causes and characteristics. The best-known ones are the radial forces with a frequency = number of impeller vanes \times rotational speed; these radial forces appear to a greater or lesser extent in all types of pumps. In diffuser-type pumps in particular (diffuser), rotating radial forces appear at part loads and when the rotor is in central position (rotating frequency approx. $1/10$ th of pump speed).

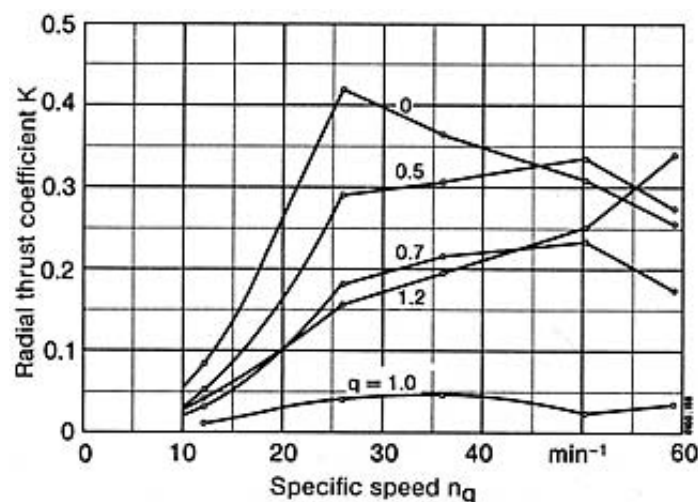


Fig. 2: Magnitude of radial force coefficient R Convolute casing pumps as a function of the specific speed n_q and the pump's rate of flow ratio q

Reactor Pump

Reaktorpumpe

Pompe de réacteur nucléaire

The r.p.'s which act as reactor coolant pumps are centrifugal pumps which circulate the coolant necessary to carry away the nuclear disintegration heat. Depending on the type of reactor plant in which the r.p. operates, one can distinguish between pressurized water r.p.'s, boiling water r.p.'s liquid metal r.p.'s, heavy water r.p.'s, and others.

Different designs of r.p.'s apply to different reactor types.

In the case of *pressurized water reactors* and *heavy water reactors*, the very high shaft power requirement imposes the use of r.p.'s with a shaft seal and with an integral bearing arrangement (thrust bearing, radial bearing, anti-friction bearing) and integrated oil supply or of closecoupled reactor pumping sets, driven by conventional electric motors (drive) (Figs. 1 and 2). The shaft seal consists either of several mechanical seals arranged in series, or of a combination of hydrostatic and mechanical seals.

Depending on the support arrangement and on the type of suspension, forces and moments have to be absorbed by the pump casing. This results in a number of different casing shapes and wall thicknesses. The r.p. casing is designed in the form of a spherical or potshaped casing (pump casing) (Fig. 3).

The design duty pressure is of the order of 170 bar, and the design duty temperature of the order of 350 °C.

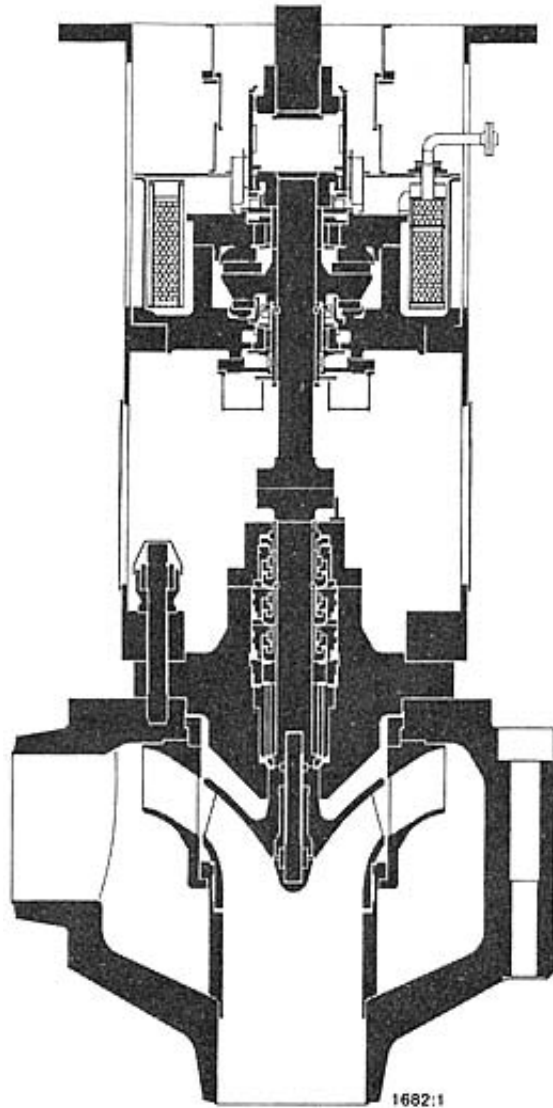


Fig. 1: Reactor coolant pump for pressurized water reactor

In the case of *boiling water reactors*, two different r.p. concepts have been adopted. On reactor pressure vessels with built-in jet pumps, the r.p.'s are welded into external piping and are designed as driving water pumps (Fig. 4). They usually have a double volute casing (pump casing, Fig. 6 under volute casing pump). Two such driving water pumps incorporated in piping loops only pump approx. one third of the local coolant flow, which drives the jet pumps, and the latter circulate the balance of the coolant flow within the pressure vessel.

The second concept involves several r.p.'s inserted into the reactor pressure vessel (insert pumps), which circulate the coolant without any necessity for external piping. Whereas the driving water pumps are equipped with shaft seals and conventional electric motor drives (Fig. 5), the insert pumps are built either in shaft sealed design or with a wet rotor motor (Fig. 6). These electric motors are speed-controlled, enabling the pump capacity to be adjusted and the reactor power to be controlled. The system pressure (design pressure) amounts to 90 bar for example, and the design temperature to 300 °C.

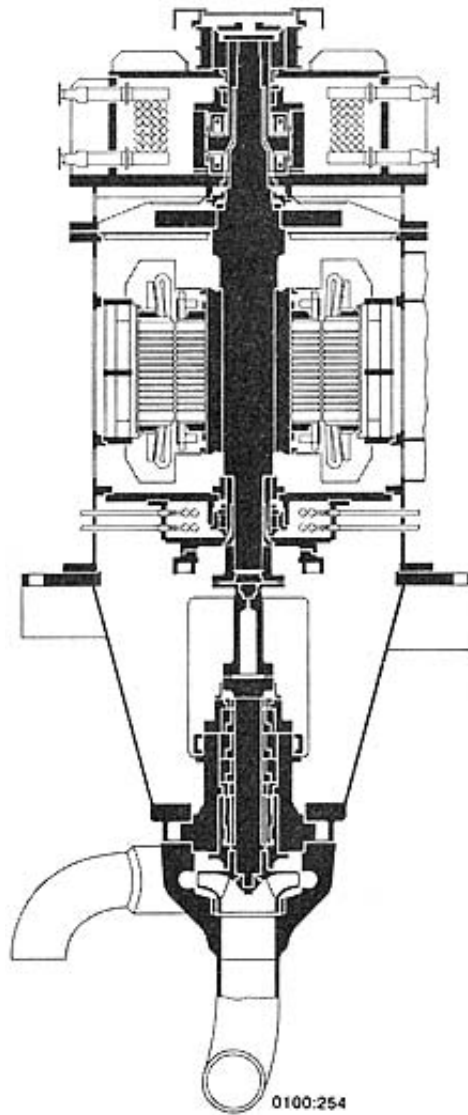


Fig. 2: Reactor coolant pump of close-coupled design

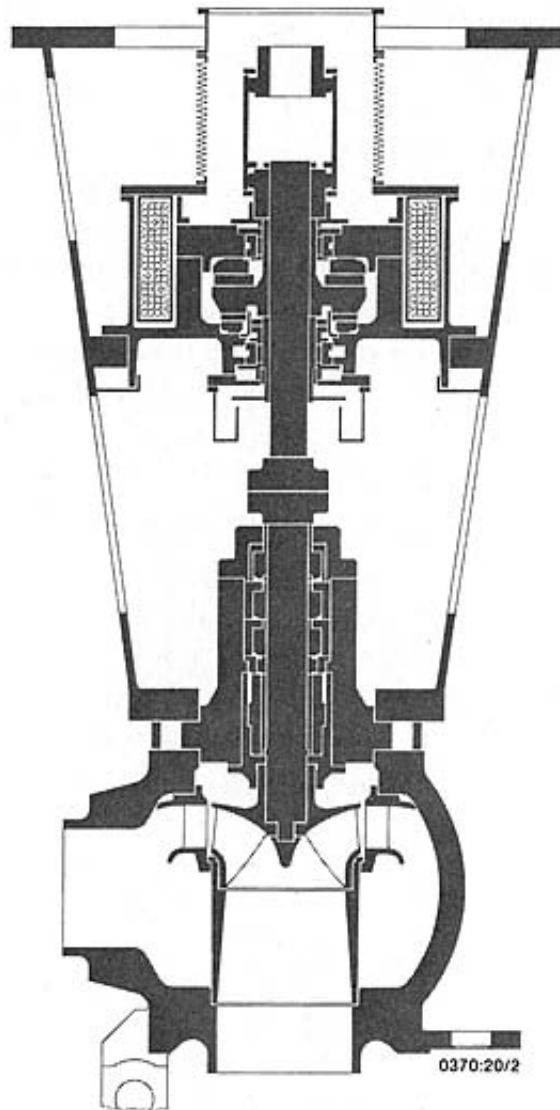


Fig. 3: Reactor coolant pump with heavy duty casing

In the case of *liquid metal cooled reactors* (sodium coolant), r.p.'s with a free liquid surface in the casing tube between the shaft seal and the pump impeller are used (Fig. 7). The free space is filled with an inert (noble) gas to prevent any sodium reactions, and the shaft seal is required to seal against this protective gas, and not against the liquid metal. The pump shaft is guided in a liquid metal lubricated hydrostatic bearing (plain bearing) situated next to the impeller. The system pressure (design pressure) in this case is 10 bar for example, and the design temperature is 580 °C.

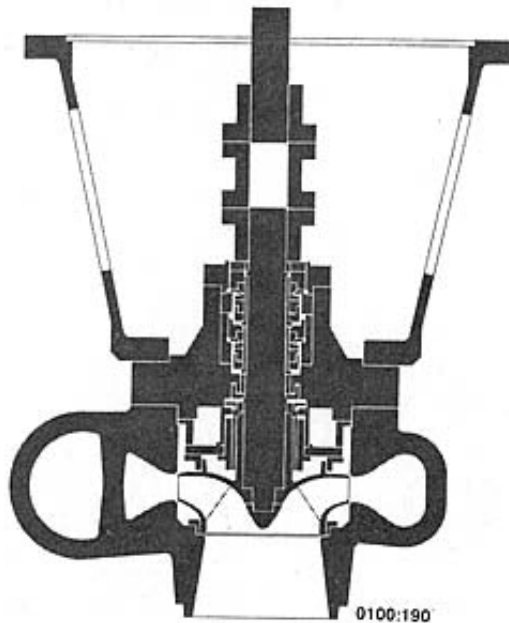


Fig. 4: Reactor coolant pump (driving water pump) for boiling water reactors

Apart from the r.p.'s proper, centrifugal pumps are also required in the auxiliary circuits and safety circuits of the various types of reactor.

Among the most important auxiliary and emergency systems are the volume control system, the decay heat removal system with high pressure and low pressure safety injection, the fuel rod storage cooling system, the secondary reactor loop, and the water treatment system, etc.

Pumps for these types of loops must meet specific requirements that usually do not need to be accounted for in normal pump construction. Among these are:

- Extremely high branch forces and moments must be transferred from the pump to the foundation (branch loading).
- The pressure retaining parts must permit 100% volumetric inspection and lend themselves to a detailed stress analysis.
- A steep QH characteristic curve with an extremely large operating range and a small $NPSH_{req}$ value (net positive suction head) must be acquired.
- Because of the radioactivity in the pumped medium, leaks in the system are not acceptable (even at temperature shock).
- - To minimize possible exposure to harmful radioactive rays, the system must be simple to service.
- - Continued operation even in cases where the building is affected by earthquakes or a plane crash. Possible results of such an incident could be flooding of the pump room, shutdown of the cooling water source for the mechanical seals (shaft seals) and the bearings, and a room temperature of 100 °C with a relative humidity of 100 %.

Figs. 8 and 9 show typical pumps for this service.

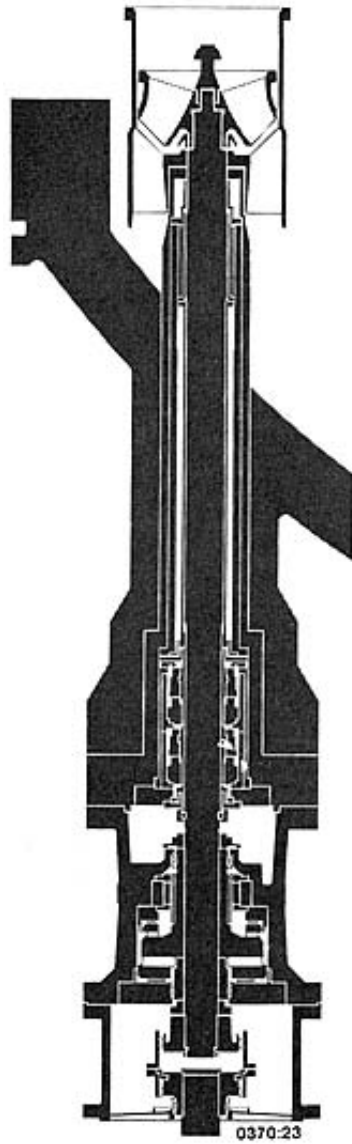


Fig. 5: Insert pump with wet rotor motor for boiling water reactors

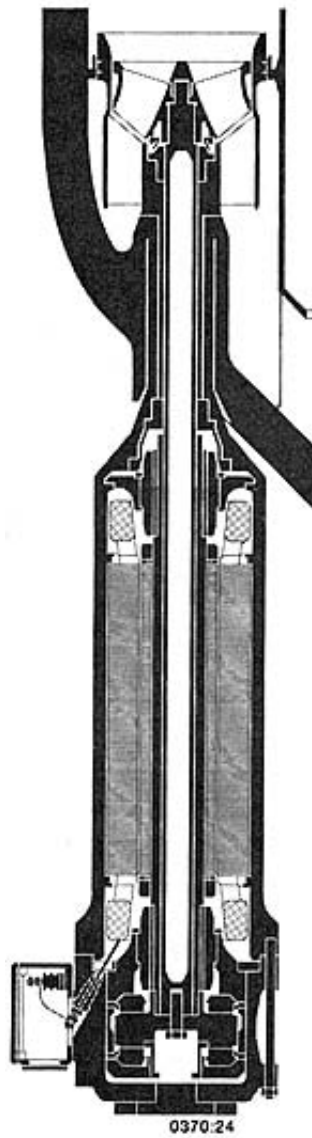


Fig. 6: Insert pump with wet rotor motor for boiling water reactor

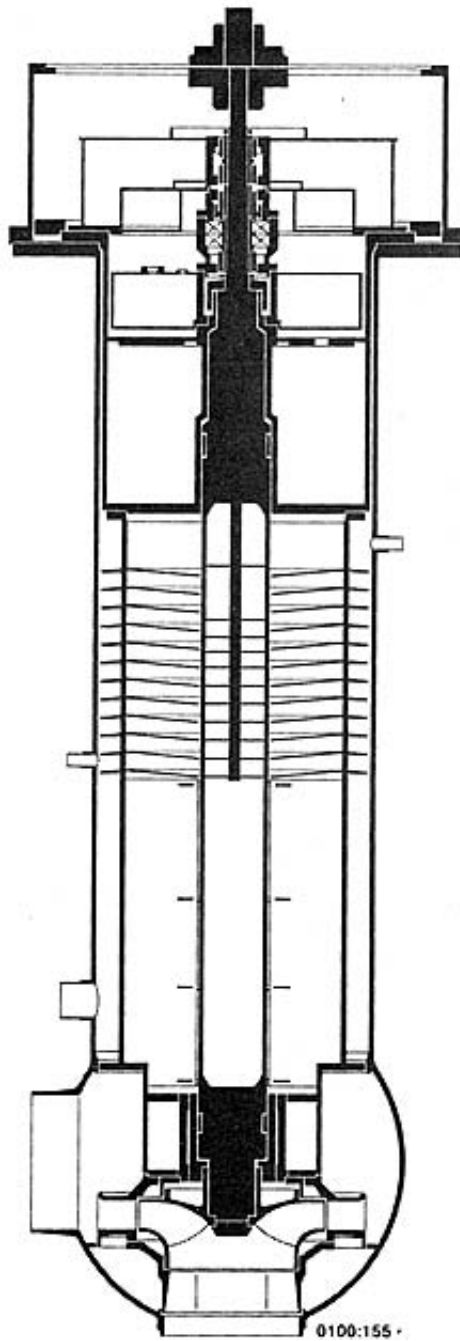


Fig. 7: Reactor coolant pump for liquid metal cooled reactor

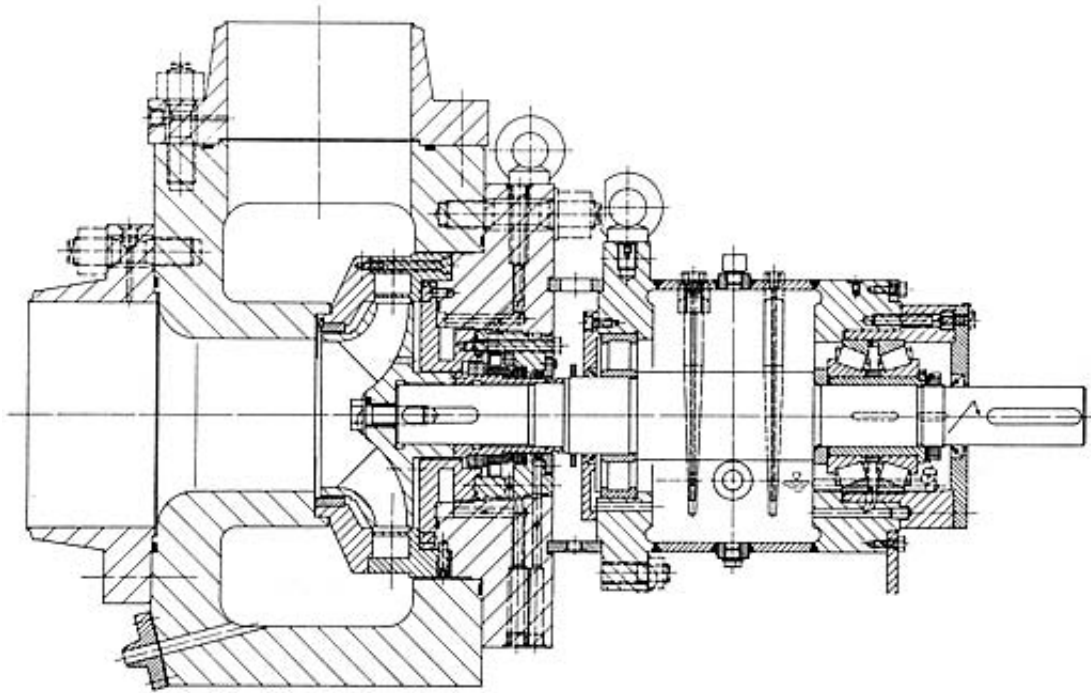


Fig. 8: Residual heat removal pump for auxiliary and safety circuits, used to supply cold borontreated water to the system so that the decay heat after reactor shutdown can be carried away

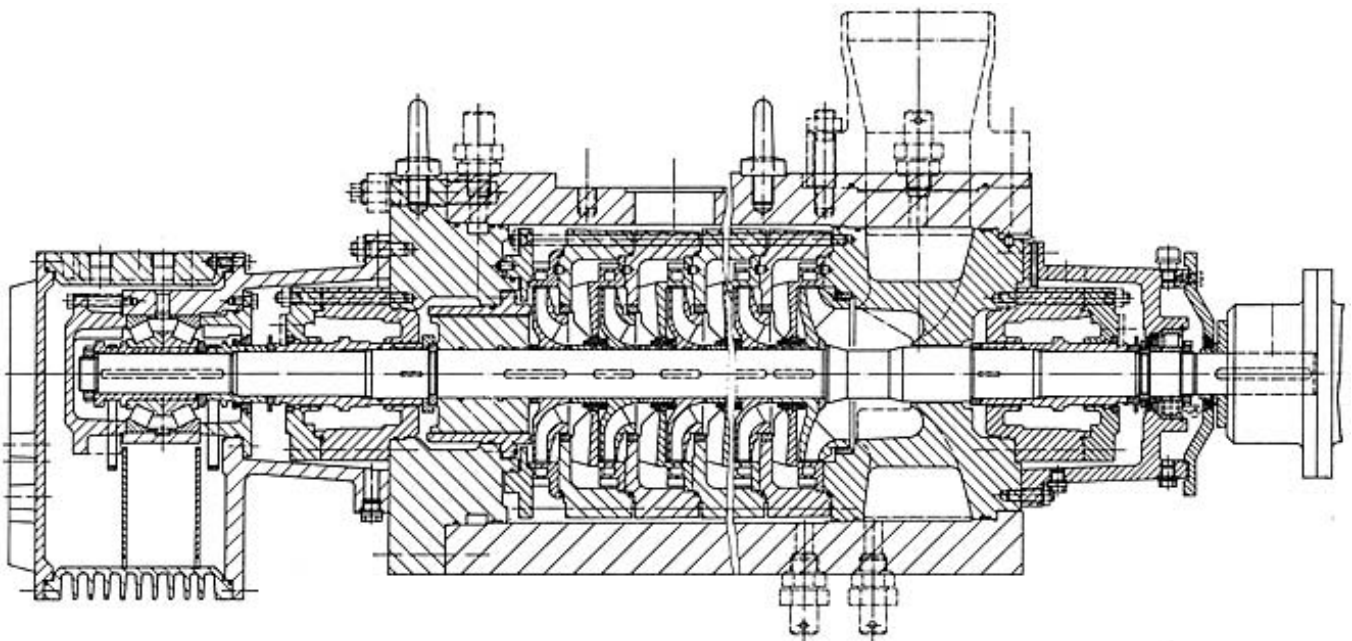


Fig. 9: High pressure safety coolant injection pump for auxiliary and safety circuits to make up for coolant losses by supplying cold, boron-treated water

Refinery Pump

Raffineriepumpe
Pompe de raffinerie

R.p.'s are pumps for the handling of crude petroleum and its refined products, as used in oil refineries, petrochemical plants and the chemical industry. R.p.'s operate at temperatures ranging from $-120\text{ }^{\circ}\text{C}$ to $+450\text{ }^{\circ}\text{C}$ and at pressures up to 65 bar approx. As the fluids handled are often highly volatile and flammable, only ductile construction materials are used for the pump components in contact with the fluid, e.g. unalloyed cast steel or cast

chrome steel, and more rarely spheroidal graphite cast iron.

The required NPSH value (net positive suction head) is of exceptional importance, and it governs the selection of the rotational speed and of the pump type.

R.p.'s are predominantly built as single stage horizontal volute casing pumps (Fig. 1) in process type construction. Depending on the operating conditions, particularly with regard to suction behaviour, horizontal or vertical multistage pumps of can-type (Fig. 2) are sometimes used, also horizontal double suction pumps with outboard bearings (Fig. 3).

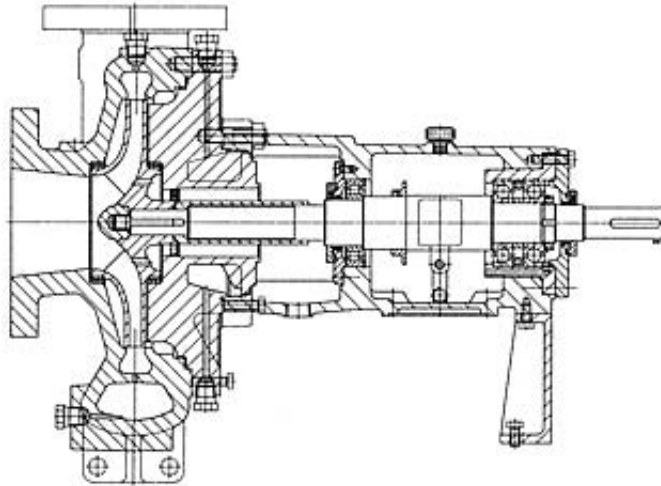


Fig. 1: Process type refinery pump without shaft seal

Certain design codes usually apply to r.p.'s. The best-known are the API 610 code of the American Petroleum Institute and the VDMA 24297 code of the German Engineering Works Association. These codes describe the r.p.'s, designated as "heavy-duty constructions" pumps, in greater detail. They lay down certain design features applying to different operating temperatures, e.g. the arrangement of the pump feet and nozzles, the mode of splitting of the casing and the sealing arrangements, also the temperature limits for cooling of the bearings.

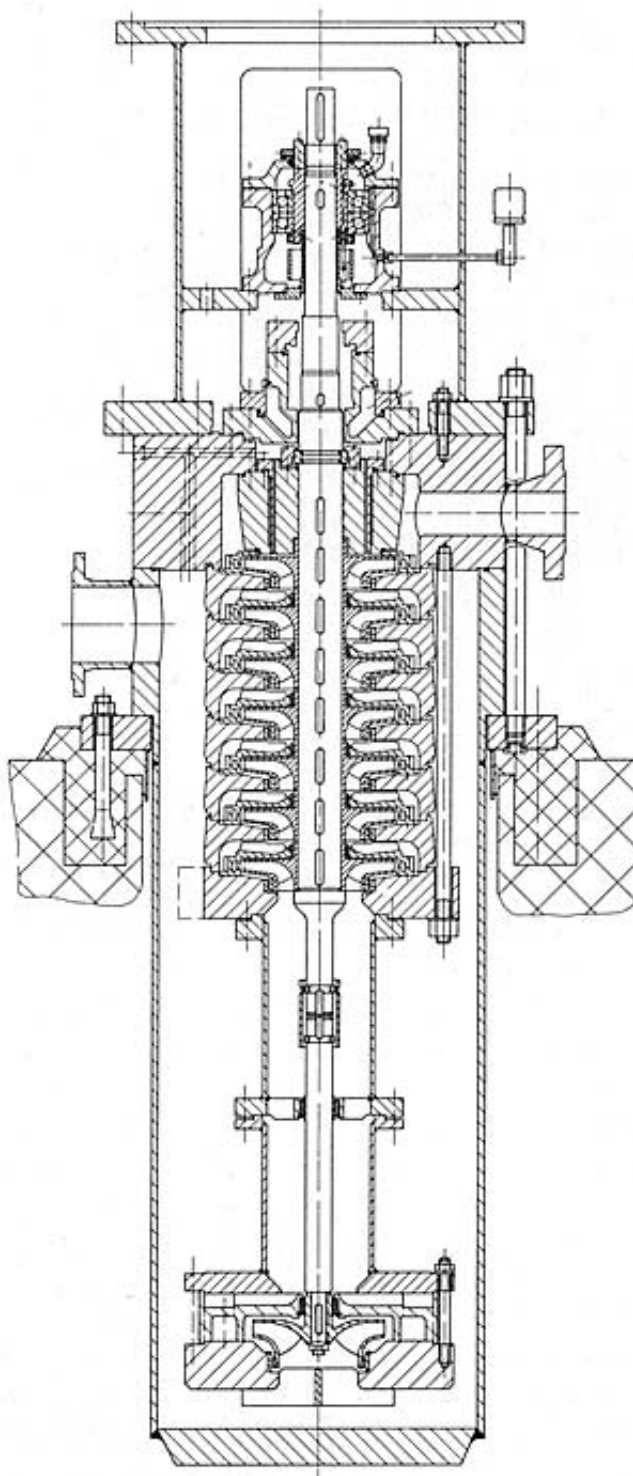


Fig. 2: Vertical can-type multistage refinery pump shown without shaft seal

The shaft seals of r.p.'s are usually mechanical seals of various types and arrangements.

Great emphasis is placed on sturdy and heavy construction, because the usually very hot pipings exerts considerable forces and moments on the r.p. and on its baseplate (pump foundation). These forces and moments must be capable of being absorbed safely without warping the pump or its baseplate (branch loading).

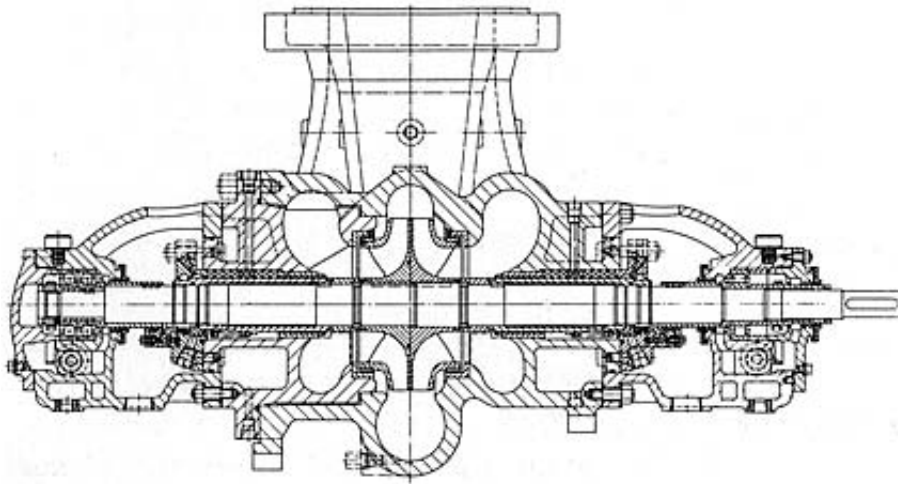


Fig. 3: Double suction refinery pump

Regulation by Inlet-Vortex

Vordrallregelung
Régulation par prérotation

see [Control](#)

Regulation by Vortex

Drallregelung
Régulation par prérotation

see [Control](#)

Relative Velocity

Relativgeschwindigkeit
Vitesse relative

The concept of r.v. is mainly used in fluid flow machine design. The r.v. w is the velocity of a fluid particle in relation to the rotating blade. The r.v., together with the absolute velocity v and the circumferential velocity u forms the velocity triangle.

Relay

Relays, Schütz
Relais, contacteur

see [Remote Control Switch, Teleswitch](#)

Remote Control Switch, Teleswitch

Fernschalter
Téléinterrupteur

R.c.s. is a switch (electrical switchgear) with a so-called power drive (in contrast to manually or pedal-operated switches). In a r.c.s., the force or torque of a motor, of a solenoid or of a pneumatically or hydraulically operated system is transmitted onto the actuating shaft of the switch. The contacted is a r.c.s., the throw-out mechanism of which resets itself automatically in the initial position when the driving force ceases to act on it (e.g. switching off by a push button switch).

In automatic pumping plants, the r.c.s. can be controlled by: float switches and water level electrodes in function of the water level; pressure switches or contact pressure gauges in function of the pressure; time switches in function of the time; differential pressure switches on flow meters (measuring technique) or switching flaps in function of the rate of flow (capacity); thermostats in function of the temperature.

Return Guide Vane

Rückführschaufel
Aube de retour

The r.g.v.'s on a multistage radial pump (boiler feed pump) form the so-called return stage, a guiding device situated downstream of the diffuser. The return stage is traversed by the fluid with a radially inward-directed flow, so that the fluid is led towards the next stage radial impeller without any swirl (rotation-free) (inlet conditions).

Reverse Flow

Rückstrom
Courant de retour

see Surge Pressure

Reverse Rotational Speed

Rücklaufdrehzahl
Vitesse de rotation inverse

R.r.s. is the highest rotational speed that a centrifugal pump can attain when "windmilling", if the medium is flowing through it in the reverse direction. This situation can come about in systems whose system characteristic curve have high static head $H_{A,0}$, but also in a centrifugal pump that is running in parallel operation: after interruption of the drive rating and when the discharge line is open, the medium will reverse its direction through the pump, and the pump rotor, after changing directions, will attain a reverse rotational speed (turbine operation, and Fig. 5 under characteristic curve) that is higher than the normal operating speed, and that, aside from the plant conditions (especially the current pressure head), is dependent on the specific speed n_q of the pump: the r.r.s. for radial flow pumps ($n_q = 40 \text{ min}^{-1}$, impeller) is approximately 25% higher than the normal operating speed of the pump, and for axial pumps ($n_q \geq 100 \text{ min}^{-1}$) it is approximately 100 % higher.

These operating conditions can also appear if a slow-closing shutoff valve is used for protection against a pressure surge (surge pressure), instead of using a check valve (valves and fittings). This allows the back-flowing medium to run through the centrifugal pump. If the backflow is caused by a power loss to the drive, the flowing medium will cause the pump shaft to run at the r.r.s., if there is no anti-reverse ratchet available.

If the back-flowing medium is near its boiling point (vapour pressure), it can vaporize in the pump or in the throttle valve on the pump discharge branch. The rate of the r.r.s., when operating with steam and liquids, can climb to dangerously high values, as a function of the square root of the ratio of the density of fluid/vapour.

If the motor of a centrifugal pump is turned on when running in the above-mentioned reverse conditions, the start-up period (starting process) of the system is lengthened considerably.

With asynchronous motors it is advisable to watch the additional temperature rise of the motor in such conditions.

Excessively high r.r.s., that can lead to damage in the equipment, can be prevented in the following manner:

The attachment of a mechanical anti-reverse ratchet on the pump shaft. The anti-reverse ratchet works according to the principle of a freewheel, and blocks the pump rotor once it starts rotating in a reverse direction.

Adding a safe automatically closing check valve (valves and fittings) in the piping system:

when the medium starts to flow backwards through the pump, reverse flow is automatically avoided.

Reverse Rotation Locking Device

Rücklaufsperre
Dispositif anti-dévireur

see Reverse Rotational Speed

REYNOLDS Number

REYNOLDS-Zahl
Nombre de REYNOLDS

see Model Laws

Ring-Section Pump

Gliederpumpe
Pomps à corps segmenté

see Multistage Pump

Roller Vane Pump

Drehflügelpumpe
Pompe à palettes entraînées

see Positive Displacement Pump

Rotational Speed

Drehzahl
Vitesse de rotation

R.s. is not a speed within the strictly technical definition of the word (i.e. distance divided by time), but is the frequency of rotation (frequency). The SI unit of r.s. is s^{-1} , but a more usual and permissible unit is min^{-1} .

The r.s. of a pump is the frequency of rotation of the pump shaft.

The r.s. of a pump is always a positive magnitude. The direction of rotation of the pump must be specified separately as clockwise or anti-clockwise. Clockwise rotation of the pump shaft presupposes that the direction of rotation is viewed from the drive end onto the pump. In case of doubt, the viewing direction should be indicated by means of a sketch. The direction of rotation of the impeller is defined independently of the direction of rotation of the pump: a right-handed rotation impeller means clockwise rotation of the impeller viewed from the impeller inlet. The pump r.s. must not be confused with the specific speed n_q .

Usual pump r.s.'s lie between 1000 and 3000 min⁻¹, but if special speed-increasing gears or a turbine drive are provided, they often attain 6000 min⁻¹ or more; in the case of very large centrifugal pumps, e.g. cooling water pumps for power stations. slow-running electrical drives tend to be very expensive. In such cases it is preferable to install a speed-reducing gear between drive and pump. The lowest pump r.s.'s arising today in such cases are situated around 200 min⁻¹.

The selection of pump r.s. is closely bound up with questions of pump hydraulics (circumferential velocity, impeller, specific speed) and also with questions of mechanical strength and of the economics of the pump and drive as a whole.

The angular velocity ω often appears in connection with the r.s., ω being the quotient of plane angle over time interval. The SI unit (unit) of ω is rad/s. 1 rad (radian) is equal to the plane angle (equivalent to 57.296°) which as sector angle (centre angle) intercepts an arc of 1 m length on a circle of 1 m radius.

The SI unit 1 rad = $\frac{1 \text{ m (arc)}}{1 \text{ m (radius)}}$ is in practice replaced by the figure 1. The following relationship applies between n and ω :

$$\omega = \frac{\pi \cdot n}{30}$$

with

ω angular velocity in s⁻¹ and
 n r.s. in min⁻¹.

Rotational Speed Transmitter

Drehzahlgeber
Capteur de finesse

see Measuring Technique

Rubber Hose

Gummischlauch
Tube flexible en caoutchouc

see Pressure Loss

Rubber-Lined Pump

Gummierte Pumpe
Pompe protégée par un revêtement

R.l.p. is a centrifugal pump, the components of which in contact with the fluid pumped are provided with a rubber lining (Fig. 1 under chemical pump). A distinction must be made between soft rubber lining (multilayer lining, mainly intended as protection against abrasion) and hard rubber lining (single layer lining, mainly intended as protection against corrosion pulp pumping).

S

Saturation Pressure

Sättigungsdruck
Pression de vapeur saturée

see [Vapour Pressure](#)

Screw Pump

Schraubenspindelpumpe
Pompe à vis

see [Positive Displacement Pump](#)

Sealing Fluid

Sperrflüssigkeit
Liquide de barrage

see [Boiler Feed Pump](#), [Shaft Seals](#)

Seals

Dichtungen
Joints d'étanchéité

S. in [centrifugal pumps](#) are generally designed to ensure separation between selected spaces or from the outside of structures in leak-tight fashion or with a minimum of leakage. In general terms, s. are still referred to as s. even if absolute leak-tightness is not achieved (throttling). Thus the concept of s. can be said to encompass in decreasing order of leak-tightness absolutely leak-tight s. (welding together of sealing faces), through non-welded s. (e.g. flat gaskets, ground s., O ring s.) down to throttling gap constructions with a given leakage rate.

All types of s. are to be found in centrifugal pumps, starting with their connection to the [piping](#), through to pump internal s. They include s. with

1. *stationary* mating sealing faces,
2. sealing faces which *slide* against one another,
3. sealing faces which *adjust* themselves in relation to one another within limits, and
4. sealing faces which *rotate* against one another.

Type 1: S. with stationary mating sealing faces can be subdivided into *non-disconnectable* s. such as welded flanges or welded sealing faces, and *disconnectable* s. such as flat gaskets (flexible materials), chambered s. (O rings etc.), convex head s. (metallic s.) and s. fed with sealing liquid (for toxic or explosive media, or if there is a danger of air ingress at pressures below atmospheric).

[Centrifugal pumps](#) are usually connected to the [piping](#) by means of s. with stationary mating sealing faces, although s. of type 2 or 4 are sometimes used.

Type 2: Sealing faces which slide against one another in order to compensate changes in length due to erection or operational reasons (temperature fluctuations) consist e.g. of two concentric tubes sealed against one another by a stuffing box.

Type 3: Sealing faces which adjust themselves in relation to one another within limits may be sealed by means of bellows, diaphragms and expansion joints (to limit the forces acting on the nozzles or to provide noise insulation of the pump casing from the pipng). This type of s. also includes connection and sealing by hoses.

Type 4: Shaft seals. Depending on the function of the sealing faces in contact with one another, such as "sealing", "centering" or "alignment" it becomes necessary to adjust the seal construction to perform these functions. Operating requirements such as thermal shocks or cold shocks must be taken into account, also the forces (which can sometimes be quite appreciable) necessary to fulfil the "sealing" function. Thus an axially sealing O ring requires a relatively small additional axial pre-stressing force, whilst a radially sealing O ring requires no such force. In contrast, an axially sealing flat gasket requires a greater axial pressing force, and a ground metallic sealing face requires the highest axial pressing force of all. S. at locations such as drain plugs have the sole function of "sealing" and can therefore be very flexible, because they have no alignment function. S. not exposed to a great pressure differential, such as s. on bearing covers of bearing brackets can be designed very simply. Rotating s. mounted between stationary sealing faces (e.g. shaft protecting sleeve against shaft) are subjected to centrifugal forces as well as to differential pressure forces, and these can be absorbed by suitable mounting features (e.g. chambered seal).

Seawater

Meerwasser
Eau de mer

see Table of Corrosion Resistance

Seawater Desalination Plant

Meerwasserentsalzungsanlage
Installation de dessalement d'eau de mer

Seawater desalination has been gaining steadily in importance in recent years. There are a number of processes in existence for the conversion of the raw water pumped out of the sea into a distillate which can be used e.g. as drinking water after suitable treatment. They include:

- distillation processes,
- freezing processes,
- solvent extraction
- reverse osmosis,
- electrodialysis,
- solar exchange processes.

Processes of technical and practical importance include the multistage flash distillation process, or MSF process for short (among the various distillation processes), and in reverse osmosis process, or RO process for short.

MSF process (Fig. 1). The raw water taken out of the sea (by means of a pump) is gradually heated up as it passes through the condenser coils (heat exchangers) of each individual stage (in Fig. 1, from stage 10 to stage 1). After leaving stage 1, the temperature of the raw water is increased still further in the brine heater (heat exchanger). The brine heater receives its thermal energy e.g. from the waste steam of a process plant or of a power station (combined heating and power station, coal-fired power station etc.), in conjunction with which the installation of a s.d.p. is undertaken in the majority of cases. The brine, which is now at a high temperature, then flows through the evaporation chambers of stages 1 to 10, releasing steam (vapour).

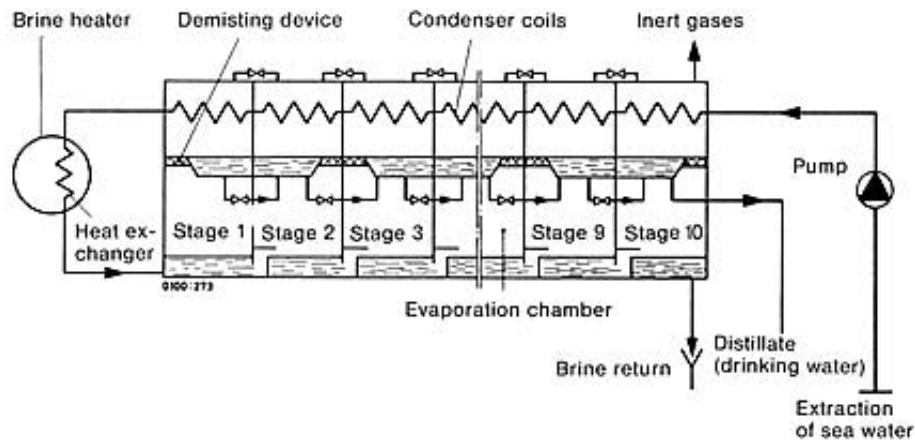


Fig. 1: Multistage flash distillation plant (MSF)

The pressure is decreased from one stage to the next, the temperature of the brine decreases accordingly, and the salt concentration increases. The vapour rises through the demisting device of each stage and condenses on the condenser coils. The condensed water (distillate) is collected in the troughs situated beneath the coils, and is led away for further treatment after the final stage. The following types of centrifugal pumps are mainly used for the above process: tubular casing pumps (as cooling water pumps and brine circulating pumps), volute casing pumps of process type design (as distillate pumps and decarbonizing pumps), double suction volute casing pumps (multisuction pump) (as brine extraction pumps) and vertical condensate pumps (as condensate dewatering pumps).

Reverse osmosis (Ro). The osmosis process is based on the following principle:

In aqueous salt solutions of differing concentrations, which are separated from one another by a semi-permeable membrane, water from the low concentration solution will penetrate through the membrane into the zone of higher concentration (Fig. 2). This phenomenon, which occurs in nature and has been known for a long time as osmosis, continues until an equilibrium has been attained between the two concentrations, or alternatively until a given pressure, the so-called osmotic pressure has built up on the side of the higher concentration.

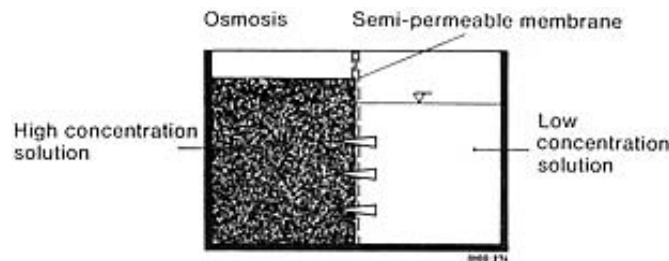


Fig. 2: Osmosis principle

If the pressure on the side of the higher concentration is increased to a value above that of the osmotic pressure, the phenomenon will occur in the reverse direction (reverse osmosis Fig. 3): very pure water with only a very low salt content will penetrate through the membrane from the side of the high concentration salt solution. This water, known as permeate, can be used as drinking water, whilst a concentrated salt solution is left on the high pressure side of the membrane.

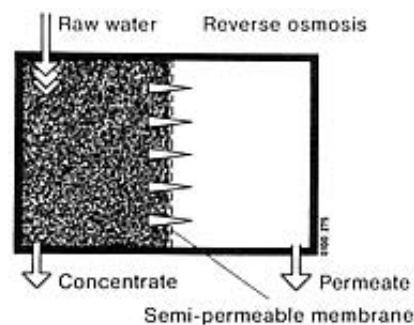


Fig. 3: Principle of reverse osmosis

In s.d.p.'s operating according to the reverse osmosis principle, the following types of centrifugal pumps are mainly used: volute casing pumps of process type design (as pressure boosting pumps, filter reverse flow flushing pumps and permeate pumps), also horizontal multistage pumps (as high pressure pumps).

Seawater Pump

Seewasserpumpe
Pompe à eau de mer

S.p. is a centrifugal pump for seawater for cooling purposes (marine pump) made of materials resistant to seawater (corrosion table of corrosion resistance, selection of materials).

Selection of Materials

Werkstoffauswahl
Sélection des matériaux

In s.o.m. for individual parts it is important to know both the *product form* and the *loading conditions*. This includes static and dynamic mechanical and thermal loading, as well as stress from erosion, abrasion, corrosion, and cavitation. Which types of loading affect Which materials can only be established by experience with pumps (application fields for pumps).

There are general Tables of individual applications for the s.o.m., which can be of help in the basic selection. Best known of these are so-called tables of corrosion resistance in which materials (mostly metallic) are listed in accordance to their resistance to corrosion in liquid media.

Aside from the product form and loading condition there are certain *economic aspects* that play a role in the s.o.m. However, individual cases are still handled from the point of *view safety* (e.g. reactor pumps for nuclear reactors).

Due to the computerized design of pumps, questions of *machinability*, and availability on the market are growing in importance.

The s.o.m. for a pump part and a particular case is usually the responsibility of the pump manufacturer and relies on his "know-how". Usage of materials may vary from manufacturer to manufacturer.

Self-Acting Circuit Breaker

Selbstschalter
Interrupteur automatique

The s.a.c.b., also called automatic circuit breaker is a type of electrical remote control switch. The s.a.c.b. incorporates a circuit breaker (electrical switchgear) and an overload protection. S.a.c.b.'s switch off automatically in the event of a short circuit or of overload, and in contrast to fuses, they are always instantly ready for switching on again.

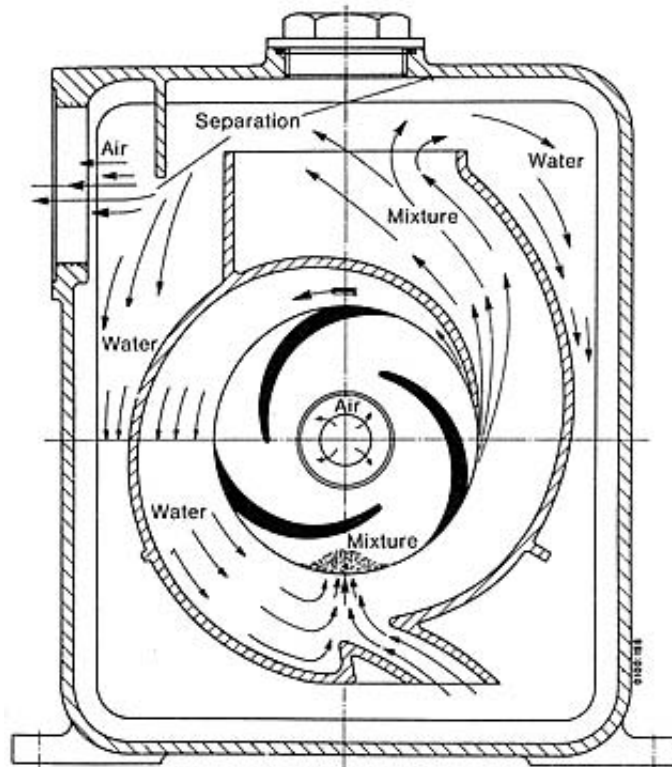
Self-Priming Pump

Selbstansaugende Pumpe
Pompe auto-amorçante

Centrifugal pumps are deemed to be selfpriming if they are capable of pumping liquids, gases and mixtures of liquids and gases; s.p.p.'s must be capable of venting the pump suction pipe (venting) of their own accord, without any external auxiliary priming devices. Centrifugal pumps equipped with an internal priming stage (e.g. a water ring pump) are also classed as s.p.p.'s.

Centrifugal pumps not equipped with an internal or external self-priming stage can be self-priming on condition that the pump is filled with fluid before the actual pumping process starts; in addition, a check valve (valves and fittings) on the suction side or a vent valve (breather) must be provided to ensure that no siphon action takes place and that the liquid remains trapped in the casing after the pump has been switched off. The centrifugal pump becomes selfpriming as a result of the pumping by the impeller of the operating fluid together with the entrained air bubbles into a separating chamber (see illustration). The air is able to escape towards the pump discharge branch whilst the operating fluid drops down again and is once more entrained by the impeller; the suction pipe is thus continuously vented. The design made necessary by the self-priming feature affects the pump efficiency adversely; in addition, the separating chamber occupies a relatively large space. Consequently this solution is only adopted on pumps with a low drive rating.

Frequently used types of s.p.p.'s include the side channel pump and the water ring pump.



Self-priming centrifugal pump without self-priming stage

Self-Regulation

Selbstregelung
Autorégulation

see Condensate Pump

Sense of Pump Rotation

Pumpendreh Sinn
Sens de rotation de la pompe

see Rotational Speed

Sensors

Sensoren
DéTECTEURS

S. transform values that are not in usable signal form into a usable form. They are built into circuits in the form of s. units, usually with additional features that allow them to function properly. These s. units are the active or passive parts that transform raw information into useful signals:

Active s. present a signal in the form of an electrical potential or a voltage or a current without the need of an electrical source on the receiving end. Passive s. change a given electrical parameter such as resistance, capacitance, or inductance. Optical s. change non-optical values into optical signals, that are then processed by onto-electronic devices.

S. systems are s. with built-in equipment to transform and process the signals.

Series Operation

Serienbetrieb
Marche en série

In the s.o. of centrifugal pumps I and II, the head H_{I+II} is the sum of the heads of the individual pumps

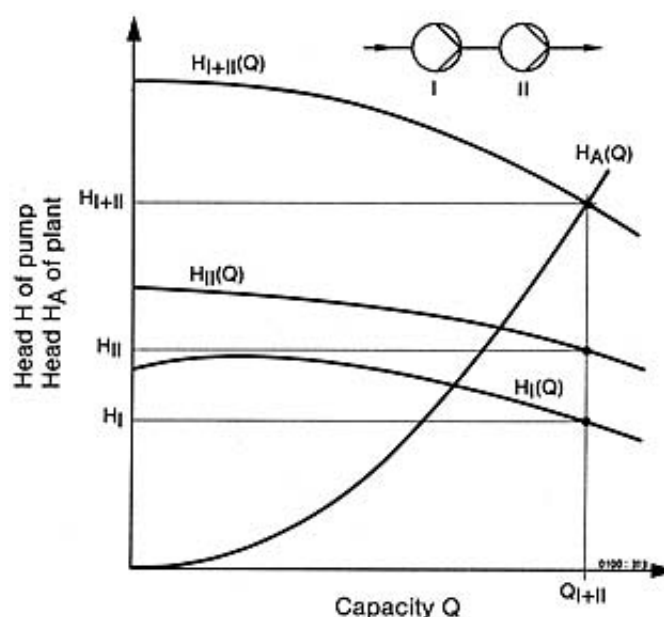
$$H_{I+II} = H_I + H_{II}$$

at the same capacity

$$Q_{I+II} = Q_I = Q_{II}.$$

S.o. has many advantages over parallel operation, if the system characteristic curve $H_A(Q)$ is steep, and the pump characteristic curve $H_{I, II}(Q)$ runs flat (characteristic curve). In the case of s.o. the addition of the head values is beneficial to the steep system characteristic curve as opposed to the addition of the capacities in a parallel system (illustration).

The characteristics of s.o. are easier to understand than those for parallel operation, and are not complicated by unstable throttling curves or varying shutoff heads (head).



Series operation of two centrifugal pumps I and II with any type of characteristic curves

In the s.o. of several centrifugal pumps one must watch that the pump casings and the shaft seals can hold up to the high pressure. The pump with the best suction behaviour should be placed in the lead position.

When starting up (starting process), the lead pump must create enough pressure before the next pump can be started up, in order to eliminate the risk of cavitation. When shutting down, no pump on the lower pressure side may be turned off, as long as the pump following it is still running, otherwise the flow will run through the non-rotating pump, as through a rotating throttle (characteristic curve), which will drive the available NPSH (net positive suction head) (for the following pump) down to an unacceptable level. Interlocking of the drive motors is sometimes recommended.

Not only the head but also the capacity, in each of the series pumps is enlarged after changing a pumping plant from single to s.o. In such a layout one must look out for a satisfactorily large NPSH value (net positive suction head).

When the before-mentioned criteria are met and the non-driven pump is allowed to be turned by the medium, or is circumvented by a by-pass, the s.o. has been economically adapted to stepwise control of the operation of centrifugal pumps.

If one does not choose the above-mentioned option of stepwise control, it is easier and cheaper to use a multistage pump, in which the impellers and diffusers of the pumps are mounted in a common pump casing instead of using several centrifugal pumps in s.o.

Sewage Pump

Abwasserpumpe

Pompe à eaux d'égout

The s.p., also called effluent or waste water pump, is a centrifugal pump designed for handling contaminated water (often containing solids), the chemical analysis of which can widely vary (organic, inorganic or mineral admixtures).

Typical fluids handled by a s.p. comprise: crude sewage (untreated sewage, faeces pump), treated sewage (mechanically cleaned water from settling tanks), sludge (activated sludge, raw sludge, sapropel, Imhoff tank sludge) and storm water. Sewage and effluent can be very corrosive in certain cases (selection of materials) or abrasive (abrasion); this must be taken into account when selecting the appropriate construction materials. Other application fields for s.p.'s are the building and construction industry (mine drainage pump, pumping of asbestos-cement water mixtures for building slabs etc.), the mining industry (mine drainage pump, pumping of mixtures of water and coal, ore or sludge, hydrotransport), the foodstuffs industry (pumping of mixtures of water and fruit, vegetables or grain), the paper industry (pulp pump, pumping of screening water, paper stock, fibrous material, mechanical wood pulp and cellulose of low pulp density, pulp pumping), the sugar industry (pumping of sugar beet, beet slices, slimy washwater, thin and thick juices of all kinds), the aluminium industry (bauxite sludges, white sludges, red sludges) and the metal electrolysis.

Thus various impeller shapes (impeller) are used, depending on the fluid pumped: the non-clogging impeller (Figs. 1 and 2), in the form of two passage and three passage impellers (either closed or open) the single vane impeller (Fig. 3) and the torque-flow impeller (Fig. 4) (torque-flow pump). The pump casing can be provided with easily renewable wear plates made from specially wear-resistant materials.

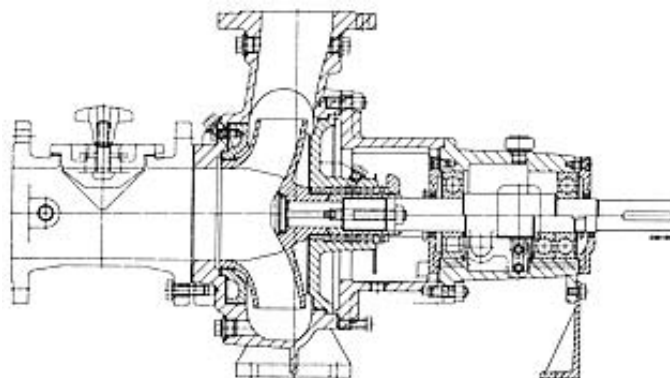


Fig. 1 : Horizontal sewage pump with open non-clogging impeller

S.p.'s are designed both as vertical and as horizontal pumps. In the case of vertical pumps, submersible motor pumps (Figs. 3 and 4) are being used in increasing numbers in recent years. The entire set is designed for underwater operation; the pumping part is separated from the dry motor by a seal-oil-filled, double-acting, hard-alloy mechanical seal (shaft seals). The unit is lowered into the sump on a guide future and connected to the discharge line by a positive-action mechanism.

S.p.'s of the submersible pump type (Fig. 2; vertical spindle sump pumps) are also known. In the case of horizontal pumps, the process type design design has gained greatly in popularity; close-coupled pumping sets are used for drive ratings up to 50 kW.

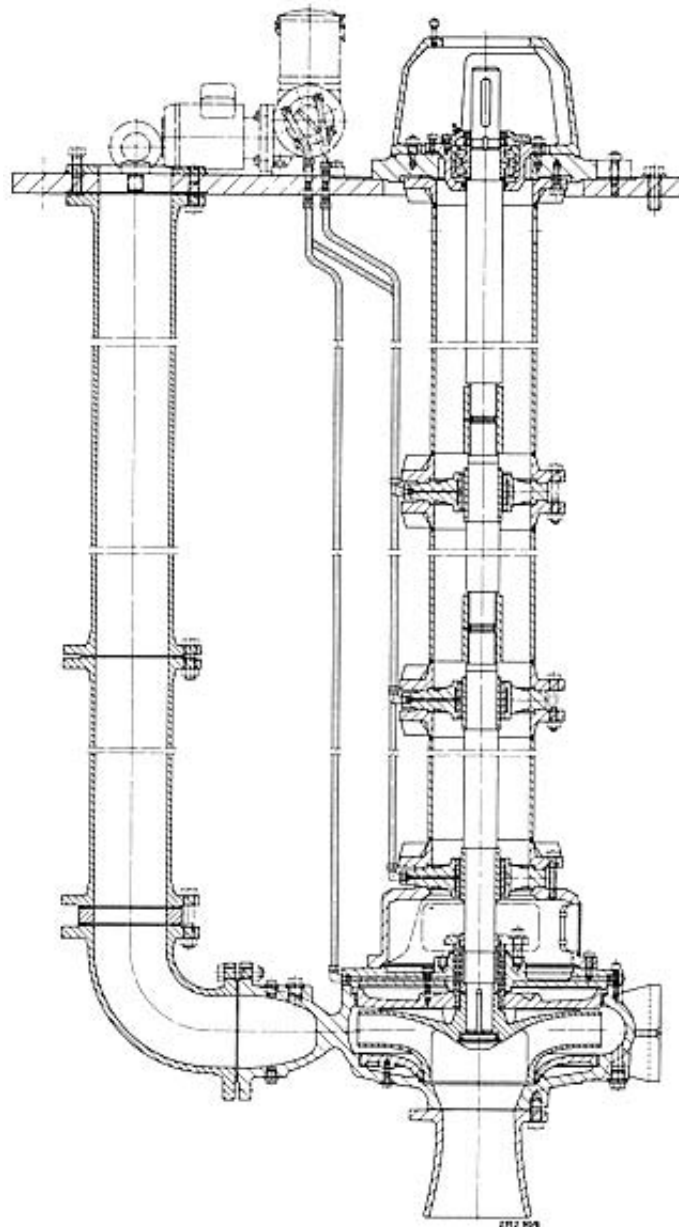


Fig. 2: Vertical sewage pump (wet installation) with closed non-clogging impeller

In the case of single vane impeller pumps (impeller), a V-belt drive (belt drive) is often adopted; this enables any required pump rotational speed (rotational speed) to be used in conjunction with a standard electric motor, and also enables the pump to be readily adapted to the required performance data. Single vane impellers can only be trimmed to a limited extent (cutdown of impellers). The nominal diameters of the pump suction and discharge branch are the same in this case.

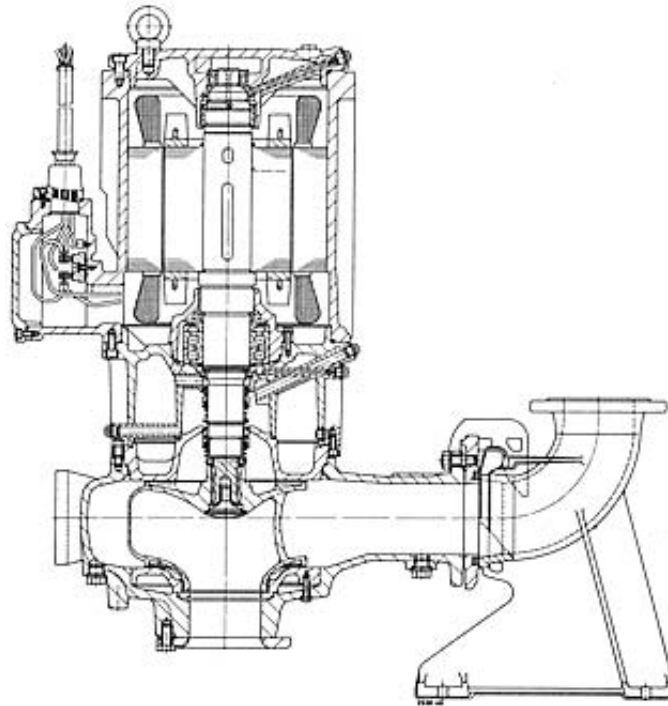


Fig. 3: Submersible motor pump with single vane impeller used as a sewage pump

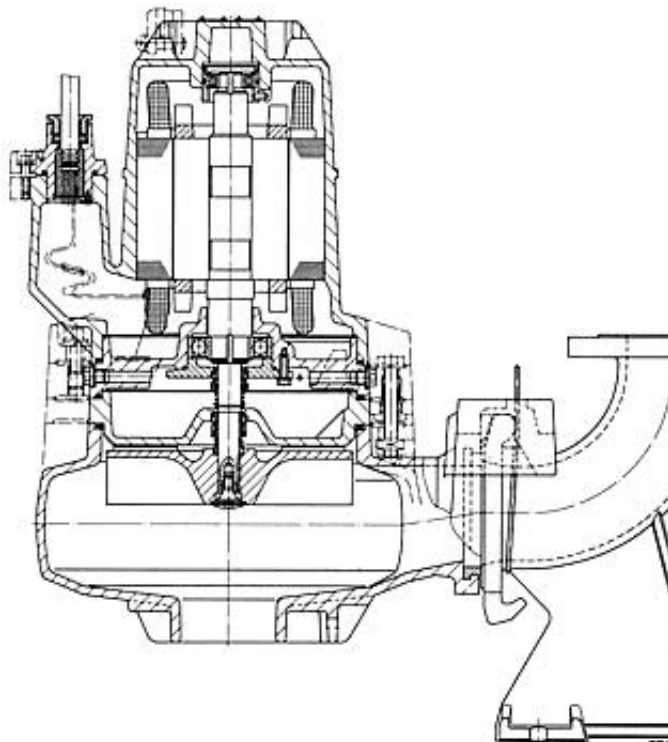


Fig. 4: Submersible motor pump with torque-flow impeller used as a sewage pump

The impellers are shaped in such way that the free passage of the medium to be handled is ensured. S.p.'s are preferably of single stage type; usually they are not self-priming (self-priming pump). Shaft sealing is effected by stuffing boxes fed with sealing water or wear-resistant mechanical seals.

The counter-part of the s.p. is the clean water pump.

Shaft Coupling

Wellenkupplung

Accouplement d'arbres

Slip-free s.c.'s used in centrifugal pump technology can be subdivided into rigid and flexible s.c.'s.

Rigid s.c.'s are mainly used to connect together shafts which are very accurately aligned. The smallest misalignment will cause considerable extra stresses in the s.c. and in the adjoining shaft ends. The following types are used (coupling alignment):

1. sleeve (thimble) couplings,
2. muff (butt) couplings,
3. serrated (splined) couplings,
4. clamp (split) couplings (DIN 115),
5. flange (face plate) couplings,
6. flange couplings.

Flexible s.c.'s. Flexible s.c.'s according to DIN 740 are flexible slip-free connecting elements between driver and driven machine which to a certain extent are capable of compensating axial, radial and angular misalignments (Fig. 1) as well as damping shock loads. The flexibility is usually achieved by the deformation of damping, rubber or metal-elastic spring elements, the service life of which is largely governed by the extent of the misalignment to be compensated.

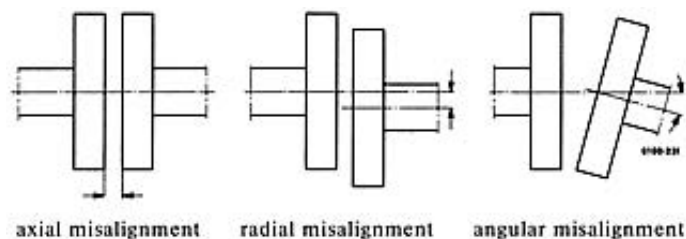


Fig. 1: Misalignments (examples)

Fig. 2 illustrates the most widely used types of flexible s.c.'s. A spacer type coupling on a pump is illustrated in Fig. 3; this type enables the pump rotor to be dismantled without disconnecting the pump casing from the piping or removing the driver from the baseplate (process type design).

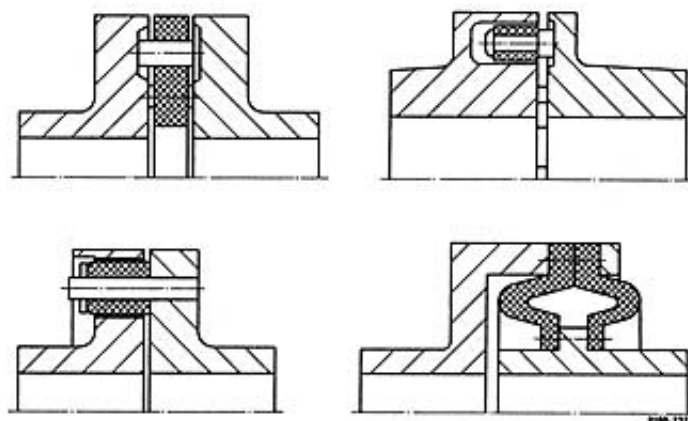


Fig. 2: Examples of coupling types

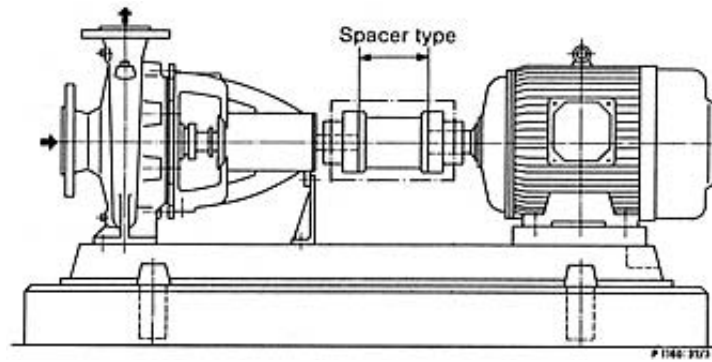


Fig. 3: Spacer type coupling

In the past s.c. were in most cases designed on the basis of the purely empirical documentation of the manufacturers. However DIN 740, Sheet 2 (1986 edition) features a method of calculation based on physical relationship. This applies in particular to the evaluation of the torque shock. If there are no precise data available on the influence of frequency of starts and of the surrounding temperature, the loading can be taken into account by suitable factors which are listed in the form of guideline values. No fixed value is laid down for the ratio of max. torque to operating torque, in order to be able to take all coupling types into consideration, according to their specific suitability. The calculation of the loading by torque shocks relates therefore to the max. torque (starting torque).

The detailed calculation method for sizing a flexible coupling given in DIN 740, Sheet 2 is only valid on the assumption that the coupling is the sole torsionally elastic member in the installation. In all other cases it will be necessary to carry out a vibration calculation.

Shaft Efficiency

Kupplungswirkungsgrad
Rendement à l'accouplement

see Pump Efficiency

Shaft Power

Leistungsbedarf
Puissance absorbée

The s.p. of a centrifugal pump is the mechanical power (measuring technique) absorbed at the pump coupling or pump shaft from the drive

$$P = \frac{\rho \cdot g \cdot Q \cdot H}{\eta}$$

with

ρ density of pumped medium,
 g gravitational constant,
 Q capacity of pump,
 H head of pump,
 η pump efficiency.

(In the event that the pumped medium exhibits any appreciable amount of compressibility, the calculation must be based on ρ and Q in the inlet cross-section of the centrifugal pump.)

The SI unit of s.p. is 1 W; it is however usual in centrifugal pump technology to adopt 1 kW as unit of s.p.

S.p. should not be confused with the drive rating, i.e. the available driving power.

Apart from the general concept of s.p., certain special concepts of s.p. appear in centrifugal pump technology:

Optimum s.p. P_{opt} is the s.p. at the operating point of optimum efficiency.

Maximum (or upper limit) s.p. P_G is the max. s.p. within the operating range mutually agreed in the supply contract.

Power absorbed at pump shut-off point P_0 is the s.p. at $Q = 0 \text{ m}^3/\text{s}$ (characteristic curve).

Shaft Protecting Sleeve

Wellenschutzhülse
Chemise d'arbre

The s.p.s is a structural element for the protection of the pump shaft against mechanical and chemical damage by shaft seals, bearing shells (plain bearing) and corrosive media (selection of materials, corrosion, erosion). The design of the s.p.s. must ensure that the medium pumped is not able to flow outwards between the shaft and the s.p.s., also that any axial thermal expansion is taken care of, and that the s.p.s. is secured against tangential and axial displacement relative to the shaft (preferably at its end closest to the impeller).

Shaft Seal Ring

Wellendichtring
Anneau d'étanchéité d'arbre

see Shaft Seals

Shaft Seals

Wellendichtungen
Garniture d'étanchéité d'arbre

As opposed to static seals, which seal nonmoving parts against gas or liquids, s.s. seal moving and non-moving parts (in this case the pump shaft) that run between areas of varying pressures so that the leakage loss or air intake are minimized. The seal should also minimize wear. The principle of the s.s. lies in the throttling of a long gap (slit seal), of minimal clearance (clearance gap width), between the rotating and non-rotating parts. This gap will reduce the pressure between the two areas of unequal pressure. There are also s.s. where this pressure difference is partially or completely neutralized by hydraulic means. Additional steps, such as introducing quench fluids or controlled collection of seal leakage, allow the s.s. to also prevent leaks to the outside, which is important in the transport of caustic chemicals, of polluting compounds, of radioactive materials and of flammable products.

The most important variants of s.s. will be discussed: one is to note the difference between contact type and non-contact type s.s.

Contact type s.s.: Lip s.s. and line contact s.s. are only suitable for very small pressure differentials, e.g. to seal off oil at the bearings (radial shaft seal rings); such s.s. are generally not adjustable (Fig. 1).

Packings (soft-packed stuffing boxes) are adjustable, and are used for larger pressure differentials and circumferential velocities of the rotating component (Fig. 2).

Mechanical seals are successfully used both for low and very high pressures and circumferential velocities (Fig. 3). Mechanical seal occupy less space than a soft-packed stuffing box and are maintenance-free (maintenance); therefore there is very little chance of mistakes in servicing being made. However they suffer considerable disadvantages from excessive wear when abrasive media are pumped. The abrasive particles can be kept clear of the sliding faces (exposed to the greatest danger of wear) by injecting clean sealing or flushing liquid (or flushing liquid cleaned by way of a cyclone).

Whether a stationary or a rotating seal ring is used depends amongst other things on circumferential velocity and type of construction of the mechanical seals.

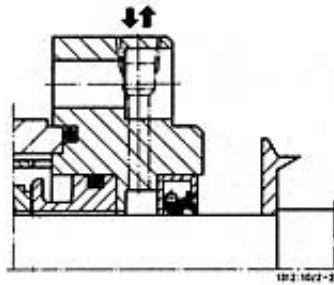


Fig. 1: Radial shaft seal ring

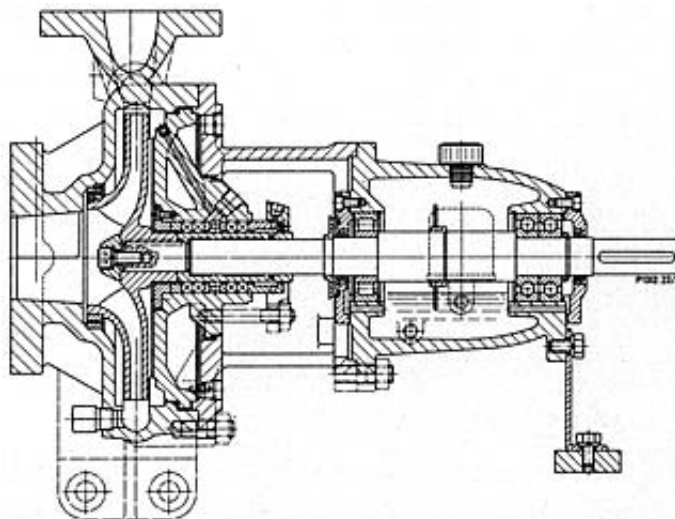


Fig. 2: Centrifugal pump with soft-packed stuffing box

The axial sealing force of balanced seals, caused by inside pressure, is lessened by smaller diameter of the pump shaft or by the shaft protecting sleeve (Fig. 3). This lessens the thermal loading of the sealing surfaces.

Mechanical seals can be arranged in different ways: single (Fig. 3) or multiple acting, faceto-face, back-to-back (Fig. 4 or as tandem arrangement Fig. 6), with or without injection of sealing liquid (for the stepwise reduction of the sealing pressure and the prevention of dry running). Dry running, which is very harmful to the mechanical seal, can arise if the pump is not primed before start-up, or if there is a sudden inrush of gas, if the gas content of the pumped fluid is high, or if the medium evaporates. Transmitters (piston transmitter) are often used for the transmission of the pressure to the sealing liquid in the mechanical seals.

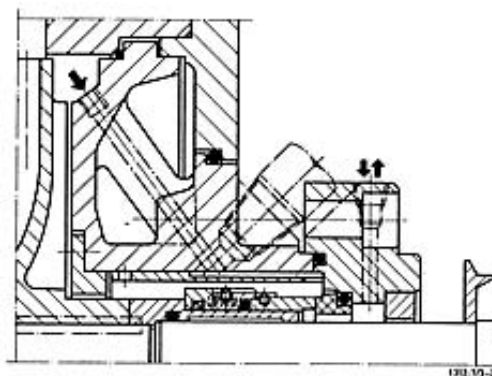


Fig. 3: Single acting balanced mechanical seal with circulation borehole and quench seal

The term "quench liquid" in conjunction with s.s. usually means a cooling fluid injected for the purpose of cooling down the fluid pumped surrounding the seal (Figs. 3 and 4). A quench seal is a compartment arranged downstream of the main seal (e.g. mechanical seal) through which water flows, and which is fitted with a simple seal (radial sealing ring or packing ring), the purpose of which is to positively lead away any leakage such as dangerous liquids or gases and also heat in some cases from the mechanical seal.

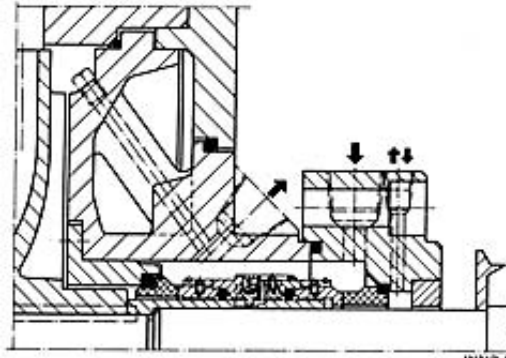


Fig. 4: Double acting balanced mechanical seal with quench seal and connections for sealing or flushing liquid

The friction heat of a mechanical seal, especially in the gap (clearance gap width) between the sealing surfaces (less than 1 μm) is usually transferred to the outside by the seal casing, but can also be carried away by the medium handled. This cooling process can be supplemented by a circulation flow, which is generated by pressure differences in the space between impeller shroud and casing (Fig. 3) or by rotating parts of the mechanical seal (so-called pump rings) or of the pump rotor (boiler feed pumps; with hot mediums, this circulating flow is led into a cooler that is either integrated into the seal casing or is externally located).

Noncontact type s.s. (preferably used for abrasive media, at very high circumferential velocities or where there is a requirement for total absence of wear): A distinction must be made here between throttling s.s. with leakage and pressure generating s.s.

In the first group we have:

- smooth surface throttling gaps with a very narrow gap width (dependent on the shaft deflection, vibrations and casing deformation caused by pressure and temperature), used e.g. for throttling on impellers and intermediate sleeves of ring section pumps (multistage pump),

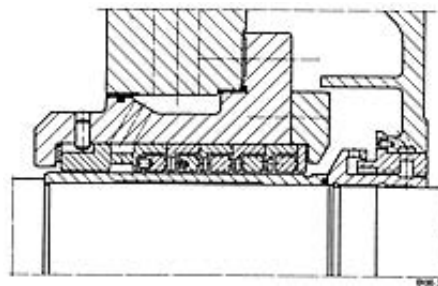


Fig. 5: Floating ring shaft seal (floating seal)

- throttling gaps with labyrinths (boiler feed pump) between shaft and casing, also shaped seal (on impellers and intermediate sleeves, also used as s.s.),
- s.s. (by float rings) which automatically throttle the leakage (floating seals, boiler feed pump) with a control gap (Fig. 5),

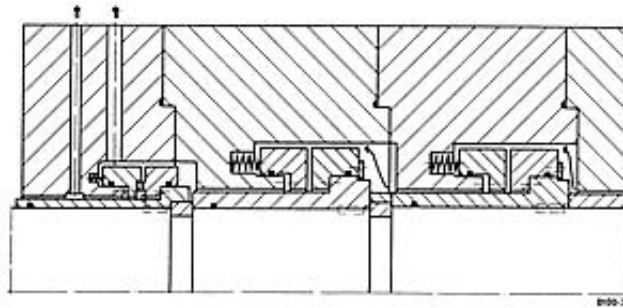


Fig. 6: Multistage hydrostatic mechanical seal for sealing high pressures

- mechanical seals with a well-defined clearance gap (control clearance gap) (slit seal) and a well-defined leakage rate (subdivision of the pressure gradient among several mechanical seals arranged in series, return of the leakage fluid to the pump suction branch or to the suction vessel), Fig. 6.

S.s. which generate a pressure to compensate for the pressure to be sealed is the other group of noncontact type s.s. Belonging to these are:

- Centrifugal seals or hydrodynamic s.s. (Fig. 7), often with a secondary seal. typically a springloaded mechanical seal disengaged by centrifugal forces at very low rotational speeds of the shaft, thus providing protection from wear. The actual hydrodynamic s.s. is non-contacting and wearfree.

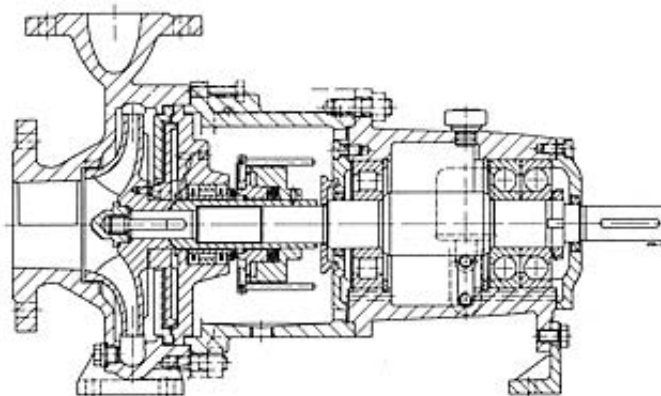


Fig. 7: Hydrodynamic shaft seal by auxiliary impeller with liquid ring and soft packing downstream to act as seal when pump is stationary

- Spiral grooves generate a pressure when the pump is in operation, and with additional seals can act as s.s. (Fig. 8). They can also be used in the decrease of pressure differences in close clearance gaps.

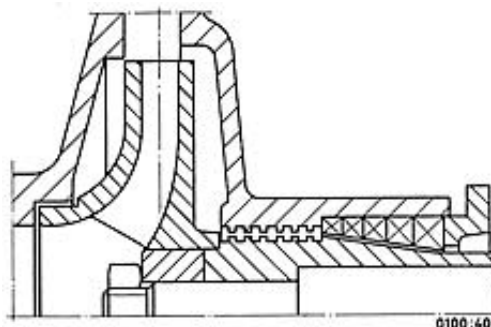


Fig. 8: Hydrodynamic shaft seal by threaded feed bushes with soft packing downstream for stationary sealing

Shock Condensation

Stoßkondensation
Condensation par choc

S.c. is the sudden and violent collapse of a vapour space in the event of cavitation. The implosion of the vapour bubbles results in the disintegration of the material akin to erosion (cavitation), sometimes accompanied by marked corrosion attacks (corrosion). S.c. occurs e.g. in the event of pressure surges (surge pressure) and of compression waves (sound velocity) in fluid flows with a high vapour content.

Shock-Free Entry

Stoßfreier Eintritt
Entrée sans choc

S.f.e. occurs in a cascade if the relative flow (relative velocity) of the medium entering the impeller, or the absolute flow upstream of the diffuser hits the blades tangentially; tangential, means in the same direction as the median line of the blade profile (Fig. 1 under cascade, flow profile). At the design duty point (operating point), s.f.e. usually occurs, but in certain special cases impellers are designed for non-s.f.e. (e.g. inducers). Nons.f.e. (which normally occurs at part loads and overloads operating behaviour) causes the so-called shock loss.

Shock Loss in Centrifugal Pumps

Stoßverlust bei Kreiselpumpen
Perte par choc dans des pompes centrifuges

If the angle of attack of a cascade deviates from the inlet angle of the (shock-free entry), the loss resulting from the forced change in velocity is designated shock loss or impact loss.

This phenomenon occurs when the capacity changes, i.e. when the meridian velocity changes (velocity triangle).

Let us assume that the calculated meridian velocity at the inlet to an impeller is $v_{m,0}$ (subscript 0 signifies: upstream of the cascade), and that the circumferential velocity is u_1 . Therefore, in the case of an irrotational approach flow, the blade is approached with a relative velocity w_0 . We also assume that the resulting relative inlet angle β_0 is identical with the direction of the blade inlet edge (Fig. 1). If we now reduce the capacity to the point where the meridian velocity becomes $v_{m,1}$, the flow is diverted from β_0 to β_1 . Related to the impeller the shock component $w_{\text{shock.La}}$ causes a head loss of

$$H_{\text{shock.La}} = \zeta_{\text{shock.La}} \cdot \frac{w_{\text{shock.La}}^2}{2g}$$

with

$\zeta_{\text{shock.La}}$ shock coefficient, related to impeller inlet,
 g gravitational constant.

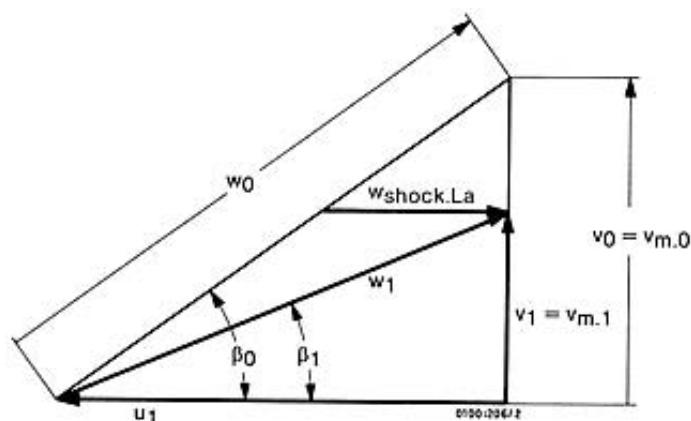


Fig. 1: Velocity triangles at impeller inlet with shock component $w_{\text{shock.La}}$

An analogous consideration can be used when evaluating the shock losses in the diffuser. In that connection, it must be noted that, contrary to the case of swirl-free flow incident to the impeller inlet, the direction of absolute flow is not independent of the capacity. In approximation, it may be assumed that the direction of the relative flow $w_{2,La}$ departing from the impeller remains constant within the range $0.75 Q_{opt} \leq Q \leq 1.25 Q_{opt}$ (Figs. 2 and 3).

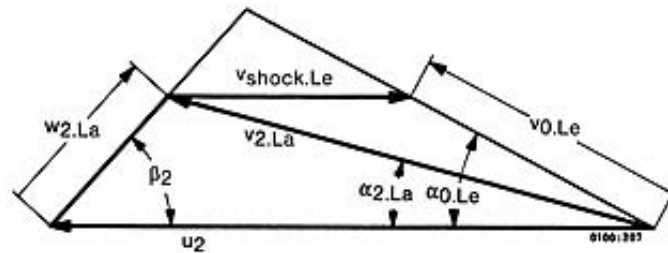


Fig. 2: Velocity triangles at impeller outlet (subscript 2. La) and upstream of diffuser inlet (subscript 0. Le) showing the shock component $v_{shock,Le}$

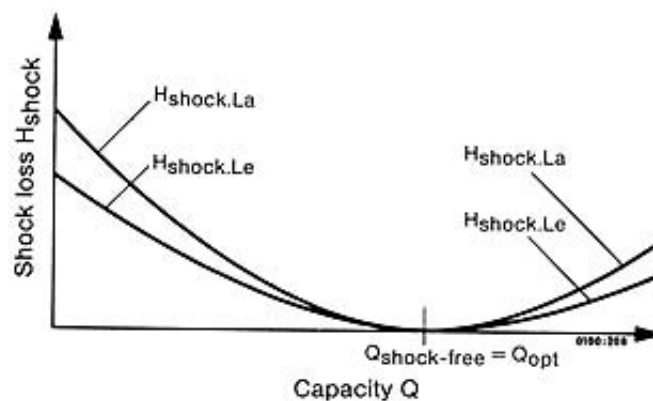


Fig. 3: Shock losses in the impeller and diffuser in function of capacity. $Q_{shock-free}$ does not necessarily correspond to the same value for the impeller as for the diffuser. $Q_{shock-free}$ will differ from Q_{opt} by a corresponding amount

When calculating the shock losses, it should be remembered that the shock coefficient in the part load region is greater than in the overload region (operating behaviour of centrifugal pumps).

This can be explained by the fact that for $Q < Q_{opt}$ separation occurs as a result of the shock approach flow, particularly in the outer region of the cascade, and this separation finally leads to a reflux of the medium at the inlet when the shock approach flow becomes extreme (operating behaviour of centrifugal pump). CARNOT's shock loss is caused by other factors.

Shut-Off Valve

Absperrorgan
Appareil d'arrêt

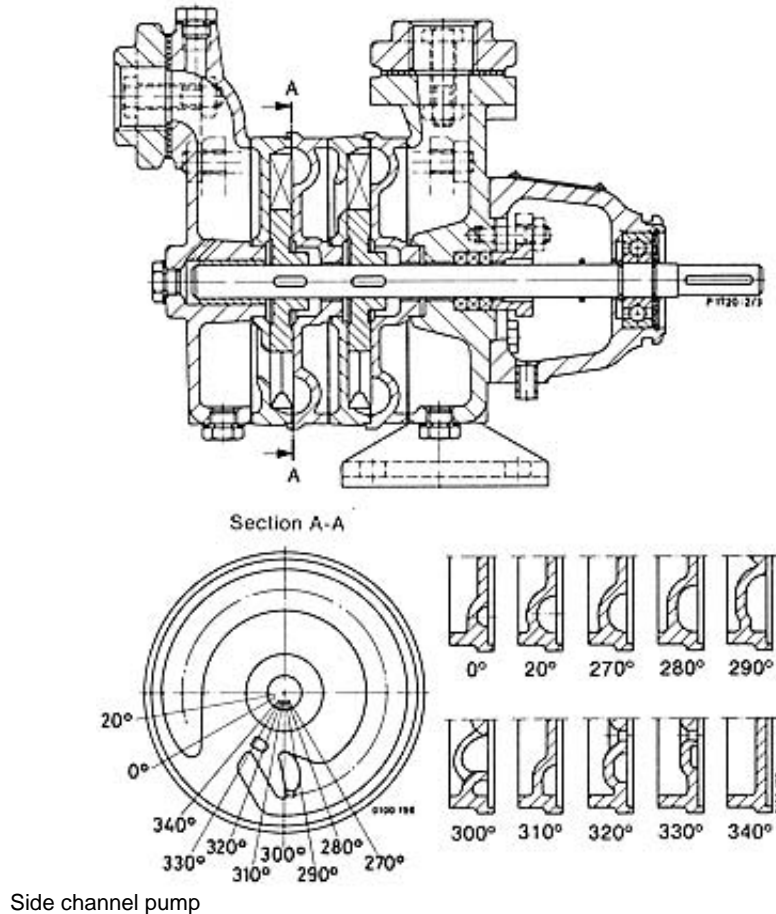
see Valves and Fittings

Side Channel Pump

Seitenkanalpumpe
Pompe à canal latéral

In a s.c.p. (see illustration), the output of a star-shaped impeller (star impeller, water ring pump) rotating concentrically in the casing is transferred to the fluid pumped in a side channel (duct) arranged next to the impeller. The star impeller has straight radial blades without any impeller side walls (shrouds) and is of very simple design; it is surrounded by the casing walls on the outside and on either side, with narrow clearance gaps. The casing has a

side channel on one side or on both sides, which stretches almost around the entire periphery, but which is interrupted at one point (between the inlet and outlet slits). During the rotation of the impeller, the fluid moves to and for several times between the cells of the star impeller and the side channel, and a considerable transfer of energy takes place by momentum exchange between the fluid rotating at high circumferential velocity in the star impeller and the fluid flowing more slowly in the side channel. This momentum exchange which occurs repeatedly at the periphery generates a very high head, which amounts to between 5 and 15 times the head generated by a radial impeller of equal size rotating at the same circumferential velocity.



Apart from this high head which can even be multiplied several times over in multistage s.c.p.'s (multistage pump), the s.c.p. has the further advantage of being self-priming (self-priming pump). Similarly to what happens in water ring pumps, the water remaining in the pump casing forms a water ring with a free surface when the star impeller rotates and when air sucked in through the suction branch (pump suction branch) penetrates inside the s.c.p. This water ring would rotate concentrically and therefore ineffectively in the casing were it not for the fact that the side channel is interrupted at one point of the periphery; the displacement action in this region of the periphery causes the inner radius of the water ring to decrease, and the air-filled pockets between the radial blades of the impeller are also reduced in volume as a consequence; the s.c.p. now operates as a positive displacement compressor, like a water ring pump.

The field of application of s.c.p.'s is governed by their self-priming capability and by their high head at low capacities, compared with conventional centrifugal pumps of the same dimensions; only the peripheral pump achieves even higher heads. S.c.p.'s of more than 4 kW drive rating are seldom built because of their relatively low pump efficiency.

Similarity Conditions

Ähnlichkeitsbedingungen
Conditions de similitude

The similarity theory imposes three sufficient conditions for hydraulic model tests, namely geometric (length similarity), kinematic (velocity similarity) and dynamic (forces similarity) similarity between the model M and the full-scale construction G.

The kinematic and dynamic similarities are grouped together under the heading of physical similarity (model laws).

In order to fulfil the condition of *geometric* similarity, all the dimensions l_M of the model pump must be in the same ratio m_l to the corresponding dimensions l_G of the full-scale construction

$$m_l = \frac{l_M}{l_G} = \text{constant.}$$

The pump and plant associated with it need only be reproduced geometrically similar on the model in so far as the flow in the section under consideration demands them to be. Thus e.g. there is no essential need for geometric similarity of the plant at the discharge end of the pump if the investigations into the flow are being confined to its suction end. The wall surface roughness of the full-scale construction can only be imperfectly reproduced on the model, i.e. the boundary layer flow (boundary layer) and the resultant pressure losses arising from skin friction can only be conditionally investigated on the model.

The *kinematic* similarity imposes the proportionality of the corresponding velocity vectors on the model v_M and on the full-scale construction v_G (velocity triangle). The requirement of a constant velocity scale

$$m_v = \frac{v_M}{v_G} = \text{constant}$$

can strictly speaking only be fulfilled in conjunction with geometric and dynamic similarity. Any deviation from geometric similarity will result in a roughly equal deviation from kinematic similarity. In model tests there is often a deviation from kinematic similarity resulting from the fact that the values of the degree of turbulence in the model flow do not correspond with those of the full-scale construction; the degree of turbulence of the flow influences the sudden change from laminar to turbulent flow conditions (boundary layer, fluid dynamics), and therefore the friction losses as a result can often not be assessed on the model with sufficient accuracy. Experience shows however that the different shapes and structures of the boundary layers on the model as related to those of the full-scale construction result in only insignificant deviations from the kinematic similarity, provided there are no significant differences in the flow separation regions present, and provided also that the investigation is not concerned with areas close to solid surfaces (blades, casing walls etc.).

In order to fulfil the requirement of *dynamic* similarity, the same scale m_f of the forces F which determine the flow phenomenon must obtain, viz.:

$$m_f = \frac{F_M}{F_G} = \text{constant.}$$

The forces F which arise in the hydraulic pump model technology (model laws) are the inertia force, the force of gravity, the pressure force and the frictional force; in this context, we neglect so called "two-phase effects" (e.g. as can happen in a flow of a steam and water mixture, two-phase flow).

The dynamic similarity in relation of the inertia force and the force of gravity on the model and on the full-scale construction is maintained by the FROUDE number Fr (model laws) being constant:

$$\text{FROUDE number} = \frac{\text{Inertia force}}{\text{Force of gravity}},$$

$$Fr = \frac{v^2}{g \cdot l},$$

$$Fr_M = Fr_G$$

where

v characteristic flow velocity of the phenomenon,
 l characteristic length for the phenomenon,
 g local gravitational constant.

The same applies to the REYNOLDS number Re (model laws):

$$\text{REYNOLDS number} = \frac{\text{Inertia force}}{\text{Frictional force}},$$

$$\text{Re} = \frac{v \cdot l}{\nu},$$

$$\text{Re}_M = \text{Re}_G$$

where

ν kinematic viscosity of the pumped medium.

A frequent deviation from dynamic similarity in the hydraulic model technology arises from the fact that the FROUDE or REYNOLDS numbers on the model and on the full-scale construction are not constant, because of technical reasons relating to the tests. Many years of experience have enabled certain ranges of FROUDE or REYNOLDS numbers to be attained on the model and on the full-scale construction without impairing the physical similarity to any great extent (efficiency re-evaluation).

Single-Phase Alternating Current

Einphasenwechselstrom
Courant alternatif monophasé

see Alternating Current

Single Vane Impeller

Einsehaufelrad
Roue à un canal

see Impeller

Siphon Flow

Heberstrom
Débit de siphonnage

see Siphoning Installation

Siphoning Installation

Heberleitung
Conduite en siphon

S.i.'s can be used in groundwater pumping stations to feed the discharge from individual wells into a collecting well, from which the water is then pumped into a reservoir by self-priming pumps. The difference in levels e (see illustration) is used as a driving gradient or head to generate the flow velocity v and overcome the head losses $\sum \zeta \frac{v^2}{2g}$ (pressure loss), with ζ resistance coefficient and g gravitational constant.

We have:

$$e = \frac{v^2}{2g} (1 + \Sigma \zeta).$$

Thus the siphon discharge Q in m³/h becomes:

$$Q = 1,25 \cdot 10^4 \cdot d^2 \sqrt{\frac{e}{1 + \Sigma \zeta}}$$

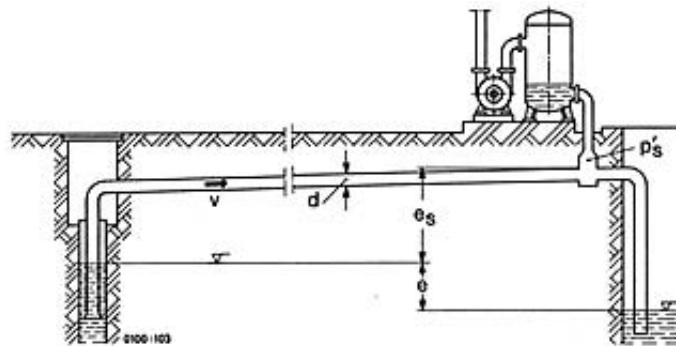
with

v flow velocity in m/s,

$\Sigma \zeta$ sum of resistance coefficients of pipe lengths elbows etc. (pressure loss),

d inner diameter of pipeline in m,

e difference between the two water levels (see illustration) in m.



Siphoning installation (diagrammatic)

The s.i. is only capable for operating if the absolute pressure p'_s at the apex of the system exceeds the vapour pressure p_D corresponding to the prevailing water temperature. This condition is satisfied if

$$e_s < 10 (p_b - p_D) - \frac{v^2}{2g} (1 + \Sigma \zeta_{e,s}) - S$$

or, if the apex is situated close to the collecting well:

$$e_s < 10 (p_b - p_D) - e - S$$

with

e_s difference between elevation of apex and water level at inlet side in m,

$\Sigma \zeta_{e,s}$ sum of resistance coefficients from the inlet to the apex of the system,

S safety margin of 1 to 2 m, depending on quality of pipeline construction,

p_D vapour pressure in bar

p_b barometric pressure in bar.

Dissolved gases (gas separation) are constantly released out of solution at the apex of the line and have to be pumped out.

Instead of s.i.'s often underwater motor pumps are used. which directly pump the water into the reservoir (see illustration under domestic water supply plant).

Slip Ring Rotor

Schleifringläufer
Rotor bobiné

see [Asynchronous Motor](#)

Slit seal

Spaltdichtung
Joint d'étanchéité par jeu

The s.s. is a form of seal ([shaft seals](#)) frequently used on [centrifugal pumps](#) between rotating and stationary components. The clearance gap is sized in its width ([clearance gap width](#)) and shape in such a way that it lets through as small a [mass flow](#) ([leakage loss](#), [clearance gap loss in centrifugal pumps](#)) as possible, e.g. in [multistage pumps](#) for the sealing of the individual [stages](#) against one another, or in [balancing devices](#), or again for the sealing of the suction side of the impeller against the discharge side ([clearance gap pressure](#)). The width of the s.s. affects both the [economic](#) efficiency and the operating reliability of a [centrifugal pump](#) and depends on the following factors: shaft sag, [vibrations](#) (including self-excited vibrations), nature of fluid pumped, degree of contamination, grain size of the dirt particles ([abrasion](#)), temperature (if the thermal expansion of the rotating components differs from that of the stationary components, the [clearance gap width](#) may change and under certain circumstances the casing may even be distorted, [boiler feed pump](#)). The shapes of s.s.'s used in centrifugal pump technology include principally smooth clearance gap seals, stepped clearance gap seals and labyrinth seals.

Slope

Steilheit
Pente

see [Characteristic Curve](#)

Slowing-Down Time

Auslaufzeit
Temps de ralentissement

see [Starting Process](#)

Sludge Pump

Schlammpumpe
Pompe à boues

see [Pulp Pump](#)

Sound Insulation

Schalldämmung
Isolation acoustique

see [Noise in Pumps](#) and [Pumping Installations](#)

Sound Level

Lautstärkepegel
Niveau d'issonie

see [Noise in Pumps](#) and [Pumping Installations](#)

Sound Pressure Level

Schalldruckpegel
Niveau de pression acoustique

see [Noise in Pumps](#) and [Pumping Installations](#)

Sound Velocity

Schallgeschwindigkeit
Vitesse du son

The s.v. a is the propagation velocity of a weak, i.e. isentropic (of equal entropy) pressure disturbance with the relationship

$$a^2 = \frac{dp}{d\rho}$$

with

dp differential pressure variation (pressure disturbance).
 $d\rho$ differential density variation of medium as a result of the pressure disturbance.

The s.v. plays a part in centrifugal pump technology particularly in the case of pressure surges in piping systems, but also when a mixture of liquid and vapour is pumped (cavitation, two-phase flow, shock condensation, condensate pump). In such cases, extremely low s.v.'s of the order of magnitude of the absolute velocity v can occur (sound limit $v/a = 1$, compression (shock) wave with sudden and abrupt disappearance of the vapour bubbles). The low s.v.'s in a mixture of liquid and vapour are explained by the fact that considerable changes in density $d\rho$ of the mixture occur in relation to the change in pressure dp , because of evaporation or condensation (particularly in the region of low pressures and low vapour contents). Unexpectedly low s.v.'s also occur in mixtures of liquid and gas (dirty water pump, pulp pump).

In pipelines the s.v. is also influenced by the material used and the ratio between pipeline diameter and wall thickness. The s.v. for water in pipelines of steel, cast iron or concrete is approx. 1000 m/s.

Specific Energy

Spezifische Förderarbeit
Énergie massique

The s.e., symbol Y (SI unit 1 N m/kg) is the useful mechanical energy transmitted by the centrifugal pump to the fluid pumped, related to the mass of the fluid pumped.

There is the following relationship between head H and s.e. Y ..:

$$Y = g \cdot H$$

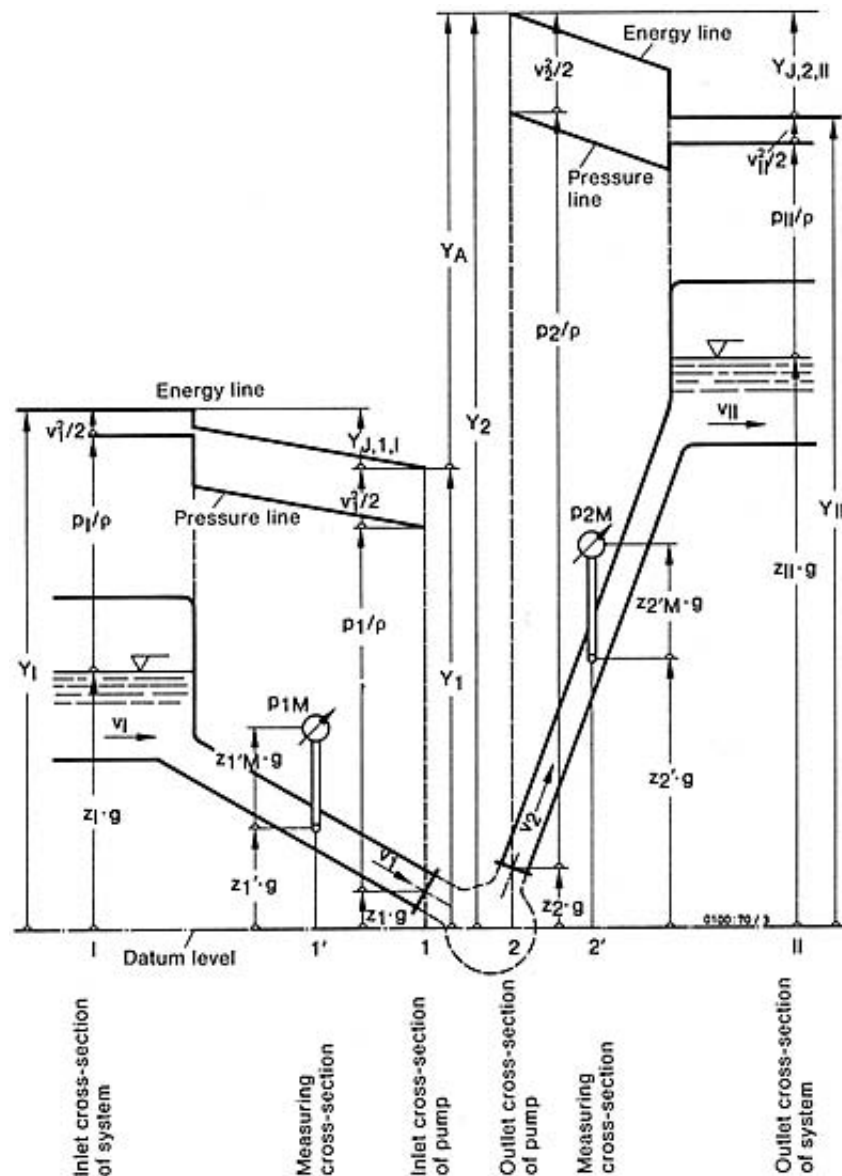
with

g gravitational constant.

The illustration below (from DIN 24260, September 1986 edition) shows the progression of the s.e. (energy line) along a pumping plant.

In that diagram:

- H head,
- p static pressure,
- v flow velocity,
- z geodetic altitude,
- ρ density of the pumped medium, and the indices:
- A system,
- J pipe friction loss,
- M water gauge above the measuring point,
- 1 inlet cross-section of pump,
- 1' measuring cross-section upstream of pump,
- 2 outlet cross-section of pump,
- 2' measuring cross-section downstream of pump,
- I inlet cross-section of the system,
- II outlet cross-section of the system.



Specific energy Y , referred to datum level BN (illustration according to DIN 24260, September 1986 edition)

Specific Gravity

Wichte

Poids volumique

S.g. is the quotient of weight force (weight) by volume. The s.g. (symbol γ) is to be replaced in the SI unit system (unit) by the product of density ρ and the gravitational constant g .

Specific Speed

Schnellläufigkeit, spezifische Drehzahl

Vitesse spécifique

If we give the following data of a centrifugal pump impeller: capacity Q , head H and rotational speed n , the s.s. of said centrifugal pump impeller is the true rotational speed of a model pump impeller (index q), similar in blade geometry and in velocity planes (velocity triangle), having the following performance data:

capacity $Q_q = 1 \text{ m}^3/\text{s}$,
head $H_q = 1 \text{ m}$.

From the relationships of similitude mechanics (similarity conditions, affinity law, model laws) it follows:

$$\frac{Q}{Q_q} = \frac{D^3 \cdot n}{D_q^3 \cdot n_q};$$

$$\frac{H}{H_q} = \frac{D^2 \cdot n^2}{D_q^2 \cdot n_q^2}.$$

Solved for n_q we obtain:

$$n_q = n \frac{(Q/Q_q)^{1/2}}{(H/H_q)^{3/4}}$$

or, with $Q_q = 1 \text{ m}^3/\text{s}$ and $H_q = 1 \text{ m}$:

$$n_q = n \frac{Q^{1/2}}{H^{3/4}}$$

with

Q capacity in m^3/s ,
 H head in m ,
 n rotational speed in min^{-1} ,
 n_q s.s. in min^{-1}

In centrifugal pump technology, it is usual to quote the s.s. in relation to the optimum capacity and head values (i.e. at the operating point of optimum efficiency η_{opt}) of the impeller, so that we have:

$$n_q = n \frac{Q_{\text{opt}}^{1/2}}{H_{\text{opt}}^{3/4}}$$

with

n_q s.s. in min^{-1} ,
 n rotational speed in min^{-1} ,
 Q_{opt} optimum capacity in m^3/s ,
 H_{opt} optimum head in m .

The s.s. n_q in the last-named version is a frequently used characteristic magnitude in centrifugal pump technology, and it is characteristic of the optimal impeller shape required to achieve optimum efficiency:

radial impeller $n_q \approx 12 \text{ to } 35 \text{ min}^{-1}$,
mixed flow impeller $n_q \approx 35 \text{ to } 160 \text{ min}^{-1}$,
axial impeller $n_q \approx 160 \text{ to } 400 \text{ min}^{-1}$, and over.

The s.s. also enables a forecast of the qualitative pattern of the characteristic curves (relevant Fig. 1).

For a case involving a given Q and a given H , the s.s. can be altered - e.g. in order to optimize the pump efficiency η - either by changing the rotational speed (to the extent permitted by net positive suction head and drive) or by splitting the head up among several stages (multistage pump).

Based on the dimensional magnitude n_q , a non-dimensional coefficient characterizing the type of construction has also been adopted in centrifugal pump technology:

$$n'_q = 333 \cdot n \frac{Q_{opt}^{1/2}}{(g \cdot H_{opt})^{3/4}}$$

with

n rotational speed in s^{-1} ,

Q_{opt} optimum capacity in m^3/s ,

g gravitational constant = $9,81 \text{ m/s}^2$,

H_{opt} optimum head in m.

The numerical values of the magnitudes n_q and n'_q are the same.

The conversion of the characteristic factor n'_q into the so-called "type number"

$$K = 2 \pi \cdot n \frac{Q^{1/2}}{(g \cdot H)^{3/4}}$$

(see ISO 2540), frequently used in English and American centrifugal pump literature (see also ISO 2540) is given by

$$K = \frac{1}{52.919} \cdot n'_q.$$

K has been introduced into the ISO 2548 and DIN 24260 and IEC/TC 4 Codes of Practice.

Further conversions:

Conversion formula	Units for n_s , n_q engl and n_q USA		
	Q	H	n
$n_s = 3,65 \cdot n'_q$	m^3/s	m	min^{-1}
$n_q \text{ engl} = 47,13 \cdot n'_q$	gallon (engl)/min	foot	min^{-1}
$n_q \text{ USA} = 51,64 \cdot n'_q$	gallon (USA)/min	foot	min^{-1}

Speed Control

Drehzahlregelung

Régulation de vitesse

see [Control](#)

Sprinkler Pump

Sprinklerpumpe
Pompe Sprinkler

Sprinkler systems are the most important and most widely used form of fire extinguishing (fire-fighting pump). The principle of the sprinkler system is as follows: all rooms to be protected are equipped with piping in which, at certain intervals, sprinklers are mounted. They open independently in cases of extreme heat. A sprinkler system is, therefore, an automatic extinguishing system.

The most used sprinkler system is the wet system. The pipes are constantly filled and under pressure, so that in case of fire, water can escape from one or more sprinklers. In dry systems, pipes that are threatened by freezing temperatures, are filled with air. For safety reasons, sprinkler systems should have two independent sources of water, so that, depending on the size of the fire, one or two centrifugal pumps can be activated. In the case of two pumps, it is recommended that each be connected to an independent energy source.

The s.p. pumps the fire-extinguishing water out of a reservoir, pond, river or lake. S.p.'s are at rest most of the time. They are only operated for checking purposes or in case of fire. Therefore special emphasis must be laid on reliable starting (starting process) and rapid attainment of the full output, even after a prolonged shutdown. To meet these requirements, s.p.'s must be constructed in accordance with special guidelines and must undergo a type acceptance test. Societies and Associations which issue the appropriate guidelines include, amongst others, the Association of Property Damage Liability Insurers (VdS) (with Headquarters in Cologne), the Factory Mutual (FM) (with Headquarters in Norwood/Mass., USA) and the National Fire Protection Association (NFPA) (with Headquarters in Boston/Mass., USA).

S.p.'s are used for capacities up to 1000 m³/h approx. and for heads up to 100 m approx. The following pump types are mainly employed: single stage volute casing pumps, double suction radial pumps (multisuction pump, centrifugal pump), ring section pumps (multistage pump), underwater motor pumps and borehole shaft driven pumps.

Squirrel-Cage Rotor Motor

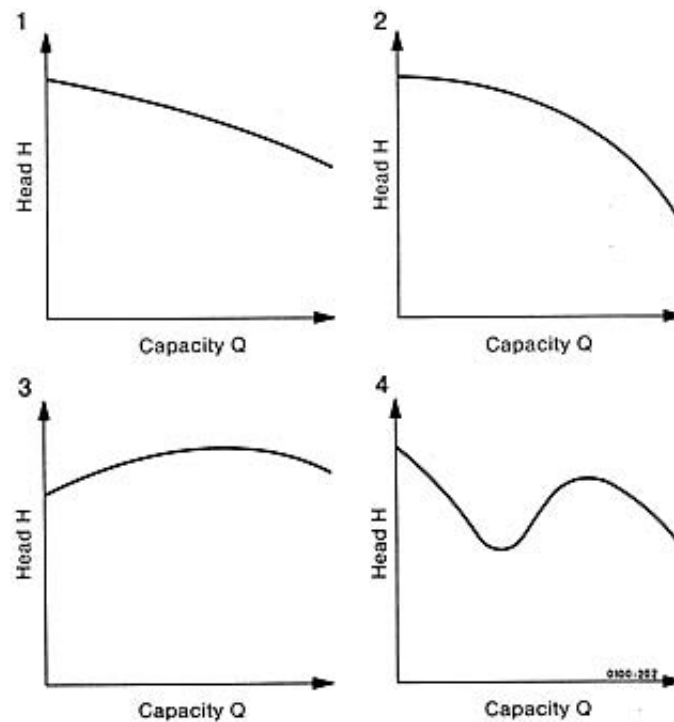
Kurzschlußläufermotor
Moteur à cage d'écureuil

see Asynchronous Motor

Stable Throttling Curve

Stabile Drosselkurve
Courbe d'étranglement stable

The throttling curve (characteristic curve) of a centrifugal pump is designated "stable" if it exhibits a continuously negative slope (negative differential quotient from head H and capacity Q) in respect of the Q axis. In the illustration, curves 1 and 2 are stable, and curves 3 and 4 unstable (unstable throttling curve).



Examples of stable (1 and 2) and of unstable throttling curves (3 and 4)

Stage

Stufe
Étage

The s. of a multistage centrifugal pump consists of the combination of an impeller with a diffuser device (diffuser, volute casing, annular casing). The characteristic feature of the s. of a centrifugal pump is the transmission of power from the pump shaft (shaft power P) to the flowing medium (pump output P_Q). Thus a so-called s. efficiency can be obtained by means of the pump output related to the s. and the shaft power, and this s. efficiency in multistage pumps is set as the highest attainable constant value over all the s.'s, unless special suction or NPSH conditions are required (inlet conditions, suction behaviour). The number of s.'s i of a multistage centrifugal pump should be selected in such a way that hydraulically favourable conditions prevail at the individual s.'s, in accordance with the type of impeller used. In this connection, the determining factor for the individual impeller is not the specific speed $n_{q,P}$ of the pump, but the specific speed $n_{q,St}$ related to the s.

$$n_{q,St} = i^{3/4} \cdot n_{q,P}$$

Standard Chemical Pump

Chemienormpumpe
Pompe normalisée pour l'industrie chimique

see Chemical Pump, Standard Pump

Standard Conditions

Normzustand
Conditions normales

According to DIN 1343, the s.c. of a solid, liquid or gaseous substance are the conditions governed by the standard temperature $T_n = 273.15 \text{ K}$ ($t_n = 0 \text{ °C}$) and the standard pressure $p_n = 101325 \text{ Pa} = 101325 \text{ N/m}^2 = 1.01325 \text{ bar}$ (unit).

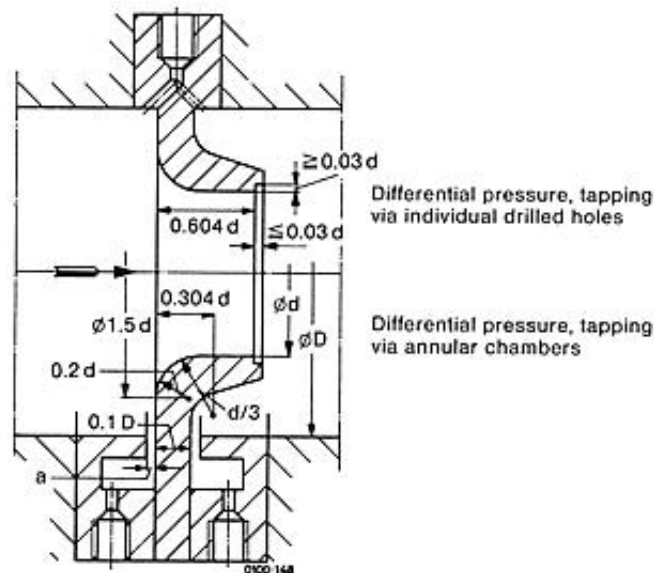
The volume of a substance under s.c. is known as the standard volume V_n . The molar standard volume of ideal gases (special symbol V_0) is $V_0 = 22.414 \cdot 10^{-3} \text{ m}^3/\text{mol}$ (unit).

Standard Nozzle

Normdüse

Tuyère normalisée

The s.n. (see illustration) is a nozzle which is in accordance with the specifications contained in a standard (e.g. DIN 1952 or ISO 5167) as regards its geometrical dimensions, its surface finish and its installation conditions. The flow coefficient α takes into account the pressure losses, which are lower than for a standard orifice of equal nominal diameter and equal diameter ratio; there is no contraction of the jet passing through the nozzle. Otherwise, what is said of the standard orifice applies similarly to the s.n. For further details, see DIN 1952, 1922 edition.



Standard nozzle (according to DIN 1952) a = aperture width (see Fig. 3 under standard orifice)

Standard Orifice

Normblende

Diaphragme normalisé

The s.o. is a measuring orifice which can be fitted in piping runs (Figs. 1 and 2), and which is in accordance with the specifications contained in a standard, as regards its geometrical dimensions, its surface finish and its installation conditions, e.g. DIN 1952, Measurement of Fluid Flow by Means of Orifice Plates Nozzles, and Venturi Nozzles (VDI rules for Measurement of Fluid); ISO 5167, Measurement of Fluid Flow by Means of Orifice Plates. Nozzles and Venturi Tubes inserted in circular Cross-section Conduits running full; VDI 2040, Principles of Calculation of Flow Measurement by Means of Throttling Devices. The concept "measuring orifice" is used to distinguish it from the use of an orifice plate to throttle the capacity (control).

A straight length of piping is artificially narrowed by the installation of a s.o.; the differential pressure head h at the s.o. is measured as shown in Fig. 1. The capacity Q (rate of flow) is obtained from the flow cross-section $d^2 \pi / 4$ of the s.o., the measured differential pressure head h and a coefficient α (flow coefficient) which, for the s.o. is a function of the

REYNOLDS number $Re = \frac{v \cdot D}{\nu}$ and of the

diameter ratio $\beta = \frac{d}{D}$ at

$$Q = \alpha \cdot \frac{d^2 \pi}{4} \cdot \sqrt{2g \cdot h}$$

with

d diameter of the throttle opening of the s.o.,
 D inside pipe diameter,
 g gravitational constant,
 h differential pressure head,
 v flow velocity in the pipe,
 Q capacity,
 α flow coefficient
 ν kinematic viscosity.

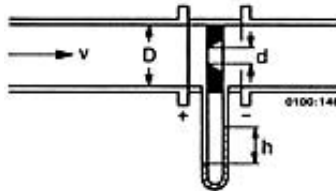


Fig. 1: Measuring orifice in a pipeline d diameter of the throttle opening; D inside pipe diameter; h differential pressure head

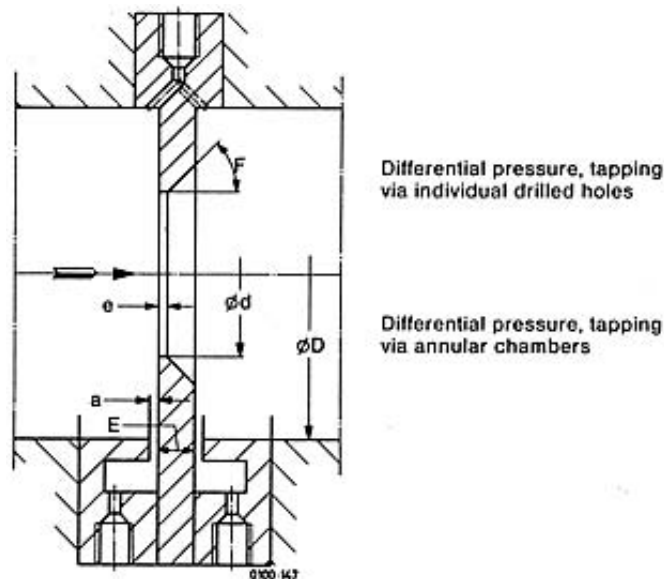


Fig. 2: Standard orifice (according to DIN 1952)

a = aperture width ($0.005 \cdot D \leq a \leq 0.03 \cdot D$; for $\beta \leq 0.65$
 $0.01 \cdot D \leq a \leq 0.02 \cdot D$; for $\beta \geq 0.65$)

e = length ($0.005 \cdot D \leq e \leq 0.02 \cdot D$)

E = thickness ($e \leq E \leq 0.05 \cdot D$)

F = beveling angle ($30^\circ \leq F \leq 45^\circ$)

The term "aperture ratio" occasionally used refers to the square of the diameter ratio β . Values for the flow coefficient a of s.o.'s in smooth bore pipes in function of the REYNOLDS number related to the diameter ratio β can be found e.g. in DIN 1952. This standard also contains all details concerning the construction and installation of the s.o. in conformity with the standard. In particular, the pipe diameter must be measured very accurately; a fairly long straight undisturbed length of piping must be provided both upstream and downstream of the device, the precise length of which will depend on the diameter ratio β of the device and on the nature of the disturbance upstream (see Table). If these straight pipe lengths cannot be provided, additional tolerances for the flow coefficients must be adopted (VDI 2040, Sheet 1). The differential pressure can be tapped via annular chambers (Fig. 2, bottom) or via individual drilled holes (Fig. 2, top), and measured by means of a U-tube (measuring technique).

Table: Minimum values of undisturbed straight pipeline lengths in multiples of pipe diameter D , valid for standard orifices standard nozzles and standard venturi nozzles with annular chambers and with individual drilled holes in accordance with DIN 1952

Diameter ratio β	Inlet side of throttling device							Outlet Side
	single 90° elbow or T-piece (flow from one side only)	two or more 90° elbows in the same plane	two or more 90° elbows in different planes	confusor (nozzle) or reducer (from 2 · D to D over a length of 1.5 · D to 3 · D)	diffuser (from 0.5 · D to D over a length of 1 · D to 2 · D)	globe valve, fully open	gate valve, fully open	any of the valves and fittings listed on the left
0.20	10	14	34	5	16	18	12	4
0.25	10	14	34	5	16	18	12	4
0.30	10	16	34	5	16	18	12	5
0.35	12	16	36	5	16	18	12	5
0.40	14	18	36	5	16	20	12	6
0.45	14	18	38	5	17	20	12	6
0.50	14	20	40	6	18	22	12	6
0.55	16	22	44	8	20	24	14	6
0.60	18	26	48	9	22	26	14	7
0.65	22	32	54	11	25	28	16	7
0.70	28	36	62	14	30	32	20	7
0.75	36	42	70	22	38	36	24	8
0.80	46	50	80	30	54	44	30	8

for all diameter ratios β	disturbances by inserted components	required straight length of piping at inlet
	sudden symmetric diameter reduction with a diameter ratio of ≥ 0.5	30
	thermometer well with a diameter of $\geq 0.03 \cdot D$ of 0.03 to 0.13 · D	5 20

Standard Pressure

Normdruck

Pression normale

see [Standard Conditions](#)

Standard Pump

Normpumpe

Pompe normalisée

The s.p. is a pump which complies with the regulations, guidelines or recommendations issued by a Standards Institute, or a Manufacturers' or Users' Association, e.g.:

ACS	American Chemical Society; standardized: American Voluntary Standard (AVS) pump,
AFNOR	Association Française de Normalisation,
ANSI	American National Standard Institute,
API	American Petroleum Institute,
BPMA	British Pump Manufacturers Association,
BSI	British Standard Institution,
DIN	Deutsches Institut für Normung,
ISO	International Organization for Standardization,
UNI	Unificazione Italiana.

Standard Temperature

Normtemperatur

Température normalisée

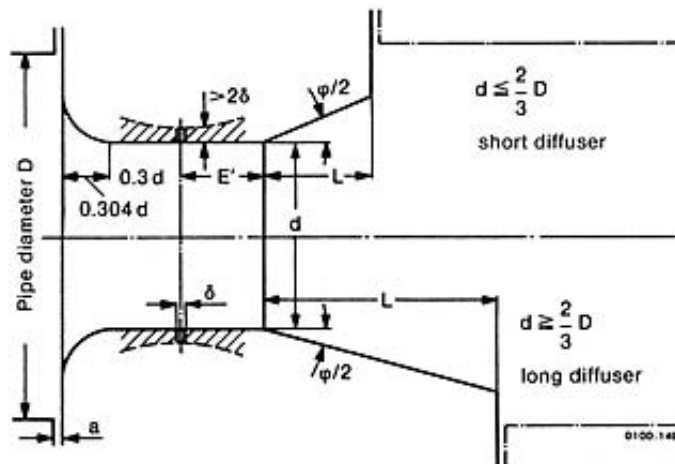
see [Standard Conditions](#)

Standard Venturi Nozzle

Normventuridüse

Venturi normalisé

The s.v.n. (see illustration) is a venturi nozzle which is in accordance with the specifications contained in a standard (e.g. DIN 1952 and ISO 5167) as regards its geometrical dimensions, its surface finish and its installation conditions. S.v.n.'s are used in cases where the pressure loss has to be kept as low as possible. The entry side is identical with that of a standard nozzle. The length L of the tapered diffuser has practically no influence on the flow coefficient α , but it has an influence on the pressure loss, in conjunction with the diffuser angle. The flow coefficient α for s.v.n.'s in smooth bore pipes is dependent on the diameter ratio β , whose significance corresponds to the diameter ratio of nozzles and orifices (standard orifice). The Table under standard orifice applies equally to the straight undisturbed lengths of pipe upstream and downstream of the s.v.n. See DIN 1952. August 1982 edition, for further details of dimensions and installation conditions.



Standard venturi nozzle (in acc. with DIN 1952)

$E' = 0.40 \cdot d$ to $0.45 \cdot d$,

$\varphi/2$ max. 15° ,

δ max. $0.04 \cdot d$; min. 2 mm

Standard Water Pump

Wassernormpumpe

Pompe standard à eau

see [Standard Pump](#)

Star-Delta Starter Circuit

Stern-Dreieck-Schaltung

Connexion étoile-triangle

S.d.s.c. is a starting circuit (electrical circuit) for three-phase motors (asynchronous motor). The s.d.s.c motor has six terminals (terminal designation). The delta circuit only applies to continuous operation (starting process).

Star Impeller

Sternrad

Roue ouverte à ailettes radiales

The s.i. is part of a side channel pump or of a liquid ring pumps (water ring pump) operating on the semi-positive displacement principle.

Fig. 1 illustrates the s.i. of a side channel pump, and Figs. 2 and 3 the s.i.'s of liquid ring pumps.

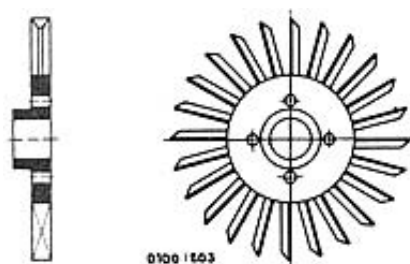


Fig. 1: Star impeller of a side channel pump

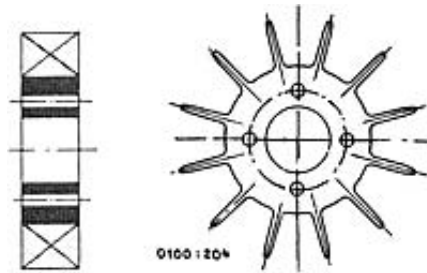
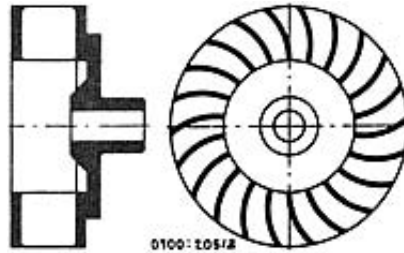


Fig. 2: Open star impeller of a liquid ring pump

Fig. 3: Closed star impeller of a liquid ring pump
(front view with cover plate removed)

Starting Process

Anlaufvorgang

Processus de mise en marche

The s.p. of a mechanical unit consisting of an electric motor and of a driven machine (e.g. a radial centrifugal pump) is influenced by the following factors:

1. Type of electric motor:

Direct current motors (except for series wound motors) have very good starting characteristics and pose no s.p. problems; asynchronous motors with slip-ring rotors which are started by means of starting rheostats also have very good starting characteristics; asynchronous motors with squirrel cage rotors exhibit different starting torque characteristics according to design and type of rotor, and must always be adapted to the particular application.

2. Type of driven machine:

Reciprocating pumps (positive displacement pump) require high starting torques. Centrifugal pumps of radial type require a relatively low starting torque; the countertorque, i.e. the torque (starting torque) developed by the pump, which act: countered the motor torque, increases with the square of the rotational speed. The magnitude of this countertorque will depend on whether the pump is started up against a closed or an open discharge valve.

3. Type of starting and type of power supply:

In the case of direct-on-line starting onto a "anonyielding mains supply" the full starting torque and full run-up torque are available; the starting current amounts to approx. 4.5 to 6 times the nominal current J_N . In the case of direct-online starting onto a "yielding mains supply", the mains voltage will drop under the effect of the starting current. The torque curve of the motor will sink roughly proportionately to the square of the voltage drop. After run-up to full operating speed, the mains voltage regains its full value as soon as the current absorbed by the motor has decreased again down to the nominal current.

If the short-circuit characteristic and the shortcircuit power factor $\cos \varphi$ of the motor are known, the voltage drop ΔU can be calculated from the mains resistances R_V and X_V :

$$\Delta U = \sqrt{3} \cdot J \cdot (R_V \cdot \cos \varphi + X_V \cdot \sin \varphi).$$

The torque curve of the motor can thus be determined, and it can be estimated whether an adequate excess torque is available to accelerate the pump and motor rotors. Switching on is effected via a star-delta starter circuit an autotransformer starter or a series resistance. All these devices are designed to limit the starting current. They all

have the effect of reducing the starting torque considerably. In most cases the starting process is also accompanied by a certain voltage drop in the power supply undertow influence of the starting current, which usually amounts to between twice and 2.5 times the nominal current.

The regulations of the local electricity supply undertakings usually forbid the direct-on-line starting of asynchronous motors above a certain rating from the public grid. In the case of drives of centrifugal pumps, it should also be borne in mind that in many instances automatic pumping plants are installed which preclude the closing in advance of the throttling valve (valves and fittings) every time the radial centrifugal pump is started up. Thus the pump must be started and run up to speed using the star circuit against the full pump torque (starting torque, case 1). If the pump cannot attain at least 90% of its full rotational speed under these conditions, the provision of a star-delta starter is useless, because a current surge almost as large as the surge occurring under direct-on-line starting will in fact arise when switching over from star to delta.

The starting curves of squirrel-cage rotor motors and of slip-ring rotor motors are discussed below.

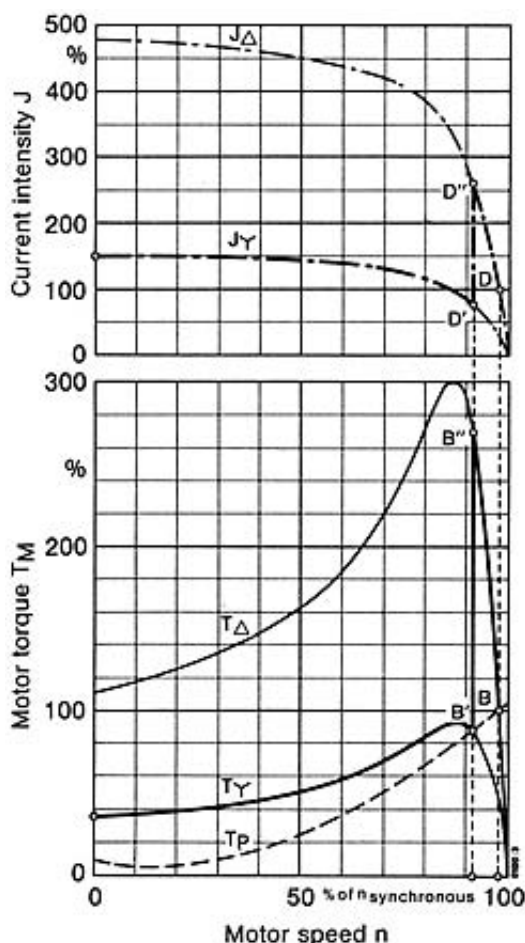


Fig.1: Starting curve for current J and torque T of a squirrel-cage rotor motor
Subscripts: Υ = star circuit; Δ = delta circuit; P = pump

Squirrel-cage rotor motors. Fig. 1 illustrates the torque curves T_Y and $T_{\Delta} = f(n)$ of a submersible motor arranged for star-delta operation, also the current curves J_Y and $J_{\Delta} = f(n)$ and the torque curve $T_P = f(n)$ of the pump operating against an open discharge valve. These curves apply equally to conventional air-cooled squirrel-cage rotor motors (asynchronous motor). After switching in the star circuit, the motor first runs up to speed along the T_Y curve, up to the intersection B'' with the torque curve T_P of the pump. The current absorbed during this run-up is illustrated by the curve J_Y . Then switching over to the delta circuit takes place, raising the torque to point B'' on curve T_{Δ} , and the run-up proceeds to the design duty point B (operating point). The absorbed current jumps from D' to D'' on switching over, then it follows the J_{Δ} curve down to the operating current value D . In this example, the max. starting current in the star circuit amounts to 150%, and the current on switch-over to delta amounts to 225% of the rated current.

In the above example, the adoption of a star-delta starter circuit makes sense, because the peak current is only about half what it would be with direct-on-line starting. Pumps with radial impellers (impeller), when started against a closed discharge valve, only require approximately 60 % of the operating torque at the nominal duty (or operating) point B during start-up (starting torque, case 3). Under these conditions, the intersection B' of the pump torque curve T_p with the motor torque curve T_Y would lie at 97% of the synchronous speed, and the current surge on switching over to delta would not greatly exceed 100% of the nominal current. In the case of propeller pumps, there is no sense in throttling the discharge during start-up, because the torque requirement under throttled conditions is greater than the nominal torque at the duty point (characteristic curve).

Fig. 2 illustrates the torque curves of a canned motor (wet rotor motor) with a squirrel-cage rotor. Motors of this type have a low starting current and a relatively low pull-out torque, which makes them ideal for direct-on-line starting. Fig. 2 illustrates the current and torque curves in function of the rotational speed. If a star-delta starter arrangement is adopted in this case, the intersection of the motor torque curve T_Y with the pump torque curve T_p already occurs at approx. 60% of the synchronous speed. As there is no excess motor torque available beyond this point onwards, the rotating assembly cannot accelerate any further (in star), and the rotational speed would remain at the low (60%) value. The current absorbed from the mains J_Y remains under these conditions at somewhat less than 100% above the nominal current J_N . Nevertheless the motor windings are overloaded because they are star connected and carry the full mains current, whereas during normal operation in delta circuit they only carry 58 % of the interlinked mains current. If at this point we switch over to delta, the motor torque jumps from B' on the T_Y curve to B'' on the T_Δ curve. We then have an excess torque available again, and the rotor accelerates towards the new intersection of T_p with T_Δ at the operating point B. On switchover from star to delta, the current jumps from D' to D'' and then decreases along the J_Δ curve to D at the operating point (nominal current). We can see therefore that the starting current in the star circuit is very low (less than 100 % of J_N), but that on switching over to delta, there occurs a peak at point D'' amounting to 255% of J_N . If direct-on-line starting had been adopted, there would have been a peak current of 290% of J_N approx. The difference between these peaks is so small that it does not pay to install expensive star-delta starter equipment in this case. Furthermore, canned motor pumping sets with direct-on-line starting only require a very short time to run up to full speed, so that this peak current is usually a negligible consideration.

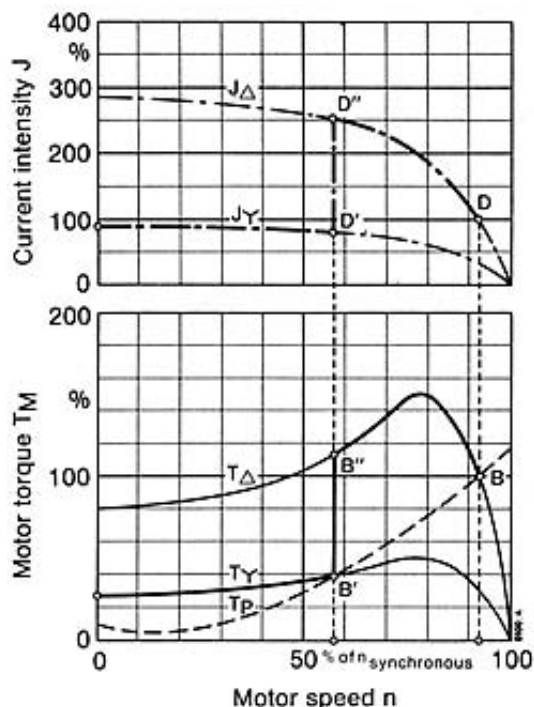


Fig. 2: Starting curve for current J and torque T of a canned motor (symbols as in Fig. 1)

Slip-ring rotor motors (asynchronous motor). The starting rheostat is switched into the rotor circuit via the slip rings during the starting period, and is subsequently short-circuited in stages. After run-up to full speed, the slip rings are often short-circuited and the carbon brushes lifted off. The pump torque T_p and the motor torque T_M (or the absorbed motor current J respectively) are

plotted against the rotational speed in Fig. 3.

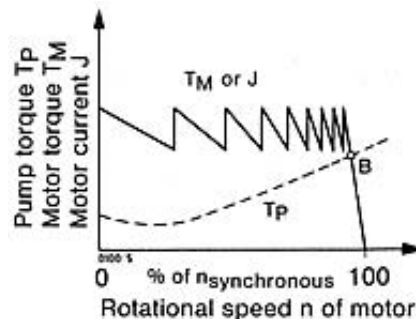


Fig. 3: Starting curve for current J and torque T of a slip ring rotor motor

The starting time t_a can be calculated by means of the average accelerating torque $T_{b,aver.}$, averaged over the rotational speed n (see Fig. 4):

$$t_a \approx \frac{\pi \cdot J \cdot n_B}{30 \cdot T_{b,aver.}}$$

where

- n_B operating rotational speed in min^{-1} ,
- $T_{b,aver.}$ average accelerating torque $T_M - T_p$ in N m,
- t_a starting time in s,
- J moment of inertia (moment of gyration) in kg m^2 .

However, the above formula only yields useful values of starting time if the accelerating torque remains reasonably constant over the entire range of speeds. If the T_M and T_p curves move close to one another in certain regions (as for example in Fig. 1, for n above 85% of $n_{asynchronous}$), the time of run-up to full speed must be partially determined by a calculation or graphical method.

For this purpose, the zone of rotational speeds is subdivided into a series of successive stages or steps Δn_i , within each of which the corresponding accelerating torque T_{bi} is assumed to have a constant value for calculation purposes (see Fig. 4). The starting time t_a is then obtained from the summation of the individual steps

$$t_a = \frac{\pi \cdot J}{30} \left(\frac{\Delta n_1}{T_{b1}} + \frac{\Delta n_2}{T_{b2}} + \frac{\Delta n_3}{T_{b3}} + \dots \right)$$

where

- J mass moment of inertia of the rotating components of the pumping set, including the liquid trapped in the impeller (moment of gyration) in kg m^2 ,
- Δn_i rotational speed range in min^{-1} ,
- T_{bi} accelerating torque in N m.

The period of time elapsing between the instant of switching off or of failure of the driver (drive) of a centrifugal pump and the instant when the pump shaft slows down to a complete stop is known as the run-down time t_{aus} . It is determined in similar fashion to the starting time. However, instead of the accelerating torque T_{bi} , the starting torque $T_{pi} = f(n)$ should be used for the calculation of the run-down time; in this instance it acts as a load torque (decelerating or retarding torque) (see Fig. 4).

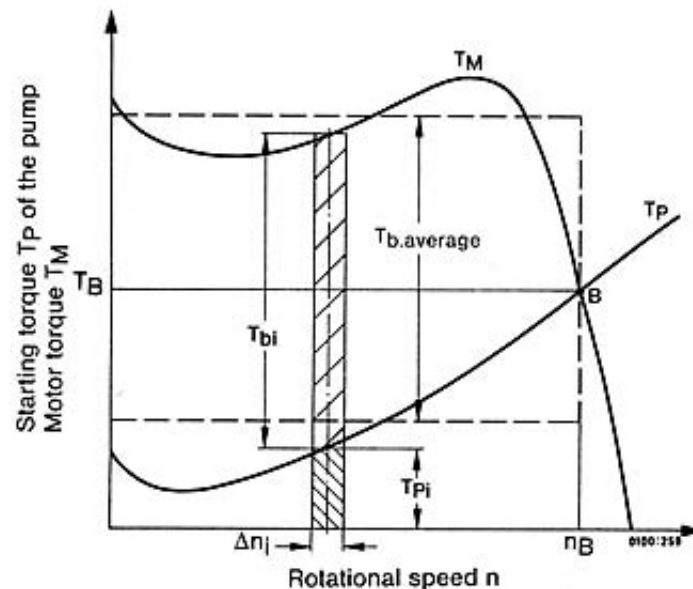


Fig. 4: Determination of starting time and run-down time of a centrifugal pump

$$t_{\text{aus}} = \frac{\pi \cdot J}{30} \left(\frac{\Delta n_1}{T_{p1}} + \frac{\Delta n_2}{T_{p2}} + \frac{\Delta n_3}{T_{p3}} + \dots \right)$$

where

T_{pi} starting torque (load torque) of the pump in N m.

Starting Torque

Anlaufdrehmoment, Anzungsmoment

Couple en cours de démarrage, couple de décollage

The s.t. is the

by the coupling during the starting process, i.e. from the instant of switching on the driver (drive) up to the instant when the operating point of the centrifugal pump has been attained. It is calculated from the ratio of power P to angular velocity ω (rotational speed) and is plotted graphically in function of the rotational speed.

$$T_P = \frac{P}{\omega}$$

The numerical value equation is:

$$T_P = 9549 \frac{P}{n}$$

with

T_P s.t. in N m,

P_P pump shaft power (absorbed power) in kW,

n rotational speed in min^{-1} .

The following factors have a decisive influence on the s.t.:

- - the shape of the characteristic curves for head and for shaft power (absorbed power) of the pump in function of capacity and of rotational speed,
- - the position of the operating point on these characteristic curves,
- - the starting behaviour of the driver (drive, starting process), characterized by the starting time t_a of the

pumping set,

- - the plant characteristic, taking into consideration any valves incorporated in the system (isolating valves, check valves or nonreturn valves),
- - the starting time t_{aQ} for the acceleration of the mass of fluid in the primed pipeline (piping)

$$t_{aQ} = \frac{3 Q}{2 g (H_0 - H_{A,0})} \cdot \frac{L}{A}$$

where

t_{aQ}	starting time of mass of fluid in the pipeline (<u>piping</u>),
Q	capacity,
H_0	shut-off head (<u>head</u>),
$H_{A,0}$	static portion of <u>system characteristic curve</u> (<u>control</u>),
g	<u>gravitational constant</u> ,
L	length of pipeline,
A	cross-sectional area of pipeline (<u>piping</u>).

The possible shape of the s.t. curve is illustrated for the case of a radial pump (low specific speed, and on the simplifying assumption that the shaft power P (absorbed power) increases linearly with increasing capacity Q) for four different types of operation.

All four s.t. curves T_p commence at the same breakaway torque T_{pL} , required to surmount the static friction of the bearings and seal, and thereafter they are composed of two processes running either simultaneously or in succession, viz.:

1. the increase of the torque with increasing rotational speed n , and
2. the increase of the torque corresponding to the increase in shaft power requirement P as the capacity Q increases.

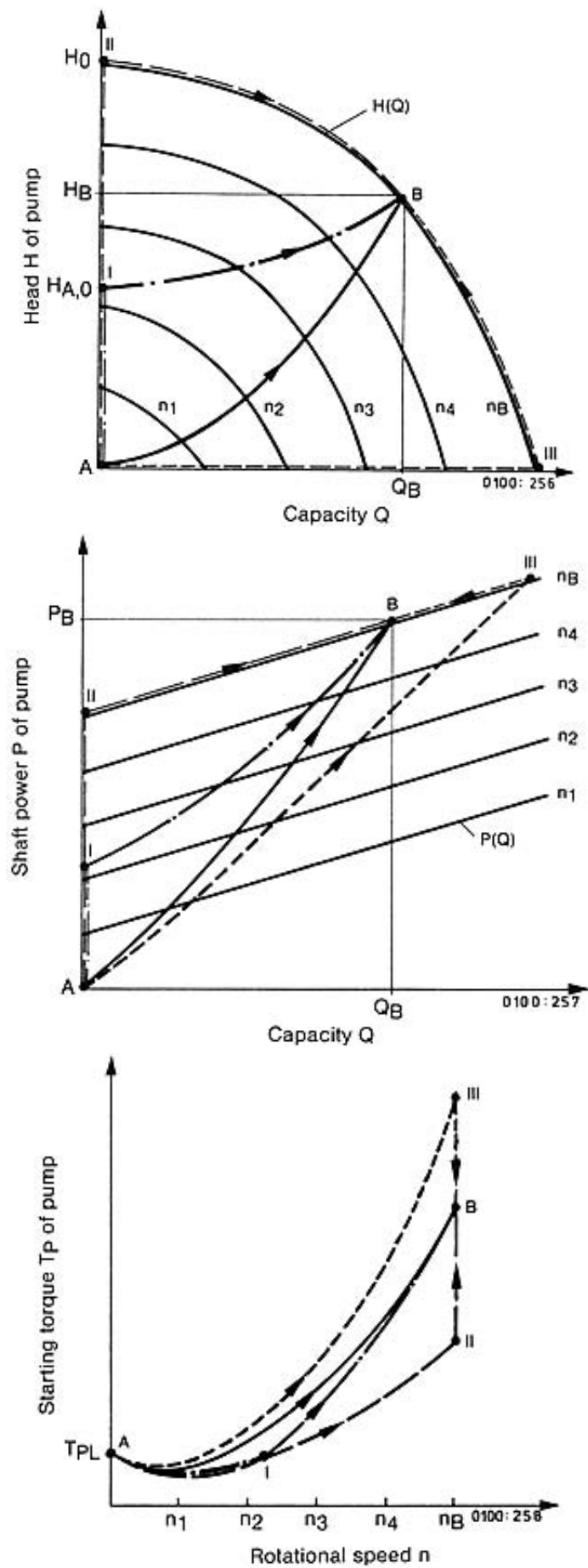
Case 1, curve A-B (see illustration).

The pump delivers into a very short, filled pipeline with open valves, the starting time t_{aQ} of the masses of liquid is negligible, no static counter-pressure $H_{A,0}$.

In this case the pump attains the operating point B in the starting time t_a of pump and motor along a direct path, without reaching the shutoff head H_0 (head).

Case 2, curve A-I-B (see illustration).

The system characteristic curve comprises a static portion $H_{A,0}$ which is checked (held) by a check valve or nonreturn valve (valves and fittings) (e.g. difference in elevations of two water levels, or boiler pressure). In the case of a primed, short pipeline, the check valve or nonreturn valve will open at I after the head $H = H_{A,0}$ has been attained, and the pump will start to deliver. The starting time t_{aQ} of the masses of liquid is negligible. The s.t. Increases along the curve I-B, the position of which is determined by the system characteristic curve.



Heads H, shaft power P and starting torque T_p for different operating conditions

Case 1, curve A-B ———
 Case 2, curve A-I-B - · - · -
 Case 3, curve A-II-B - - - -
 Case 4, curve A-III-B - - - -

Case 3, curve A-II-B (see illustration).

The pump delivers into a long pipeline (piping) filled with liquid, and the starting time t_a of the masses of liquid is considerably greater than the starting time t_a (starting process) of the pump and motor.

If we consider the extreme case where the acceleration of the liquid only starts after the starting time t_a has elapsed, the pump will attain the operating speed n_B (rotational speed) at point II, after the starting time t_a has elapsed. Thereafter the pump will attain the operating point B in the starting time t_{aQ} at an increasing s.t. and an approximately constant rotational speed n_B .

The extreme case described here also corresponds to starting up the pump against a closed discharge valve, which is only opened when the pump has attained its operating speed at point II.

Depending on the ratio of the starting times t_a and t_{aQ} and on the valves and fittings incorporated in the system, the s.t. will adopt a pattern situated between the extreme cases 1 and 3 (i.e. between the curves A-B and A-II-B). Within these limits the power ratio P_0/P_B (P_0 is the power at speed n_B and capacity 0) and the head ratio $H_{A,0}/H_B$, being parameters of the pump and plant, determine the exact curve.

Case 4, curve A-III-B (see illustration).

The pump delivers into a pressurized empty pipeline. It will therefore attain first of all the overload point III at a head $H = 0$, and from there it will gradually attain the operating point B as the pipeline fills up and the head increases, whilst the s.t. decreases at the same time.

In the case of centrifugal pumps with a high specific speed (e.g. propeller pumps), the pattern of the s.t. will be different, due to the gradually decreasing shaft power absorbed as the capacity increases (characteristic curve). During start-up against a closed discharge valve, a higher s.t. will be required in this case (in the $T_p = f(n)$ graph, points I and II will be situated above the A-B curve), whereas during start-up into an empty, pressurized pipeline, the s.t. will be smaller (point III will be situated below B). This characteristic must be taken into consideration during the start-up procedure (propeller pump).

Starting Transformer

Anlaßtransformator

Transformateur de démarrage

Three-phase transformer (three-phase current). The s.t. enables asynchronous motors with squirrel-cage rotor and synchronous motors to start smoothly at a reduced voltage. S.t.'s are normally designed as autotransformers with a single winding tapped at several points.

The load on the mains and the starting torque (starting process) decrease roughly proportionately to the square of the decrease in voltage.

Start-Up

Anfahren

Démarrage

see Starting Process

Start-Up Time

Anlaufzeit
Temps de démarrage

see [Starting Process](#)

Static Pressure

Statischer Druck
Pression statique

see [Pressure](#)

Steady Flow

Stationäre Strömung
Écoulement permanent

The flow of a [fluid](#) is said to be steady if its velocity, [pressure](#) and all the numerical values relating to its substance (e.g. [density](#), [viscosity](#)) are independent of time at every point of the flow field. A s.f. is in principle only possible in equilibrium (steady state) condition. A turbulent flow ([fluid dynamics](#)) is strictly speaking a [unsteady flow](#), because the turbulent fluctuations are statistically irregular, time dependent phenomena; however, a turbulent flow can be treated as a s.f. with reasonably good accuracy, if the values of velocity, [pressure](#) etc. are averaged over time and these values are taken into account. The same applies to the [absolute velocity](#) in close proximity to and in the [impellers](#) of [centrifugal pumps](#), which strictly speaking, is unsteady due to the finite number of vanes; however, it is considered to be steady by taking the values averaged over time into account.

Storm Water Pump

Regenwasserpumpe
Pompe d'eau pure

see [Drainage Pump](#), [Land Reclamation Pump](#)

Strain Gauge Technology

Dehnungsmeßtechnik
Technique extensométrique

see [Measuring Technique](#)

Stress Corrosion Cracking

Spannungsrißkorrosion
Corrosion mécanique

see [Corrosion](#)

Stripping System

Restlenzanlage

Installation d'assèchement complémentaire

The s.s. is an auxiliary device on cargo oil pumps which enables the continuous discharging (unloading) of the oil in tankers to take place without any manual intervention right up to the end of the discharging operation.

The purpose of the s.s. is to prevent the breaking off of the flow by gas intrusions into the suction pipe, which occur e.g. during residual pump-out operations. This is achieved by incorporating a gas separator in the suction pipe (see illustration), in which any gas which has penetrated into the suction pipe is separated out and sucked away by a vacuum plant (vacuum pumping station and vacuum tank). If the accumulating volume of gas exceeds a preset value (level controller), the capacity of the cargo oil pump is throttled (throttling valve) as the liquid level in the gas separator decreases.

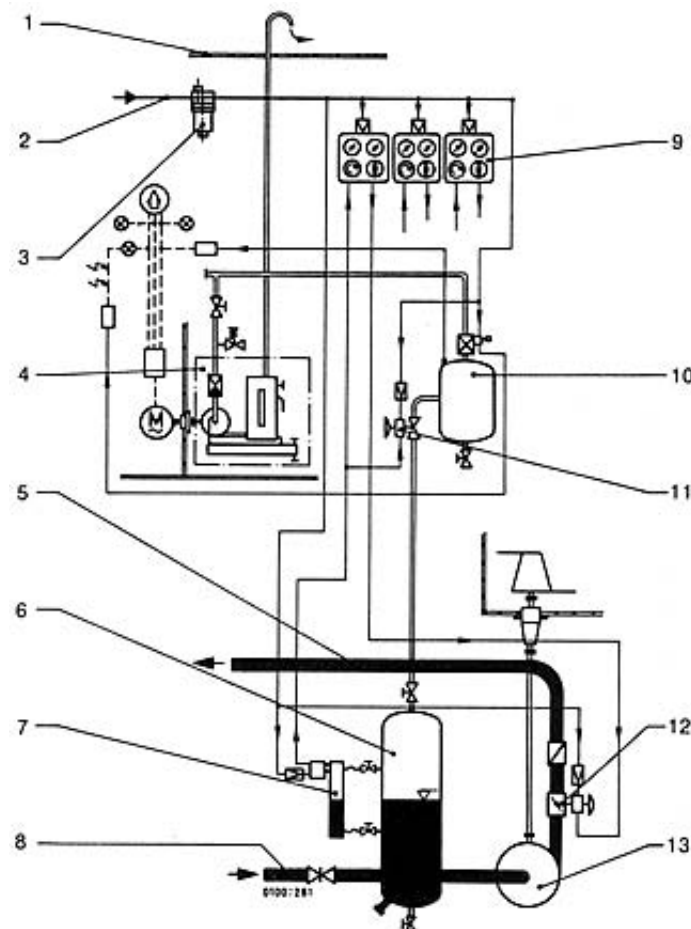


Diagram of a residual pump-out device for a cargo oil pump on a product tanker 1 ships deck; 2 compressed air; 3 compressed air cleaner; 4 vacuum station; 5 discharge line (on shore); 6 gas separator; 7 level controller; 8 suction pipe (out of the tank); 9 manual/automatic station; 10 vacuum tank; 11 control valve; 12 throttling valve; 13 cargo oil pump

Stuffing Box

Stopfbuchse

Presse-étoupe

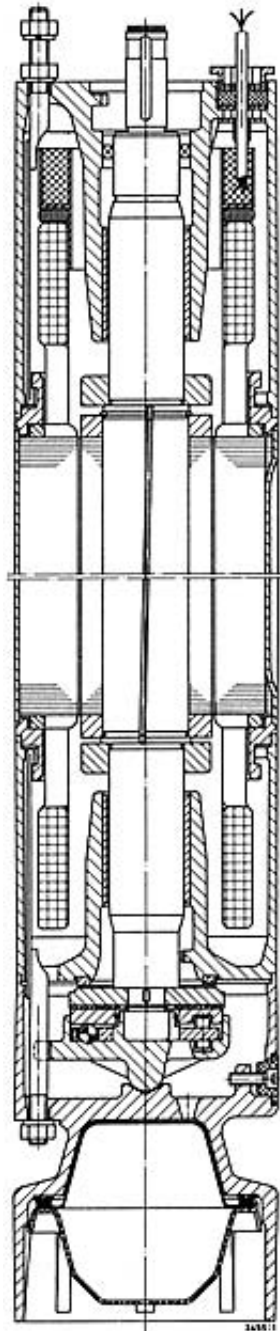
see Shaft Seals

Submersible Motor

Unterflüssigkeitsmotor

Moteur immergé dans un liquide

A. s.m. (when pumping water also called an underwater motor) is an asynchronous motor in squirrel cage design, that is wetted on the *inside and outside*, or rather is completely submerged in and filled with the medium (illustration). It can be used to power most vertical multistage pumps in drilling holes, shafts, welts, containers or bodies of water, and as opposed to air-filled s.m.'s (submersible motor pump) are designed for depths up to several kilometers (underwater motor pump).



Submersible motor for driving an underwater motor pump

Based on the slender shape designed to work in narrow shafts, the friction losses of the rotor in the medium (proportional D^4) are minimized (impeller side friction). The plain bearings of the rotor are lubricated by the liquid in the motor. The stator windings must be insulated by water resistant materials if they are not protected by a stator can (canned motor pump). The heat generated by the s.m. is passed into the medium. S.m. have power ratings

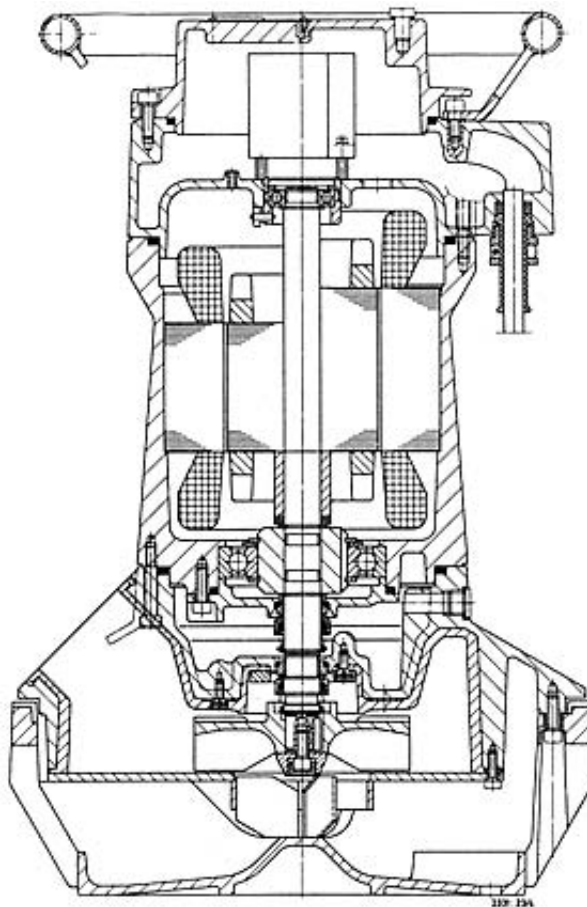
from fractional kW to 3500 kW and rated voltages as high as 10 kV for large motors. As opposed to s.m., wet rotor motors are wetted only *on the inside*.

Submersible Motor Pump

Tauchmotorpumpe

Groupe immergé

A s.m.p. is a pump with an asynchronous motor with the external motor housing designed to be submerged, so that the motor can be cooled by the surrounding liquid handled. Along with underwater motor pumps, whose motors are filled with liquid (submersible motor) and are designed to work at great depth there are s.m.p.'s whose motors are filled with air (sometimes but seldom with oil) which are designed to work at lesser depths (up to 80 m depth). Some are only temporary submerged in the liquid. These last, semi-submerged pumps are cooled partially by heat conduction to the medium and possibly over the large surfaces of the motor casing to the atmosphere by convection or by forced ventilation. S.m.p.'s can also be built in explosion protected design.



Submersible motor pump used as a sewage pump

A vertically positioned s.m.p. (illustration) is used for cases of flooding, sewage (sewage pump), as well as for drainage (Fig. 2 under drainage pump) of flooded basements, ditches, but also used (with more powerful motor) in place of tubular casing pumps with dry motors (Fig. 5 under cooling water pump, or Fig. 1 under irrigation pump). They are usually constructed in a single stage and do not require a suction pipe since the medium enters from all sides. Smaller units are transportable, and most units are equipped with an automatic start and stop switch that respond to water level. The materials of construction of s.m.p.'s must be matched up to the contents of the medium, such as sand and other impurities.

Submersible Pump

Tauchpumpe
Pompe immergée

The s.p. is a centrifugal pump with a pump casing flooded by (submerged in) the medium pumped (wet installation). Therefore a s.p. usually has no suction pipe (pumping plant). In the case of submersible motor pumps, both the pump casing and the motor are flooded, in contrast to s.p.'s.

Suction Behaviour

Saugverhalten
Comportement à l'aspiration

The s.b. of a centrifugal pump, i.e. the reaction of the pump to changing NPSH value of the installation (net positive suction head) at the operating point is determined by cavitation in the suction part of the pumping plant and in the pump itself. Within the meaning of s.b., a centrifugal pump can be operated continuously at the rotational speed n_L , capacity Q_L and head H_L and with the medium specified in the supply contract, if the NPSH value of the installation is greater than, or equal to the NPSH value of the pump (net positive suction head), i.e. if

$$\text{NPSH}_{\text{av}} \geq \text{NPSH}_{\text{req.}}$$

The s.b. of a centrifugal pump can be plotted as shown in Fig. 8 under measuring technique. As the NPSH value of the installation decreases, the head H will decrease more or less abruptly as a result of cavitation, when the pump is operating at a given operating point characterized by a capacity Q (controlled at a constant value). In order to evaluate the s.b., plotted curves of the capacity Q or of the pump efficiency η in function of the NPSH value of the installation are often used, and in many cases Q and H are replaced by the analogue characteristic numbers Φ and Ψ .

Suction Bell

Saugglocke
Tulipe d'aspiration

see Entry Nozzle

Suction Branch

Saugstutzen
Tubulure d'aspiration

see Pump Suction Branch

Suction Casing

Sauggehäuse
Corps d'aspiration

see Pump Casing

Suction Chamber

Saugkammer
Chambre d'aspiration

see [Intake Chamber](#)

Suction Head

Sanghöhe
Hauteur d'aspiration

see [Suction Behaviour](#)

Suction Impeller

Sauglaufrad
Roue aspiratrice

S.i., on [multistage pumps](#) is the first [stage impeller](#). The s.i. may for reasons of [suction behaviour](#) be shaped differently from the impellers of the other [stages](#) (see Fig. 4 under marine pump). The s.i. is *not* the same thing as an [inducer](#).

Suction Number

Saugzahl
Vitesse spécifique d'aspiration

The s.n., introduced by PFLEIDERER, is a [characteristic number](#) for the evaluation of the NPSH value ([net positive suction head](#)) of a [centrifugal pump](#) impeller, and is expressed by the following numerical value equation:

$$S = \left(\frac{n}{100} \right)^2 \cdot \frac{Q}{k \cdot \text{NPSH}_{\text{req}}^{3/2}}$$

with

n	rotational speed in min ⁻¹ ,
Q	capacity in m ³ /s,
NPSH _{req}	minimum positive suction head of pump (net positive suction head) in m,
k	= 1 - (D _{1,i} /D _{1,a}) ² ,
D _{1,i} /D _{1,a}	hub ratio at impeller inlet (subscript 1),
D _{1,a}	outer diameter at impeller inlet,
D _{1,i}	hub diameter at impeller inlet,
S	s.n. for axial impellers , 2.5 approx.; for radial impellers , 3.0 approx. (much higher s.n.'s are attained in the case of special designs).

In centrifugal pump technology, a different s.n. is often used, akin to the [specific speed](#), and expressed by the following equation:

$$S_q = n \frac{Q^{1/2}}{(g \cdot \text{NPSH}_{\text{req}})^{3/4}}$$

with

g gravitational constant.

In English and American literature, a so-called "suction specific speed" value is often used:

$$n_{qs} = n \frac{Q^{1/2}}{(NPSH_{req})^{3/4}}$$

with

n rotational speed,
 Q capacity,
 NPSH_{req} NPSH value of pump (net positive suction head),
 n_{qs} suction specific speed,

and different units are used in this connection.

Suction Pipe

Saugleitung
Conduite d'aspiration

see Pumping Plant

Suction Strainer Basket

Saugkorb
Crépine

see Pressure Loss, Valves and Fittings

Suction Volume

Absaugvolumen
Volume d'aspiration

see Venting

Super Pressure Pump

Höchstdruckpumpe
Pompe à très haute pression

The s.p.p. is a centrifugal pump with a head in excess of 1200 m (e.g. a boiler feed pump).

In contrast, we have low pressure, medium pressure and high pressure pumps.

Surface Protection

Oberflächenschutz
Protection de surface

S.p. is governed by the following factors:

- Nature of attack (corrosion, cavitation, erosion, abrasion etc.).
- Duration of attack (temporary protection during transport and storage, long term protection effect in the event of continuous stressing).
- Application (e.g. pumping of drinking water, reactor technology).
- Economics (e.g. single or multilayer coating).
- Safety (e.g. danger of secondary damage caused by the peeling off of coatings).

Painting is the most widespread form of s.p. for unalloyed iron and steel materials. Its efficacy depends largely on the pretreatment of the surface (e.g. sandblasting).

It is advisable to consult the pump manufacturer for the best protection in relation to the product application.

Surge Pressure

Druckstoß

Coup de bélier d'onde

A s.p. is generally understood to involve the more or less rapid change in static pressure in a flow in a pipeline (water flow) as a result of a more or less rapid change in the mass flow. The elasticity of the water and of the pipe wall play a very important part. If a change dv in the flow velocity v takes place at a point (disturbance point) in the piping (e.g. by actuating a valve, or as a result of the starting process or of the control or shutdown of a centrifugal pump, due to a pipe fracture), the resulting transformation of kinetic energy into other forms of energy causes the static pressure p to alter by an amount dp . This pressure change dp , according to JOUKOWSKY, amounts to:

$$dp = \rho \cdot a \cdot dv \quad (1)$$

with

- ρ density of flowing medium, and
 a sound velocity in the flowing medium.

The pressure and velocity alteration in accordance with equation (1) does not remain confined to the disturbance point, but spreads at the speed of sound a (sound velocity) upstream and downstream (direct wave). At points of discontinuity such as pipe branches, changes of cross-section vessels, pumps or valves and fittings in the line, the waves are reflected to a greater or lesser extent, depending on the boundary conditions, with a reversal of phase and amplitude (indirect waves). The condition at a given point and a given instant in time will be obtained by superposition of all the waves which have arrived at the given point up to the instant of time considered. The full pressure alteration $\Delta p = \rho \cdot a \cdot \Delta v$ corresponding to a full velocity alteration Δv can only develop if

$$t_s \leq t_r = \frac{2 \cdot L}{a} \quad (2)$$

where t_s is the period of the velocity fluctuation, thistle time taken for the reflected wave to return and L is the distance between the closest reflection point and the disturbance generation point. Under these conditions, the reflected waves arrive too late at the disturbance generation point to be able to have any attenuating influence on the disturbance condition.

The pressure fluctuations themselves, and the maximum pressures which arise can lead to excessive stresses on the installation. The minimum pressure can sink as low as the vapour pressure of the fluid. Cavitation can then occur locally, separating the liquid column into two separate columns (separation). A subsequent velocity reversal causes the separated columns to move towards one another and collide at high speed (water hammer). This is a pressure surge in the strict sense, shock condensation.

Unduly high or low-pressures can often only be avoided by means of suitable *pressure surge protection devices*.

According to (1) the pressure change is small if ρ , a and Δv remain small; therefore at low density fluids, low sound velocities, low flow velocities (e.g. by means of large bore pipelines). However, in general these magnitudes can be influenced only slightly or with relatively high expenditure.

Relationship (2) states implicitly that the pressure changes Δp become smaller in proportion to the increase in switchover time t_s in relation to the reflection time t_r of the installation.

There are two cases to be considered here:

1. t_s is fixed. Attempts must be made to modify t_r to the extent that t_r is about one fifth of t_s . Such attempts include: as short a piping run as possible; provision of intermediate reflection points in long pipelines (surge tank at the apex, stand pipes at intermediate points and apexes, venting the line at apexes, downstream from suction tank).
2. t_r is fixed. Attempts must be made to modify t_s to the extent that it is approximately 5 times longer than t_r . Such attempts include: valves with a suitably selected closing characteristic; prolongation of the rundown (to a standstill) time of the pump after switching off (flywheel masses); make-up liquid out of a reservoir (pressure vessel, stand pipe, pressureless vessels, downstream from suction tank); leading away of liquid masses which tend to dam up suddenly (by-passes, relief valves, reflux through pump, by-pass).

The selection and dimensioning of the required s.p. protection devices are so complex that in general they have to be calculated by EDP methods; please also refer to DVGW notice W 303 "Dynamic Pressure Variations in Water Supply Plants."

Swing-Type Check Valve

Rückschlagklappe
Clapet anti-retour

see [Valves and Fittings](#)

Swing-Type Valve

Klappe
Clapet battant

see [Valves and Fittings](#)

Switching Frequency

Schaltzahl
Nombre de démarrages

see [Pressure Vessel](#)

Switching-On Pressure

Einschaltdruck
Pression de mise en circuit

see [Pumping Plant](#)

Switch-Off Pressure

Ausschaltdruck
Pression de mise hors circuit

see [Pumping Plant](#)

Synchronous Speed

Synchrondrehzahl
Vitesse synchrone

S.s. is the rotational speed of synchronous machines (three-phase generators and synchronous motors, drive). The s.s. is given by the frequency f of the power supply and by the number of poles (number of pole pairs p):

$$n_{\text{synchronous}} = \frac{f}{p}$$

(Table, see under number of poles).

System Characteristic Curve

Anlagenkennlinie
Courbe caractéristique de réseau

S.c.c., also called "pipeline characteristic", is a curve (often parabola-shaped) which represents the relationship between the head H_A of the plant and the capacity Q .

The intersection of the throttling curve $H(Q)$ (characteristic curve) specific to the pump with the characteristic curve $H_A(Q)$ specific to the plant determines the operating point. The illustrations 1 to 4 under the heading operating point depict parabola-shaped s.c.c.'s $H_A(Q)$ which do not pass through the origin of the QH coordinate system and which fan out progressively as the throttling increases, as illustrated in Fig. 4 under the heading operating point. The shape and position of the s.c.c. result from the equation for the head H_A of the plant:

$$H_A = \frac{p_a - p_e}{\rho \cdot g} + \frac{v_a^2 - v_e^2}{2g} + z_a - z_e + H_{v,e,s} + H_{v,d,a}$$

where

p static pressure,
 v flow velocity,
 z geodetic altitude,
 H_v head loss (pressure loss, pressure head),
 ρ density of pumped medium,
 g gravitational constant.

The subscripts define the corresponding locations:

e defines the level of the inlet cross-section of the plant (e.g. on Fig. 2 under head, a cross-section through the suction vessel),
 a defines the develop the outlet cross-section of the plant (as under e above, but a crosssection through the discharge vessel),
 s pump suction branch,
 d pump discharge branch,
 e,s relate to the suction side of the plant (see Fig. 2 under head), i.e. to the portion between the cross-sections e and s ,
 d,a relate to the discharge side of the plant, between the cross-sections d and a .

The term $(v_a^2 - v_e^2)/2$ is a negligible quantity if the cross-sections of the plant at e and a are of adequate size, or of approximately the same size. In practice, this expression is seldom of any significance (except in the case of high discharge velocities v_a , e.g. in the case of tire-fighting equipment).

The terms $(p_a - p_e)/\rho g$ and $(z_a - z_e)$ are independent of the capacity Q of the pump. Therefore the relationship between the head H_A of the plant and the capacity Q is evidenced mainly in the head losses H_v , which can be calculated by means of the formula

$$H_v = \zeta \frac{v^2}{2g}$$

where

ζ loss coefficient (pressure loss),

v flow velocity in a characteristic cross-section (of cross-sectional area A).

Because $v = Q/A$, and assuming a constant value for ζ if the REYNOLDS numbers (model laws) are high enough, we have: $H_v \sim Q^2$.

The reason for the parabola shape of the s.c.c. thus becomes clear. At the apex of the s.c.c., for $Q = 0$, we have:

$$H_{A,0} = \frac{p_a - p_e}{\rho \cdot g} + z_a - z_e .$$

From the above equation, it follows that the s.c.c. can assume any arbitrary position within the QH_A system of coordinates, if the vessel system pressures p_a and p_e vary, and hence the so-called geodetic head $H_{geo} = z_a - z_e$ varies. $H_{A,0}$ is often referred to as H_{stat} in the technical literature.

Thus for instance, we have the following equations for a cooling water system (cooling water pump) consisting of a suction pipe drawing water out of a river, a cooling water pump and a discharge line leading into a cooling tower basin (Fig. 1):

$$\begin{aligned} H_A &= z_a - z_e + H_{v,e,s} + H_{v,d,a} , \\ H_{A,0} &= z_a - z_e . \end{aligned}$$

where

$p_a = p_e = p_b$ barometric pressure,

$v_a = 0$ (negligible flow velocity in the cooling tower basin),

$v_e = 0$ (negligible inlet velocity into the inlet duct from the river),

H_v head loss resulting from pressure losses at inlet and outlet, pressure losses caused by pipe friction, passage through the condenser, through valves and fittings, through elbows, by abrupt changes of cross-section etc.,

$z_a - z_e$ difference in geodetic altitude of the water level in the cooling tower basin and in the river bed.

As the water level in the river z_e fluctuates, the s.c.c.'s will be displaced on the diagram as illustrated in Fig. 2.

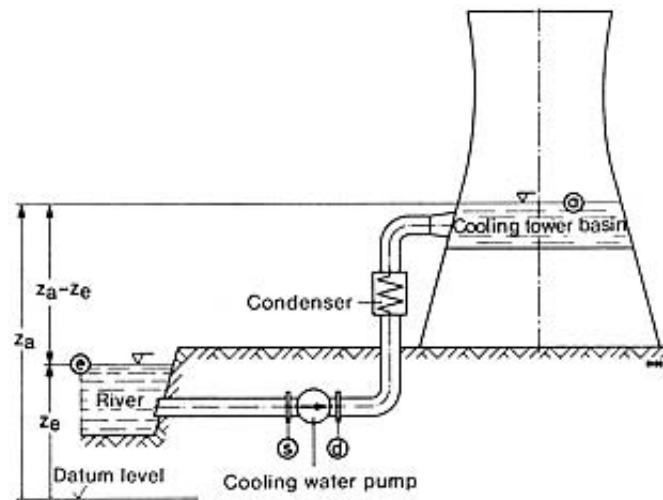


Fig. 1: Diagram of a cooling tower system

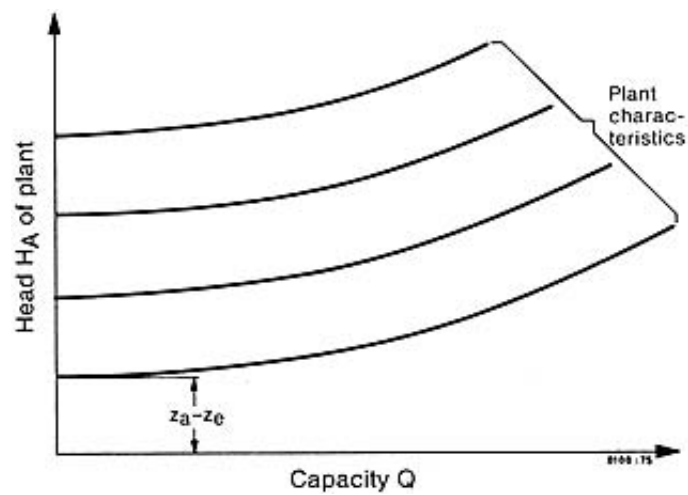


Fig. 2: System characteristics of cooling water system illustrated in Fig. 1 (diagrammatic representation)

T

Table of Corrosion Resistance

Beständigkeitstabelle

Table de résistance à la corrosion

During the selection of materials for centrifugal pump parts, the durability of the individual materials against attacks by corrosion is often a significant criteria. As a source of orientation, t.o.c.r. are often used to judge metallic materials. These charts should not, however, be used as allknowing "recipe" books, because the complex stressing of various pump materials, requires knowledge through many years of experience.

Thus, the following t.o.c.r. is meant only as a general *orientation for selecting materials only with respect to corrosive resistance*.

The symbols in the charts have the following meanings:

- "+" The material is useable in this medium throughout the entire concentration range up until the saturation point of the liquid, that is, the corrosion rate amounts to < 0.1 mm/a, and localized peak values, due to pitting, crevice, stress or vibration corrosion (corrosion) are no longer expected. Where temperatures and concentrations are given this material is only useable up until these boundaries.
- "-" The material is not to be used in this medium, because either the corrosion rate is > 0.1 mm/a or other local types of corrosion with handicap to the function could have an effect.
- "o" The material is only to be used under extremely specific conditions, for example at selected temperature or concentration levels of the medium. In such cases, one must consult an expert.

An "empty field" indicates:

The material has no practical use in this medium, its corrosion resistance is thus of no importance. In case of doubt, questions can be directed to pump material specialists. (This deals in particular with the corrosion resistant materials ERN, NORIHARD® NH 153 and NORLOY® NL 252.)

The Table covers cast-metal materials (materials) exclusively, as they are used by KSB in the manufacture of pump housings and impellers. For the selection of materials in a different manufactured form we recommend looking at other technical literature (such as DECHEMATables). This also goes for synthetic, and ceramic materials.

In review, the t.o.c.r. deals only with certain criteria of corrosion resistance. Other aspects and criteria such as economics are not touched upon in this section (selection of materials).

Table of corrosion resistance

Pumped medium	GG-25	GGG-40; GGG-40.3	GS-C 25	ERN	Ni-Resist D2	NORIHARD® NH 153	NORLOY® NL 25 2	I.4008	I.4308	I.4408	I.4500	NORICID® 9.4306	NORIDUR® 9.4460	NORICLOR® NC 24 6	G-CuAl 10 Ni	Special materials / remarks

KSB Bombas Hidráulicas S. A.

443

444

[illegible]

KSB Bombas Hidráulicas S. A.

Centrifugal Pump Lexicon																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																			
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need isocorrosion curves!

hard rubber

Hastelloy C

Hastelloy C, hard
rubber (with
limitation)

- inflammable

 $\leq 130\text{ }^{\circ}\text{C}$

449

Tank Farm Pump

Tanklagerpumpe

Pompe de parcs à réservoirs de stockage

T.f.p.'s are centrifugal pumps installed in tank farms for the pumping of liquid crude, intermediate and finished products in chemical and petrochemical plants and in oil refineries. Various pump types are used as t.f.p.'s, depending on the required performance data, and on whether the liquid pumped is highly volatile, flammable or toxic: horizontal pumps (single or multistage pump), single or double suction pumps (multisuction pump), close-coupled pump sets in the form of inline pumps (Fig. 1), grandness pumps with canned motors (wet rotor motor), vertical submersible pump (shaft sump pump) as illustrated in Fig. 2. In many cases, the adoption of a particular pump type is demanded in order to comply with official national regulations. Thus e.g. several countries demand that tanks containing flammable or toxic liquids must be equipped with submersible pumps with a pump shaft sealed (shaft seals) leaktight at its passage out of the tank. or with self-priming pumps arranged on the tank, or again with submersible motor pumps.

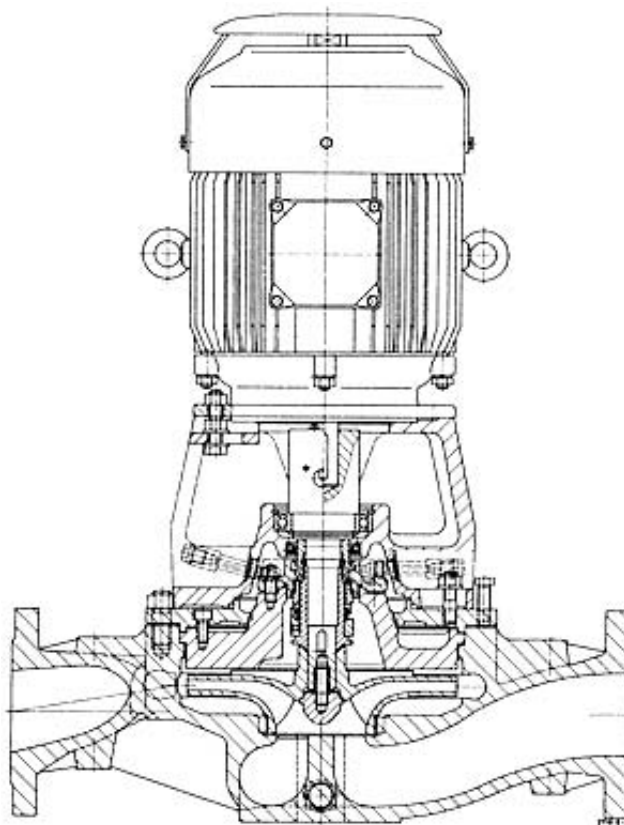


Fig. 1: Chemical inline pump (tank farm pump)

Where highly volatile and flammable fluids are pumped, ductile materials such as cast steel and spheroidal graphite cast iron are adopted generally for safety reasons (selection of materials, corrosion).

With the exception of submersible pumps with a separate discharge line (pumping plant), on which the shaft seal is not in contact with the fluid pumped during operation, mechanical seals are generally fitted to the other pump types equipped with conventional drives. Submersible pumps are generally so designed that they can be inserted in standard size manholes, and the bookplate of the pump then seals the manhole.

The use of submersible pumps becomes problematical when the height of the tank requires very great installation depths. Quite apart from the more difficult maintenance of such pumps which can only be pulled out of the tank with the aid of portable hoists, difficulties arise in connection with the large number of intermediate bearings lubricated (plain bearing) by the product pumped; this happens particularly when the liquid level is low and when operation is intermittent, or when the liquid pumped is very corrosive or tends to crystallize (e.g. caustic soda and caustic potash solutions).

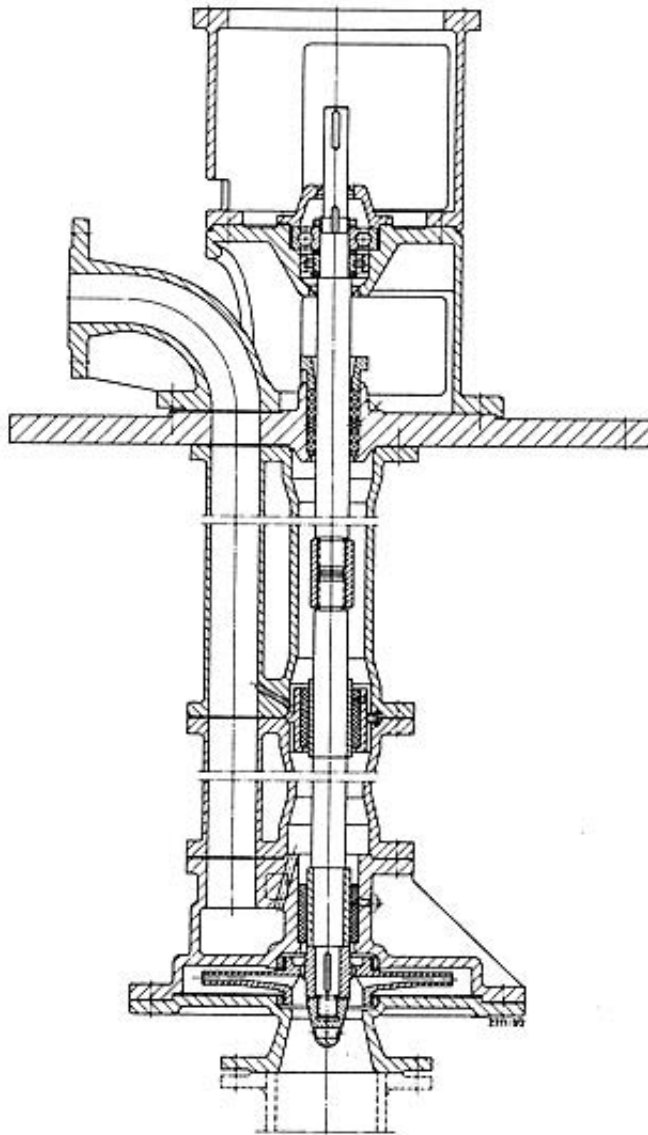


Fig. 2: Chemical submersible pump (tank farm pump)

When melts or viscous liquids (viscosity) have to be pumped, the t.f.p. is often fitted with a heating jacket, in order to keep the liquid within the pump in pumpable condition. A special construction is necessary for liquefied gas tank farms (liquefied gas pump).

Tank Farm Submersible Pump

Tanklager-U-Pumpe

Pompe submersible de parcs Á réservoirs de stockage

Fig. 1: Cavern installation with fixed water bed

The t.f.s.p. is a special type of submersible motor pump with submersible motor. It is used in underground caverns for the storage of motor fuels, heating oils and liquefied gases. Fig. 1 illustrates the layout of such a cavern installation. So-called t.f.s.p.'s are used to operate these installations. According to their application, these pumps are subdivided into:

Product pumps, which pump the product out of the cavern. The density and viscosity of the product must be taken into account when designing (design duty point) pump and motor.

Leakage water pumps, which deal with the leakage water penetrating into the cavern from the surrounding rocks. This water accumulates at the bottom of the cavern because of its higher density in comparison with the product, and must not be allowed to exceed a certain level. The leakage water pumps are designed to maintain a

reasonably constant water level (Fig. 1). Because the caverns are often situated near the sea, the leakage water is often seawater, and the pumps must be made of seawater-resistant bronze or chromenickel steel (seawater pump).

In order to keep the viscosity (Fig. 4 under viscosity) at a low level for an economic operation when pumping heavy, highly viscous heating oils the oil is heated up to 60 to 95 °C. There are two possibilities for heating the product in heat exchangers outside the cavern: either the product is circulated by means of circulating pumps similar to the product pumps, or the leakage water is maintained at a higher temperature (it is also referred to as bed water in this case) by circulating it through a heat exchanger by means of a submersible motor pump. In isolated cases, superheated steam is fed into the cavern. In this case, the condensate is pumped out of the cavern together with the leakage water.

The electric cables leading to the water-filled submersible motors of the t.f.s.p.'s are led in cable protecting pipes which are also waterfilled. The cable protecting pipes are firmly attached to the motor and are led out of the cavern (though the cavern coverplate) to a water reservoir in an antechamber. An electrode switching device inside the water reservoir ensures that any leakage at the motor or at the cable protecting pipes is detected in good time.

The application limits of the submersible motor of a t.f.s.p. are set by the temperature of the surroundings and the cooling facilities (Fig. 2). A distinction is made between:

- a) cooling by means of a cooling jacket (the product pumped is led around the motor to remove the motor heat losses),
- b) cooling by means of the surrounding leakage water (a suction pot must in this case be fitted around the product pump to separate the oil from the water) and
- c) flow cooling (the motor is e.g. connected to a cooling circuit via the cable protecting pipe).

The permissible temperature is governed by the permissible loading on the plastic-insulated winding wires, switching points and lead-in.

Higher temperatures are permissible if oil-filled motors are used. Since the insulating materials for the winding wires have to be oilproof, but not necessarily waterproof, the service life of oil-filled motors is heavily dependent on the reliability of the shaft seal between the motor and pump (underwater motor pump).

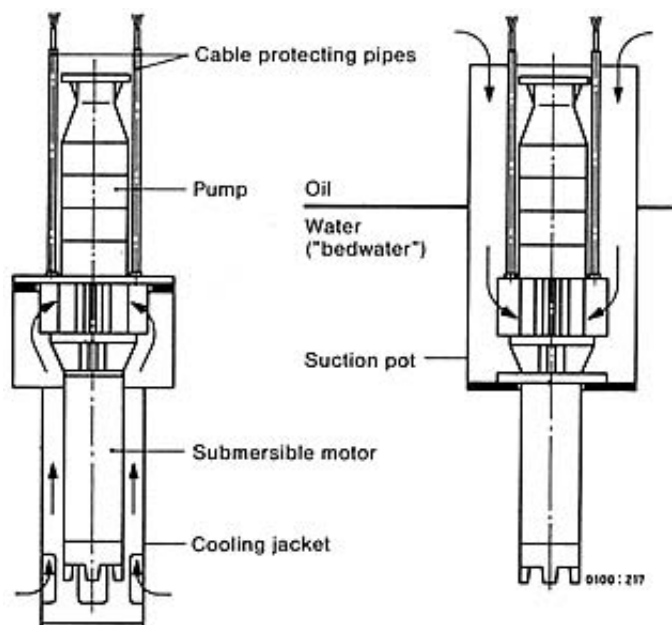


Fig. 2: Cooling of submersible motors of tank farm submersible pumps

Telemetry

Fernmessung
Téledétection

T. is a method used measuring technique for the measurement of physical magnitudes (e.g. pressure, temperature), which transmits the measured values over long distances. For this purpose, the measured magnitude is converted at the place of measurement into an analogue direct current (load-independent or impressed current or voltage), into an analogue frequency or into a digital signal.

Temperature

Temperatur
Température

see Measuring Technique, Unit

Terminal Designation

Klemmenbezeichnung
Désignation des bornes

The t.d.'s of electrical machines and appliances (motors, drive, transformers, instrument transformers, starters, positioners, power supply cables etc.) are the connection designations laid down in German standards DIN 42 400, DIN 42 402, DIN 42 403, DIN 42 404 and DIN 46 199. These standards replace the stipulations in accordance with VDE 0570. Figs. 1 to 3 indicate the t.d.'s on three-phase (asynchronous motor) and direct current motors, and the outdated designations in accordance with VDE 0570 are indicated in brackets.

The t.d.'s for three-phase motors are:

L1, L2, L3 (R, S, T)	busbars,
U1, V1, W1 (U,V,W)	stator winding, feeds,
U2, V2, W2 (X,Y,Z)	stator winding, ends,
u, v, w	rotor winding of three-phase motor and terminals of secondary starter.

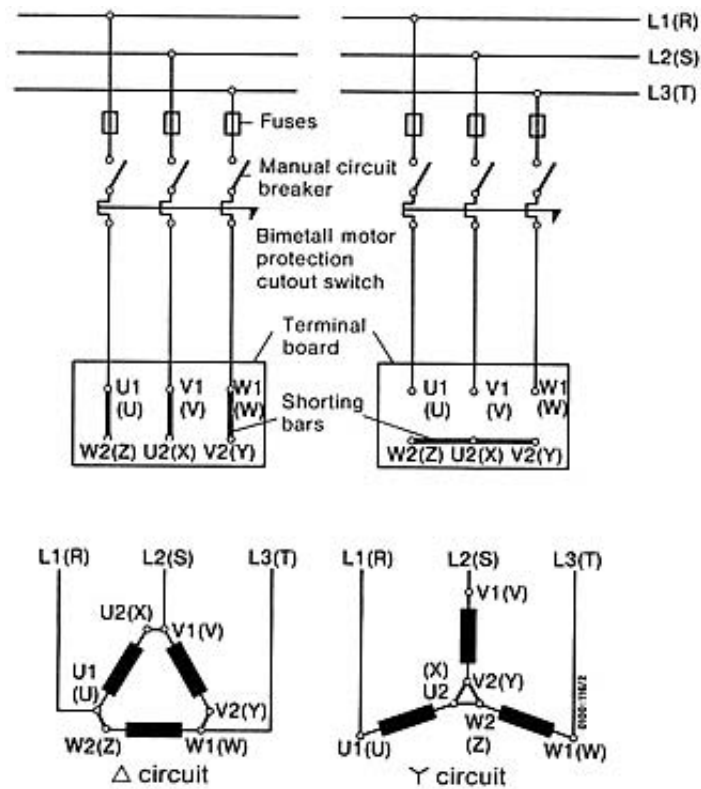


Fig. 1: Three-phase squirrel-cage rotor motor for direct-on-line starting
 Note: On the star-delta starter switch all six terminals U1, V1, W1, U2, V2, W2 lead to the switch; the shorting bars on the terminal board can be dispensed with

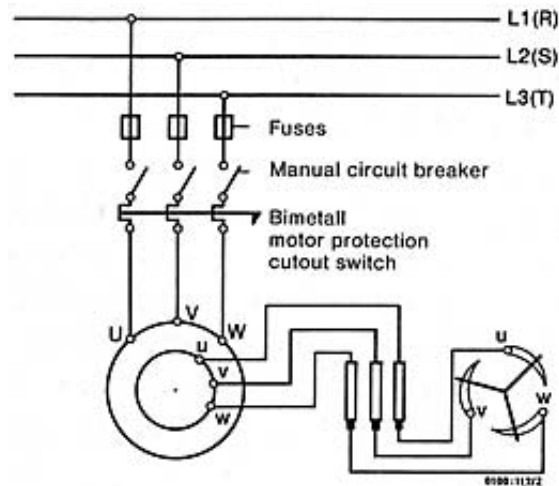


Fig. 2: Three-phase slip-ring motor with rotor rheostat (starter controller)

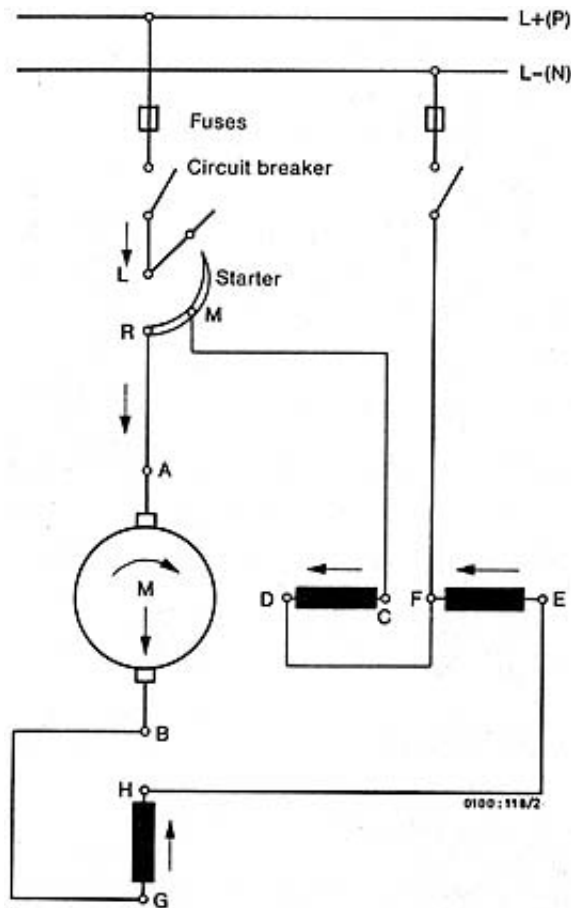


Fig. 3: Direct current compound-wound motor with starter

The t.d.'s for direct current motors are:

L + (P), L - (N) busbars,

A-B armature winding,

C-D shunt winding,

E-F series winding for exciter coil with own armature current,

G, H interposes or compensating winding, or interpose and compensating windings,

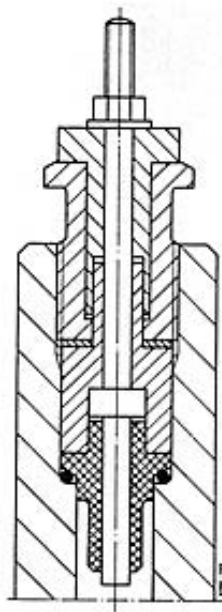
L, M, R starter.

Terminal Gland

Leiterdurchführung

Passage de conducteur

The t.g., earlier known as the cable gland, directs the conductor through the wall of the motor housing into the windings of the motor. A pressure sealed t.g., such as for submerged wet rotor motors (e.g. in grandness pumps) consists of a copper connector bolt, insulated from the pump housing, sealed by an O-ring and pressure sealed by a threaded bushing (illustration). It is not to be confused with a cable lead-in.



Terminal gland of a glandless pump with wet motor winding (wet rotor motor)

Test Bed

Prüffeld
Champs d'essai

see [Pump Test Bed](#)

Theorem of Momentum

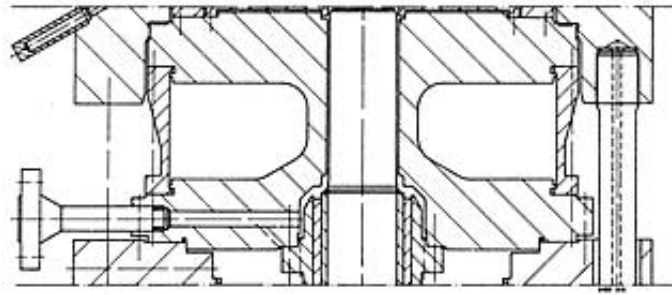
Impulssatz
Théorème des quantités de mouvement

see [Fluid Dynamics](#)

Thermal Barrier

Wärmesperre
Barrière Thermique

There is a t.b. between the pump and the motor of glandless circulating pump to prevent the heat of the hot pump from transferring to the cold motor, so that the insulation of the stator winding, (asynchronous motor), which has a limited temperature resistance, is not threatened. A t.b. can be cooled with water, but to prevent cooling water and energy losses, it is desirable to not cool with water, but rather to decrease the heat transfer. It is, therefore, important, when designing the pump, that the cross sections transferring the heat be built as small as possible, with air pockets (see illustration), and that poor heat conducting materials are used in construction. The t.b. must be able to resist mechanical strain, since the tie bolts clamp the t.b. between the motor casing and the pump casing.



Air-cooled thermal barrier of a grandness circulating pump

Thermoelectric Series

Thermoelektrische Spannungsreihe

Chaîne des forces électromotrices

The heating up of the soldered or welded thermojunction of two metals (thermo-couple, e.g. for temperature measurements, Fig. 16 under measuring technique) with cold free ends produces a so-called thermoelectric voltage. This voltage is the difference between the values given in the list below, which are related to platinum as zero point and to a 100 °C temperature difference:

Bismuth \parallel axis	-7.70 mV
Bismuth \perp axis	-5.20 mV
Constantan	-3.47 to -3.04 mV
Cobalt	-1.99 to -1.52 mV
Nickel	- 1.94 to - 1.20 mV
Mercury	- 0.07 to + 0.04 mV
Platinum	\pm 0 mV
Graphite	0.22 mV
Carbon	0.25 to 0.30 mV
Tantalum	0.34 to 0.51 mV
Tin	0.40 to 0.44 mV
Lead	0.41 to 0.46 mV
Magnesium	0.40 to 0.43 mV
Aluminium	0.37 to 0.41 mV
Tungsten	0.65 to 0.90 mV
Rhodium	0.65 mV
Silver	0.67 to 0.79 mV
Copper	0.72 to 0.77 mV
Steel	0.77 mV
Zinc	0.60 to 0.79 mV
Manganin	0.57 to 0.82 mV
Iridium	0.65 to 0.68 mV
Gold	0.56 to 0.80 mV
Cadmium	0.85 to 0.92 mV
Molybdenum	1.16 to 1.31 mV
Iron	1.87 to 1.89 mV
Chrome-nickel	2.20 mV
Antimony	4.70 to 4.86 mV
Silicium	44.80 mV
Tellurium	50.00 mV

Conventional thermocouples include:

Copper-Constantan	up to 500 °C
Nickel-Chrome-Constantan	up to 800 °C
Iron-Constantan	up to 900 °C
Nickel-Chrome-Nickel	up to 1200 °C
Platinum-Rhodium-Platinum	up to 1600 °C
Platinum-Rhodium-Platinum-Rhodium	up to 1800 °C
Tungsten-Tantalum*)	up to 2000 °C
Tungsten-Molybdenum*)	up to 2500 °C
Tantalum-Molybdenum*)	up to 2500 °C
Tungsten-TungstenMolybdenum*)	up to 2900 °C

*) only for laboratory tests, not generally available commercially

Three-Phase Current

Drehstrom

Courant triphasé

T.p.c. is the electrical alternating current in a so-called three-phase system. Three-phase systems consist of the three main conductors (outside wire) L1, L2, L3. In addition, the center conductor N (neutral conductor Mp) can also be laid as a fourth conductor. It is connected to the neutral or star point of the generator or of the low-tension transformer.

Fig.1 illustrates a three-phase generator diagrammatically, with the three windings a, b, c spaced 120° apart, and normally stationary in the stator. The starts of these three windings are connected to the terminals L1, L2, L3 of the network (mains) and their rear ends are joined together to the star point, to which the center conductor can also be joined (terminal designation). The d.c. current-excited rotor of the synchronous generator builds up a magnetic field which issues from the north pole and permeates the stator, returning into the south pole of the rotor at the other end. If this rotor is now rotated, the magnetic field will rotate with it at the same speed, and induce tensions (voltages) in the three stator windings; during one revolution of a two-pole generator, these voltages start at zero, reach a positive maximum value, then decrease again to zero, and subsequently rise to a negative maximum value, finally decreasing again to zero. Thus one cycle of the induced alternating voltage in each winding corresponds to one 360° revolution of the rotor. But as the windings are offset 120° to one another, the time sequence of the voltage generation described above is also offset by one-third of a cycle from one winding to the next. Now the magnitude of the generated voltage can be measured in terms of the star voltage (U_1, U_2, U_3) between the terminals 1, 2, 3 and the center conductor N. The line-to-line (interlinked) voltages U_{12}, U_{23}, U_{31} are measured between the terminals 1, 2, 3; they do not amount to twice the star voltage, but only to $\sqrt{3}$ times the star voltage. The reason is the 120° phase displacement of the individual voltages.

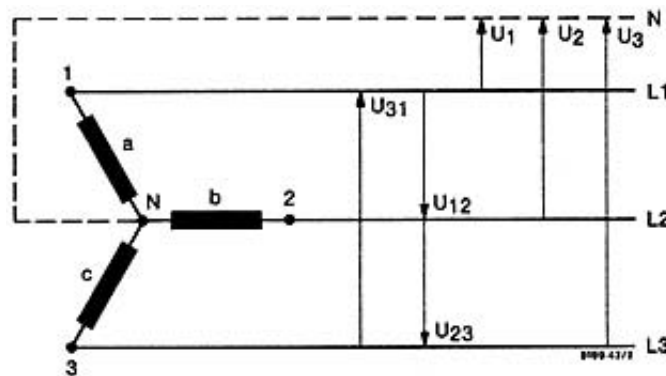


Fig. 1: Three-phase generator (diagrammatic)

Fig. 2 illustrates how the line-to-star voltages resolve into line-to-line voltages. We can see that the line-to-line voltages are effectively phase-displaced by 30° (or $1/12$ cycle) in relation to the line-to-star voltages. The magnitude of the line-to-line voltage is given by the following relationship:

$$\begin{aligned}
 U_{12} &= 2 \cdot U_1 \cdot \cos 30^\circ, \\
 &= \sqrt{3} \cdot U_1, \\
 U_1 &= U_2 = U_3, \\
 U_{12} &= U_{23} = U_{31}.
 \end{aligned}$$

If the center conductor is used in a three-phase network, the line-to-line voltage can be tapped as three-phase voltage between the three main conductors, and in addition there is single-phase voltage available of the magnitude of the line-to-line voltage between any two main conductors, and also single-phase voltage of the magnitude of the line-to-star voltage between any one main conductor and the center conductor.

If the three main conductors are the only ones used, the available three-phase and single-phase voltages are only those of the magnitude of the line-to-line voltage.

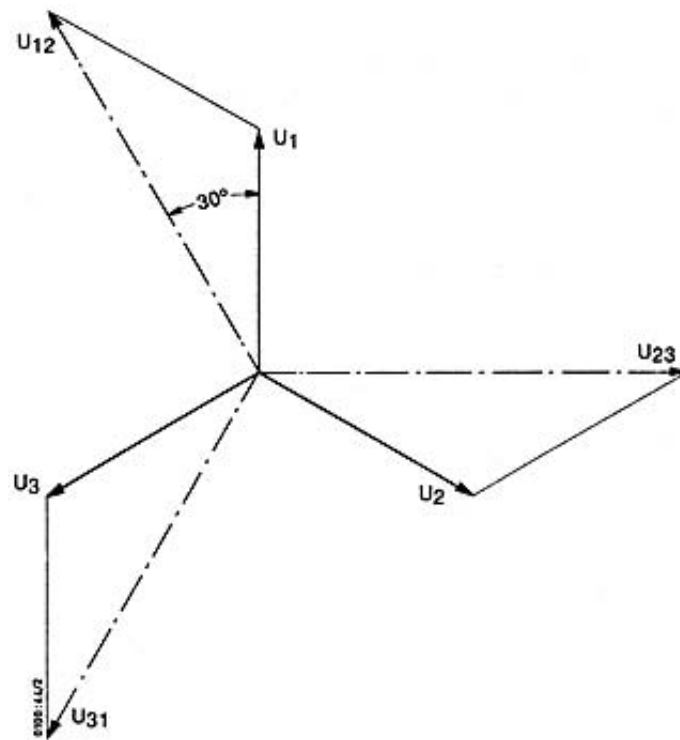


Fig. 2: Line-to-line voltages (— · —), line-to-star voltages (——)

Three-phase machinery, e.g. three-phase motors (asynchronous motor) can be connected to the mains as two different circuits: star and delta (star-delta circuit). Then, the small t.p.c. generates a rotating magnetic field (asynchronous motor) in the stator windings of three-phase motors.

Three-Phase Motor

Drehstrommotor
Moteur triphasé

see Asynchronous Motor

Three-Phase System

Dreiphasensystem
Système triphasé

see Three-phase Current

Throttling Coefficient

Drosselbeiwert
Coefficient d'étranglement

see Loss Coefficient

Throttling Control

Drosselregelung
Régulation par laminage

see [Control](#)

Throttling Curve

Drosselkurve

Courbe d'étranglement

see [Characteristic Curve](#)

Throughway Valve

Durchgangsventil

Soupape ordinaire

see [Valves and Fittings](#)

Thyristor

Thyristor

Thyristor

see [Electrical Switchgear](#)

Time

Zeit

Temps

see [Unit](#)

Torque

Drehmoment

Couple

The t. (symbol T) designates a force couple acting on a rotatable body ([impeller](#) of a [centrifugal pump](#), armature of an electric motor), which either accelerates ([starting process](#)) or decelerates said body, or act: in opposition to a reactive t. of the same magnitude [starting torque](#). The t. is the product of [force](#) and lever arm. The SI unit ([unit](#)) of t. is 1 Nm.

Torque Curve of Electric Motors

Drehmomentverlauf von E-Motoren

Allure de couple moteur

see [Starting Process](#)

Torque-Flow Pump

Freistrompumpe

Pompe à vortex

In the t.f.p., the output (head) is transmitted to the fluid pumped by a rotating disc provided with ribs (impeller) (Fig. 5 under faeces pump). The so-called torque-flow impeller is particularly suitable for pulp pumps and dirty water pumps (Fig. 4 under sewage pump). As the impeller only acts indirectly on the fluid pumped, the risk of clogging and the sensitivity towards gas contents are all reduced.

Torque Measurement

Drehmomentmessung
Mesure de couple

see Measuring Technique

Torque Meter

Torsionsdynamometer
Torsionmètre

see Measuring Technique

Torsion Rod

Drehstab
Tortilleur

see Measuring Technique

Total Head

Energiehöhe
Hauteur totale de charge

The t.h. H_{BN}, calculated from a datum level BN, is the mechanical energy of the fluid pumped, per unit weight. The unit of t.h. is 1 m. The t.h. is expressed by the equation (head)

$$H_{BN} = z + \frac{p + p_b}{\rho \cdot g} + \frac{v^2}{2g}$$

with

- z static head of point under consideration above BN,
- p_b barometric pressure (atmospheric pressure),
- p overpressure (positive) or underpressure (negative) in relation to p_b,
- ρ density of pumped media,
- g gravitational constant,
- v flow velocity.

A graphic representation of the t.h. along the length of a pumping plant can be found e.g. in DIN 24260 (specific energy).

Total Pressure

Gesamtdruck
Pression totale

see [Pressure](#)

T-Piece

T-Stück
Pièce en T

see [Fittings](#)

Transmitter

Transmitter
Transmetteur

see [Shaft Seals](#)

Traveling Screen

Siebband
Bande perforée

The t.s. is a running, belt-like, fine filter that is mainly used in cooling water cleaning of thermal power-stations; it is positioned between the rake screen and the [intake chamber](#) of the [cooling water pump](#) (Fig. 1 under intake elbow). The particles that become lodged in the screen are later removed with spraying.

Trip Speed

Durchgangsdrehzahl
Vitesse d'emballement

T.s. is the max. [rotational speed](#) attainable by the drive of the [centrifugal pump](#). The t.s. has a certain significance in the case of steam turbine-driven [boiler feed pumps](#). At very low [capacities](#), the output of the drive ([drive rating](#)), which is converted into heat, may tend to evaporate the relatively small volume of water contained inside the [boiler feed pump](#). This in turn reduces the load on the turbine set, and causes the turbine to speed up rapidly. If the boiler feed pump is driven by the turbogenerator set, the t.s. can also be reached at full load on the pump, if the steam load on the turbine is reduced due to a sudden release of the generator and if the quick-acting check valve interrupts further steam intake at overspeed. The maximum [pressure](#) at t.s. is a decisive factor for dimensioning the [pump casing](#).

Tubular Casing

Rohrgehäuse
Corps tubulaire

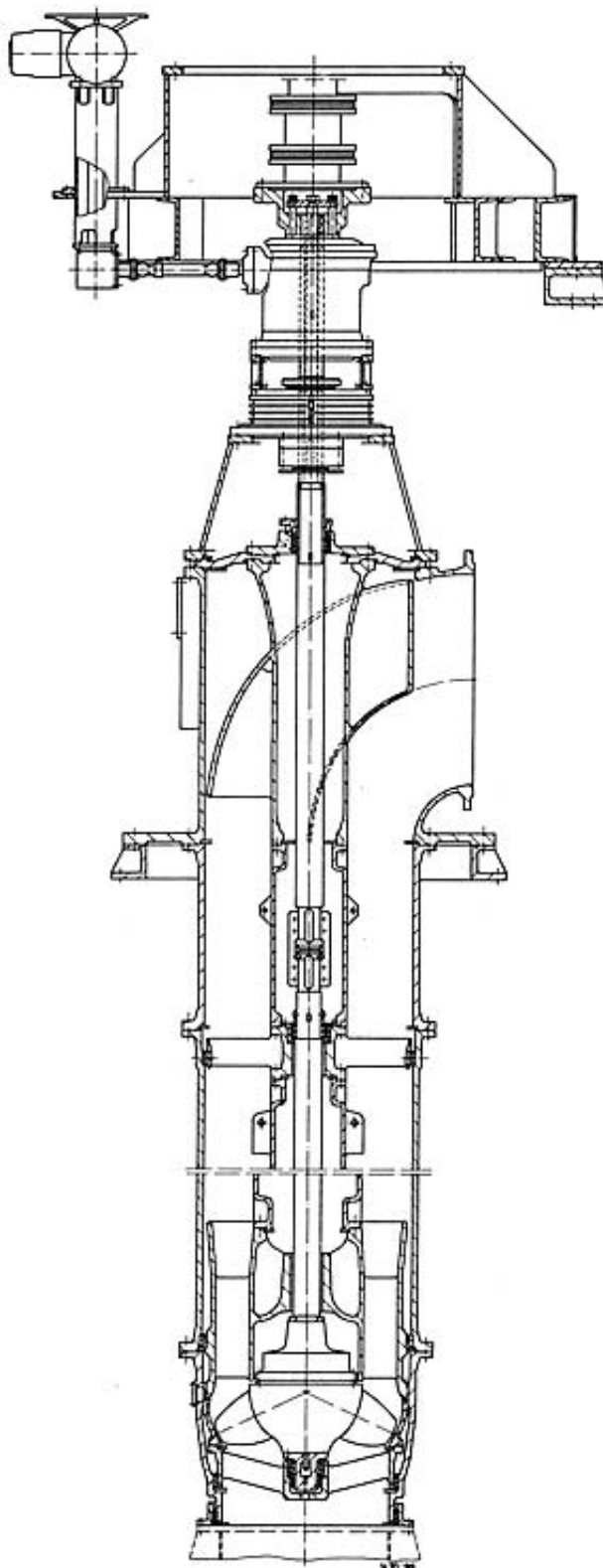
see [Pump Casing](#)

Tubular Casing Pump

Rohrgehäusepumpe
Pompe à corps tubulaire

T.c.p., a [centrifugal pump](#) in which the fluid pumped, after having passed through the [impeller](#) and the [diffuser](#), flows through the tube-shaped [pump casing](#) (concentric with the [pump shaft](#). "rising main" in the case of vertical t.c.p.'s) (see illustration). The most common t.c.p.'s are [vertical pumps](#), which in the case of [wet installation](#) have

their entry nozzle submerged in an intake chamber, which can be either open at the top or covered over, and which in the case of dry installation are bolted onto an intake elbow at their suction end, or are connected in leak-tight fashion to the cover of an intake chamber. In all cases, the inlet conditions of t.c.p.'s must be carefully studied.



Tubular casing pump with back pull out rotor

The tubular casing, including the discharge elbow, can be made of concrete for reasons of economy in certain instances (Fig. 7 under cooling water pump).

The impeller is either an axial or a mixed flow impeller, depending on the specific speed. The diffuser leads into the rising main (see illustration). Depending on the installed depth, several lengths of rising main can be screwed into one another in succession, and the pump shaft is lengthened accordingly and supported on several bearings. The discharge elbow at the end of the rising main guides the flow towards the pump discharge branch.

The shaft is led out of the tubular casing (pump casing) at the discharge elbow via a stuffing box (shaft seals). The drive stool arranged above accommodates the thrust bearing (plain bearing, anti-friction bearing) and provides access to the shaft coupling between pump and driver (drive) or gearbox (gear drive for pumps).

The drive for the control of the pump is often also accommodated in the drive stool: the impeller blade pitch adjustment of propeller pumps is usually actuated through the hollow drive shaft, and the pre-rotational swirl adjustment of mixed flow pumps is usually actuated from this spot via special universal joint shafts (cardan shafts).

The shaft guide bearings consist of grease or, more frequently, water-lubricated plain bearings. Maintenance-free hard-metal or ceramic bearings do not require a supply of clean external water or filtered pumped medium for lubrication. They can even get along on heavily contaminated, unfiltered pumped medium as lubricant.

Because dismantling and reassembly of the tubular casing is rather cumbersome in view of the weight and bulky size of the items involved, so-called back pull out t.c.p.'s (see illustration) offer the possibility of withdrawing the hydraulic components (impeller and diffuser with complete shaft and bearings) out of the tubular casing (back pull out pump). For this purpose, the discharge elbow must be designed with an aperture at the drive end having a diameter equal to the tubular casings and sealed by a cover. Then an inspection or renewal of the rotor only requires the dismantling of the driver and drive stool, and, after the cover bolts have been unscrewed, the complete internals can be pulled out. The discharge end connection, and if applicable also the suction end connection, of the tubular casing remain undisturbed. The diffuser and any intermediate bearing supports within the rising main are provided with appropriate guides (slides) at the inner wall of the tubular casing. Back pull out t.c.p.'s are preferred, particularly for the larger nominal diameters. If several identical pumps are installed, a complete spare rotor (insert) is often kept in stock for renewal purposes.

The installation possibilities of t.c.p.'s are varied, and usually governed by site conditions and characterized by the types of installation which the pump manufacturer can offer (installation of centrifugal pumps). The pump impeller is continuously flooded by the fluid pumped, in order to avoid the necessity for special suction devices or booster pumps; therefore, in the case of dry installation, the pump must be connected in watertight fashion to the intake elbow or to the cover of the intake chamber. All this amounts to additional capital investment costs, which are however compensated by the advantages that the pump casing can be very short (and therefore require only one submerged bearing in certain cases), and that the pump is accessible for inspection from the outside at all times. A t.c.p. for wet installation is submerged in the liquid at its inlet end. Because its installation is much simpler, these are to be found in greater numbers than dry installation t.c.p.'s.

Another aspect to be determined is whether the pump discharge branch of the t.c.p. should be situated above (illustration) or below floor level (Fig. 2 under irrigation pump). There are special types of elbow available for both these arrangements, and the decision on this point rests with the building site management.

Finally it must be decided how the weight of the driver is to be supported: the drive stool can be placed directly on the discharge elbow situated above floor level (provided that the complete pumping set does not become too "top-heavy" as a result), or the motor weight can be cantilevered over the discharge elbow without exerting any load on it (cantilevered drive stool), or again the drive stool can be installed on a floor of its own at the next higher floor level of the engine room (illustration; in this case the cardan shaft between the floors must be protected by a protective pipe). In certain rare cases (for the smaller nominal diameters and short overall lengths) the complete pumping set can be installed on the floor of the intake chamber by means of a special pump foot.

T.c.p.'s are mainly used as land reclamation pumps and cooling water pumps.

Tuft

*Fadensonde
Sonde de fil*

see [Measuring Technique](#)

Turbine Driven Pump

Turbopumpe
Turbo-pompe

The t.d.p. is a [centrifugal pump](#) driven by a turbine ([pump types](#)).

Turbine Operation of Centrifugal Pumps

Turbinenbetrieb von Kreiselpumpen
Marche des pompes centrifuges en turbines

Centrifugal pumps can be used as turbines without having to change the design of the [impeller](#) or the [pump casing](#). In t.o.o.c.p. the machine is run in reverse, and the direction of rotation of the pump shaft is reversed.

At the point of best efficiency, that is at the [design duty point](#) of the pump turbine, the flow enters the impeller nearly shock-free ([shock-free entry](#)), and the [efficiency](#) achieved are equally as high values as during normal pump operation.

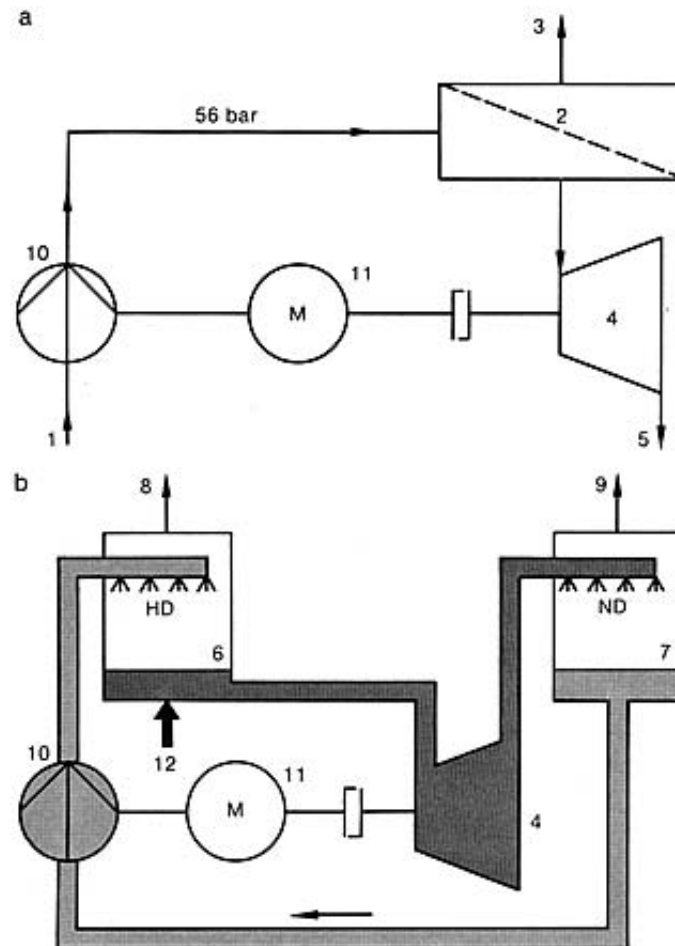
One reaches the optimal layout for t.o.o.c.p. at equal [rotational speed](#), though reversed, as normal pump operation, and select the [capacity](#) Q and the [head](#) H 20 to 60% higher than for pump operation.

Because of the higher energy concentration in turbine operation, more is demanded of the bearings and shafts of the machine. Usually standard pumps can be used in such a situation however.

In pump storage stations the machines run alternately in the forward and reverse direction. In highly technical processes, such as gas washers or reverse osmosis ([seawater](#) desalination plant), the turbine operation is used for energy recovery (see illustration).

The combination of a forwards and backwards operating pump, in one unit, can be used in the role of energy converter ([fluid coupling](#)).

When connected to a generator, a centrifugal pump in tubing operation can be used to generate electricity of less than 100 kW. [Asynchronous motors](#), such as the ones used to power [centrifugal pumps](#), allow themselves to be used for power generation ([three-phase current](#)). The direction of rotation can be reversed by switching the phases on the connection to the grid. When the rotational speed is raised above the level of the [synchronous speed](#), the asynchronous motor becomes a generator. The slip can reach values of 1 to 6% at rated load, so that the asynchronous generator holds the turbine at a speed slightly above the [synchronous speed](#) (e.g. 3000 min⁻¹ on a two pole machine in a 50 Hz grid). When changing the power given off by the turbine only the torque and the amperage fed into the grid change, whereas the rotational speed and voltage remain unchanged. Asynchronous generators require reactive power consumption ([power factor cos \$\varphi\$](#)), which can be compensated for with use of a condenser.



Examples of energy conservation in the process where centrifugal pumps are in turbine operation:

a reverse osmosis, b gas washers

1 ocean water; 2 membrane; 3 drinking water; 4 conservation unit; 5 sole; 6 absorption; 7 desorption; 8 gas release, phase 1; 9 gas release, phase 2., 10 pump; 11 electromotor; 12 mixed-gas entry, phase 1 and 2;
HD = high pressure, ND = low pressure

Asynchronous generators in independent operation (that is, without a grid providing frequency or voltage) require sophisticated controls, since the potential, current rotational speed and frequency change greatly as a function of the power available, the load and the capacity of the condenser setup.

Turbulent Flow

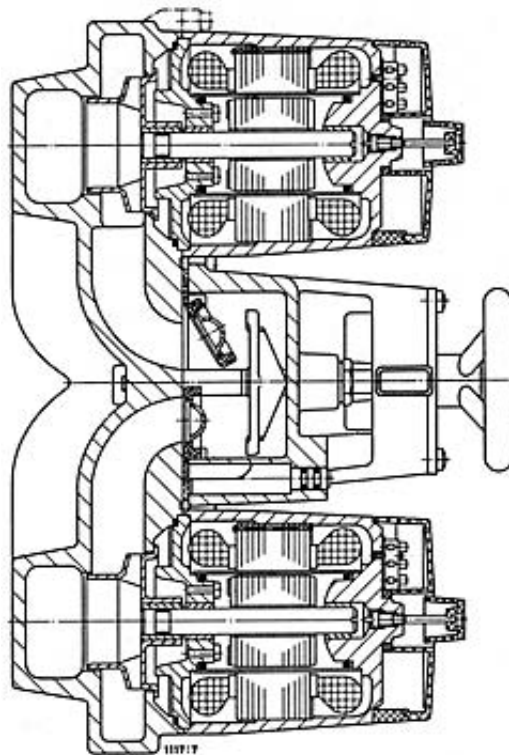
Turbulente Strömung
Écoulement turbulent

see Fluid Dynamics

Twin Pumping Set

Zwillingspumpe
Pompe jumelle

T.p.s.'s are particularly suitable as circulating pumps for hot water central heating plants (central heating circulating pump) in which a standby pump always ready for instant start-up is specified or desired for reasons of reliability or of comfort. The t.p.s. has only two flanged or screwed pipe connections; the gate valves which would normally be fitted (valves and fittings) can be dispensed with, thanks to the integrated change-over and isolating systems (see illustration).



Twin pumping set

Two-Phase Flow

Zweiphasenströmung

Écoulement diphasique

The t.p.f. is a flow in which two different aggregate states of a substance or of two different substances are simultaneously present. The possible combinations include gaseous/liquid (gas content of pumped medium), gaseous/solid and liquid/solid (hydrotransport). Examples of t.p.f. are transport processes of one medium in another (hydraulic or pneumatic conveying), and the frequently undesired entrainment of gas or vapour bubbles (cavitation) which have come out of solution in liquid flows. In t.p.f.'s, additional influence magnitudes (e.g. the concentration ratio of the two phases) are involved as well as the flow parameters (e.g. the REYNOLDS number, model laws).

The concentration ratio represents an important influence magnitude, and it can be expressed in the form of volume ratio or mass ratio of the two phases. Various forms of flow or of phase distribution can occur in a t.p.f., and these are influenced by the concentration ratio of the phases, and also by the so-called "phase slippage" i.e. by the difference between the flow velocities of the two portions, and also by the orientation of the pipings (horizontal, vertical). Some of the forms of flow of gaseous/liquid t.p.f.'s are e.g. bubble flow, slug flow and film flow.

Various mainly empirical formulae for the calculation of the pressure loss of a t.p.f. in straight pipelines are used, some of which take into consideration the influence of the form of flow and of the orientation (horizontal, vertical) of the pipeline. In the case of gaseous/liquid t.p.f.'s, a distinction must be made between flows in which the gaseous phase consists of the vapour state of aggregation of the liquid which is simultaneously present (e.g. water/water vapour) and flows in which the gaseous phase consists of a different substance (e.g. water/air), because in the first of these two cases phase change phenomena (evaporation, condensation) may play a part.

Because of the different densities of the two phases, a phase separation (e.g. settling of solid particles, rising of bubbles) may occur in t.p.f.'s at low flow velocities under the influence of gravity. Such a separation will also occur to an even more marked extent under the influence of centrifugal forces (e.g. in elbows, impellers), and at higher flow velocities also, with the constituents of lower density being subjected to the action of forces which drive them towards the centre of the bend or centre of rotation. This effect can lead to the breakaway of the delivery flow (operating behaviour) of a centrifugal pump handling a t.p.f. In the case of liquid/solid t.p.f.'s containing abrasive (abrasion) solid particles, the possibility of wear on components exposed to the flow must be taken into account;

this is particularly likely to take place in the region of curved flow lines (hydrotransport, pulp pumping).

The buoyancy, i.e. the uplift effect of t.p.f. involving a mixture of water and air, is exploited in connection with vertical underwater piping with airlift pumps (mammoth pump pump types) or as three-phase flow in the case of hydrotransport (airlift).

Types of Protection

Schutzarten

Types de protection

The t.o.p. for electrical equipment are laid down in DIN 40050, 1980 edition. They encompass protection against accidental contact, protection against foreign bodies and protection against water, and are designated by means of the symbol IP (international protection) where the internationally agreed t.o.p. are used. Electric motors used as drives of centrifugal pumps usually have one of the t.o.p. listed in the Table. Special t.o.p. are designated by the letters W (weather-proof) and R (pipe ventilated, i.e. pipe connection for motor cooling).

Table: Types of protection for electric motors in accordance with DIN 40 050

Type of protective enclosure	Protection against accidental contact	Protection against foreign bodies	Protection against water
IP 00	none	none	none
IP 02			dripwater up to 15° from the vertical
IP 11	large surface contact	large solid foreign bodies above 50 mm ø	vertical dripwater
IP 12			dripwater up to 15° from the vertical
IP 13			spray water up to 60° from the vertical
IP 21	accidental contact with fingers	solid foreign bodies of medium size above 12 mm ø	vertical dripwater
IP 22			dripwater up to 15° from the vertical
IP 23			spray water up to 60° from the vertical
IP 44	accidental contact with a. tool or similar object	small solid foreign bodies above 1 mm ø	splash water from all directions
IP 54	complete protection against accidental contact	dust deposits liable to cause damage	
IP 55			water jets from all directions
IP 56			temporary flooding

U

Unbalance of Centrifugal Pumps

Unwucht bei Kreiselpumpen

Balourd de pompes centrifuges

The rotor of a centrifugal pump, i.e. the impeller, pump shaft and all other rotating components on the shaft are carefully balanced mechanically before the pump is assembled. Thus the geometric mass distribution is corrected in such a way that the rotation axis of the pump rotor is a so-called free axis (no free inertia effects during rotation).

A distinction is made between static and dynamic balancing. In static balancing, the centre of gravity of mass of the pump rotor is placed as accurately as possible in the rotation axis of the pump shaft, and, consequently, the resultant of the centrifugal forces (not to be confused with the resultant moment) disappears (balancing in **single** plane, usually applied when the axial dimension of the rotating mass is substantially smaller than its diameter). In dynamic balancing on so-called balancing machines, the rotation axis of the pump rotor is made into a principal axis of inertia; in consequence not only does the resultant, but also the resultant static moment of the centrifugal forces disappear (balancing in **two** planes).

A pump rotor which is carefully balanced mechanically before commissioning of the pump can develop symptoms of unbalance during operation. This disturbance of the balance condition often rotates at the frequency of the rotational speed and impairs the quietness of centrifugal pumps. The causes of u.o.c.p. reside in the mechanical unbalance resulting from asymmetrical wear and in the hydraulic unbalance resulting from asymmetrically acting hydraulic forces and moments of each blade or of each blade channel (vibration).

Mechanical unbalance can arise during operation as a result of

1. mechanical fouling and removal of material (clearance gap width),
2. asymmetrical erosion as a result of abrasive pumped media,
3. asymmetrical corrosion as a result of corrosive pumped media,
4. asymmetrical deposition of crystallizing media or media prone to sedimentation (deposits in pumps),
5. asymmetrical material removal as a result of cavitation at the various impeller regions.

Hydraulic unbalanced may arise during operation as a result of

1. uneven action of blades or channels caused by geometrical unevenness (shape, roughness) or more generally as a result of unequal energy transmission (unbalance rotating at rotational frequency);
2. blades of differing cavitation behaviour (rotating unbalance).

All the unbalanced can arise together, in individual groups or singly.

Uncertainty of Measurement

Meßunsicherheit

Incertitude de mesure

The u.o.m. encompasses the accidental errors of all individual magnitudes from which the measurement result is calculated, and in addition any systematic errors which have not been determined because they are not measurable and can only be estimated (see DIN 1319, Sheet 3).

Every measurement is subject to u.'s o.m., even when the measuring methods, instruments and evaluation rules strictly comply with the appropriate acceptance test codes. When comparing the test results with the guaranteed values, this u.o.m. must be suitably allowed for. The u.o.m. is independent of the pump and of the guaranteed values (see DIN 1944, ISO 2548 and 3555).

The guideline values *f* (quoted below) for the u.o.m. of individual measured magnitudes in accordance with DIN 1944, October 1968 edition, apply to acceptance tests of degree of accuracy I. For an acceptance test of degree of accuracy II, these values should be multiplied by 1.5, and for an acceptance test of degree of accuracy III, they

should be multiplied by 2.

1. *U.o.m. of capacity Q*

a) with a tank, and a duration of measurement of 50 s, and

- using a standpipe or a float gauge to measure the difference in levels Δz in m, the filling jet being swung in and out

- observing the rising liquid level between two marks at a distance Δz in m from one another

b) with a paddle wheel counter within the permissible range of REYNOLDS numbers, for a measurement duration of 50 s

c) with a float element flowmeter

d) with a throttling measuring device in accordance with DIN 1952 (standard orifice, standard nozzle)

2. *U.o.m. of pressure head $p/\rho g$ ($p/\rho g$ should be inserted in m in the items below)*

a) with liquid manometers for $(p/\rho g)_{\max} = 1.5$ m; within the range $0.1\text{m} < (p/\rho g)_{\max} \leq 1.5$ m:

- for fluctuations Δh of $\pm 10^{-3}$ m max.

- for greater fluctuations Δh in m

b) with liquid manometers for $(p/\rho g)_{\max} > 1.5$ m

- for fluctuations Δh of $\pm 10^{-3}$ m max.

- for greater fluctuations Δh in m

c) with calibrated spring pressure gauges, e.g. of accuracy final value category 0.6

3. *U.o.m. of velocity head $v^2/2g$ (flow velocity v)*

a) determination by means of capacity and flow cross-section

- in the case of circular cross-sections and regular cross-sections which can be accurately measured

- in the case of irregular cross-sections

b) measurement with the aid of stagnation pressure tubes (probes)
+ 1,5 %

f_q

$$\pm \frac{0.3}{\Delta z} \%$$

$$\pm \frac{0.5}{\Delta z} \%$$

$$\pm 1.5 \%$$

$$\pm 2.0 \%$$

$$\pm 1.0 \text{ to } 1.5 \%$$

f_p

$$\pm \frac{0.1}{p/\rho g} \%$$

$$\pm 10^2 \frac{\Delta h}{p/\rho g} \%$$

$$\pm \frac{0.2}{p/\rho g} \%$$

$$\pm 2 \cdot 10^2 \frac{\Delta h}{p/\rho g} \%$$

$$\pm 0.6 \frac{\text{final value}}{\text{value of read.}} \%$$

f_v

$$\pm 1.5 \cdot f_q \%$$

$$\pm 2.0 \cdot f_q \%$$

$$\pm 1.5 \%$$

4. U.o.m. of rotational speed n

a) with hand tachometers

b) with electronic counters

c) with slip meters

5. U.o.m. shaft power P a) from the absorbed electric power of a three-phase motor (asynchronous motor)

- for a nominal rating

$P_N \leq 25 \text{ kW}$	$\pm 1.5 \%$
$25 \text{ kW} < P_N \leq 250 \text{ kW}$	$\pm 1.0 \%$
$P_N > 250 \text{ kW}$	$\pm 0.8 \%$

b) from the torque and rotational speed- of a pendulum type electric motor

- of a torsion dynamometer for a twist angle

$\alpha \geq 0.75 \alpha_{\max}$	$\pm \sqrt{1.2^2 + f_n^2}$
$0.50 \alpha_{\max} < \alpha < 0.75 \alpha_{\max}$	$\pm \sqrt{1.5^2 + f_n^2}$

6. U.o.m. of density ρ of water up to a temperature of 100 °C f_n $\pm 0.5 \%$ $\pm 0.1 \%$ $\pm 0.5 \%$ f_p $\pm 1.5 \%$ $\pm 1.0 \%$ $\pm 0.8 \%$ $\pm \sqrt{1.0^2 + f_n^2}$ $\pm \sqrt{1.2^2 + f_n^2}$ $\pm \sqrt{1.5^2 + f_n^2}$ $\pm 0.1 \%$

As a result of the u.'s o.m. of the individual measured magnitudes, the u.'s o.m. for capacity, head and pump efficiency are as follows (the formula symbols are those used in DIN 1944, October 1968 edition):

7. U.o.m. for measured capacity Q

$$f_Q + f_q.$$

8. U.o.m. for measured head H

$$f_H \approx \pm \sqrt{\left(\frac{z_d - z_s}{H}\right)^2 \cdot f_z^2 + \left(\frac{p_d}{\rho \cdot g \cdot H}\right)^2 \cdot f_{pd}^2} + \sqrt{\left(\frac{p_s}{\rho \cdot g \cdot H}\right)^2 \cdot f_{ps}^2 + \frac{(v_d - v_s)^4}{(g \cdot H)^2} \cdot f_v^2}$$

with

 z elevation of pump inlet and outlet cross-sections f_z u.o.m. of z ,subscript s inlet cross-section (suction branch) of pump,subscripts d outlet cross-section (discharge branch) of pump.

For pumps with a very high head, and with small values of p_s/p_d , one can assume:

$$f_H \approx f_p.$$

9. U.o.m. of pump efficiency η

$$f_\eta = \pm \sqrt{f_Q^2 + f_H^2 + f_P^2 + f_\rho^2}$$

If the measured performance values Q and H used for the determination of the throttling curve $H(Q)$ (characteristic curve) are converted to the rotational speed specified in the supply contract (subscript u), we have:

10. U.o.m. for converted capacity Q_u

$$f_{Q_u} = \pm \sqrt{f_Q^2 + f_n^2},$$

11. U.o.m. for converted head H_u

$$f_{H_u} = \pm \sqrt{f_H^2 + 4 \cdot f_n^2}.$$

For this purpose, the rotational speed n may deviate from the rotational speed n_L agreed in the supply contract as follows:

- when testing the performance figures from +5% to -50%,
- when testing the efficiencies from +5% to -10 %.

In accordance with the affinity law, the measured values are converted as follows to the rotational speed specified in the supply contract:

$$Q_u = Q \cdot \frac{n_L}{n},$$

$$H_u = H \cdot \left(\frac{n_L}{n} \right)^2,$$

$$P_u = P \cdot \left(\frac{n_L}{n} \right)^3,$$

$$\eta_u = \eta.$$

If the rotational speed deviations are larger, a conversion formula must be agreed upon to calculate the efficiency.

The max. permissible u.'s o.m. (overall tolerance) are dependent on the agreed degree of accuracy according to DIN 1944, and are laid down as shown in Table 1.

Table 1: Maximum permissible uncertainty of measurement in accordance with DIN 1944 (October 1968 edition)

Magnitude		Maximum permissible uncertainties of measurement for degree of accuracy		
		III	II	I
capacity Q :	f_Q perm. =	$\pm 3.0\%$	$\pm 2.0 \%$	$\pm 1.5\%$
head H :	f_H perm. =	$\pm 2.0\%$	$\pm 1.5\%$	$\pm 1.0\%$
shaft power P :	f_P perm. =	$\pm 2.0\%$	$\pm 1.5\%$	$\pm 1.0\%$
pump efficiency η :	f_η perm. =	not applicable	$\pm 3.0 \%$	$\pm 2.0\%$

In an $H(Q)$ diagram (Fig. 1), an $\eta(Q)$ diagram (Fig. 2) or a $P(Q)$ diagram each measurement point appears in the shape of an ellipse, if the u.'s o.m. for both coordinate magnitudes are taken into account. The semi-axes of the ellipse are for:

- the capacity $f_Q \cdot Q$,
- the head $f_H \cdot H$,
- the shaft power $f_P \cdot P$,
- the efficiency $f_\eta \cdot \eta$.

It is customary in centrifugal pump technology to indicate the surface of each u.o.m. ellipse by the four end points of the two semi-axes (Figs. 1 and 2). The characteristic curves $H(Q)$ and $\eta(Q)$ then appear in the form of a band limited by an upper (a) and a lower (b) limit curve of uniform aspect; these limit curves should be so plotted that each curve either intersects or is in contact with at least one of the two semi-axes at each measurement point (see Figs. 1 and 2).

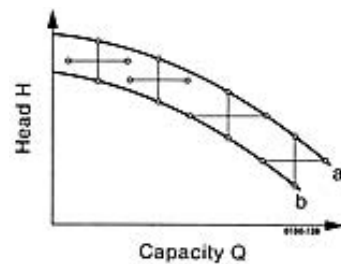


Fig. 1: Taking the uncertainties of measurement into account on the $H(Q)$ curve (a, b see Fig. 7)

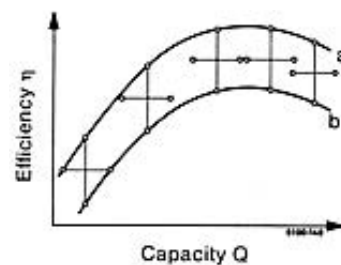


Fig. 2: Taking the uncertainties of measurement into account on the $\eta(Q)$ curve (a, b see Fig. 7)

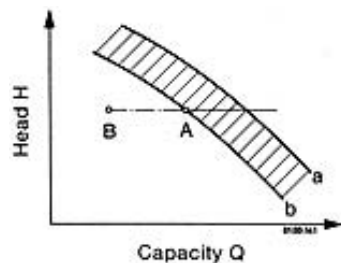


Fig. 3: The lower limit throttling curve b passes through the max. value of guaranteed capacity at A (a, b see Fig. 7)

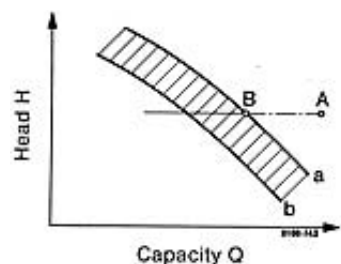


Fig. 4: The upper limit throttling curve a passes through the min. value of guaranteed capacity at B (a, b see Fig. 7)

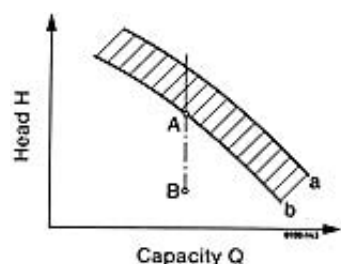


Fig. 5: The lower limit throttling curve b passes through the max. value of guaranteed head at A (a, b see Fig. 7)

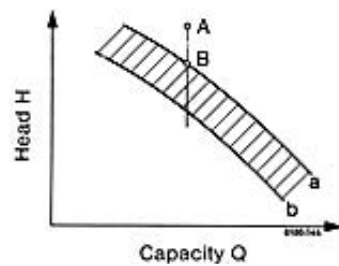


Fig. 6: The upper limit throttling curve a passes through the min. value of guaranteed head at B (a, b see Fig. 7)

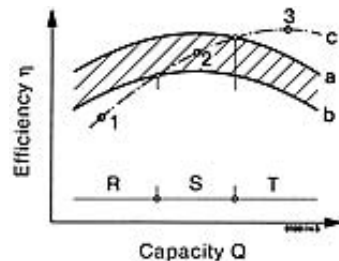


Fig. 7: Checking the efficiency guarantee

- a upper efficiency limit curve,
- b lower efficiency limit curve,
- c efficiency curve through guarantee points 1, 2, 3,
- R zone in which the guarantee is exceeded,
- S zone in which the guarantee is fulfilled,
- T zone in which the guarantee is not fulfilled

The guarantee for the performance data Q_L , H_L specified in the supply contract is *fulfilled* according to DIN 1944 (October 1968 edition) if the range AB which represents the guaranteed capacity or the guaranteed head (manufacturing tolerance) is intersected by, or in contact with at least one of the two limit throttling curves a and b; this is illustrated in Figs. 3 to 6; a similar argumentation applies to the efficiency guarantee, Fig. 7.

In the ISO acceptance test code for pumps, only max. permissible u.'s o.m. for capacity, head, shaft power of pump, rotational speed and pump efficiency are listed in Table 2 (acceptance test codes, overall tolerance). Guideline values for u.'s o.m. in respect of the individual measurement methods are absent in the ISO acceptance test code.

Table 2: Maximum permissible uncertainties of measurement according to ISO acceptance test code for pumps (ISO 2548 and 3555)

Magnitude	Maximum permissible uncertainties of measurement for accuracy category *)	
	C	B
capacity Q	± 3.5 %	± 2.0 %
head H	± 3.5 %	± 1.5 %
shaft power P of pump	± 3.5 %	± 1.5 %
absorbed power PM of electric motor (for determination of efficiency of pump set)	± 3.5 %	± 1.5 %
rotational speed n	± 2.0 %	± 0.5 %
efficiency or of pump set (group consisting of pump and motor)	± 4.5 %	± 2.5 %
pump efficiency	± 5.0 %	± 2.8 %

*) no values have yet been laid down for accuracy category A

Under-Filing of Impellers

Ausfeilen von Laufrädern

Affûtage des subes

see [Back Filing](#)

Underwater Motor Pump

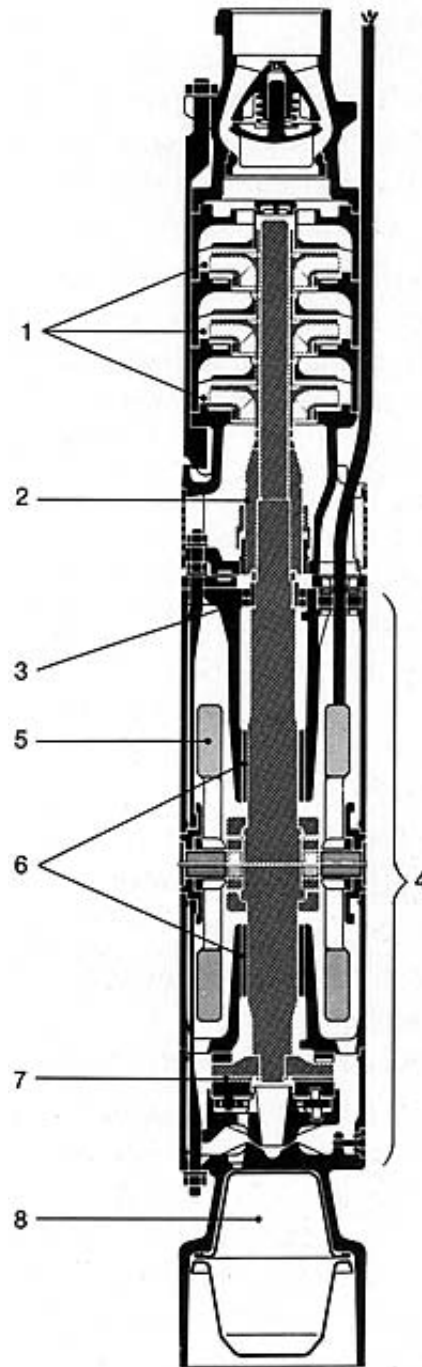
Unterwassermotorpumpe

Électro-pompe submersible

The u.m.p. is a centrifugal pump unit, that is powered by a submersible motor draws in medium directly, without use of a suction pipe, and must contently be submerged in the medium. The single or multistage pump is directly coupled to the submersible motor, where depending on the use, the motor can be either mounted at the top or bottom.

U.m.p.'s are often used as borehole pumps (see illustration), and have the advantage over borehole shaft driven pumps that they can be installed at great submerged depths without difficulty. U.m.p.'s are conditioned in their design by the physical dimensions of the borehole, i.e. their diameter is restricted and their length is extended. The motor is usually installed beneath the pump in this case; it is a squirrel-cage rotor motor (asynchronous motor, submersible motor), with a water fill or an oil fill for the cooling of the winding and for bearing lubrication, which is sealed off from the pump by a mechanical seal (shaft seals) or similar seal. The suction strainer is arranged between pump and motor.

As the pumping set is inaccessible during operation, it must be able to run entirely maintenance-free. The radial and thrust bearings must be carefully designed; the plain bearings lubricated by the pumped medium or by the motor fill liquid are often made of carbon, bronze, asbestos impregnated with synthetic resin, ceramics or tungsten-carbide. The pump casing, and the impellers and diffusers of the pump are made of cast iron, bronze, chromenickel steel or plastic, according to requirements. The connection at the top of the pumping set includes a nonreturn valve (valves and fittings), through which the capacity enters the riser, upon which the entire pumping unit hangs. The power cable is routed down the riser, along the outside of the pump, and into the motor through a cable lead-in.



Underwater motor pump with water-filled (wet) motor serving as a borehole pump
 1 multistage ring-section pump; 2 shaft coupling; 3 shaft seal; 4 water-filled underwater motor; 5 plastic-insulated wet winding; 6 oil-lubricated radial bearing; 7 water-lubricated thrust bearing (for pump and motor); 8 pressure-equalizing diaphragm

The principal applications for u.m.p.'s, are irrigation, dewatering in open-cast and underground mining operations, water-supply wells and the offshore, geothermic and deep-sea mining sectors.

The sets can in certain cases be designed as portable sets, depending on their application, and their pump materials (materials) must be suitably adapted to the medium pumped, which may contain sand and dirt.

Underwater Pump

Unterwasserpumpe
Pompe submerssible

see [Submersible Pump](#)

Undisturbed Length of Piping

Ungestörte Rohrstrecke

Longeur de tuyauterie sans trouble

This concertos mainly used for the measurement (measuring technique) of volume flow (capacity) by means of throttling devices (standard orifice, standard nozzle, standard venturi nozzle). A measuring point for pressure or velocity requires an upstom-side u.l.o.p., if acertain degree of measuring accuracy (uncertainty of measurement) is specified. In order to achieve optimal operating behaviour of a centrifugal pump, the pump suction branch must also have an upstream-side u.l.o.p.; this applies in particular to pumps with high specific speeds (inlet conditions).

As a general rule, an u.l.o.p. comprises a length of piping meeting the following requirements:

- constant pipe diameter D;
- straight length (visually straight) of at least $20 \cdot D$ long (this is a rough approximation; more accurate data, depending on the aperture ratio and on disturbances upstream of the measuring device are given in the Table under standard orifice);
- smooth internal pipe surface (relative roughness peaks for large diameter ratios approximately 0.4, for smaller diameter ratios up to approximately 4 of the pipe diameter D, standard orifice).

Unit

Einheit

Unité

U., or "physical unit" is a well-defined and established physical dimension (e.g. the metre) used as reference dimension for quantitative representations. A physical dimension is expressed as

physical dimension = numerical value \times unit

e.g. length 1 = 1.83 m.

An international unit system, the so-called "Système International d'Unités", abbreviation SI, Was officially adopted by the "10e Conférence Générale des Poids et Mesures" in 1954. In the SI, the so-called SI basic units (Table 1) are laid down for seven basic entities, and these, together with the derived u.'s (SI units) became binding and compulsory, in accordance with the "Law on Units in the Field of Measurement" which came into force in 1970 in the German Federal Republic, and in accordance with DIN 130, 1985 edition.

Table1: SI basic dimensions and SI units

Basic dimension	Basic unit	
	name	symbol
length	metre	m
mass	kilogramme	kg
time	second	s
electrical current intensity	Ampere	A
thermodyn. temperature	Kelvin	K
light intensity	candela	cd
quantity of substance	mol	mol

Table 2: Extract of important legalized units for centrifugal pumps

Physical dimension	Formula symbol	Legalized units			No longer admissible units or notations	Recommended units	Remarks
		SI units		further legal units (not complete)			
length	l	m	metre	km, dm, cm, mm, μm , ...		m	basic unit
volume	V	m^3		dm^3 , cm^3 , mm^3 , ... litre (l 1 = 1 dm^3)	cbm, cdm, ...	m^3	
capacity, volume flow	Q, \dot{V}	m^3/s		m^3/h , l/s		l/s and m^3/s	
time	t	s	second	s, ms, μs , ns, ... min, h, d		s	basic unit
rotational speed	n	s^{-1}		min^{-1}		s^{-1} and min^{-1}	
mass	m	kg	kilogramme	g, mg, μg , ... ton (1 t = 1000 kg)	pound, hundred-weight	kg	basic unit; the mass of a commercial commodity is described as weight
density	ρ	kg/m^3		kg/dm^3		kg/dm^3 and kg/m^3	the designation "specific gravity" must no longer be employed, because it is ambiguous (see DIN 1305)
moment of inertia	J	kg m^2				kg m^2	
mass flow	\dot{m}	kg/s			t/s, t/h, kg/h	kg/s and t/s	
force	F	N	Newton (= kg/s^2)	kN, mN, μN , ...	kp, Mp, ...	N	1 kp = 9.81N. The weight force is the product of mass m by the local gravitational constant g
pressure	p	Pa	Pascal (= N/m^2)	bar (1 bar = 10^5 Pa)	kp/cm^2 , at, m WS, Torr, ..	bar	1 at = 0.981 bar = $9.81 \cdot 10^4$ Pa 1 mm Hg = 1.333 mbar 1 mm w.c. = 0.098 mbar
mechanical stress (strength)	σ , τ	Pa	Pascal (= N/m^2)	N/mm^2 , N/cm^2 , ..	kp/cm^2 , ...	N/mm^2	1 kp/mm^2 = 9.81 n/mm^2
bending moment, torque	M, T	N m			kp m, ...	N m	1 kp m = 9.81 N m
energy, work, quantity of heat	E, A, Q	J	Joule (= N m = W s)	kJ , W s , kW h, ...; 1 kW h = 3600 kJ	kp m, kcal, cal	J and kJ	1 kp m = 9.81 J 1 kcal = 4.1868 kJ
head	H	m	metre		m.c.l.	m	the head is the work done in $\text{J} = \text{N m}$ applied to the mass unit of the medium pumped, related to the weight force of this mass unit in N
power	P	W	Watt (= $\text{J}/\text{s} = \text{N m}/\text{s}$)	MW, kW, ...	kp m/s, PS, HP	kW	1 kp m/s = 9.81 W; 1 PS = 736 W
temperature difference	T	K	Kelvin	$^{\circ}\text{C}$	$^{\circ}\text{K}$, deg.	K	basic unit
kinematic viscosity	ν	m^2/s			St (Stokes), $^{\circ}\text{E}$, ...	m^2/s	1 ST = 10^{-4} m^2/s
dynamic viscosity	η	Pa s	Pascal second (= $\text{N s}/\text{m}^2$)		P (poise)	Pa s	1 P = 0.1 Pa s

Table 2 gives an extract of the most important legalized u.'s for centrifugal pumps. A complete tabulation of these legalized u.'s is contained in the 1974 publication of the VDMA (Association of German Machinery Manufacturers) entitled "Gesetzliche Einheiten im Pumpenbau" (legalized units in pump manufacture). For a summary of unit symbols and conversion factors of British and USA units, [on appendix](#).

Unstable Throttling Curve

Instabile Drosselkurve

Courbe d'étranglement instable

The throttling curve (characteristic curve) of a centrifugal pump is said to be unstable in a given capacity (rate of flow) zone if its slope (differential quotient of head H over capacity Q) is positive in said zone (see illustration under stable throttling curve). Under certain operating conditions, e.g. during parallel operation, centrifugal pumps with u.t.c. may cause operating problems (unsteady flow).

Unsteady Flow

Instationäre Strömung

Écoulement non permanent

The flow of a fluid is unsteady if the flow magnitudes such as velocity and pressure are dependent not only on the position in the coordinate system used to describe the field of flow, but also on time. There is a distinction made between three different types of u.f. phenomena:

1. statistically irregular phenomena. e.g. turbulent fluctuations (fluid dynamics),
2. run-up to and run-down from full speed phenomena, e.g. when switching a centrifugal pump on or off (starting process, starting torque),
3. periodic phenomena, e.g. pulsations, such as arise in case of pressure surges (surge pressure) in pipng, in centrifugal pumps with unstable characteristic curves or under the influence of the rotating impeller.

As a result of velocity changes at a given spot, additional mass forces are created in a u.f. because of local accelerations or decelerations, and these cause corresponding changes in pressure. Typical consequences are e.g. the considerable short duration pressure rise which occurs when an isolating valve (valves and fittings) is suddenly closed in a long pipng run through which fluid flows (surge pressure), or the increased pressure loss caused by pulsating flow in a pipe.

If the frequency of the changes of condition in a periodic u.f. phenomenon is low enough, said flow phenomenon can often be treated as a quasi-steady phenomenon, for which the same flow condition applies as in a steady flow, with the appropriate instantaneous values.

The flow through a rotating set of beading (impeller) is strictly speaking always a u.f. if viewed from a stationary coordinate system (absolute velocity, steady flow); the velocity and pressure change periodically at a given spot under observation as the blade spacings pass by the spot. Despite this, the flow in an impeller and its immediate surroundings is a steady flow, if one adopts a system of coordinates which rotates with the impeller to describe it (relative velocity); however the centrifugal and CORIOLIS forces arising in the relative system must be taken into account.

Useful Power Output

Nutzleistung

Puissance utile

see [Pump Output](#)

U-Tube

U-Rohr

Tube en U

see Measuring Technique

V

Valve

Ventil
Soupape

see [Valves and Fittings](#)

Valves and Fittings

Armaturen
Robinetterie

V.a.f. are components of piping; particular examples are shut-off valves and control valves. V.a.f. are more usually classed according to application, e.g. v.a.f. for boilers, central heating or plumbing. This classification does not however define the type of construction of the component concerned, and its function is only characterized in a general way by the field of application. It is preferable to classify v.a.f. by their constructive features, into

- *globe valves* (straight-line motion of the closure element parallel to the local direction of flow),
- *gate valves* (straight-line motion of the closure element at right angles to the direction of flow),
- *cocks and flap valves* (rotary or swinging motion of the closure element).

G l o b e v a l v e s (Fig. 1). These can be classified into subgroups according to the shape of the housing or body (straight-through or throughway valves, angle valves, axial valves etc.) or according to their function (shut-off valves, control valves etc.). Various factors govern the choice of body, such as the possibility of fitting a control type plug or disc (throughway valves), the piping layout (angle valves), the need for a low pressure drop (axial valves). In the case of large nominal diameters or high nominal pressures, the forces acting on the valve stem become appreciable, and they can be kept within reasonable bounds by providing a pilot compensated slider valve (i.e. internal balancing). This can however only be done if a back pressure can build up in the adjoining system. Similar considerations have led to the adoption of double seats on control valves. The plug contours also vary according to applications, the main types being quick opening, linear and equal percentage plugs.

Nonreturn valves represent a special type of valve. They operate automatically, opening in the direction of flow and closing when the flow ceases or starts to flow in reverse. In order to make them independent of their location in the pipeline they can be provided with a closing spring. They can also be provided with an internal damping device or dashpot. The closing speed of the plug or disc is controllable, by providing a displacement chamber between the plug and plug guide which gradually fills with liquid via narrow gaps or small bores. By fitting a nonreturn valve in the discharge line, any reflux of liquid into the pump is prevented when the latter is shut down, thus avoiding any reverse rotation or turbining of the pump (reverse rotational speed, turbine operation of centrifugal pumps), e.g. in the case of two pumps operating in parallel, one of which is switched off. It is advisable to fit a nonreturn valve upstream of a shut-off valve (gate valve) so that the pump and nonreturn valve can be overhauled without having to drain the discharge line.

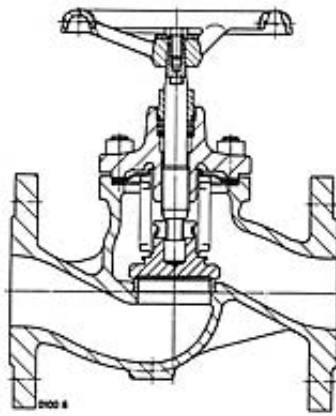


Fig. 1: Globe valve with bellows

The automatic recirculation valve (Fig. 2) represents a special type of nonreturn valve. It is usually mounted on the discharge side of a boiler feed pump and is designed to maintain a minimum flow of fluid through the pump at very low loads (partial load operation operating behaviour). At very low loads, the efficiency of the pump is low, and most of the absorbed energy is converted into heat, which may result in the partial evaporation of the water volume trapped in the pump. Insufficient water lubrication, e.g. at the balance disc (axial thrust), will lead to damage due to metal-to-metal contact. The valve cone or plug of the automatic recirculation valve sinks when the capacity of the pump decreases, opening a by-pass in the process, through which a given flow proportion (capacity), the so-called minimum flow rate is by-passed back into the suction vessel.

By fitting a nonreturn valve in the suction pipe, the emptying of the suction pipe after the pump has been switched off can be prevented. If the nonreturn valve is leak-tight, the pump need not be primed again before the next start. This so-called foot valve should be mounted at an adequate distance above the floor of the pump sump, and at an adequate distance beneath the lowest water level likely to occur in the sump. It often has the form of an axial valve, which ensures an even and symmetrical flow through the valve.

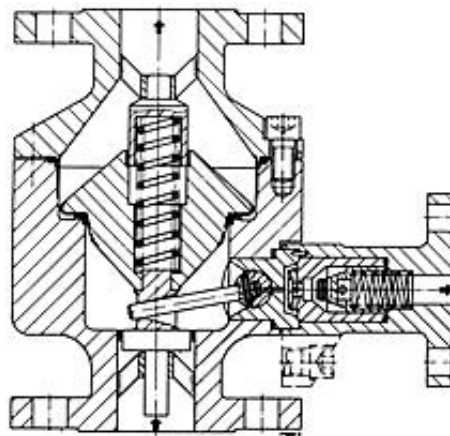


Fig. 2: Automatic recirculation valve

There are other special types of globe valve, viz. relief valves and overflow valves (spill valves). A spring or weight-loaded plug or disc opens when a given overpressure has been attained, and relieves the system. A globe valve can also be combined with a nonreturn valve in a so-called stopcheck valve.

G a t e v a l v e s (Fig. 3). This type of valve is characterized by a low pressure drop (approx. 1:30 as compared with valves). because the fluid flows straight through it. Such gate valves are built up to very large nominal diameters and up to the highest nominal pressures. In contrast to a globe valve, a gate valve of the same seat diameter and same differential pressure requires a much lower effort on the stem, amounting to 30% approx. only of that of a globe valve. As a general rule, gate valves are designed to permit flow from either direction under equal operating conditions, a state of affairs which cannot always be achieved with globe valves. As regards the seat arrangement, two types are available viz. wedge gate valves and parallel slide gate valves.

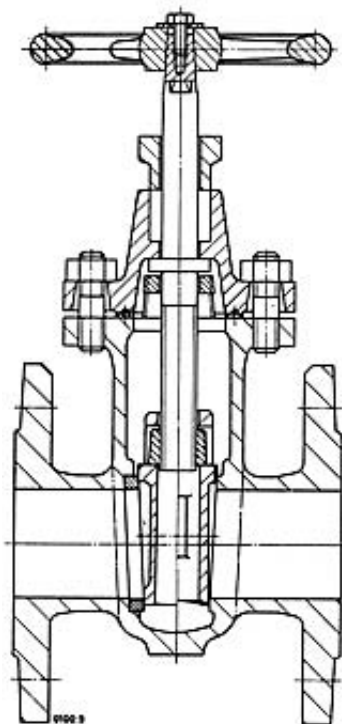


Fig. 3: Wedge gate valve

Wedge gate valves have either a rigid or a flexible wedge, or a double wedge disc. Parallel slide gate valves have either a beam-type disc or a double disc. Wedge gate valves have the following advantages in comparison with parallel slide gate valves: superior sealing force as a result of the wedge action (and therefore good sealing action even at low pressure differentials) and quick withdrawal of the wedge from its seat (thereby avoiding a sliding motion with undesirable side effects such as scratches by foreign particles or seizure).

In order to ensure satisfactory sealing at low operating pressures in the case of parallel slide valves, it is necessary to provide springs to press the discs against their seats (the leak-tightness becomes a problem at high temperatures). As a result of the sliding action of the sealing faces against one another, a certain self-cleaning effect at the sealing faces is achieved.

Gate valves are not generally used as control valves. There are exceptions however, which are usually only adopted in cases where the pressure differential is low. The valve travel is appreciable, of the same order as the nominal diameter of the pipe, and gate valves are therefore only used as quick-closing valves to a limited extent. Globe valves are used far more widely as quick-closing valves, because their travel is generally limited to approx. one quarter of the nominal diameter. The longer valve travels of gate valves also mean greater overall valve heights than in the case of globe valves. Special construction types include rotary plate gate valves or radial gate valves.

C o c k s (Fig. 4). Together with weirs and plugs, the cock is probably the oldest form of a shut-off device. Bronze and lead cocks were used by the ancient Romans in large numbers in the centuries B.C. Wood-carved cocks (usually on barrels) have been used from the Middle Ages to the present day. Metal gas cocks have been in use domestically since the turn of the century. Cocks, like gate valves, are characterized by a low pressure drop. As the closure element of a cock is rotated within the flow chamber, the overall height is usually small, and the max. setting or travel distance does not exceed one quarter turn. In the case of small cocks not fitted with a reduction gear, the position of the actuating lever indicates the position of the cock itself. The closure element of a cock can be a spherical, tapered or cylindrical plug. Cocks are mainly used as shut-off devices (open or closed) and more seldom as control devices. A plastic material (e.g. PTFE) is usually adopted for the joint rings of cocks, which limits their upper operating temperatures. Some cocks have metal-to-metal sealing faces, and an appropriate lubricant is required for these. In certain instances (e.g. in long natural gas pipelines), cocks of very large nominal diameters are used. Most applications however involve cocks of small sizes. Special types include three-way and fourway cocks.

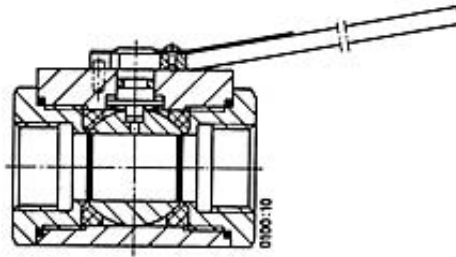


Fig. 4: Ball valve

Butterfly valves (Fig. 5). Butterfly valves are a relatively recent form of valves, characterized by their simple design. Butterfly valves are used both as shut-off hand as control devices, and are built mainly in the larger nominal diameters from medium sizes to the very largest. When used as tight closure shut-off valves, they are usually fitted with a seat of soft material and are therefore limited as regards max. application temperature. For higher temperatures and pressures (pressure), butterfly valves with a metal-to-metal sealing closure are available on the market today. As in the case of cocks, the angle of rotation is 90° . The most commonly used types of butterfly valve differ by virtue of the location of the flap relative to the axis of rotation, viz. concentric, eccentric and double eccentric types. Butterfly valves used as control devices are usually of the concentric type. Metal-to-metal sealing butterfly valves are usually designed with a double eccentric pivoting of the flap disc.

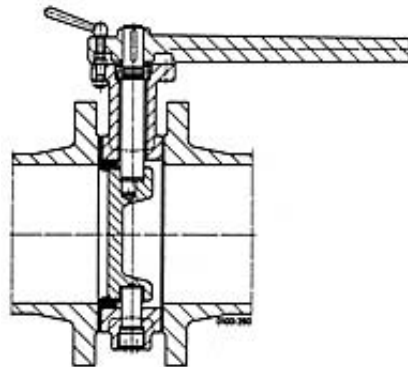


Fig. 5: Shut-off butterfly valve of eccentric type with hand lever, for clamping between flanges

Swing check valves (Fig. 6) represent a special type of butterfly valve. Whereas nonreturn valves are mainly adopted as nonreturn devices for the smaller nominal diameters, for economic reasons, swing check valves are mainly used for the medium up to the very largest nominal diameters. The pressure loss across the check valve is low, because there is only a small deflection of the flow path. The loss coefficient varies with the flow velocity, because the latter governs the position of the flap. Whereas spring-loaded nonreturn valves can be mounted in the piping in almost any position, swing check valves, which are dependent on gravity for their operation, must be mounted horizontally, or vertically with the flow approach from below. Swing check valves can as a general rule only be damped by means of a device mounted outside the valve, which poses problems in the case of pulsating flows. The position of the fulcrum in relation to the centre of gravity of the flap is important, to ensure that the flap disc lies square on its seat in the dosed position, or is pressed against its seat by its weight. The arrangement of a swing check valve in the discharge line or suction pipe is the same as that applying to a nonreturn valve.

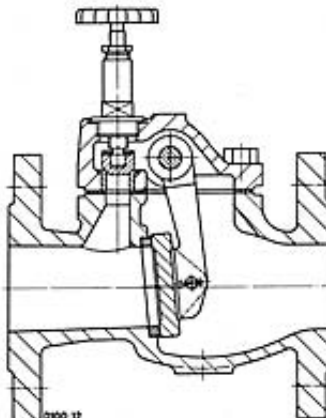


Fig. 6: Swing check valve

Another special type is the diaphragm nonreturn valve (Fig. 7) with elastic sealing elements that facilitate smoother closing.

Globe valves, gate valves, cocks and butterfly valves are mainly used as shut-off devices. The concept of leak-tightness is intimately bound up with the act of closure. Apart from conventional units such as "drops per unit of time" used to define this, the unit Pascal \times litres/second is often used, particularly in vacuum technology. It refers to a rate of leakage which causes the pressure in an evacuated vessel of one litre content to rise by one Pascal per second.

In the wider sense, filters, particularly filters for liquids belong to the grouping designated as v.a.f. The purpose of a filter for liquids is to separate out the almost insoluble solids from the liquid pumped. Various media can be used for filtration, such as beds of quartz gravel (rubble filters), paper, metal gauzes, fibrous materials etc. Other types of filters include edge filters (streamline filters), with gaps between adjoining discs of less than 0.1mm, and magnetic filters for iron contaminants. The suction strainer basket at the mouth of the suction pipe of a centrifugal pump represents a special type of filter. The basket is perforated with slots or round holes, with a total hole area several times larger than the pipe cross-sectional area (pressure loss). A foot valve arranged inside the suction strainer basket prevents the emptying of the suction pipe when the pump is switched off.

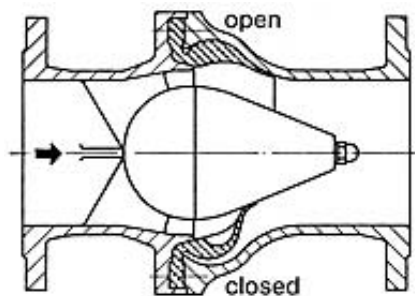


Fig. 7: Diaphragm nonreturn valve

Vaporization Pressure

Verdampfungsdruck
Pression d'évaporation

see Vapour Pressure

Vapour Pressure

Dampfdruck
Pression de vapeur

V.p., symbol p_D , also known as vaporization or saturation pressure, is the absolute pressure adopted by the vapour of a substance in equilibrium with its liquid or solid phase.

In centrifugal pump technology, v.p. refers to the pressure at which the vapour and liquid phases are in equilibrium (net positive suction head. cavitation); in this context, the boiling point curve represents the v.p. in function of temperature, starting from the triple point (equilibrium condition of all three possible phases: vapour, liquid, solid) right up to the so-called critical point (no longer any phase difference between vapour and liquid phase).

The SI unit (unit) of v.p. is 1 Pascal; the more widely adopted SI unit in pump technology is the bar or mbar.

Table 1 gives the v.p. p_D and the density ρ of water in function of the temperature in degrees centigrade and degrees Kelvin. starting from the triple point condition up to the critical condition. Data on the v.p.'s of various fluids prevalent in chemical pump technology (chemical pump) are listed in Table 2.

See following Tables 1 and 2 ([see diagram](#)).

Variable Pitch Blade

Verstellbare Schaufel
Pale orientable

see [Blade](#)

V-Belt Drive

Keilriementrieb
Commande à courroie trapézoïdale

see [Belt Drive](#)

Velocity

Geschwindigkeit
Vitesse

see [Flow Velocity](#)

Velocity Measurement

Geschwindigkeitsmessung
Mesure de vitesse

see [Measuring Technique](#)

Velocity Triangle

Geschwindigkeitsdreieck
Triangle des vitesses

V.t. is the vectorial representation of the equation

$$\vec{u} + \vec{w} = \vec{v}$$

where

\vec{u} vectorial circumferential velocity,
 \vec{v} vectorial absolute velocity,
 \vec{w} vectorial relative velocity.

(A vector is a directional magnitude.)

Fig. 1 illustrates this relationship on the so-called velocity parallelogram of a liquid particle, and this is transposed in the form of a v.t. in Fig. 2, which is often designated also as "velocity diagram" (as distinct from the "Forces diagram" or "blade diagram"). The flow angles α and β also appear on Fig. 2

- α angle between v and positive u direction,

- β angle between w and negative u direction.

The position of one v.t. in relation to another v.t. does not indicate whether the flow velocities plotted on these triangles are in fact situated in the same positions in relation to each other in the actual flow. This is made clear by Figs. 4 and 6. Fig. 7 illustrates the subdivision of a velocity into its axial, radial and peripheral components. The peripheral components (circumferential velocity) of the absolute and relative velocity are also illustrated in Fig. 2.

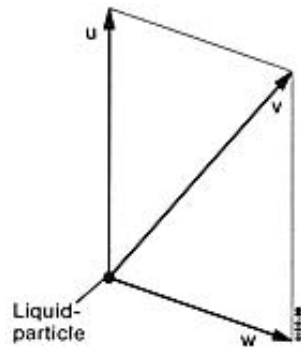


Fig. 1: Velocity parallelogram u , v , w

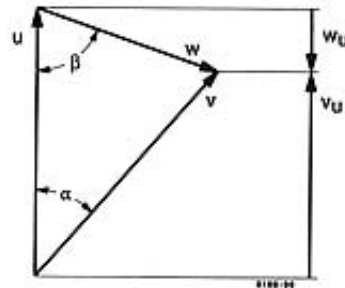


Fig. 2: Velocity triangle

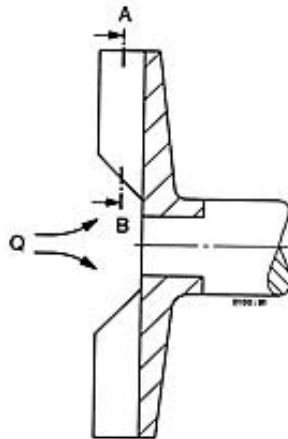


Fig. 3: Meridian section through a radial impeller (longitudinal section through pump shaft)

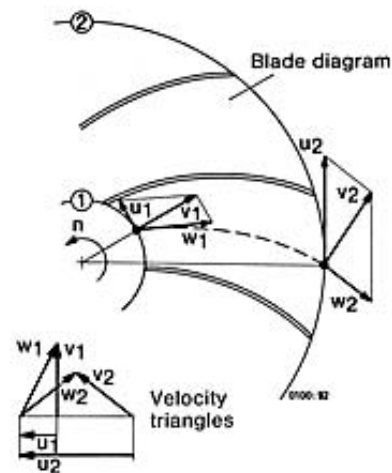


Fig. 4: Blade diagram and velocity triangles for section A-B through the radial impeller of Fig. 3

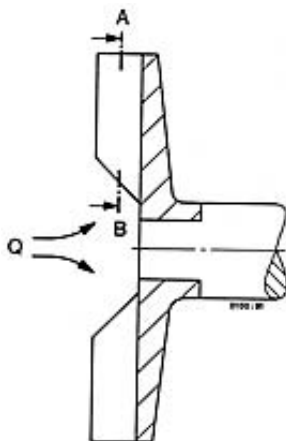


Fig. 5: Meridian section through an axial impeller

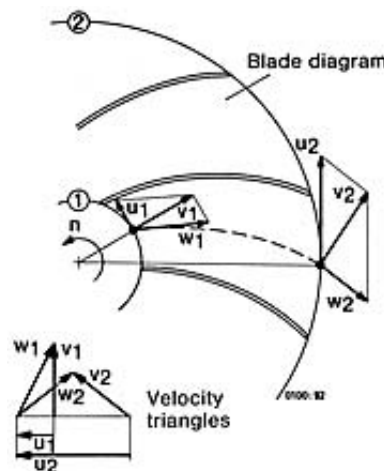


Fig. 6: Blade diagram and velocity triangles for the cylindrical section A-B through the axial impeller of Fig. 5

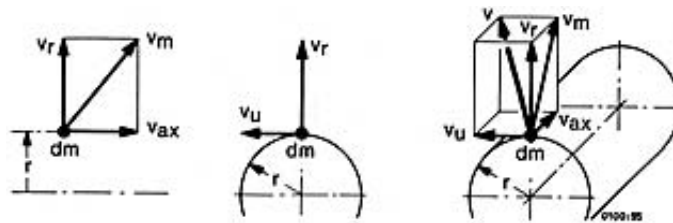


Fig. 7: Subdivision of a velocity into an axial, radial and peripheral component, illustrated in the example of the absolute velocity v of a liquid particle dm

Table 1: Vapour pressure p_D and density ρ of water (diagram)

t	T	p_D	ρ	t	T	p_D	ρ	t	T	p_D	ρ
°C	K	bar	kg/dm ³	°C	K	bar	kg/dm ³	°C	K	bar	kg/dm ³
0	273.15	0.00611	0.9998					138	411.15	3.414	0.9276
1	274.15	0.00657	0.9999	61	334.15	0.2086	0.9826	140	413.15	3.614	0.9258
2	275.15	0.00706	0.9999	62	335.15	0.2184	0.9821	145	418.15	4.155	0.9214
3	276.15	0.00758	0.9999	63	336.15	0.2286	0.9816	150	423.15	4.760	0.9168
4	277.15	0.00813	1.0000	64	337.15	0.2391	0.9811	155	428.15	5.433	0.9121
5	278.15	0.00872	1.0000	65	338.15	0.2501	0.9805	160	433.15	6.181	0.9073
6	279.15	0.00935	1.0000	66	339.15	0.2615	0.9799	165	438.15	7.008	0.9024
7	280.15	0.01001	0.9999	67	340.15	0.2733	0.9793	170	443.15	7.920	0.8973
8	281.15	0.01072	0.9999	68	341.15	0.2856	0.9788	175	448.15	8.924	0.8921
9	282.15	0.01147	0.9998	69	342.15	0.2984	0.9782	180	453.15	10.027	0.8869
10	283.15	0.01227	0.9997	70	343.15	0.3116	0.9777	185	458.15	11.233	0.8815
								190	463.15	12.551	0.8760
11	284.15	0.01312	0.9997	71	344.15	0.3253	0.9770	195	468.15	13.987	0.8704
12	285.15	0.01401	0.9996	72	345.15	0.3396	0.9765	200	473.15	15.50	0.8647
13	286.15	0.01497	0.9994	73	346.15	0.3543	0.9760	205	478.15	17.243	0.8588
14	287.15	0.01597	0.9993	74	347.15	0.3696	0.9753	210	483.15	19.077	0.8528
15	288.15	0.01704	0.9992	75	348.15	0.3855	0.9748	215	488.15	21.060	0.8467
16	289.15	0.01817	0.9990	76	349.15	0.4019	0.9741	220	493.15	23.198	0.8403
17	290.15	0.01936	0.9988	77	350.15	0.4189	0.9735	225	498.15	25.501	0.8339
18	291.15	0.02062	0.9987	78	351.15	0.4365	0.9729	230	503.15	27.976	0.8273
19	292.15	0.02196	0.9985	79	352.15	0.4547	0.9723	235	508.15	30.632	0.8205
20	293.15	0.02337	0.9983	80	353.15	0.4736	0.9716	240	513.15	33.478	0.8136
								245	518.15	36.523	0.8065
21	294.15	0.02485	0.9981	81	354.15	0.4931	0.9710	250	523.15	39.776	0.7992
22	295.15	0.02642	0.9978	82	355.15	0.5133	0.9704	255	528.15	43.246	0.7916
23	296.15	0.02808	0.9976	83	356.15	0.5342	0.9697	260	533.15	46.943	0.7839
24	297.15	0.02982	0.9974	84	357.15	0.5557	0.9691	265	538.15	50.877	0.7759
25	298.15	0.03166	0.9971	85	358.15	0.5780	0.9684	270	543.15	55.058	0.7678
26	299.15	0.03360	0.9968	86	359.15	0.6011	0.9678	275	548.15	59.496	0.7593
27	300.15	0.03564	0.9966	87	360.15	0.6249	0.9671	280	553.15	64.202	0.7505
28	301.15	0.03778	0.9963	88	361.15	0.6495	0.9665	285	558.15	69.186	0.7415
29	302.15	0.04004	0.9960	89	362.15	0.6749	0.9658	290	563.15	74.461	0.7321
30	303.15	0.04241	0.9957	90	363.15	0.7011	0.9652	295	568.15	80.037	0.7223

								300	573.15	85.927	0.7122
31	304.15	0.04491	0.9954	91	364.15	0.7281	0.9644	305	578.15	92.144	0.7017
32	305.15	0.04753	0.9951	92	365.15	0.7561	0.9638	310	583.15	98.700	0.6906
33	306.15	0.05029	0.9947	93	366.15	0.7849	0.9630	315	588.15	105.61	0.6791
34	307.15	0.05318	0.9944	94	367.15	0.8146	0.9624	320	593.15	112.89	0.6669
35	308.15	0.05622	0.9940	95	368.15	0.8453	0.9616	325	598.15	120.56	0.6541
36	309.15	0.05940	0.9937	96	369.15	0.8769	0.9610	330	603.15	128.63	0.6404
37	310.15	0.06274	0.9933	97	370.15	0.9094	0.9602	340	613.15	146.05	0.6102
38	311.15	0.06624	0.9930	98	371.15	0.9430	0.9596	350	623.15	165.35	0.5743
39	312.15	0.06991	0.9927	99	372.15	0.9776	0.9586	360	633.15	186.75	0.5275
40	313.15	0.07375	0.9923	100	373.15	1.0133	0.9581	370	643.15	210.54	0.4518
								374.15	647.15	221.2	0.3154
41	314.15	0.07777	0.9919	102	375.15	1.0878	0.9567				
42	315.15	0.08198	0.9915	104	377.15	1.1668	0.9552				
43	316.15	0.08639	0.9911	106	379.15	1.2504	0.9537				
44	317.15	0.09100	0.9907	108	381.15	1.3390	0.9522				
45	318.15	0.09582	0.9902	110	383.15	1.4327	0.9507				
46	319.15	0.10086	0.9898								
47	320.15	0.10612	0.9894	112	385.15	1.5316	0.9491				
48	321.15	0.11162	0.9889	114	387.15	1.6362	0.9476				
49	322.15	0.11736	0.9884	116	389.15	1.7465	0.9460				
50	323.15	0.12335	0.9880	118	391.15	1.8628	0.9445				
				120	393.15	1.9854	0.9429				
51	324.15	0.12961	0.9876								
52	325.15	0.13613	0.9871	122	395.15	2.1145	0.9412				
53	326.15	0.14293	0.9866	124	397.15	2.2504	0.9396				
54	327.15	0.15002	0.9862	126	399.15	2.3933	0.9379				
55	328.15	0.15741	0.9857	128	401.15	2.5435	0.9362				
56	329.15	0.16511	0.9852	130	403.15	2.7013	0.9346				
57	330.15	0.17313	0.9846								
58	331.15	0.18147	0.9842	132	405.15	2.8670	0.9328				
59	332.15	0.19016	0.9837	134	407.15	3.041	0.9311				
60	333.15	0.19920	0.9832	136	409.15	3.223	0.9294				

Table 2: Vapour pressure of various liquids (see diagram)

Temperature	Ethane C_2H_6	Acetone $(CH_3)_2CO$	Ammonia NH_3	Ethyl alcohol C_2H_5OH	n-Butane C_4H_{10}	i-Butane C_4H_{10}	Benzol C_6H_6	Aniline $C_6H_5NH_2$	Ether $C_2H_5OC_2H_5$	Formic acid CH_2O_2	Acetic acid $C_2H_4O_2$	n-Propane C_3H_8	Methanol CH_3O	Sulphur dioxide SO_2	Sulphurous acid H_2SO_3	Carbon disulphide CS_2	Carbon tetrachloride CCl_4	Toluene C_7H_8
t °C T K																		
-50	223	5.517	0.00319	0.409	0.103				0.0127			0.707		0.1157				
-45	228	6.574		0.545								0.890		0.1598				
-40	233	7.776		0.718	0.179				0.0255			1.115		0.2157				
-35	238	9.129		0.932								1.379		0.2883				
-30	243	10.65	0.0149	1.195	0.294	0.483			0.050			1.672		0.3805	0.0335			
-25	248	12.34		1.516								2.017		0.4942				
-20	253	14.23	0.0293	1.902	0.469	0.748			0.883			2.423		0.6355	0.0609			0.0129
-15	258	16.31		2.363								2.889		0.8071				0.0180
-10	263	18.59	0.0516	2.909	0.691	1.103			0.150			3.405		1.014	0.1047			0.0246
-5	268	21.10		3.549								4.015		1.261				0.0330
±0	273	23.76	0.0856	4.294	0.0159	1.039	1.613	0.0354	0.247		0.0044	4.684	0.0381	1.554	0.1697			0.0439
5	278	26.86	0.115	5.157					0.311			5.453		1.899				0.0576
10	283	30.16	0.1542	6.149	0.0306	1.50	2.201	0.0606	0.389	0.0245	0.0085	6.339	0.0699	2.302	0.2648	0.017		0.0746
15	288	33.76	0.196	7.283					0.481			7.298		2.768				0.0956
20	293	37.75	0.246	8.572	0.0568	2.069	3.119	0.0996	0.589	0.0419	0.0156	8.334	0.1227	3.305	0.3996	0.0298		0.1213
25	298	42.15	0.306	10.03					0.716			9.489		3.920				0.1527
30	303	47.07	0.377	11.67	0.1008	2.824	4.232	0.1578	0.864	0.0688	0.0275	10.807	0.2068	4.619	0.5848	0.0489		0.1907
35	308		0.462	13.50								12.219		5.411				0.2349
40	313		0.562	15.54	0.1722	3.765	5.609	0.2412	1.228	0.1097	0.0464	13.739	0.336	6.303	0.8306	0.0784		0.2876
45	318		0.681	17.81								15.455		7.303				0.3499
50	323		0.817	20.33	0.2836	4.98	7.257	0.3589	0.00319	1.702	0.1696	0.0754	17.269	0.5283	8.417	1.1466	0.121	0.4228
55	328																	0.5057
60	333		1.118		0.4519	6.37	9.267	0.5188	0.0075	2.306	0.2549	0.1186	20.89	0.8095		1.549	0.1863	0.6010
65	338																	0.7078
70	343		1.55		0.6979	8.14	11.719	0.7301	0.0139	3.061	0.3733	0.1812	25.79	1.1954			0.2689	0.8296
75	348																	
80	353		2.08		1.047	10.20		1.0052	0.0239	3.991	0.533	0.269	31.38	1.7298		2.700	0.3818	1.1169
85	358												34.13					
90	363		2.76		1.531	12.55		1.355	0.0389	5.121	0.7439	0.3915	36.58	2.445			0.5369	1.4828
95	368												39.91					
100	373		3.60		2.184	15.40		1.795	0.0609	6.478	1.0159	0.556		3.384		4.333	0.7354	1.9505
105	378																	
110	383		4.65		3.045	18.34		2.331	0.0922	8.092		0.774		4.595			0.9924	2.5164
115	388																	
120	393		5.89		4.159	21.77		2.984	0.1327	9.992		1.059		6.131		6.999	1.267	3.1911
125	398																	
130	403		5.572		5.572	25.69		3.766	0.1926	12.209		1.423		8.050			1.7407	3.956
135	408																	
140	413						4.694	0.2719	14.768		1.885					10.399	2.2457	4.945
145	418																	
150	423								17.711		2.499						2.824	6.073

Venting

Entlüftung
Purge d'air

Centrifugal pumps must be primed with liquid before start-up (starting torque). Therefore pumps with an overhead suction vessel must be provided with a manually operated vent cock or with a vent valve at the apex of the pump casing.

Sewage pumps are continuously vented via an open vent line of at least 25 mm ø the leakage water flow is allowed for in this arrangement, and is led back to the intake chamber or similar.

Centrifugal pumps that are set up above the water leveling whose suction pipes are equipped with a foot valve, are filled manually, with use of a filling funnel, at the pump suction branch or through a v. system, when they are not capable of creating their own suction (self-priming pump, filling-up of centrifugal pumps).

The v. systems are systems based on water ring pumps or jet pumps (deep well suction device). There are automatics. systems that keep the pumps ready for operation, and start-up v. systems that prime the suction pipe and the pump just prior to start-up.

When calculating the v. time of suction pipes and siphoning installations, the air volume to be drawn off should be calculated separately for the continuously rising portion and the horizontal portion of the piping, and the two added together.

The following equations apply:

$$a) T \cdot Q_s = V_1 \left(2 - \frac{p_2}{p_1 - p_2} \cdot \ln \frac{p_1}{p_2} \right)$$

for rising pipes,

$$b) T \cdot Q_s = V_2 \left(\ln \frac{p_1}{p_2} + 1 \right)$$

for horizontal pipes.

The expressions in brackets are combined in a factor f which is plotted in illustration in function of the geodetic suction lift Hs.geo (head).

Then

$$T = \frac{V}{Q_s} \cdot f$$

with

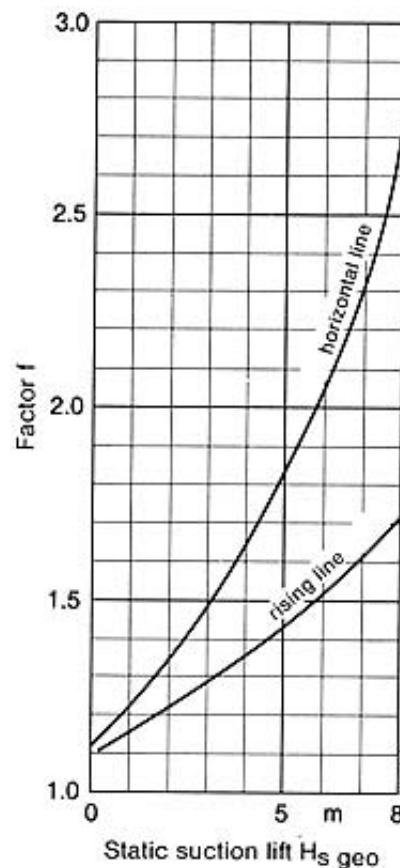
T vent time in s,

V gas-filled volume of rising line or horizontal line in m³. The gas volume within the pump should be added to that of the horizontal line.

Q_s capacity of vent pump in m³/s,

p₁ pressure in bar at which the v. process of the pipeline gas volume V starts,

p₂ pressure in bar at which the pipeline volume under consideration in each case is filled with water.



Factor f for determination of suction volume of vent pumps for suction pipe and siphoning installations. Factor f incorporates a 10 % safety margin to take into account the gas and air content of the water

Venturi Nozzle

Venturidüse

Tuyère de venturi

see [Standard Venturi Nozzle](#)

Vent Valve

Belüftungsventil, Entlüftungsventil

Vanne d'isolement de casse-vide, soupape d'évacuation

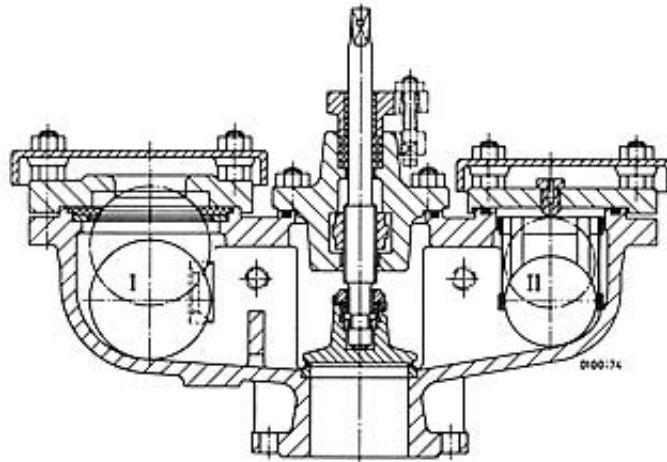
The v.v. (sometimes also referred to as float valve), also known as float valve, should remain open for as long as there is air in the space to be vented; it should close as soon as water starts to flow outwards. V.v.'s are fitted to facilitate rapid venting during start-up (starting torque) of a pump, and to lead away accumulations of air in hydraulic pipes. Both functions are taken care of by the double valve (see illustration), for example, which is connected to the apex of the space to be vented.

In the venting space I, a large aperture is provided for the expulsion of the air, and this aperture is sealed by a ball float when water starts to stream through. The ball float only sinks again when the pressure in the line drops below atmospheric. The pump and suction pipe are then vented.

The ball in the venting space II also floats on the water and closes the air exit aperture. This ball is however sized in such a way that it sinks again because of its weight, when air accumulates in the valve housing, despite the pressure in the piping or pump. Thus small accumulations of air can be continuously vented and led away while the pump is running.

When starting up the pump it must be seen to it that the v.v. does not close too quickly, since otherwise pressure surges (surge pressure) may occur.

The v.v. may also be used as aerating valve.



Double valve for venting flow spaces filled with water, with additional shut-off device.
Part I is shown on the left, Part II on the right

Vertical Can-Type Pump

Topfpumpe

Pompe à cuve de charge

see Condensate Pump, Refinery Pump

Vertical Pump

Vertikalpumpe

Pompe verticale

V.p. is a centrifugal pump with a vertical drive shaft.

Vibration

Schwingung

Vibration

V. of a physical quantity is said to occur, when its time trace shows regular or irregular reversal points.

Mechanical v.'s are almost unavoidable with rotating or oscillating machinery, and with flowing medium. As with all fluid flow machines, v. are evident in centrifugal pumps and machines that are influenced by the fluid handled (quietness of centrifugal pumps).

The simple v. measurements, such as types and frequency, are described in international literature, with examples being VDI 2056, VDI 2059, ISO 2372, ISO 5199, VDMA 24297, and API 610. Also given are general values for the overall assessment of v.'s in rotating machinery.

There is even information about the effects of v. on the human being, as found DIN 45675, VDI 2057, and ISO 2631.

V.'s are like a warning sign when watching the operating behaviour of machinery.

When planning the layout and construction of machinery, as well as the layout of an entire system, one main objective is to eliminate causes of v., especially resonances (critical speed of rotation, unbalance of centrifugal pumps).

Viscosity

Viskosität

Viscosité

V. is generally understood to mean the property of a fluid to resist the relative displacement of adjoining layers (internal friction). The physical definition of v. originates in the so-called NEWTON formulation of the shear stress (NEWTONian liquid):

$$\tau = \eta \frac{\partial v_x}{\partial y}$$

with

τ shear stress,

v_x local flow velocity,

$\partial v_x / \partial y$ velocity gradient in y direction, i.e. perpendicular to direction of flow,

η dynamic v.

The v. curve in the shear diagram $\tau = f(\partial v_x / \partial y)$ is, therefore, a straight line (Fig. 1a under pulp pumping). All other curves denote non-NEWTONian liquids whose effects on the operation of centrifugal pumps cannot easily be calculated.

The coefficient of proportionality η between shear stress and velocity gradient is named dynamic v. (unit) because its unit of $1 \text{ N s/m}^2 = 1 \text{ Pa s}$ contains a unit of force.

In practice, it is usual to quote the viscosity-density ratio, which is named kinematic v. because of its unit m^2/s

$$\nu = \frac{\eta}{\rho}$$

with

ν kinematic v.,

η dynamic v.,

ρ density of pumped medium.

The SI unit of kinematic v. is $1 \text{ m}^2/\text{s}$, (e.g. water at 20°C has a kinematic v. of $\nu = 1.002 \cdot 10^{-6} \text{ m}^2/\text{s}$), but most measurements are still made in centi-Stokes (e.g. in the UBBELOHDE viscometer). This unit cm^2/s is called a Stokes (St) in honour of the English physicist STOKES, and the hundredth part thereof is a centi-Stokes (cSt). The unit Stokes ceased to be valid on 1977. Tables 1 and 2 are provided for the purpose of converting Stokes and other viscosity units still frequently used today but no longer valid due to the adoption of SI units, into the SI unit m^2/s . Table 1 lists the conversions of the various units of kinematic v. Table 2 shows the numeric values for various units in comparison to one another. These conventional units of v. are unsuitable as a basis for calculation, but are still widely used in commerce. These conventional v. units include: degrees Engler ($^\circ\text{E}$) in Germany, Saybolt seconds (S") in USA, and Redwood seconds (R") in Britain.

Table1: Conversion of units for kinematic viscosity ν

	German units ¹⁾				British units	
	$\frac{\text{m}^2}{\text{s}}$	$\frac{\text{m}^2}{\text{h}}$	Stokes	Centistokes	$\frac{\text{ft}^2}{\text{s}}$	$\frac{\text{ft}^2}{\text{h}}$
$1 \frac{\text{m}^2}{\text{s}} =$	1	$0.36 \cdot 10^4$	10^4	10^6	$0.10764 \cdot 10^2$	$3.8751 \cdot 10^4$
$1 \frac{\text{m}^2}{\text{h}} =$	$2.7778 \cdot 10^4$	1	2.7778	$2.7778 \cdot 10^2$	$0.299 \cdot 10^{-2}$	$0.10764 \cdot 10^2$
$1 \text{ Stokes} = 1 \frac{\text{cm}^2}{\text{s}} =$	10^{-4}	0.36	1	10^2	$0.10764 \cdot 10^{-2}$	3.8751
$1 \text{ Centistokes} = 1 \text{ cSt} =$	10^{-6}	$0.36 \cdot 10^{-2}$	10^{-2}	1	$0.10764 \cdot 10^{-4}$	$3.8751 \cdot 10^{-2}$
$1 \frac{\text{ft}^2}{\text{s}} =$	$9.2903 \cdot 10^{-2}$	$3.3445 \cdot 10^2$	$9.2903 \cdot 10^2$	$9.2903 \cdot 10^4$	1	$0.3600 \cdot 10^4$
$1 \frac{\text{ft}^2}{\text{h}} =$	$0.25806 \cdot 10^{-4}$	$9.2903 \cdot 10^{-2}$	0.25806	$0.25806 \cdot 10^2$	$2.7778 \cdot 10^{-4}$	1

1) SI unit m^2/s ; Stokes ceased to be valid on 1.1.1978

Table 2 is applicable up to 1000 cSt. If higher values are required, the data given from 60 cSt upward can be multiplied accordingly by 10, 100, 1000 etc. The conversion of conventional measures into cSt is inaccurate, particularly within the 1 to 9.5 cSt range (see DIN 1342, viscosity of NEWTONian liquids).

The dependence of ν on temperature is shown for water on diagram; for mineral oil distillate it is displayed in Fig. 1, where the axes were chosen as to provide straight lines. According to DIN 51 519, there exist 18 classes of ν , whose numerical values, give the middle ν points in mm^2/s at 40 °C. The allowable boundaries of each class are $\pm 10\%$ of these middle points; the middle points at 40 °C are marked with circles in Fig. 1. Table 3 gives the requirements for the ν classes of various types of mineral oil. The remaining effect of temperature can only be determined generally, according to DIN 51 502, and is shown in Fig. 1. This diagram can also be used for other liquids (as an approximation) as long as two points on the viscosity-temperature line of the liquid are known. These two points should be entered in the diagram and joined by a straight line to obtain the desired relationship. If the pair of values of only one point are known, the point can be entered on the diagram and a strange line which fits in with the pattern of the existing lines drawn, to give an approximate relationship. All the values on this diagram are approximate only.

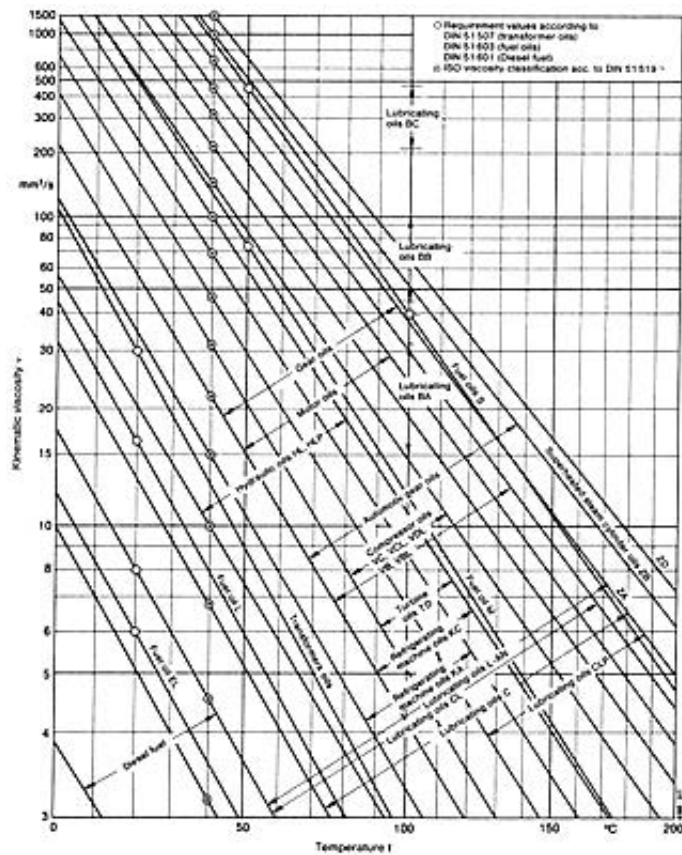


Fig. 1: Correlation between kinematic viscosity of various mineral oil distillate and temperature. See Table 3 and enlarged diagram

Table 3: Viscosity classes of liquid lubricants (to Fig.1)

DIN	Name, use, requirement	Symbol	Viscosity in mm ² /s (at t ≠ 40 °C) resp. viscosity class ISO VG (t = 40 °C)	Reference temperatur in °C
51 501	Lubricating oils L-AN (minimum requirements)	L-AN	5 to 680	40
51 503	Refrigerating machine oils for compressors with ammonia Refrigerating machine oils for compressors with halogenated hydrocarbons	KA KC	15 to 68 22 to 100	40 40
51 506	Lubricating oils for compressor and vacuum pump	VB,VBL VC,VCL VDL	22 to 460 32 to 150	40 40
51 509	Lubricating oils for gears (selection)		22 to 680	40
51 510	Lubricating oils Z for superheated steam cylinder	ZA ZB ZD	≧ 30 (ISO VG 680) ≧ 40 (ISO VG 1000) ≧ 60 (ISO VG 1500)	100 100 100
51 511	Lubricating oils for Otto carburetor engine and Diesel engine	SAE	3.8 to 21.9	100
51 512	Lubricating oils for automotive gears	SAE	≧ 4.2 to ≧ 43	98.9
51513	Bituminiferous lubricating oils B (for hand-,traversing- and splash lubrication) specially in gears, gliding plane etc.	BA BB BC	15 to 32 46 to 100 220 to 460	100 100 100
51 515, T.1	Lubricating oils and governor oils L-TD for steam turbines and other turbo machines	TD	32 to 100	40
51 517, T.1	Lubricating oils C, non-aging, without active component additions	C	7 to 680	40
51 517, T.2	Lubricating oils CL, with active components for anticorrosion and resistance to aging	CL	5 to 460	40
51 517, T.3	Lubricating oils CLP, as CL, however, with active components against mixed friction wear	CLP	46 to 680	40
51 524, T.1	Hydraulic oils HL, with active components as lubricating oil CL	HL	10 to 100	40
51 524, T.2	Hydraulic oils HLP, as HL, however, with active components against mixed friction wear	HLP	10 to 100	40
51 601	Diesel fuel	DK	2 to 8	20
51 603, T.1	Fuel oil EL	EL	< 6	20
51 603, T.2	Fuel oils L, M and S	L M S S	< 17 < 75 < 450 < 40	20 50 50 100

Example: given is an oil with $\nu_1 = 13 \text{ mm}^2/\text{s}$ at 50°C , sought kinematic ν . ν_2 at 20°C , answer $\nu_2 \approx 45 \text{ mm}^2/\text{s}$.

With increasing ν of the medium, the pressure losses rise in the piping, and the pump head falls in the centrifugal pump, as is qualitatively shown in Fig. 2, at a constant rotational speed n , capacity Q , and head H ; because of the increased impeller side friction, the shaft power P increases. The peak of the sinking efficiency curve moves over towards the part load region and the shaft power curve is displaced upwards almost parallel to the original one. The head at pump shut-off point remains almost the same.

When dealing with the transport of NEWTONian liquids, two techniques are used for the conversion of the characteristic curves of centrifugal pumps: The method of the Standards of Hydraulic Institute, New York, USA (1983) (Fig. 3), that had already been experimentally tried in 1955 on volute casing pumps of nominal diameters 50 to 200 mm and a specific speed of approximately $n_q = 20 \text{ min}^{-1}$. The other method, developed by KSB (Fig. 4), that is based on measurements on standardized chemical pumps (impeller, with back shroud blades) at specific rotational speeds ranging from 6.5 to 45 min^{-1} . Both methods use graphs that differ due to varying specific speeds, but are basically used in the same manner.

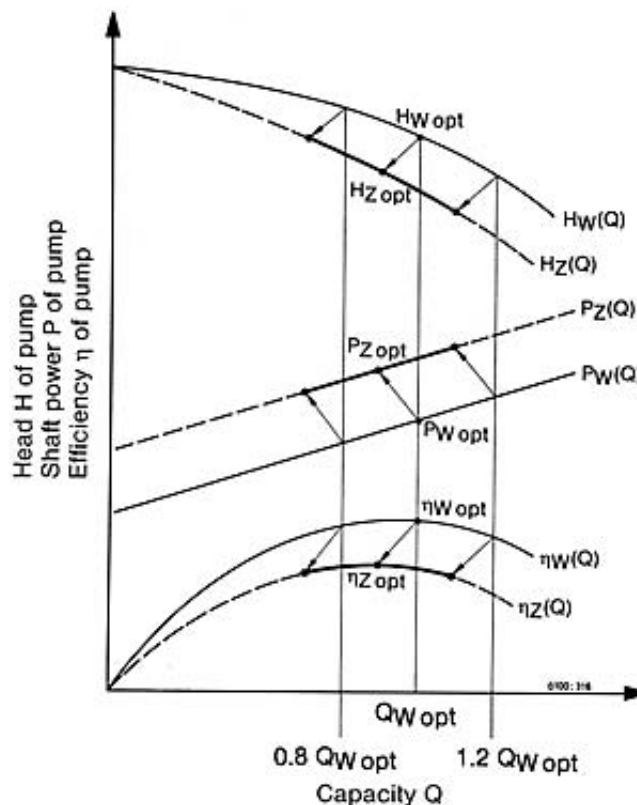


Fig. 2: Characteristic curves of a centrifugal pump when pumping water (subscript W) and a viscous liquid (subscript Z)

In the following remarks, subscript W represents the values for pumping water and subscript Z the values for pumping a viscous liquid. If the characteristic curves $H_w(Q_w)$, $P_w(Q_w)$ and $\eta_w(Q_w)$ for a single stage centrifugal pump with a radial impeller are known for water transport, values can be calculated for a viscous liquid using correction factors f_H , f_Q and f_η from Fig. 3 and 4:

$$\begin{aligned} Q_Z &= f_Q \cdot Q_W, \\ H_Z &= f_H \cdot H_W, \\ \eta_Z &= f_\eta \cdot \eta_W. \end{aligned}$$

Use the head per stage for H_w in multistage centrifugal pumps; in this case f_Q and f_H can be taken from Fig. 4 with some estimation, whereas a value of about $0.4 + 0.6 \cdot f_\eta$ is required for the conversion of the efficiency (which is more realistic for all impellers without back shroud blades).

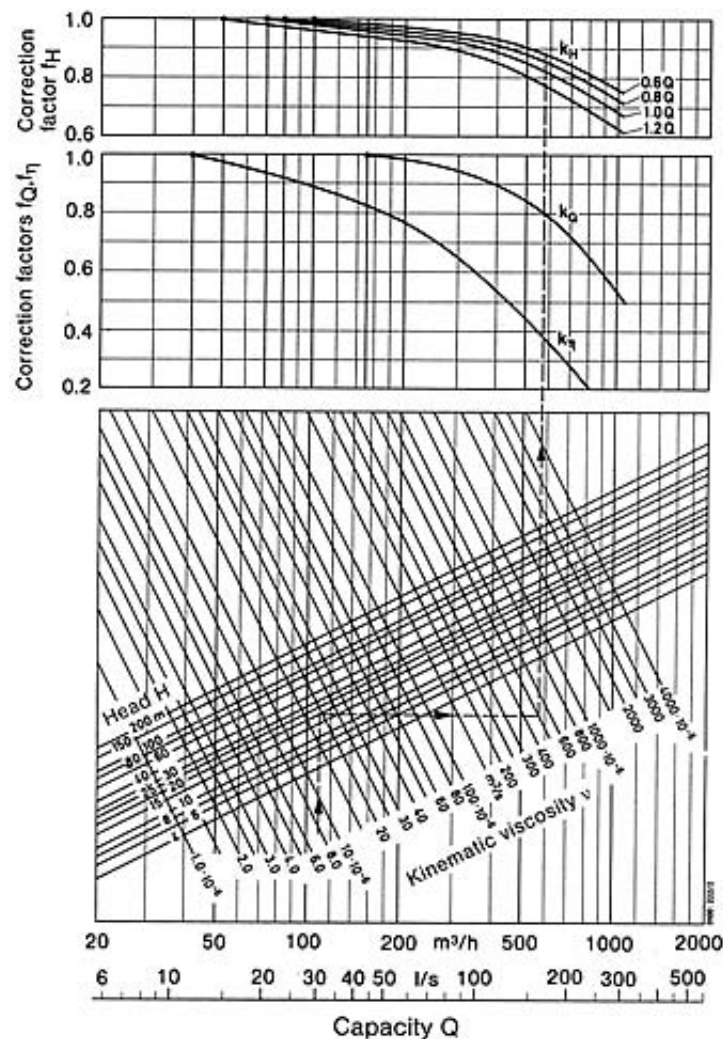


Fig. 3: Diagram for the conversion of characteristic curves of centrifugal pumps with radial impellers for pumping viscous liquids (from Standards of Hydraulic Institute, New Yorks USA 1983) (See enlarged diagram)

The provided correction factors are valid for the point of best efficiency H_{Wopt} , Q_{Wopt} , and η_{Wopt} , they can also be used with $Q = 0.8 \cdot Q_{opt}$ and with $Q = 1.2 \cdot Q_{opt}$ (but only if, with $Q = 0.8 \cdot Q_{opt}$, the average value f_H grows by about 3%, but does not exceed 1.0). At $Q = 0$, $H_w = H_z$. In this manner four points on the characteristic curve can be converted, as is demonstrated:

Case 1: The characteristic curves for pumping water are given, as are also the ν and density ρ_z of a viscous liquid. We want to obtain the characteristic curves for the viscous liquid.

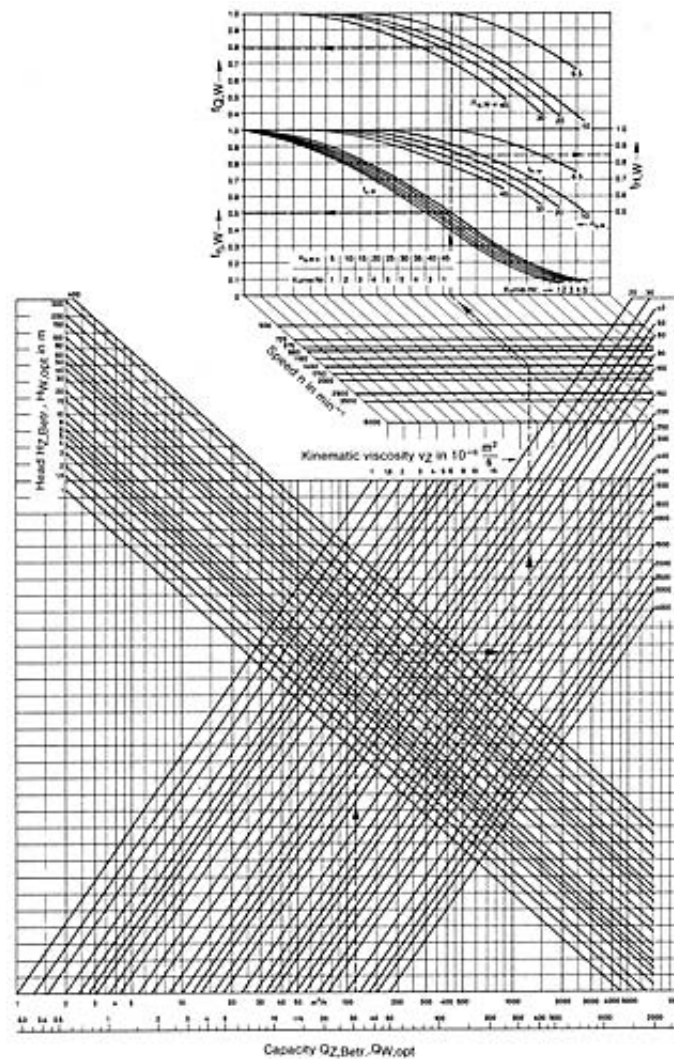


Fig. 4: Diagramm for the conversion of characteristic curves of centrifugal pumps with radial impellers for pumping viscous liquids (from KSB) (See enlarged diagram)

From the characteristic curve for pumping water we select four points for Q_w , H_w and η_w at 0, 0.8, 1.0 and 1.2 times the optimum capacity. Then the correction factors f_Q , f_H and f_η are obtained from Fig. 4 (example see Table 4 or Fig. 3 by starting from the given capacity value Q_w , moving vertically upwards to the intersection with the given H_w value, then horizontally to the right or left to the intersection with the given v value and, then vertically upwards again to the intersections with the curves of the various correction factors. In Fig. 4 one finds an intersection with corresponding rotational speed curves where one follows the frame up and left to the intersection with the next abscissa. In Fig. 3 rotational speed has no effect; even here one obtains an intersection with the abscissa and, in both graphs, follows vertically up to the intersection with the curves of the various correction factors, if applicable according to the various specific speeds or the relative capacity. Multiply the values Q_w , H_w and η_w in Table 4 by these correction factors, so that you obtain the new values Q_z , H_z , and η_z , as explained earlier.

The shaft power P_z is finally calculated from

$$P_z = \rho_z \cdot g \cdot Q_z \cdot \frac{H_z}{\eta_z}$$

with

- η_z pump efficiency,
- ρ_z density of pumped medium,
- g gravitational constant.

Thus the characteristic curves of the pump when pumping the viscous liquid can now be plotted. With Q_z , H_z , η_z and P_z one has four points on the $Q_z H_z$ and $Q_z \eta_z$ line and three points on the $Q_z P_z$ line. These characteristic curves are to be entered over Q_z .

In a quantitative example, according to Table 4, and according to the KSB method (Fig. 4), the power P_z is 2.6 to 2.9 kW smaller, that is, the power of the drive motor can be about 20% less than if using Table 3.

Case 2: The operating data Q_z and H_z for the viscous liquid, and its ν and density ρ are given. Sought is the appropriate pump size for this purpose, but first one must find the correct values for the transport of water (subscript W).

One takes Q_z and H_z as water values Q_{Wopt} and H_{Wopt} , and then using Fig. 3 or 4, one determines a general rotational speed n , the correction factor f_Q and (f_H , as in case 1. Then one must use:

$$Q_w = \frac{Q_z}{f_Q} \quad \text{and} \quad H_w = \frac{H_z}{f_H}$$

to obtain better values after a second liberation from the graph, one can usually already determine the final correction factors and water values Q_w and H_w . With this data, one can choose the proper pump from a diagram or performance chart.

Example: given $Q_z = 170 \text{ m}^3/\text{h}$, $H_z = 30 \text{ m}$, kinematic $\nu = 206 \cdot 10^{-6} \text{ m}^2/\text{s}$ (previously 27 °E), density $\rho = 900 \text{ kg/m}^3$.

From Fig. 3 (or Fig. in the same manner) choose the factors

$$\begin{aligned} f_Q &= 0.94, \\ f_H &= 0.92 \text{ (for } Q/Q_{opt} = 1.0), \\ f\eta &= 0.64. \end{aligned}$$

From the above, we obtain the values for pumping water:

$$\begin{aligned} Q_w &= Q_z / f_Q = 170 / 0.94 = 181 \text{ m}^3/\text{h}, \text{ and} \\ H_w &= H_z / f_H = 30 / 0.92 = 32.6 \text{ m}, \end{aligned}$$

Using these values one pulls the final factors from Fig. 3:

$$\begin{aligned} f_Q &= 0.95, \\ f_H &= 0.92, \text{ and} \\ f\eta &= 0.65 \end{aligned}$$

and calculates the resulting values for the transport of water

$$\begin{aligned} Q_w &= 179 \text{ m}^3/\text{h}, \text{ and} \\ H_w &= 32.6 \text{ m} \end{aligned}$$

according to which the appropriate pump size is selected.

If the centrifugal pump has an efficiency $\eta_w = 80\%$ at the operating point when pumping water, its efficiency when pumping the viscous liquid will be reduced to

$$\eta_z = \eta_w \cdot f\eta = 80\% \cdot 0.65 = 52\%.$$

With this one can even calculate the shaft power P_z of the pump, as in the last section of Table 4.

Table 4: Quantitative example according to Fig. 4 (values in parentheses are from Fig. 3) for an impeller with back shroud blades

Given: Operating data for operation in water with subscript W

Desired: Operating data for operation with viscous liquid with subscript Z

Given are:

capacity	$Q_{W,opt}$	31	1/s
head	$H_{W,opt}$	20	m
efficiency	$\eta_{W,opt}$	0.78	-
rotational speed	n	1450	min^{-1}
kinematic v.	ν_z	$500 \cdot 10^{-6}$	m^2/s
density	ρ_z	0.897	kg/dm^3
gravitational constant	g	9.81	m/s^2
specific speed ¹⁾	$n_{q,W}$	27	min^{-1}

Found:

Q_W/Q_{opt}		0	0.8	1.0	1.2	-
Q_W	given for 4 points on the characteristic curve	0	24.8	31	37.2	1/s
H_W		25	21.6	20	18.2	m
η_W		0	0.74	0.78	0.73	-
f_Q		0.78 (0.80)				-
f_H	from Fig.4 (values in parentheses are from Fig.3)	1.0	0.83 (0.86)	0.83 (0.83)	0.83 (0.78)	-
$f\eta$		0.49 (0.38)				-
$Q_Z = Q_W \cdot f_Q$		0	19.3 (19.8)	24.2 (24.8)	29.0 (29.8)	1/s
$H_Z = H_W \cdot f_H$ * $= H_W \cdot f_H \cdot 1.03 < H_W$		25	(18.6) 18.5*	16.6 (16.6)	15.1 (14.2)	m
$\eta_Z = \eta_W \cdot f\eta$		0	0.36 (0.28)	0.38 (0.30)	0.36 (0.28)	-
$P_Z = \frac{\rho_Z \cdot g \cdot H_Z \cdot Q_Z}{\eta_Z \cdot 1000}$			8.7 (11.6)	9.3 (12.1)	10.7 (13.3)	kW

1) Calculation see under specific speed

* only valid for $Q_W/Q_{opt} = 0.8$

Table 2: Units of kinematic viscosity ν of 1 to 1000 mm^2/s which are no longer admissible

cSt (centistokes), **°E** (degrees Engler), **S"** (Saybolt Universal Viscosity Sec., at 54.5 °C), **R"** (Redwood No. 1 Viscosity Sec., at 60 °C)

Conversion: 1 cSt = 10^{-6} m²/s = 1 mm²/s

cSt	°E	S"	R"	cSt	°E	S"	R"	cSt	°E	S"	R"
1.00	1.000	28.8	27.7	3.00	1.217	36.07	33.45	4.10	1.315	39.49	36.20
1.20	1.027	29.5	28.2	3.02	1.219	36.13	33.50	4.12	1.317	39.55	36.25
1.40	1.052	30.3	28.7	3.04	1.220	36.20	33.55	4.14	1.319	39.62	36.30
1.60	1.075	31.1	29.2	3.06	1.222	36.26	33.60	4.16	1.321	39.68	36.35
1.80	1.098	31.8	29.7	3.08	1.224	36.33	33.65	4.18	1.322	39.75	36.40
2.00	1.119	32.66	30.95	3.10	1.226	36.39	33.70	4.20	1.324	39.81	36.45
2.02	1.121	32.73	31.00	3.12	1.228	36.45	33.75	4.22	1.326	39.87	36.50
2.04	1.123	32.80	31.05	3.14	1.230	36.52	33.80	4.24	1.328	39.94	36.55

2.06	1.125	32.88	31.10	3.16	1.232	36.58	33.85	4.26	1.329	40.00	36.60
2.08	1.127	32.96	31.15	3.18	1.233	36.65	33.90	4.28	1.331	40.07	36.65
2.10	1.129	33.02	31.20	3.20	1.235	36.71	33.95	4.30	1.333	40.14	36.70
2.12	1.131	33.09	31.25	3.22	1.237	36.77	34.00	4.32	1.334	40.20	36.75
2.14	1.133	33.16	31.30	3.24	1.239	36.84	34.05	4.34	1.336	40.27	36.80
2.16	1.136	33.24	31.35	3.26	1.241	36.90	34.10	4.36	1.338	40.33	36.85
2.18	1.138	33.31	31.40	3.28	1.242	36.97	34.15	4.38	1.340	40.40	36.90
2.20	1.140	33.38	31.45	3.30	1.244	37.03	34.20	4.40	1.341	40.46	36.95
2.22	1.142	33.45	31.50	3.32	1.246	37.09	34.25	4.42	1.343	40.52	37.00
2.24	1.144	33.52	31.55	3.34	1.248	37.16	34.30	4.44	1.345	40.59	37.05
2.26	1.146	33.60	31.60	3.36	1.250	37.22	34.35	4.46	1.347	40.65	37.10
2.28	1.148	33.67	31.65	3.38	1.252	37.29	34.40	4.48	1.348	40.72	37.15
2.30	1.150	33.74	31.70	3.40	1.253	37.35	34.45	4.50	1.350	40.78	37.20
2.32	1.152	33.81	31.75	3.42	1.255	37.41	34.50	4.52	1.352	40.84	37.25
2.34	1.154	33.88	31.80	3.44	1.257	37.48	34.55	4.54	1.354	40.91	37.30
2.36	1.156	33.96	31.85	3.46	1.259	37.55	34.60	4.56	1.355	40.97	37.35
2.38	1.158	34.04	31.90	3.48	1.261	37.61	34.65	4.58	1.357	41.04	37.40
2.40	1.160	34.10	31.95	3.50	1.261	37.67	34.70	4.60	1.359	41.10	37.45
2.42	1.162	34.17	32.00	3.52	1.264	37.73	34.75	4.62	1.360	41.16	37.50
2.44	1.164	34.25	32.05	3.54	1.266	37.79	34.80	4.64	1.362	41.23	37.55
2.46	1.165	34.32	32.10	3.56	1.268	37.85	34.85	4.66	1.364	41.29	37.60
2.48	1.167	34.39	32.15	3.58	1.269	37.91	34.90	4.68	1.366	41.36	37.65
2.50	1.169	34.46	32.20	3.60	1.271	37.97	34.95	4.70	1.367	41.42	37.70
2.52	1.171	34.52	32.25	3.62	1.273	38.03	35.00	4.72	1.369	41.48	37.75
2.54	1.173	34.59	32.30	3.64	1.275	38.09	35.05	4.74	1.371	41.55	37.80
2.56	1.175	34.65	32.35	3.66	1.277	38.15	35.10	4.76	1.372	41.61	37.85
2.58	1.177	34.72	32.40	3.68	1.278	38.21	35.15	4.78	1.374	41.68	37.90
2.60	1.179	34.78	32.45	3.70	1.280	38.27	35.20	4.80	1.376	41.74	37.95
2.62	1.181	34.84	32.50	3.72	1.282	38.33	35.25	4.82	1.378	41.80	38.00
2.64	1.183	34.91	32.55	3.74	1.284	38.39	35.30	4.84	1.379	41.87	38.05
2.66	1.185	34.96	32.60	3.76	1.285	38.45	35.35	4.86	1.381	41.93	38.10
2.68	1.187	35.04	32.65	3.78	1.287	38.51	35.40	4.88	1.383	42.00	38.15

2.70	1.189	35.10	32.70	3.80	1.289	38.57	35.45	4.90	1.384	42.06	38.20
2.72	1.191	35.16	32.75	3.82	1.291	38.63	35.50	4.92	1.386	42.12	38.25
2.74	1.192	35.23	32.80	3.84	1.292	38.69	35.55	4.94	1.388	42.19	38.30
2.76	1.194	35.29	32.85	3.86	1.294	38.75	35.60	4.96	1.390	42.25	38.35
2.78	1.196	35.36	32.90	3.88	1.296	38.81	35.65	4.98	1.391	42.32	38.40
2.80	1.198	35.43	32.95	3.90	1.298	38.87	35.70	5.00	1.393	42.38	38.45
2.82	1.200	35.49	33.00	3.92	1.300	38.93	35.75	5.05	1.397	42.54	38.58
2.84	1.202	35.56	33.05	3.94	1.301	38.99	35.80	5.10	1.402	42.70	38.72
2.86	1.204	35.62	33.10	3.96	1.303	39.05	35.85	5.15	1.406	42.86	38.85
2.88	1.206	35.69	33.15	3.98	1.305	39.11	35.90	5.20	1.410	43.02	38.99
2.90	1.207	35.75	33.20	4.00	1.307	39.17	35.95	5.25	1.414	43.18	39.12
2.92	1.209	35.81	33.25	4.02	1.308	39.23	36.00	5.30	1.419	43.34	39.26
2.94	1.211	35.88	33.30	4.04	1.310	39.30	36.05	5.35	1.423	43.50	39.39
2.96	1.213	35.94	33.35	4.06	1.312	39.36	36.10	5.40	1.427	43.66	39.53
2.98	1.215	36.01	33.40	4.08	1.314	39.43	36.15	5.45	1.432	43.82	39.66
cSt	°E	S"	R"	cSt	°E	S"	R"	cSt	°E	S"	R"
5.50	1.436	43.98	39.80	8.25	1.673	52.95	47.06	12.0	2.020	66.03	58.10
5.55	1.440	44.15	39.92	8.30	1.678	53.12	47.19	12.1	2.029	66.40	58.41
5.60	1.444	44.31	40.05	8.35	1.682	53.29	47.33	12.2	2.039	66.77	58.72
5.65	1.449	44.47	40.17	8.40	1.687	53.46	47.47	12.3	2.049	67.14	59.03
5.70	1.453	44.63	40.30	8.45	1.691	53.63	47.61	12.4	2.059	67.51	59.34
5.75	1.457	44.79	40.42	8.50	1.696	53.80	47.75	12.5	2.069	67.88	59.65
5.80	1.461	44.95	40.55	8.55	1.700	53.97	47.89	12.6	2.079	68.25	59.98
5.85	1.466	45.11	40.67	8.60	1.704	54.15	48.02	12.7	2.089	68.62	60.31
5.90	1.470	45.27	40.80	8.65	1.709	54.32	48.16	12.8	2.098	68.99	60.64
5.95	1.474	45.43	40.92	8.70	1.713	54.49	48.29	12.9	2.108	69.36	60.97
6.00	1.479	45.59	41.05	8.75	1.718	54.66	48.42	13.0	2.118	69.73	61.30
6.05	1.483	45.75	41.18	8.80	1.722	54.83	48.56	13.1	2.128	70.11	61.62
6.10	1.487	45.91	41.32	8.85	1.727	55.00	48.69	13.2	2.138	70.49	61.94
6.15	1.491	46.07	41.46	8.90	1.731	55.17	48.83	13.3	2.148	70.78	62.26
6.20	1.496	46.23	41.59	8.95	1.736	55.34	48.97	13.4	2.158	71.25	62.58
6.25	1.500	46.39	41.73	9.00	1.740	55.51	49.10	13.5	2.168	71.64	62.90
6.30	1.504	46.55	41.86	9.05	1.745	55.68	49.24	13.6	2.178	72.02	63.23

6.35	1.509	46.71	42.00	9.10	1.749	55.58	49.39	13.7	2.188	72.40	63.56
6.40	1.513	46.87	42.13	9.15	1.754	56.02	49.53	13.8	2.198	72.79	63.88
6.45	1.517	47.03	42.26	9.20	1.758	56.19	49.68	13.9	2.208	73.16	64.22
6.50	1.521	47.19	42.20	9.25	1.763	56.36	49.82	14.0	2.218	73.54	64.55
6.55	1.526	47.35	42.53	9.30	1.767	56.53	49.97	14.1	2.228	73.92	64.89
6.60	1.530	47.51	42.66	9.35	1.772	56.70	50.11	14.2	2.239	74.30	65.23
6.65	1.534	47.67	42.79	9.40	1.776	56.87	50.26	14.3	2.249	74.68	65.57
6.70	1.539	47.83	42.92	9.45	1.781	57.04	50.40	14.4	2.259	75.06	65.91
6.75	1.543	47.99	43.05	9.50	1.785	57.21	50.55	14.5	2.269	75.44	66.25
6.80	1.547	48.15	43.18	9.55	1.790	57.38	50.70	14.6	2.280	75.83	66.59
6.85	1.557	48.31	43.31	9.60	1.794	57.55	50.84	14.7	2.290	76.21	66.93
6.90	1.556	48.47	43.44	9.65	1.799	57.72	50.98	14.8	2.300	76.59	67.27
6.95	1.560	48.63	43.57	9.70	1.803	57.89	51.13	14.9	2.311	79.97	67.61
7.00	1.564	48.79	43.70	9.75	1.808	58.06	51.28	15.0	2.32	77.35	67.95
7.05	1.569	48.95	43.83	9.80	1.813	58.23	51.42	15.1	2.33	77.74	68.29
7.10	1.573	49.12	43.96	9.85	1.817	58.40	51.55	15.2	2.34	78.13	68.63
7.15	1.577	49.28	44.09	9.90	1.822	58.57	51.71	15.3	2.35	78.52	68.97
7.20	1.582	49.45	44.22	9.95	1.826	58.74	51.85	15.4	2.36	78.91	69.31
7.25	1.586	49.61	44.35	10.0	1.831	58.91	52.00	15.5	2.37	79.30	69.65
7.30	1.590	49.78	44.48	10.1	1.840	59.26	52.29	15.6	2.38	79.69	70.00
7.35	1.595	49.95	44.61	10.2	1.849	59.61	52.58	15.7	2.39	80.08	70.35
7.40	1.599	50.11	44.74	10.3	1.859	59.96	52.87	15.8	2.41	80.47	70.70
7.45	1.603	50.27	44.87	10.4	1.868	60.31	53.17	15.9	2.42	80.86	71.05
7.50	1.608	50.44	45.00	10.5	1.877	60.66	53.47	16.0	2.43	81.25	71.40
7.55	1.612	50.61	45.13	10.6	1.887	61.02	53.76	16.1	2.44	81.65	71.74
7.60	1.616	50.78	45.27	10.7	1.896	61.37	54.07	16.2	2.45	82.05	72.08
7.65	1.621	50.94	45.41	10.8	1.905	61.72	54.38	16.3	2.46	82.45	72.42
7.70	1.625	51.11	45.54	10.9	1.915	62.07	54.69	16.4	2.47	82.85	72.76
7.75	1.630	51.28	45.69	11.0	1.924	62.42	55.00	16.5	2.48	83.26	73.10
7.80	1.634	51.44	45.81	11.1	1.934	62.78	55.31	16.6	2.49	83.66	73.45
7.85	1.638	51.60	45.94	11.2	1.943	63.14	55.62	16.7	2.50	84.06	73.80
7.90	1.643	51.77	46.08	11.3	1.953	63.50	55.93	16.8	2.51	84.46	74.15
7.95	1.647	51.94	46.22	11.4	1.962	63.86	56.24	16.9	2.52	84.86	74.50

8.00	1.651	52.10	46.35	11.5	1.972	64.22	56.55	17.0	2.53	85.26	74.85
8.05	1.656	52.27	46.49	11.6	1.981	64.59	56.86	17.1	2.54	85.67	75.21
8.10	1.660	52.44	46.63	11.7	1.991	64.95	57.17	17.2	2.56	86.08	75.57
8.15	1.665	52.61	46.77	11.8	2.000	65.31	57.48	17.3	2.57	86.49	75.93
8.20	1.669	52.78	46.91	11.9	2.010	65.67	57.79	17.4	2.58	86.90	76.29
cSt °E S" R"				cSt °E S" R"				cSt °E S" R"			
17.5	2.59	87.32	76.65	26.0	3.58	123.5	108.6	37.0	4.95	172.5	152.2
17.6	2.60	87.73	77.01	26.2	3.60	124.4	109.4	37.2	4.98	173.4	153.0
17.7	2.61	88.14	77.37	26.4	3.62	125.3	110.1	37.4	5.00	174.3	153.8
17.8	2.62	88.55	77.73	26.6	3.65	126.2	110.9	37.6	5.03	175.2	154.6
17.9	2.63	88.96	78.09	26.8	3.67	127.1	111.7	37.8	5.05	176.1	155.4
18.0	2.64	89.37	78.45	27.0	3.70	127.9	112.5	38.0	5.08	177.0	156.2
18.1	2.65	89.78	78.81	27.2	3.72	128.8	113.3	38.2	5.10	177.9	157.0
18.2	2.67	90.19	79.17	27.4	3.75	129.7	114.1	38.4	5.13	178.8	157.8
18.3	2.68	90.60	79.53	27.6	3.77	130.6	114.9	38.6	5.15	179.7	158.6
18.4	2.69	91.01	79.89	27.8	3.80	131.5	115.7	38.8	5.18	180.6	159.1
18.5	2.70	91.42	80.25	28.0	3.82	132.4	116.5	39.0	5.21	181.5	160.3
18.6	2.71	91.84	80.62	28.2	3.85	133.2	117.3	39.2	5.23	182.4	161.1
18.7	2.72	92.25	80.99	28.4	3.87	134.1	118.0	39.4	5.26	183.3	161.9
18.8	2.73	92.66	81.36	28.6	3.89	135.0	118.8	39.6	5.28	184.2	162.7
18.9	2.74	93.07	81.73	28.8	3.92	135.9	119.6	39.8	5.31	185.2	163.5
19.0	2.75	93.48	82.10	29.0	3.94	136.8	120.4	40.0	5.33	186.0	164.3
19.1	2.77	93.90	82.47	29.2	3.97	137.6	121.2	40.2	5.36	187.0	165.1
19.2	2.78	94.32	82.84	29.4	3.99	138.5	122.0	40.4	5.39	187.8	165.9
19.3	2.79	94.74	83.21	29.6	4.02	139.4	122.8	40.6	5.41	188.8	166.7
19.4	2.80	95.16	83.58	29.8	4.04	140.3	123.6	40.8	5.44	189.7	167.5
19.5	2.81	95.58	83.95	30.0	4.07	141.2	124.4	41.0	5.46	190.6	168.3
19.6	2.82	96.01	84.31	30.2	4.09	142.0	125.2	41.2	5.49	191.5	169.1
19.7	2.83	96.43	84.67	30.4	4.12	142.9	126.0	41.4	5.51	192.4	169.9
19.8	2.85	96.85	85.03	30.6	4.14	143.8	126.7	41.6	5.54	193.3	170.7
19.9	2.86	97.27	85.39	30.8	4.17	144.7	127.5	41.8	5.57	194.2	171.5
20.0	2.87	97.69	85.75	31.0	4.19	145.6	128.3	42.0	5.59	195.1	172.3

20.2	2.89	98.53	86.49	31.2	4.22	146.5	129.1	42.2	5.62	196.0	173.1
20.4	2.91	99.37	87.23	31.4	4.24	147.3	129.9	42.4	5.64	196.9	173.9
20.6	2.94	100.2	87.98	31.6	4.27	148.2	130.7	42.6	5.67	197.8	174.8
20.8	2.96	101.0	88.74	31.8	4.29	149.1	131.5	42.8	5.69	198.7	175.6
21.0	2.98	101.9	89.50	32.0	4.32	150.0	132.3	43.0	5.72	199.6	176.4
21.2	3.01	102.8	99.26	32.2	4.34	150.9	133.1	43.2	5.75	200.5	177.2
21.4	3.03	103.6	91.02	32.4	4.37	151.8	133.9	43.4	5.77	201.4	178.0
21.6	3.05	104.5	91.77	32.6	4.39	152.7	134.7	43.6	5.80	202.3	178.8
21.8	3.08	105.3	92.51	32.8	4.42	153.6	135.5	43.8	5.82	203.3	179.6
22.0	3.10	106.2	93.25	33.0	4.44	154.5	136.3	44.0	5.85	204.2	180.4
22.2	3.12	107.1	94.01	33.2	4.47	155.4	137.1	44.2	5.87	205.1	181.2
22.4	3.15	107.9	94.77	33.4	4.49	156.3	137.9	44.4	5.90	206.0	182.0
22.6	3.17	108.8	95.53	33.6	4.52	157.2	138.6	44.6	5.93	207.0	182.9
22.8	3.19	109.6	96.29	33.8	4.54	158.1	139.4	44.8	5.95	207.9	183.7
23.0	3.22	110.5	97.05	34.0	4.57	159.0	140.2	45.0	5.98	208.8	184.5
23.2	3.24	111.4	97.81	34.2	4.60	159.9	141.0	45.2	6.00	209.7	185.3
23.4	3.26	112.2	98.57	34.4	4.62	160.8	141.8	45.4	6.03	210.6	186.1
23.6	3.29	113.1	99.34	34.6	4.65	161.7	142.6	45.6	6.06	211.5	186.9
23.8	3.31	114.0	100.1	34.8	4.67	162.6	143.4	45.8	6.08	212.5	187.7
24.0	3.34	114.8	100.9	35.0	4.70	163.5	144.2	46.0	6.11	213.4	188.5
24.2	3.36	115.7	101.7	35.2	4.72	164.4	145.0	46.2	6.13	214.3	189.3
24.4	3.38	116.5	102.4	35.4	4.75	165.3	145.8	46.4	6.16	215.2	190.1
24.6	3.41	117.4	103.2	35.6	4.77	166.2	146.6	46.6	6.18	216.2	191.0
24.8	3.43	118.3	103.9	35.8	4.80	167.1	147.4	46.8	6.21	217.1	191.8
25.0	3.46	119.1	104.7	36.0	4.82	168.0	148.2	47.0	6.23	218.0	192.6
25.2	3.48	120.0	105.5	36.2	4.85	168.9	149.0	47.2	6.26	218.9	193.4
25.4	3.50	120.9	106.2	36.4	4.87	169.8	149.8	47.4	6.29	219.8	194.2
25.6	3.53	121.8	107.0	35.6	4.90	170.7	150.6	47.6	6.31	220.8	195.0
25.8	3.55	122.7	107.8	36.8	4.93	171.6	151.4	47.8	6.34	221.7	195.8
cSt	°E	S"	R"	cSt	°E	S"	R"	cSt	°E	S"	R"
48.0	6.37	222.6	196.6	72.5	9.56	335.6	296.2	100	13.17	462.9	408.2
48.2	6.39	223.5	197.4	73.0	9.63	337.9	298.2	101	13.30	467.5	412.3
48.4	6.42	224.5	198.2	73.5	9.69	340.2	300.2	102	13.43	472.1	416.4

48.6	6.44	225.4	199.1	74.0	9.76	342.5	302.2	103	13.56	476.8	420.4
48.8	6.47	226.3	199.9	74.5	9.82	344.8	304.3	104	13.69	481.4	424.5
49.0	6.50	227.2	200.7	75.0	9.89	347.1	306.1	105	13.83	486.1	428.6
49.2	6.52	228.2	201.5	75.5	9.95	349.5	308.1	106	13.96	490.7	432.7
49.4	6.55	229.1	202.3	76.0	10.02	351.7	310.2	107	14.09	495.3	436.8
49.6	6.57	230.0	203.1	76.5	10.08	354.1	312.3	108	14.22	500.0	440.9
49.8	6.60	230.9	203.9	77.0	10.15	356.3	314.3	109	14.35	504.6	444.9
50.0	6.62	231.8	204.7	77.5	10.22	358.7	316.4	110	14.48	509.2	449.0
50.5	6.69	234.1	206.7	78.0	10.28	361.1	318.4	111	14.61	513.8	453.1
51.0	6.75	236.4	208.8	78.5	10.35	363.4	320.5	112	14.75	518.4	457.2
51.5	6.82	238.8	210.8	79.0	10.41	365.7	322.5	113	14.88	523.1	461.3
52.0	6.88	241.1	212.8	79.5	10.48	368.0	324.6	114	15.01	527.7	465.3
52.5	6.95	243.4	214.8	80.0	10.54	370.3	326.6	115	15.14	523.3	469.4
53.0	7.01	245.7	216.9	80.5	10.61	372.7	328.6	116	15.27	536.9	473.5
53.5	7.08	248.0	219.0	81.0	10.68	375.0	330.6	117	15.40	541.5	477.6
54.0	7.14	250.3	221.0	81.5	10.74	377.2	332.7	118	15.53	546.2	481.6
54.5	7.21	252.6	223.0	82.0	10.81	379.5	334.7	119	15.67	550.8	485.7
55.0	7.28	254.9	225.0	82.5	10.87	381.9	336.8	120	15.80	555.4	489.8
55.5	7.34	257.2	227.1	83.0	10.94	384.2	338.8	121	15.93	560.0	493.9
56.0	7.41	259.5	229.1	83.5	11.00	386.5	340.8	122	16.06	564.7	498.0
56.5	7.47	261.8	231.1	84.0	11.07	388.8	342.9	123	16.19	569.4	502.1
57.0	7.54	264.1	233.2	84.5	11.13	391.1	344.9	124	16.32	574.0	506.2
57.5	7.60	266.4	235.2	85.0	11.20	393.4	347.0	125	16.45	578.7	510.3
58.0	7.67	268.7	237.2	85.5	11.27	395.7	349.0	126	16.59	583.3	514.4
58.5	7.73	271.0	239.2	86.0	11.33	398.0	351.1	127	16.72	587.9	518.5
59.0	7.80	273.3	241.2	86.5	11.40	400.3	353.1	128	16.85	592.6	522.6
59.5	7.86	275.6	243.3	87.0	11.46	402.6	355.2	129	16.98	597.2	526.7
60.0	7.93	277.9	245.3	87.5	11.53	404.9	357.2	130	17.11	601.8	530.7
60.5	7.99	280.2	247.3	88.0	11.59	407.3	359.2	131	17.24	606.4	534.7
61.0	8.06	282.5	249.4	88.5	11.66	409.6	361.3	132	17.39	611.0	538.8
61.5	8.12	284.8	251.4	89.0	11.73	411.9	363.3	133	17.51	615.7	542.9
62.0	8.19	287.2	253.5	89.5	11.79	414.2	365.4	134	17.64	620.3	547.0

62.5	8.25	289.4	255.5	90.0	11.86	416.6	367.4	135	17.77	624.8	551.1
63.0	8.32	291.8	257.5	90.5	11.92	419.0	369.4	136	17.90	629.5	555.2
63.5	8.38	294.1	259.6	91.0	11.99	421.2	371.5	137	18.03	634.1	559.3
64.0	8.45	296.4	261.6	91.5	12.05	423.5	373.5	138	18.16	638.8	563.4
64.5	8.51	298.7	263.7	92.0	12.12	425.8	375.6	139	18.30	643.4	567.4
65.0	8.58	301.0	365.7	92.5	12.18	428.2	377.6	140	18.43	648.1	571.5
65.5	8.65	303.3	267.7	93.0	12.25	430.5	379.6	141	18.56	652.7	575.6
66.0	8.71	305.6	269.8	93.5	12.32	432.8	381.7	142	18.69	657.3	579.7
66.5	8.78	307.9	271.8	94.0	12.38	435.1	383.7	143	18.82	662.0	583.8
67.0	8.84	310.2	273.8	94.5	12.45	437.4	385.8	144	18.95	666.6	587.8
67.5	8.91	312.5	275.8	95.0	12.51	439.7	387.6	145	19.08	671.2	591.9
68.0	8.97	314.8	277.9	95.5	12.58	442.0	389.9	146	19.22	675.8	596.0
68.5	9.04	317.1	279.9	96.0	12.64	444.3	391.9	147	19.35	680.4	600.1
69.0	9.10	319.4	281.9	96.5	12.71	446.6	394.0	148	19.48	685.2	604.2
69.5	9.17	321.7	284.0	97.0	12.78	448.9	396.0	149	19.61	689.7	608.3
70.0	9.23	324.0	286.0	97.5	12.84	451.2	398.0	150	19.74	694.4	612.3
70.5	9.30	326.3	288.0	98.0	12.91	453.6	400.1	151	19.87	699.0	616.4
71.0	9.37	328.6	290.1	98.5	12.97	455.9	402.1	152	20.01	703.6	620.4
71.5	9.43	331.0	292.1	99.0	13.04	458.2	404.1	153	20.14	708.3	624.5
72.0	9.50	333.3	294.1	99.5	13.10	460.5	406.2	154	20.27	712.9	628.6
cSt	°E	S"	R"	cSt	°E	S"	R"	cSt	°E	S"	R"
155	20.40	717.5	632.7	220	28.9	1018.4	898.0	330	43.4	1527.6	1347.1
156	20.53	722.1	636.8	222	29.2	1027.6	906.2	332	43.7	1536.8	1355.3
157	20.66	726.7	640.9	224	29.5	1036.9	914.4	334	43.9	1546.1	1363.4
158	20.79	731.4	645.0	226	29.7	1046.1	922.6	336	44.2	1555.3	1371.8
159	20.93	736.0	649.1	228	30.0	1055.4	930.8	338	44.5	1564.6	1379.7
160	21.06	740.6	653.2	230	30.3	1064.7	938.9	340	44.7	1573.8	1387.9
161	21.19	745.3	657.3	232	30.5	1073.9	947.1	342	45.0	1583.0	1396.1
162	21.32	749.9	661.4	234	30.8	1083.1	955.3	344	45.3	1592.3	1404.3
163	21.45	754.5	665.5	236	31.1	1092.3	963.4	346	45.5	1601.5	1412.5
164	21.58	759.2	669.6	238	31.3	1101.7	971.5	348	45.8	1610.7	1420.6
165	21.71	763.8	673.6	240	31.6	1111.0	979.7	350	46.1	1620.1	1428.7
166	21.85	768.4	677.7	242	31.8	1120.2	987.9	352	46.3	1629.4	1436.9

167	21.98	773.0	681.7	244	32.1	1129.5	996.1	354	46.6	1638.7	1445.1
168	22.11	777.7	685.8	246	32.4	1138.7	1004.3	356	46.8	1648.0	1453.3
169	22.24	782.3	689.8	248	32.6	1148.0	1012.4	358	47.1	1656.2	1461.5
170	22.37	786.9	693.9	250	32.9	1157.3	1020.5	360	47.4	1666.4	1469.6
171	22.50	791.6	698.0	252	33.2	1166.5	1028.7	362	47.6	1675.6	1477.8
172	22.64	796.2	702.1	254	33.4	1175.8	1036.9	364	47.9	1684.9	1486.0
173	22.77	800.8	706.2	256	33.7	1185.0	1045.1	366	48.2	1694.0	1494.1
174	22.90	805.4	710.3	258	33.9	1194.3	1053.3	368	48.4	1703.4	1502.2
175	23.03	810.2	714.4	260	34.2	1203.5	1061.4	370	48.7	1712.7	1510.3
176	23.16	814.8	718.5	262	34.5	1212.7	1069.6	372	48.9	1721.9	1518.5
177	23.29	819.4	722.6	264	34.7	1222.0	1077.8	374	49.2	1731.2	1526.7
178	23.42	824.1	726.7	266	35.0	1231.2	1086.0	376	49.5	1740.5	1534.9
179	23.56	828.7	730.8	268	35.3	1240.5	1094.1	378	49.7	1749.8	1543.1
180	23.69	833.3	734.8	270	35.5	1249.8	1102.2	380	50.0	1759.0	1551.2
181	23.82	837.9	738.9	272	35.8	1259.0	1110.3	382	50.3	1768.4	1559.4
182	23.95	842.6	743.0	274	36.1	1268.3	1118.4	384	50.5	1777.6	1567.6
183	24.08	847.3	747.1	276	36.3	1277.5	1126.6	386	50.8	1786.8	1575.8
184	24.21	851.8	751.1	278	36.6	1286.8	1134.8	388	51.1	1796.0	1583.9
185	24.35	856.4	755.2	280	36.8	1296.1	1143.0	390	51.3	1805.3	1592.0
186	24.48	861.0	759.2	282	37.1	1305.3	1151.2	392	51.6	1814.6	1600.2
187	24.61	865.7	763.3	284	37.4	1314.6	1159.4	394	51.8	1824.1	1608.3
188	24.74	870.3	767.4	286	37.6	1323.8	1167.5	396	52.1	1833.5	1616.6
189	24.87	874.9	771.5	288	37.9	1333.1	1176.7	398	52.4	1843.0	1624.8
190	25.00	879.5	775.6	290	38.2	1342.4	1183.8	400	52.6	1852	1633
191	25.13	884.1	779.4	292	38.4	1351.6	1192.0	402	52.9	1861	1641
192	25.27	888.7	783.7	294	38.7	1360.9	1200.1	404	53.2	1870	1649
193	25.40	893.4	787.8	296	38.9	1370.1	1208.5	406	53.4	1879	1657
194	25.53	898.0	791.9	298	39.2	1379.4	1216.5	408	53.7	1898	1665
195	25.66	902.6	796.0	300	39.4	1388.7	1224.6	410	53.9	1898	1674
196	25.79	907.2	800.1	302	39.7	1397.9	1232.8	412	54.2	1907	1682
197	25.92	911.8	804.2	304	40.0	1407.2	1240.9	414	54.5	1916	1690
198	26.06	916.5	808.3	306	40.3	1416.4	1249.1	416	54.7	1926	1698
199	26.19	921.1	812.4	308	40.5	1425.8	1257.1	418	55.0	1935	1706

200	26.3	925.8	816.4	310	40.8	1435.0	1265.2	420	55.3	1944	1714
202	26.6	935.0	824.5	312	41.1	1444.2	1273.4	422	55.5	1953	1723
204	26.8	944.3	832.7	314	41.3	1453.5	1281.6	424	55.8	1963	1731
206	27.1	953.4	840.8	316	41.6	1462.7	1289.8	426	56.1	1972	1739
208	27.4	962.7	849.0	318	41.8	1472.1	1298.0	428	56.3	1981	1747
210	27.6	972.0	857.2	320	42.1	1481.3	1306.2	430	56.6	1990	1755
212	27.9	981.3	865.3	322	42.4	1490.5	1314.4	432	56.8	2000	1763
214	28.2	990.5	873.5	324	42.6	1499.8	1322.6	434	57.1	2009	1772
216	28.4	999.7	881.4	326	42.9	1509.0	1330.8	436	57.4	2018	1780
218	28.7	1009.1	889.9	328	43.2	1518.4	1339.0	438	57.6	2028	1788
cSt	°E	S"	R"	cSt	°E	S"	R"	cSt	°E	S"	R"
440	57.9	2037	1796	490	64.5	2268	2000	700	92.1	3240	2857
442	58.2	2046	1804	492	64.7	2277	2008	710	93.4	3287	2898
444	58.4	2055	1812	494	65.0	2287	2017	720	94.7	3333	2939
446	58.7	2066	1821	496	65.3	2296	2025	730	96.1	3379	2980
448	58.9	2074	1829	498	65.5	2305	2033	740	97.4	3425	3021
450	59.2	2083	1837	500	65.8	2315	2041	750	98.7	3472	3062
452	59.5	2092	1845	510	67.1	2361	2082	760	100.0	3518	3102
454	59.7	2102	1853	520	68.4	2407	2123	770	101.3	3564	3143
456	60.0	2111	1861	530	69.7	2453	2163	780	102.6	3611	3184
458	60.3	2120	1870	540	71.1	2500	2204	790	103.9	3657	3225
460	60.5	2129	1878	550	72.4	2546	2245	800	105.3	3703	3266
462	60.8	2139	1886	560	73.7	2592	2286	810	106.6	3749	3306
464	61.1	2148	1894	570	75.0	2639	2327	820	107.9	3796	3347
466	61.3	2157	1902	580	76.3	2685	2368	830	109.2	3842	3388
468	61.6	2166	1910	590	77.6	2731	2408	840	110.5	3888	3429
470	61.8	2176	1919	600	78.9	2777	2449	850	111.8	3935	3470
472	62.1	2185	1927	610	80.3	2824	2490	860	113.2	3981	3511
474	62.4	2194	1935	620	81.6	2870	2531	870	114.4	4027	3551
476	62.6	2203	1943	630	82.9	2916	2572	880	115.8	4074	3592
478	62.9	2213	1951	640	84.2	2963	2612	890	117.1	4120	3633
480	63.2	2222	1959	650	85.5	3009	2653	900	118.4	4166	3674

482	63.4	2231	1968	660	86.8	3055	2694	925	121.7	4282	3776
484	63.7	2240	1976	670	88.2	3101	2735	950	125.0	4398	3878
486	63.9	2250	1984	680	89.5	3148	2776	975	128.3	4513	3980
488	64.2	2259	1992	690	90.8	3194	2817	1000	131.6	4629	4082

Viscous Liquid

Zähe Flüssigkeit
Liquide visqueux

see [Viscosity](#)

Voltage Drop

Spannungsabfall
Chute de tension

The following Table gives the v.d. per 100 m cable length for various copper conductor sections, depending on the type of current and the max. current carrying capacity when the max. current is applied.

Volume Flow

Volumenstrom
Débit

see [Capacity](#)

Volute Casing

Spiralgehäuse
Volute

see [Pump Casing](#)

Volute Casing Pump

Spiralgehäusepumpe
Pompe à volute

V.c.p.'s are the most common type of [centrifugal pump](#). Their characteristic feature is the volute-shaped [pump casing](#), which makes this [pump type](#) instantly recognizable from the outside. Fig. 1 illustrates a construction type with dimensions standardized in accordance with DIN 24255 and 24256/ISO 2858, but this standardization still leaves ample room for many variations of design.

V.c.p.'s are usually single stage, but occasionally two stage ([multistage pump](#), Fig. 2 under pipeline pump).

Both single suction and double suction v.c.p.'s ([multisuction pump](#)) are common. Corresponding to a [specific speed](#) n_q ranging from approx. 12 to 80 min⁻¹ (in special cases up to 100 min⁻¹ and over), the [impellers](#) can be of either radial or mixed flow type. The [axial thrust](#) can be absorbed by a thrust bearing or balanced by holes drilled in the impeller back-plate (balance holes) (Figs. 1, 2 and 5; see also Figs. 7, 8 and 10 under axial thrust), or by [back shroud blades](#) (Fig. 9 under axial thrust), or, in the case of two stage or double suction v.c.p.'s, by a "mirror image"

(back-to-back, Figs. 3 or 4 under axial thrust) arrangement of the impellers. The pump casings can be radially (Figs. 1, 2 and 5) or axially split at shaft centreline (radially split casing, axially split casing).

V.c.p.'s are built in the form of vertical and horizontal pumps, the pump shaft can be supported on anti-friction or plain bearings on one side only or on either side of the impeller. Horizontal v.c.p.'s with an overhung impeller can have their bearings arranged either in a bearing bracket (Figs. 1 and 4) or in a bearing pedestal (Fig. 5) or in the driver (close-coupled pumping set). The bearing bracket is in accordance with the process type design; however the bearing pedestal has the advantage of being able to transmit the radial and axial forces (axial thrust, radial thrust) from the impeller via the shaft direct onto the pump foundation, and enables smaller nameplates to be used.

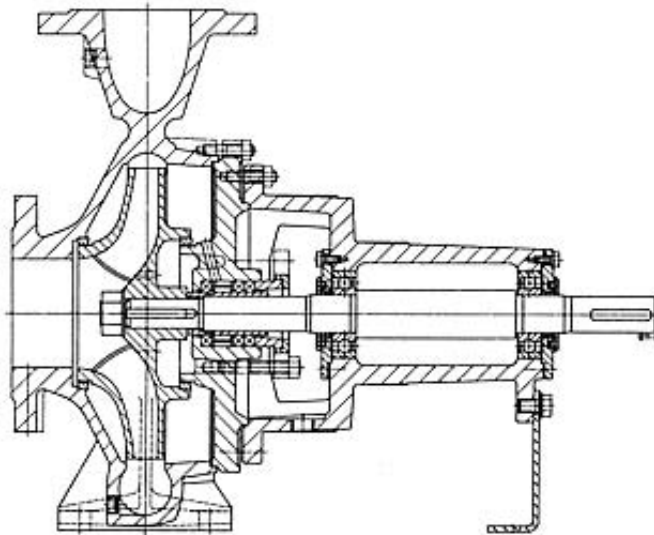


Fig. 1: Volute casing pump in accordance with DIN 24255 (see also Fig. 4)

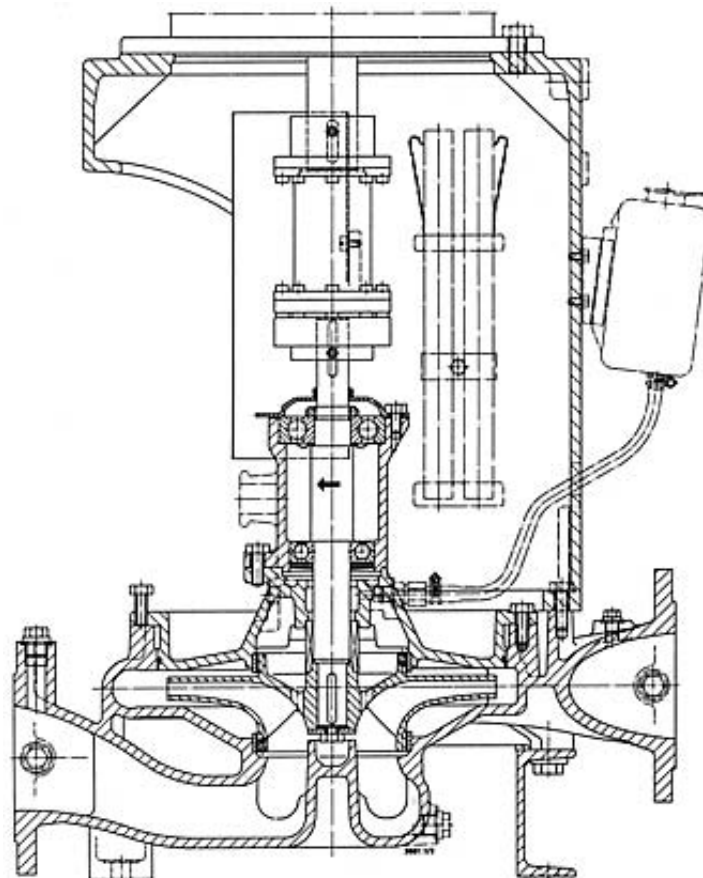


Fig. 2: Radially split casing vertical volute casing pump with radial impeller

The pump discharge branch can be attached tangentially to the volute casing, or arranged radially in the plane of the shaft, by providing a swanneck; the pump discharge branch can be arranged at the top, bottom or sides of the volute casing, providing it does not interfere with the pump feet on the casing (if provided). The pump suction branch is frequently arranged axially (end suction) in the case of v.c.p.'s with an overhung impeller, and either radially or tangentially in the case of inline pumps or v.c.p.'s with bearings at either side of the impeller.

In lieu of radial or mixed flow impeller, special impellers, e.g. single or multi-passage impellers (impeller) can be fitted if required. Occasionally a diffuser is fitted between the impeller and volute of large v.c.p.'s to improve the pump efficiency and balance the radial thrust. A double volute, i.e. two axially symmetrical volutes that are staggered by 180° but normally have a common outlet branch (Fig. 6), also serves in balancing the radial thrust.

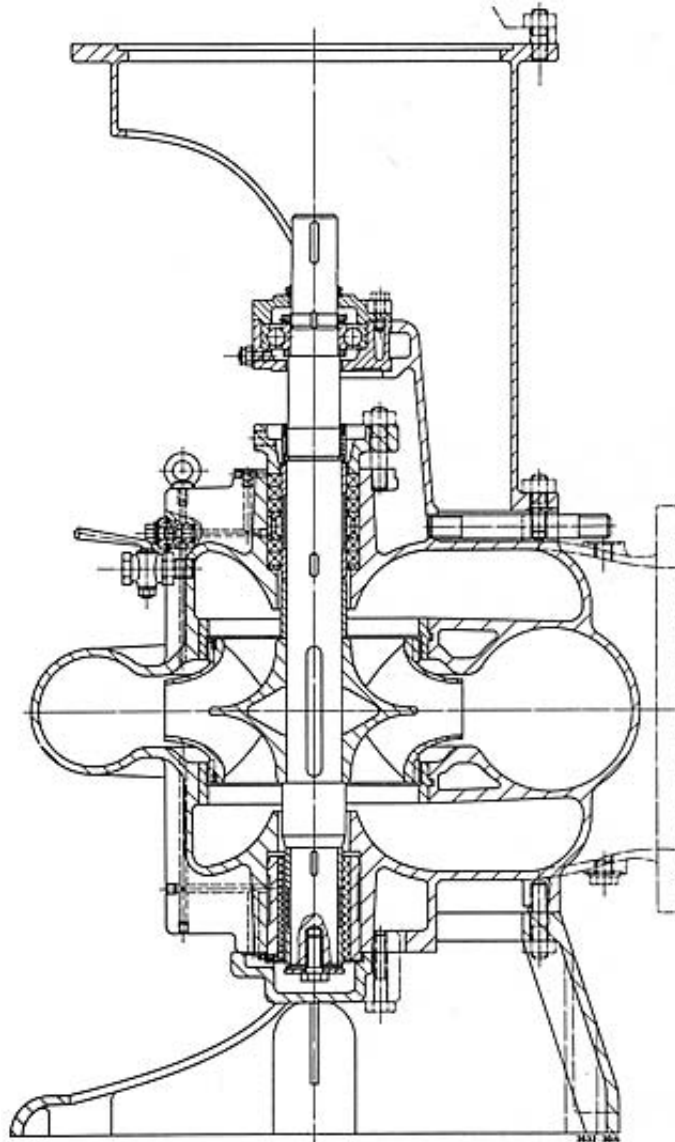


Fig. 3: Axially split casing double suction vertical volute casing pump

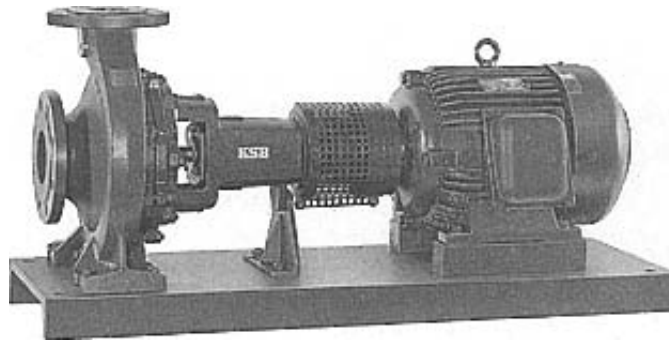


Fig. 4: Volute casing pump with bearing bracket (see also Fig. 1)

Depending on the fluid pumped or the required degree of freedom from maintenance, all types of shaft seals can be fitted. It is possible to heat or cool the v.c.p., e.g. when used for chemical processes (Fig. 12 under pump casing).

V.c.p.'s are often designated in accordance with other criteria, e.g. according to their drive (canned motor pump, close-coupled pumping set) or according to their application (water supply pump, marine pump, chemical pump, fire-fighting pump), or according to the medium pumped (sewage pump, pulp pump, heat transfer pump), or again according to the volute casing material (plastic v.c.p. concrete v.c.p.). The concept v.c.p. can therefore only express one of several aspects.

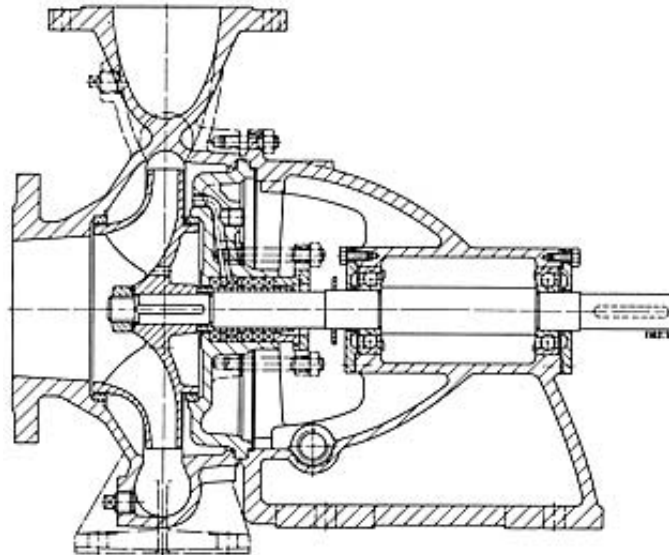


Fig. 5: Volute casing pump with bearing pedestal

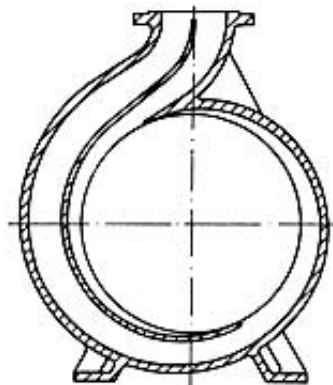


Fig. 6: Double volute casing

Vortex Flow

Drallströmung

Écoulement tourbillonnaire

The v.f. is a rotation-symmetrical flow (fluid dynamics) with peripheral or tangential components (circumferential velocity). In many instances, e.g. v.f. in pipes, there is an axial component of the flow velocity present in addition to the peripheral component; thus the flowing particles move along helical paths. For this reason, a v.f. in a pipe has a higher pressure loss than a swirl-free flow of the same flow velocity, because the liquid particles, in the case of v.f., have to travel along a longer path along the pipe wall for a given length of piping.

The peripheral component v_u at radius r in a v.f. usually follows the law

$$v_u \cdot r = \text{constant},$$

applicable to potential flow, and the flow is then known as a potential v.f. The above relationship is however not valid for very small radii because in this case the peripheral component of the v.f. would have to increase to infinity as the radius decreases. In fact, a vortex core is present at the centre of a v.f. below a given limit diameter, and the peripheral component v_u of flow velocity in this core follows the law

$$\frac{v_u}{r} = \text{constant},$$

whereas the axial component (in the case of v.f. in a pipe) is much lower in the core than in the region of the outer potential vortex (dead water core). The static pressure in a v.f. increases with the radius.

A v.f. is always present at the outlet of a rotating impeller, and the peripheral components of this flow can be partially converted into static pressure by suitable components arranged downstream (e.g. diffuser). In a v.f. encompassed within solid walls in a pipe or annular chamber, the intensity of the swirl decreases slowly along its travel path (as the distance increases) due to wall friction. Undesirable v.f. in pipes, e.g. upstream of a measuring point, or in the approach flow to a pump (inlet conditions) can be suppressed by fitting cruciform flat plate flow straighteners e.g. in the form of a crosspiece made of plate.

Vortex Pump

Wirbelpumpe

Pompe tourbillon trapézoïdale

V.p. see torque-flow pump (but not peripheral pump, which occasionally is also erroneously called v.p.).

W

Warranty Range

Gewährleistungsfeld

Zone de garantie

W.r. is the range created around the guarantee (or warranty) point by the uncertainties of measurement and by the manufacturing tolerances.

The DIN 1944 acceptance test code, October 1968 edition, no longer accepts this concept, in contrast to the former editions of this Standard. According to the October 1968 edition, one must proceed as follows: if the uncertainties of measurement are taken into account, an ellipse will be formed around to each of the measured points, so that the measured throttling curve (characteristic curve) in reality becomes a band or range situated between an upper and a lower limit curve. The manufacturing tolerance (warranty tolerance) for capacity or head forms a segment of straight line through the guarantee point (see relevant diagram under uncertainty of measurement).

In the ISO acceptance test code, the w.r. is described as an ellipse by the equation

$$\left(\frac{H_L \cdot x_H}{\Delta H}\right)^2 + \left(\frac{Q_L \cdot x_Q}{\Delta Q}\right)^2 \geq 1$$

(see "overall tolerance" for explanation of symbols), without any reference to the concept "warranty range" however.

Warranty Tolerance

Gewährleistungstoleranz

Tolérance de garantie

W.t. is the permissible deviation from the guaranteed value, or "warranted" value (warranty range). In the acceptance test code DIN 1944, October 1968 edition, the w.t. is equivalent to the manufacturing tolerance, whereas in the ISO acceptance test code, the w.t. is equivalent to the overall tolerance (designated in the code as "acceptance tolerance").

Water Consumption

Wasserverbrauch

Consommation d'eau

The following Table gives a selection of guideline values (in litres):

Table: Water Consumption (in litres)

Garden sprinkling per m ² and day	1.5 to 3.0
Dairies per litre milk	3 to 6
Market halls per m ² and day	5
Schools per pupil and per school day	5
Gas works per m ³ of gas produced	5 to 8
Fire hydrants per s	5 to 10
Vegetable gardens per m ² and day	5 to 10
Calves, sheep per animal and day	8 to 10

Washing of laundry per inhabitant and day	10 to 15
W.C. flushing, once	10 to 15
Drinking, cooking, cleaning per inhabitant and day	20 to 30
Cooling water for Diesel engines, per kWh	25 to 50
Domestic dishwasher, depending on washing programme	30 to 60
Stores and commercial establishments without restaurant facilities per employee and sales day	32 to 148
Large cattle, per animal and day	40 to 60
Office administrative buildings, per employee and calendar day	40 to 60
Laundries per kg washing	40 to 80
Shower in private dwelling	50 to 75
Village per day and head*	60 to 80
Town up to 50 000 inhabitants, per day and head*	80 to 120
Town with more than 50 000 inhabitants, per day and head*	120 to 200
Manufacture of 1 kg of sugar	100
Inns, per guest and day with full board	100 to 130
Barracks, per head and day	100 to 150
Hospitals and nursing homes per head and day	100 to 650
Private household, per head and day	120
Manufacture of 1 m ³ compressed concrete	125 to 150
Domestic washing machine depending on wash programme	130 to 200
Stores and commercial establishments with restaurant facilities, per employee and sales day	130 to 620
Bath in private dwelling	150 to 200
Abattoirs, per small animal	150 to 300
Washing a private car	200 to 300
Washing a van or lorry	200 to 400
Hotels (from simple hotels and boarding houses to first class hotels) per hotel guest and day	250 to 830
Public baths, per bath, including cleaning	300
Abattoirs, per large animal	300 to 400
Manufacture of 1 kg newsprint	400 to 600
Public swimming baths, per m ³ contents and day	500
Brewery, per hl beer (without cooling)	500 to 750
Public baths, per Turkish bath with cleaning	up to 700
Laying of 1000 bricks including mortar manufacture	750
Processing of 1 kg wool to cloth	1000
Manufacture of 1 kg fine paper	1500 to 3000
Brewery, per hl beer (with cooling)	1700 to 2250
Cleaning out of a railway goods wagon (cattle wagon)	2000 to 2500
Cooling water for fossil-fuelled power stations with fresh water cooling per MW h	100 000 to 120000
Cooling water for nuclear power stations with fresh water cooling per MWh	140000 to 160 000

* maximum daily water consumption per head is approximately double the above quoted values, and maximum hourly consumption per head is approximately 15 % of the above values

Water Hammer

Wasserchlag
Coup de bélier d'onde

see [Surge Pressure](#)

Water Hardness

Wasserhärte
Dureté d'une eau

The hardness of water is defined as the content of calcium (Ca^{2+}) and magnesium ions (Mg^{2+}) (DIN 38 409, Part 6, 1926 edition).

In certain individual cases, such as, for example, seawater, it is better to include barium (Ba^{2+}) and strontium (Sr^{2+}) as "hardeners". This needs to be distinguished.

Hydrogen carbonate hardness (known until now as carbonate hardness, temporary hardness) is the amount of "hardness-ions" which are equivalent to the amount of hydrogen carbonate and carbonate ions in water (i.e. the amount of $\text{Ca}(\text{HCO}_3)_2$).

The hardness of water is given as the concentration of hardness-ions in the water, mmol/l.

The correlation between the modern legal unit mmol/l and the formerly nationally and internationally used degrees of hardness can be seen in the [Table](#).

The hardness of water is important for forming the protective carbonate-calcium-rust layer ([protected layer](#)) on low alloy iron base alloys and for the formation of calcium carbonate deposits in boilers.

In tap and cooling water, the formation of deposits can be avoided by reducing the hardness, i.e. taking the hardeners out of the water chemically, or adding polyphosphates and/or ions which form complexes.

Water Jet

Wasserstrahl
Jet d'eau

In fire-fighting, it is customary to test the ability to function of a [fire-fighting pump](#) according to the height or horizontal range of the w.j. issuing from the nozzle of the jet pipe. In order to achieve an adequate fire-extinguishing effect, the [pressure](#) (i.e. the gauge pressure above atmospheric) at the jet pipe should amount to 4 bar at least. In the Table "Height of jet h and horizontal range w of w.j.'s issuing from jet pipes", the symbols used have the following meanings:

- p [pressure](#) in jet pipe,
- h height of w.j. (to the highest drop),
- w horizontal range of w.j. (to the farthest drop),
- Q [capacity](#) of [fire-fighting pump](#),
- D jet diameter at nozzle (mouthpiece).

The data in the [Table](#) are guideline values (assuming no wind).

Table: Conversion of degrees and units for water hardness (according to DIN 38 409, Part 6)

Units of water hardness	mval/l	German °d	French °e	English °e	American ppm	Legal unit mmol/l
	28 mg CaO or 50 mg CaCO ₃ per 1 000 ml water	10 mg CaO per 1 000 ml water	10 mg CaCO ₃ per 1 000 ml water	1 grain CaCO ₃ per gallon = 14.3 mg CaCO ₃ per 1 000 ml water	1 part CaCO ₃ per million = 1 mg CaCO ₃ per 1000 ml water	100 mg CaCO ₃ per 1 000 ml water
1 mval/l	1	2.8	5.00	3.51	50	0.5
1 °d	0.357	1	1.786	1.250	17.86	0.1786
1 °f	0.20	0.5599	1	0.700	10.00	0.10
1 °e	0.285	0.7999	1.429	1	14.29	0.1429
1 ppm	0.02	0.056	0.10	0.070	1	0.01
1 mmol/l	2	5.6	10.00	7.0	100	1

Example: 1 °d equivalent to 0.1786 mmol/l

Table: Height of jet h and horizontal range w of water jets issuing out of jet pipes

p in bar	h in m w in m Q in m ³ /h	D in mm																
		8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40
3.0	h	15.5	16	17	18	19	20	21	22	23	24	24	25	25	25	26	26	26
	w	21.0	22	23	24	26	27	28	29	30	31	32	33	34	35	35	36	36
	Q	4.3	6.8	9.7	13.3	17.4	22.0	27.2	33	39.3	46.2	53.6	61.5	70.0	79.0	88.6	98.7	109.4
3.5	h	17.0	17	18	19	20	21	22	23	24	25	26	27	27	28	28	29	29
	w	23.0	23	24	25	27	28	30	31	32	33	35	36	37	38	38	39	40
	Q	4.6	7.3	10.5	14.3	18.8	23.8	29.4	35.6	42.5	49.9	57.9	66.4	75.6	85.4	95.6	106.6	118.1
4.0	h	18.5	18	19	20	21	22	23	25	26	27	29	30	30	31	31	32	33
	w	24.0	24	25	26	28	29	32	33	34	36	39	40	41	41	42	43	44
	Q	4.9	7.8	11.3	15.4	20.1	25.4	31.4	38.2	45.4	53.3	61.9	71	80.8	91.3	102.4	114	126.3
4.5	h	20.0	18	19	20	21	23	24	26	27	29	31	32	33	34	35	36	37
	w	25.0	25	26	27	29	30	33	35	36	38	41	43	44	45	46	47	48
	Q	5.2	8.3	11.9	16.3	21.3	27	33.4	40.4	48.2	56.6	65.4	75.4	85.7	96.8	108.5	120.9	134.0
5.0	h	20.5	19	20	21	22	24	26	28	29	31	33	34	35	36	37	38	39
	w	25.6	26	27	28	30	32	34	37	38	41	44	46	47	48	49	51	53
	Q	5.5	8.7	12.6	17.2	22.4	28.4	35.2	42.7	50.8	59.6	69.2	79.4	90.4	102.1	114.5	127.4	141.2
5.5	h	21.0	20	21	22	23	25	27	29	30	32	34	36	37	38	39	40	41
	w	26.5	27	28	29	31	33	35	38	40	43	46	49	50	51	53	54	56
	Q	5.8	9.1	13.2	18	23.5	29.8	36.9	44.7	53.3	62.6	72.5	83.3	94.7	107	120	133.6	148.1
6.0	h	21.7	21	22	23	24	26	28	30	31	34	37	38	39	40	42	43	44
	w	27.5	28	29	30	32	34	37	40	42	45	49	52	53	55	57	59	61
	Q	6.1	9.5	13.8	18.8	24.6	31.1	38.5	46.7	55.6	65.3	75.8	87.0	99.0	111.8	125.3	139.6	154.7
6.5	h		21	22	23	25	27	29	31	32	35	38	40	41	42	44	45	46
	w		29	30	31	33	35	38	41	43	47	50	53	55	57	59	61	63
	Q	6.3	10.0	14.3	19.6	25.6	32.5	40.1	48.6	57.9	68.0	78.9	90.5	103.0	116.3	130.4	145.3	161.0
7.0	h		22	23	24	26	28	30	32	33	37	40	42	43	44	45	47	48
	w		30	32	33	35	37	40	43	45	49	52	55	58	61	63	65	67
	Q	6.5	10.3	14.9	20.3	26.6	33.7	41.6	50.5	60.1	70.6	82.0	94.0	106.9	120.7	135.4	150.8	167.0

7.5	h		22	23	24	27	29	31	33	34	38	41	43	44	45	46	47	48
	w		30	32	33	36	38	41	42	46	50	53	56	60	62	64	66	68
8.0	Q	6.8	10.7	15.4	21	27.5	34.9	43.1	52.2	62.2	73.1	84.7	97.3	110.7	125.0	140.1	156.1	170.0
	h		23	24	25	28	30	32	34	36	39	43	45	46	47	49	50	51
8.5	w		31	33	34	38	40	43	46	48	52	54	58	62	64	66	68	70
	Q	7.0	11.0	15.9	21.7	28.4	36	44.5	53.9	64.3	75.5	87.6	100.5	114.3	129.1	144.7	161.2	178.6
9.0	h			24	25	28	30	31	35	37	40	44	46	47	48	49	50	51
	w			33	34	39	41	44	47	49	51	55	59	63	65	67	69	71
9.5	Q	7.2	11.3	16.4	22.4	29.3	37.1	45.8	55.6	66.2	77.8	90.2	103.6	117.8	133.0	149.0	166.2	184.1
	h			25	26	29	31	34	36	38	41	45	47	48	49	50	51	52
10.0	w			34	35	40	42	45	48	51	54	56	60	65	67	69	71	72
	Q	7.4	11.7	16.9	23	30.1	38.2	47.2	57.2	68.2	80.1	92.9	106.6	121.3	136.9	153.5	171.0	189.5
10.5	h				27	29	31	34	37	39	41	45	47	49	50	51	52	52
	w				35	40	42	46	48	51	54	57	60	66	68	70	72	73
11.0	Q	7.6	12	17.3	23.6	31	39.2	48.5	58.8	70.0	82.3	95.4	109.5	124.6	140.6	157.7	175.7	194.6
	h				28	30	32	35	38	40	42	46	48	50	51	51	52	53
11.5	w				36	41	43	47	49	52	55	58	61	67	69	71	73	74
	Q	7.8	12.3	17.8	24.2	31.7	40.2	49.7	60.3	71.8	84.5	97.9	112.3	127.8	144.3	161.8	180.2	199.7
12.0	h					31	33	36	39	41	43	47	49	51	52	52	53	54
	w					42	44	48	50	53	56	59	62	68	70	72	74	76
12.5	Q	8.2	13	18.7	25.4	33.3	42.4	52.2	63.2	75.4	88.5	102.6	117.8	134.0	151.3	169.7	189.1	209.5
	h						34	37	40	42	44	48	50	52	53	53	54	55
13.0	w						45	49	51	54	57	60	63	69	72	74	76	78
	Q	8.6	13.5	19.5	26.6	34.9	44.1	54.5	66.1	78.7	92.5	107.2	121.3	140.0	158.1	177.2	197.5	218.8

Water Jet Pump

Wasserstrahlpumpe
Hydro-éjecteur

see Deep Well Suction Device

Water Requirement

Wasserbedarf
Besoin en eaux

see Water Consumption

Water Ring Pump

Wasserringpumpe
Pompe à anneau d'eau

The w.r.p. (Fig. 1), and more generally the liquid ring pump operates according to the positive displacement principle (positive displacement pump); therefore w.r.p.'s have a very good selfpriming capability (self-priming pump, siphoning installation). A star-shaped impeller (star impeller) rotates eccentrically in a casing partially filled

with liquid, forming a rotating liquid ring with a free surface. The blades of the star impeller dip into this liquid ring (often a water ring) at varying depths during the course of a revolution. This causes the formation of gas-filled cavities between adjoining blades and the surface of the liquid, which alternately increase and decrease in volume. In this operating condition, the w.r.p. is capable of pumping (i.e. discharging) gas.

When the gas feed ceases (e.g. when the suction pipe is completely vented), the w.r.p. starts pumping liquid, usually with only a moderate pump efficiency. For this reason, self-priming aids of this type, fitted e.g. to self-priming marine pumps are switched off by hand or automatically when the venting process is completed, i.e. they are isolated from the suction pipe, which is now filled with water (pumping plant), and ventilated, so that the star impellers continue rotating whilst requiring a minimum of shaft power. The effectiveness of a w.r.p. is very much dependent on an efficient sealing action between star impeller and casing (pump casing). Thus very narrow clearance gaps at the two flat end faces of the casing which accommodate the suction and discharge apertures are absolutely essential.

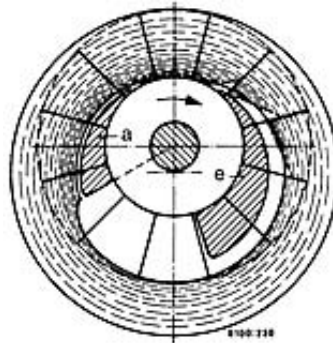


Fig. 1: Water ring pump, position of the water surfaces in the individual blade cells
a outlet; e inlet

Liquid ring pumps are not only used as selfpriming devices for centrifugal pumps, but also as vacuum pumps and as compressors in the chemical industry; they can be used to great advantage where gases have to be compressed in moist condition or without any temperature increase, e.g. in the chlorelectrolysis process, where chlorine gas is compressed isothermally and where it can simultaneously be cleaned by the concentrated sulphuric acid used as a liquid ring. Figs. 2 and 3 illustrate such a chlorine gas compressor with a casing shaped eccentrically in mirror image at top and bottom, so that the star impeller with forward curvature blades aspirates and compresses twice per revolution; the compressed gas issuing from the outlet branch (pump discharge branch) entrains a certain amount of liquid, which is separated from the gas in a separator and cooled in a heat exchanger before being returned to the compressor to maintain the liquid ring.

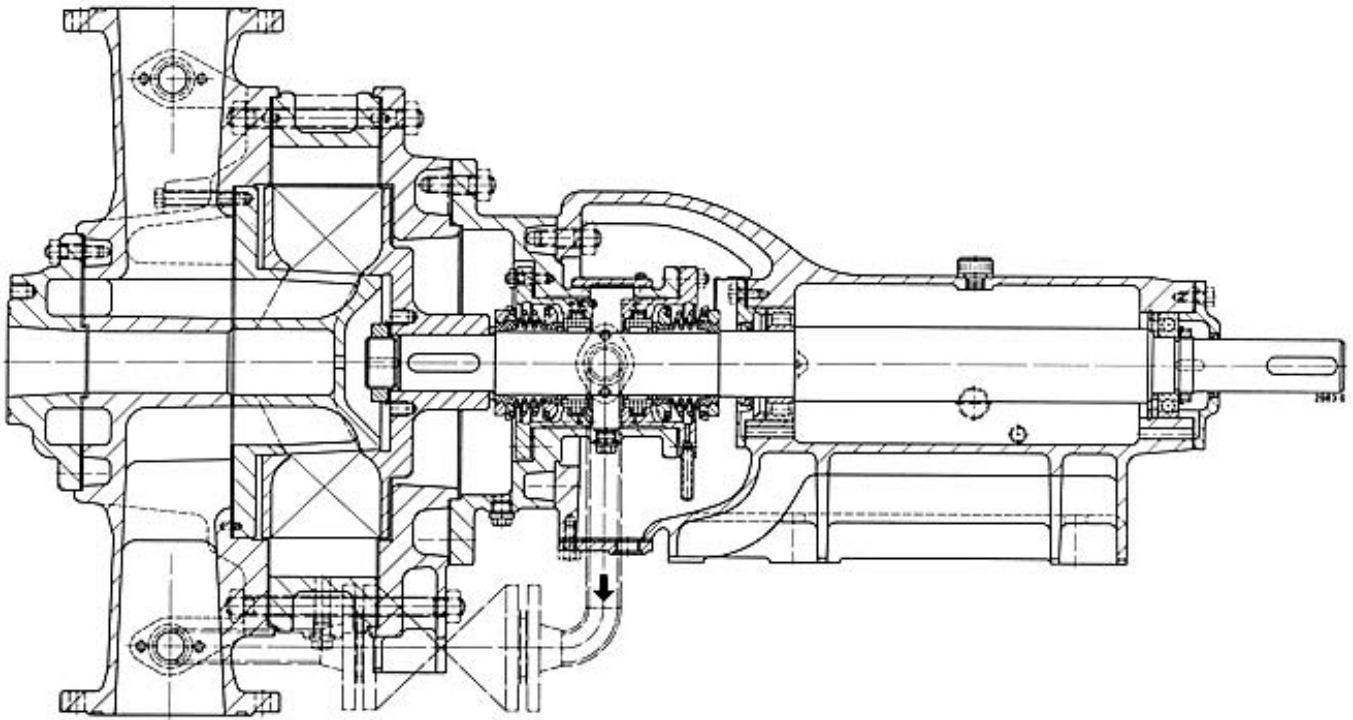


Fig. 2: Liquid ring compressor (chlorine gas compressor)

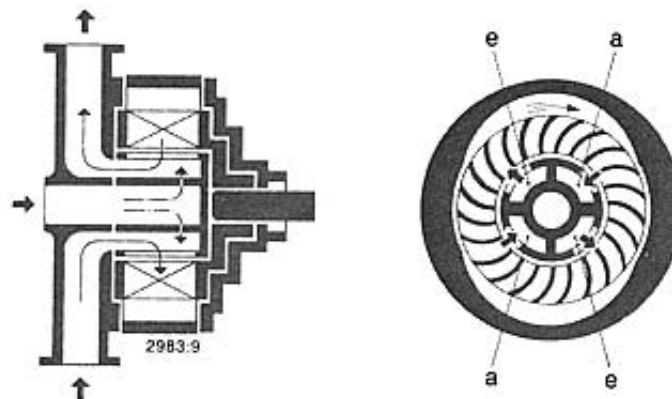


Fig. 3: Operating principle diagram relating to Fig. 2 a outlet; e inlet

Water Supply Pump

Wasserversorgungspumpe

Pompe pour services d'eau

W.s.p.'s are all centrifugal pumps which pump drinking or industrial water directly into a supply network, or which transport the water to a supply area through long-distance pipelines. As a general rule, w.s.p.'s have to satisfy the following requirements:

- high efficiency (continuous operation),
- reliable bearing arrangement which does not impair the quality of the water,
- low noise level.

Depending on the capacity, head and installation conditions, various pump types are used; they are listed below, together with their appropriate operating ranges:

1. Underwater motor pumps, installed in wells and pumping directly into the network if no water treatment is required:

Q up to 3000 m³/h H up to 1400 m.

2. Vertical wet installation borehole shaft driven pumps (vertical pump, wet installation):

Q up to 2800 m³/h, H up to 160 m.

3. Vertical, wet installation, multistage tubular casing pumps (mixed flow pump, multistage pump):

Q = 800 to 30 000 m³/h, H up to 140 m.

4. Single stage, single suction volute casing pumps:

Q = up to 36 000 m³/h, H up to 140 m.

5. Single stage, single suction volute casing pumps with diffuser:

Q = 500 to 10 000 m³/h, H up to 210 m.

6. Multistage pumps:

Q = 20 to 500 m³/h, H up to 500 m,

Q = 500 to 3500 m³/h, H up to 350 m.

7. Double suction, single stage volute casing pumps (multisuction pump):

Q = 100 to 30000 m³/h, H up to 500 m.

8. Double suction, single stage volute casing pumps with diffuser:

Q = 800 to 20 000 m³/h, H up to 700 m.

Waterworks Pump

Wasserwerkspumpe

Pompe d'adduction d'eau

The w.p. is a centrifugal pump for the public supply of drinking water to cities and communities. W.p.'s are characterized by low maintenance costs, high operational safety and reliability lasting for decades of service, and a design which ensures that the water pumped is hygienically pure. W.p.'s belong to the category of water supply pumps.

Water Yield

Wasseranfall

Arrivée d'eau intempestive

W.y. is a concept applying to land drainage, i.e. dewatering through an underground pipe system. The my. gives the volume flow (capacity) per unit area of land to be drained, e.g. for polders (dyked land), the w.y. amounts to 0.7 to 0.8 litres per second and hectare (10 000 m²) approx.; max. values are 2 l/s ha approx.

Wear

Verschleiß
Usure

W., according to DIN 50 320, is a continuing loss of material from the surface of a solid body, due to mechanical causes, i.e., contact and movement against another solid, liquid or gaseous body (abrasion).

The most important forms of w. that appear in centrifugal pumps are listed in the Table (as taken from DIN 50 320).

Weight

Gewicht
Poids

In accordance with the regulation relating to the "Law on Units used in the Field of Measurement" (which has had force of law in the German Federal Republic since 1970), the designation "weight" shall be understood to mean a mass. Thus the SI unit of w. is 1 kg.

The ambiguous term "specific gravitaty" (for ρ and previously for $\rho \cdot g$) should no longer be used in centrifugal pump technology (density of pumped media, specific gravity).

Whereas the w. (= mass m) is independent of location, the "gravitational force" $G = m \cdot g$ exerted by the mass is dependent on location, because of the gravitational constant g .

Wet Installation

Naßaufstellung
Installation non à l'air libre

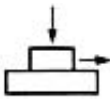
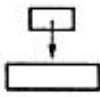
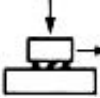


The concept of w.i. is mainly used in conjunction with large vertical pumps. On a w.i. centrifugal pump, the pump casing can or must be partly or wholly flooded (submersible pump). The opposite is dry installation.

Wet Rotor Motor

Naßläufermotor
Moteur étanche

W.r.m., usually an asynchronous squirrel-cage rotor motor (asynchronous motor) with a rotor and bearings running immersed in liquid.

Table: Forms of wear in centrifugal pumps (according to DIN 50 320)

System structure	Tribological loading		Type of wear	Examples
		symbol		
Solid-solid (for solid-state friction, boundary lubrication and mixed lubrication)	sliding		sliding wear	packing on shaft sleeve
	impact, shock		impact wear, shock wear	solids on front side of blade
Solid-solid and particles	sliding		particle friction wear	particles between packing and sleeve
Solid-liquid with particles	flowing		hydroabrasive wear (erosion)	particles on blade surface
Solid-liquid	flowing, vibration		material cavitation, cavitation erosion	bubble trail on guide vane

There is a distinction made between the type with dry stator winding (canned motor) and the type with wet stator winding (wet motor).

The canned motor is a liquid-filled asynchronous squirrel-cage rotor motor, with a rotor and bearings immersed and operating in the fluid pumped by the pump driven by said motor (canned motor pump). The stator winding is however sealed off against the fluid in the rotor compartment by a high-grade steel tube or can. This high grade steel can is very thin-walled and has a low electrical conductivity, and is situated in the "air gap" of the motor.

The output range of canned motors ranges between a few Watts and 2000 kW approx.

The application fields range from central heating circulating pumps via process pumps for the chemical and process industry to reactor circulating pumps (reactor pump), and comprise all applications where hot, corrosive, toxic or radioactively contaminated fluids have to be pumped, and where emphasis is consequently placed on hermetically sealed pumping sets without a stuffing box or mechanical seal (shaft seals), i.e. grandness (Fig. 1).

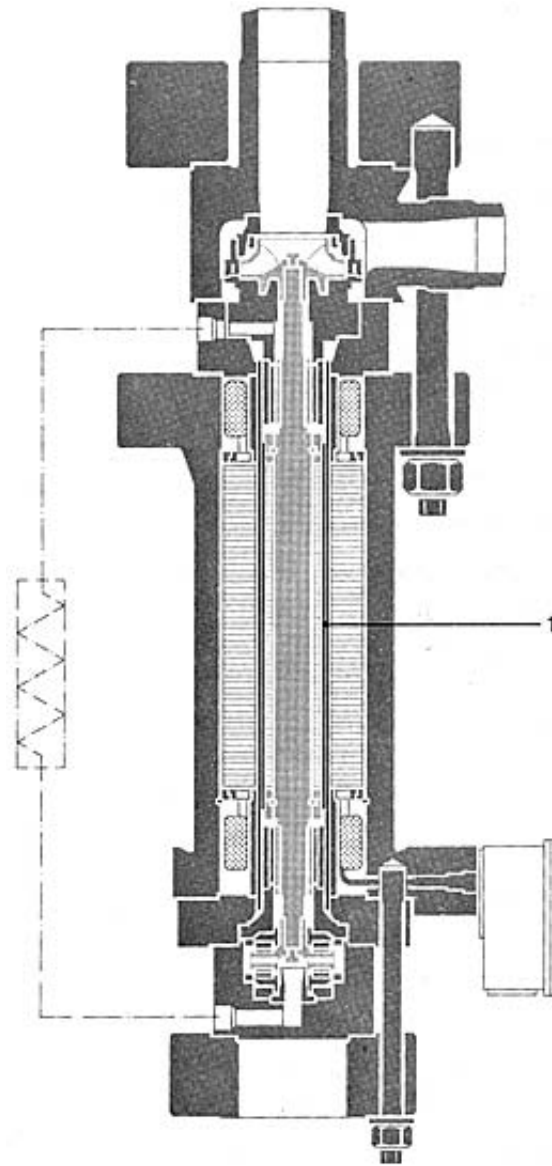


Fig. 1: Diagram of a glandless circulating pump with canned motor for high-pressure process engineering 1 can

The wet motor is completely filled with fluid (preferably water). Not only the rotor and bearings, but also the stator lamination pack and winding, including the supply connections, are immersed in the fluid. A prerequisite is a waterand pressure-tight insulation of all live components.

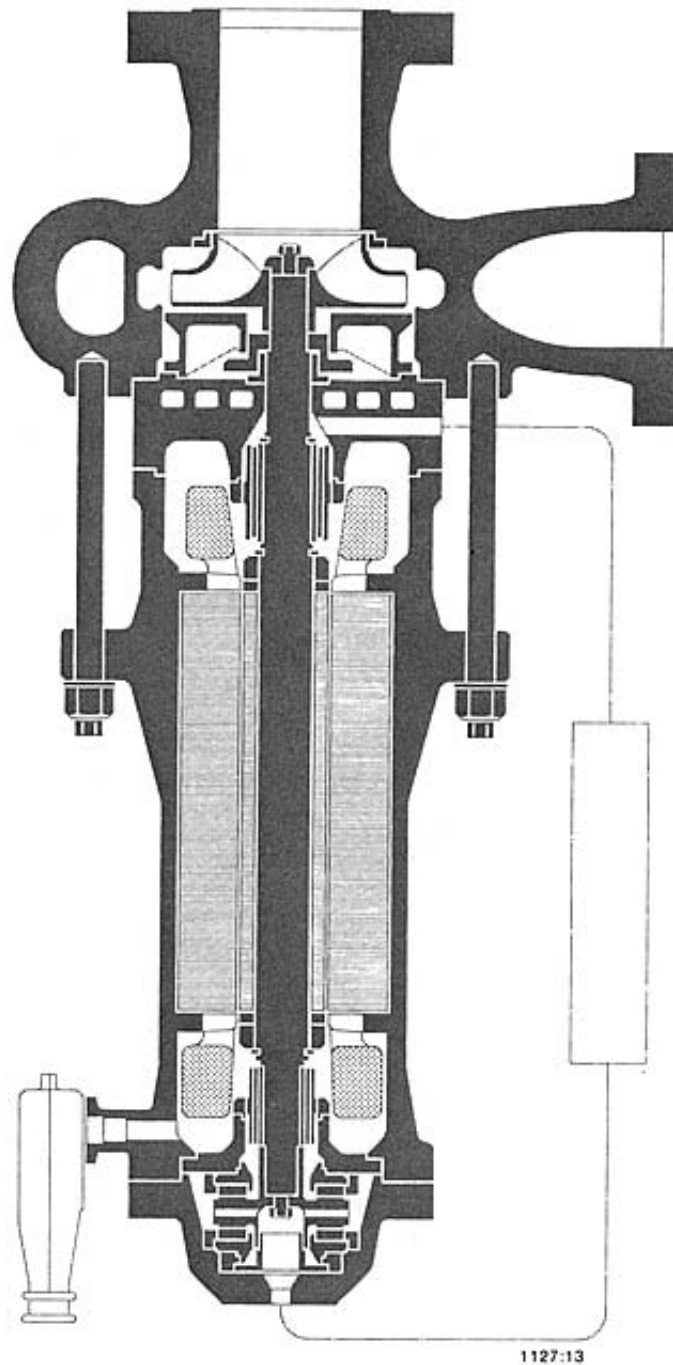


Fig. 2: Diagram of a glandless circulating pump with wet motor and water-cooled heat barrier for steam power stations (cf. Fig. 3 under circulating pump)

Currently outputs range between 1 kW approx. and 2500 kW approx. (glandless circulating pump) or 3500 kW (submersible motor) for operating voltages of up to 6.6 kV in standard cases and up to 10 kV in special cases. Wet motors are used as drives for submersible motor pumps in wells, and as driver motors for glandless circulating pumps in conventional power stations (Fig. 2).

Wood Pulp

Holzstoff

Pâte de bois

see Fibrous Material

Work

Arbeit
Travail

W., in the sense of mechanical w., is a physical entity, viz. the scalar product of force and distance (energy). Mechanical w. (symbol A) is done when a body is displaced over a distance s by a force F acting in the direction of the displacement:

$$A = F \cdot s.$$

When a body is rotated, the mechanical w. is:

$$A = T \cdot \Phi$$

where

T torque in plane of action,
 Φ angle of rotation in radians (rad rotational speed).

W., energy and quantity of heat are entities of the same nature in the general technical terminology, defined by the same SI unit, viz. 1 Joule (J, unit).

Y

Y-Branch

Abzweigstück

Pièce de dérivation

see Pressure Loss

APPENDIX

1. Unit symbols

Symbol	Name (dimension)	Remark
a	are (area of a parcel of land)	2, 4
A	Ampere (electric current)	1, 3, 4
at	technical atmosphere (pressure)	5
ata	absolute atmosphere (pressure)	5
atm	physical atmosphere (pressure)	5
atü	atmosphere's over-pressure (pressure)	5
A/kg	Ampere divided by kilogramme (ion dose rate)	2
A/m	Ampere divided by metre (magnetic field intensity)	2
bar	bar (pressure)	2, 4
C	Coulomb (quantity of electricity, electrical charge)	2, 4
cal	calorie (quantity of heat)	5
cd	candela (light intensity)	1, 3, 4
cd/m ²	candela divided by square metre (luminous density, radiant intensity per unit area)	2
C/kg	Coulomb divided by kilogramme (ion dose)	2
C/m ²	Coulomb divided by square metre (electrical flux density)	2
d	day (interval of time)	2
dpt	diopetre (optical refraction)	2
eV	electron volt (atomic physics energy)	1, 3
F	farad (electrical capacitance)	2, 4
g	gramme (mass)	2, 4
h	hour (interval of time)	2
H	Henry (inductance)	2, 4
HP	horse power (power)	5
Hz	Hertz (frequency)	2, 4
J	Joule (energy, work, quantity of heat)	2, 4
J/kg	Joule divided by kilogramme (energy per units mass)	2
K	Kelvin (temperature)	1, 3
kg	kilogramme (mass)	1, 3
kg/m	kilogramme divided by metre (mass per unit length)	2
kg/m ²	kilogramme divided by square metre (mass per unit area)	2
kg/m ³	kilogramme divided by cubic metre (density)	2
kg/s	kilogramme divided by second (mass flow)	2
kg/mol	kilogramme divided by mol (mass related to quantity of substance, molar mass)	2
Kt	metric carat (weight of precious stones)	2
l	litre (volume)	2, 4
lm	lumen (luminous flux)	2, 4
lx	lux (illumination, luminous intensity)	2, 4
m	metre (length)	1, 3, 4

m ²	square metre (area)	2
m ² /s	square metre divided by second (kinematic viscosity)	2
m ³	cubic metre (volume)	2
m ³ /s	cubic metre divided by second (volume flow rate)	2
m/s	metre divided by second (velocity)	2
m/s ²	metre divided by second square (acceleration)	2
min	minute (interval of time)	2
mm Hg	millimetre column of mercury (pressure)	5
mol	mol (quantity of substance in atomic physics)	1, 3
mol/m ³	mol divided by cubic metre (quantity of substance concentration, polarity, molar concentration)	2
m w.c.	metre column of water (pressure)	5
N	Newton (force)	2, 4
p	pond (force)	5
P	poise (dynamic viscosity)	5
Pa	Pascal (pressure, mechanical stress)	2, 4
Pa s	Pascal second (dynamic viscosity)	2
rad	radian (plane angle)	2
rad/s	radian divided by second (angular velocity)	2
rad/s ²	radian divided by second square (angular acceleration)	2
s	second (interval of time)	1, 3, 4
S	Siemens (electrical conductance)	2, 4
s ⁻¹	reciprocal second (activity of a radio-active substance)	2
sr	steradian (dihedral angle)	2
St	Stokes (kinematic viscosity)	5
t	ton (mass)	2, 4
T	Tesla (magnetic flux density, induction)	2, 4
tex	tex (mass of textile fibres related to length)	2, 4
Torr	Torr (pressure)	5
u	atomic mass unit (particle mass in atomic physics)	1, 3
V	volt (electric potential difference, electromotive force)	2, 4
VA	volt-ampere (apparent electrical power)	2
var	var (wattless or reactive electrical power)	2
Vs	Volt-second (magnetic flux)	2
V/m	Volt divided by metre (electrical field intensity)	2
W	Watt (energy flux, power heat flow)	2, 4
Wb	Weber (magnetic flux)	2, 4
W/kg	Watt divided by kilogramme (energy rate per unit mass)	2
Ω	Ohm (electrical resistance)	2, 4
°	degree (plane angle)	2
°C	degree centigrade (temperature)	2
°E	degree Engler (kinematic viscosity)	5
°F	degree Fahrenheit (temperature)	5
'	minute (plane angle)	2

"	sekond (plane angle)	2
"	inch (length)	5
⊥	right angle (plane angle)	2

The numerals under the heading "Remark" signify:

1 Legal unit in accordance with "Law relating to Units used for Measurements, dated 2nd July 1969", Federal Republic of Germany.

The field of application extends to commercial and official traffic from and to the Member Countries of the *European Community*, and to imports and exports to and from said countries.

2 Legally derived unit in accordance with "Implementation Decree of 26th June 1970" relating to 1. Legally derived units in this context also encompass all other dimensional products, quotients and powers formed from legal units; e.g. for the volume flow m^3/s , m^3/h , l/min

3 Basic units of international unit system (SI), see unit.

4 Decimal multiples and decimal parts of this unit are also legal units; they must be noted by means of the following prefix symbols (see opposite Table) immediately before the unit symbol.

5 No longer authorized.

	Power of ten	Prefix	Prefix symbol
Decimal multiple	10^{12}	tera	T
	10^9	giga	G
	10^6	mega	M
	10^3	kilo	k
	10^2	hecto	h
	10^1	deka	da
Decimal part	10^{-1}	deci	d
	10^{-2}	centi	c
	10^{-3}	milli	m
	10^{-6}	micro	μ
	10^{-9}	nano	n
	10^{-12}	pico	p

2. Convention of British and USA units (unit)

				British	USA
Length	1 mil			25.4 μm	25.4 μm
	1 point			0.3528 mm	0.3528 mm
	1 line			0.635 mm	0.635 mm
	1 inch	(in)		25.4 mm	25.4 mm
	1 hand	(li)		10.16 cm	10.16 cm
	1 link	(ft)		20.1168 cm	20.1168 cm
	1 span	(yd)	= 12 in	22.86 cm	22.86 cm
	1 foot	(fath)	= 3 ft = 36 in	0.3048 m	0.3048 m
	1 yard	(rd)	= 2 yd	0.9144 m	0.9144 m
	1 fathom	(ch)	= 1760 yd	1.8288 m	1.8288 m
	1 rod	(fur)		5.0292 m	5.0292 m
	1 chain	(mi)		20.1168 m	20.1168 m
	1 furlong			201.168 m	201.168 m
	1 mile (statute mile)			1.6093 km	1.6093 km
	1 nautical mile			1.8532 km	1.8532 km

Area	1 circular mil		506.709 μm^2	506.709 μm^2
	1 circular inch		5.067 cm^2	5.067 cm^2
	1 square inch		6.4516 cm^2	6.4516 cm^2
	1 square link	(sq in)	404.687 cm^2	404.687 cm^2
	1 square foot	(sq li)	929.03 cm^2	929.03 cm^2
	1 square yard	(sq ft)	0.8361 m^2	0.8361 m^2
	1 square rod	(sq yd)	25.2929 m^2	25.2929 m^2
	1 square chain	(sq rd)	404.686 m^2	404.686 m^2
	1 rood	(sq ch)	1011.7 m^2	1011.7 m^2
	1 acre	(sq mi)	4046.86 m^2	4046.86 m^2
	1 square mile		2.59 km^2	2.59 km^2
Volume basic unit gallon for fluids basic unit bushel for dry goods	1 cubic inch		16.387 cm^3	16.387 cm^3
	1 board foot		2.3597 dm^3	2.3597 dm^3
	1 cubic foot		28.3268 dm^3	28.3268 dm^3
	1 cubic yard	(cu in)	0.7646 m^3	0.7646 m^3
	1 register ton	(fbm)	2.8327 m^3	2.8327 m^3
	1 British shipping ton = 42	(cu ft)	1.1897 m^3	-
	1 US shipping ton = 40	(cu yd)	-	1.1331 m^3
	1 minim	(RT) = 100 cu ft	59.1939 mm^3	61.6119 mm^3
	1 fluid scruple	cu ft	1.1839 cm^3	-
	1 fluid drachm	cu ft	3.5516 cm^3	-
	1 fluid dram	(min)	-	3.6967 cm^3
	1 fluid ounce	(fl. dr.)	28.4131 cm^3	29.5737 cm^3
	1 gill	(fl. dr.)	142.065 cm^3	118.2948 cm^3
	1 pint	(fl. oz.)	0.5683 dm^3	0.4732 dm^3
	1 quart	(git)	1.1365 dm^3	0.9464 dm^3
	1 pottle	(liq pt)	2.2730 dm^3	-
	1 gallon	(liq qt)	4.5460 dm^3	3.7854 dm^3
	1 peck	(gal)	9.0922 dm^3	-
	1 bushel	(für Rohöl)	36.3687 dm^3	-
	1 US oil-barrel	(dry pt)	-	0.159 m^3
	1 quarter	(dry qt)	0.291 m^3	-
	1 chaldron	(pk)	1.3093 m^3	-
	1 dry pint	(bu)	-	0.5506 dm^3
	1 dry quart	(bbl)	1.1012 dm^3	-
	1 peck		-	8.8098 dm^3
	1 bushel		36.3687 dm^3	335.2393 dm^3
	1 dry barrel		-	0.1156 m^3
Mass and Weight avoirdupois system (trade and commerce weights) troy system (for precious metals)	1 grain		64.7989 mg	64.7989 mg
	1 dram		1.7718 g	1.7718 g
	1 ounce	(gr)	28.3495 g	28.3495 g
	1 pound	(dr avdp)	0.4536 kg	0.4536 kg
	1 stone	(oz avdp)	6.3503 kg	-
	1 quarter	(lb)	12.7006 kg	-
	1 central	(sh cwt)	45.3592 kg	-
	1 short hundredweight	(cwt)	-	45.3592 kg
	1 hundredweight	(1 cwt)	50.8024 kg	-
	1 long hundredweight	(sh tn)	-	50.8024 kg
	1 short ton	(ltn)	-	907.1849 kg
	1 ton	(dwt)	1016.0470 kg	-
	1 long ton	(oz tr)	-	1016.0470 kg
	1 pennyweight	(lb t)	1.5552 g	1.5552 g
	1 troy ounce		31.1035 g	31.1035 g
	1 troy pound		-	0.3732 kg

Density	1 ounce (av) per cubic-foot	(oz/cu ft)	0.0010 kg/dm ³	0.0010 kg/dm ³
	1 pound per cubic-foot	(lb/cu ft)	0.0160 kg/dm ³	0.0160 kg/dm ³
	1 ounce (av) per cubic-inch	(oz/cu in)	1.7300 kg/dm ³	1.7300 kg/dm ³
	1 pound per cubic-inch	(lb/cu in)	27.6799 kg/dm ³	27.6799 kg/dm ³
	1 short ton per cubic-yard	(sh tn/cu yd)	-	1.1865 kg/dm ³
	1 long ton per cubic-yard	(ltn/cu yd)	-	1.3289 kg/dm ³
	1 pound per gallon	(lb/gal)	0.09978 kg/dm ³	0.1198 kg/dm ³
Velocity	1 foot per second	(ft/s)	0.3048 m/s	0.3048 m/s
	1 foot per minute	(ft/min)	0.00508 m/s	0.00508 m/s
	1 yard per second	(yd/s)	0.9144 m/s	0.9144 m/s
	1 yard per minute	(yd/min)	0.01524 m/s	0.01524 m/s
Capacity (rate of volume flow)	1 gallon per second		4.5460 l/s	3.7854 l/s
	1 gallon per minute	(gpm)	0.07577 l/s	0.06309 l/s
	1 cubic foot per second	(cused)	28.3268 l/s	28.3268 l/s
	1 cubic yard per second		0.7646 m ³ /s	0.7646 m ³ /s
Mass Row	1 ounce per second	(oz/s)	28.3495 g/s	28.3495 g/s
	1 ounce per minute	(oz/min)	0.4725 g/s	0.4725 g/s
	1 pound per second	(lb/s)	0.4536 kg/s	0.4536 kg/s
	1 pound per minute	(lb/min)	0.00756 kg/s	0.00756 kg/s
	1 short ton per hour	(sh tn/h)	-	0.2520 kg/s
	1 ton per hour	(ltn/h)	0.2822 kg/s	-
	1 long ton per hour		-	0.2822 kg/s
Force (weight force)	1 ounce (force)	(oz)	0.2780 N	0.2780 N
	1 pound (force)	(lb)	4.4483 N	4.4483 N
	1 short ton (force)	(sh tn)	8.8964 kN	8.8964 kN
	1 long ton (force)	(ltn)	9.9640 kN	9.9640 kN
Pressure	$1 \frac{\text{pound (force)}}{\text{square foot}}$	$\left(\frac{\text{lb (force)}}{\text{sq ft}} \right)$	47.88025 Pa	
	$1 \frac{\text{pound (force)}}{\text{square inch}}$	$\left(\frac{\text{lb (force)}}{\text{sq in}} \right) \cdot (\text{psi})$	68.9476 mbar	
	$1 \frac{\text{short ton (force)}}{\text{square inch}}$	$\left(\frac{\text{sh tn (force)}}{\text{sq in}} \right)$	137.8951 bar	
	1 inch H ₂ O	(in H ₂ O)	2.4909 mbar	2.4909 mbar
	1 foot H ₂ O	(ft H ₂ O)	29.8907 mbar	29.8907 mbar
Mechanical stress	1 inch Hg	(in Hg)	33.8663 mbar	33.8663 mbar
	$1 \frac{\text{pound (force)}}{\text{square inch}}$	$\left(\frac{\text{lb (force)}}{\text{sq in}} \right)$	$0.006895 \frac{\text{N}}{\text{mm}^2}$	$0.006895 \frac{\text{N}}{\text{mm}^2}$
	$1 \frac{\text{short ton (force)}}{\text{square inch}}$	$\left(\frac{\text{sh tn (force)}}{\text{sq in}} \right)$	$13.78951 \frac{\text{N}}{\text{mm}^2}$	$13.78951 \frac{\text{N}}{\text{mm}^2}$
Work, energy, quantity of heat, internal (intrinsic) energy and enthalpy	1 foot-pound	(ft lb)	1.3558 J	1.3558 J
	1 horse power hour	(Hp h)	2.6841 MJ	2.6841 MJ
	1 British Thermal Unit	(BTU)	1.0558 kJ	1.0558 kJ

Power (heat flow)	1 foot-pound (av) per second	$\left(\frac{\text{ft lb}}{\text{s}}\right)$	1.3558 W	1.3558 W
	1 horse power	(Hp)	0.7457 kW	0.7457 kW
	1 British Thermal Unit per second	$\left(\frac{\text{BTU}}{\text{s}}\right)$	1.0558 kW	1.0558 kW
Dynamic viscosity	$1 \frac{\text{pound (mass)}}{\text{foot} \times \text{second}}$	$\left(\frac{\text{lb (mass)}}{\text{ft s}}\right)$	1.4882 Pa s	1.4882 Pa s
	$1 \frac{\text{pound (force)} \times \text{second}}{\text{square foot}}$	$\left(\frac{\text{lb (force) s}}{\text{sq ft}}\right)$	47.8803 Pa s	47.8803 Pa s
Temperature	Conversion of temperature points:			
	$T = \frac{5}{9} t_F + 255.37;$	$t = \frac{5}{9} (t_F - 32)$		
	$T = \frac{5}{4} t_R + 273.15;$	$t = \frac{5}{4} t_R$		
	Conversion of temperature differences:			
	$\Delta T = \Delta t = \frac{5}{9} \Delta t_F$	$\Delta T = \Delta t = \frac{5}{4} \Delta t_R$		
	where: T thermodynamic temperature t Celsius temperature tF Fahrenheit temperature tR Réaumur temperature	in K in °C in °F in °R		

3. Working diagrams (enlargements)

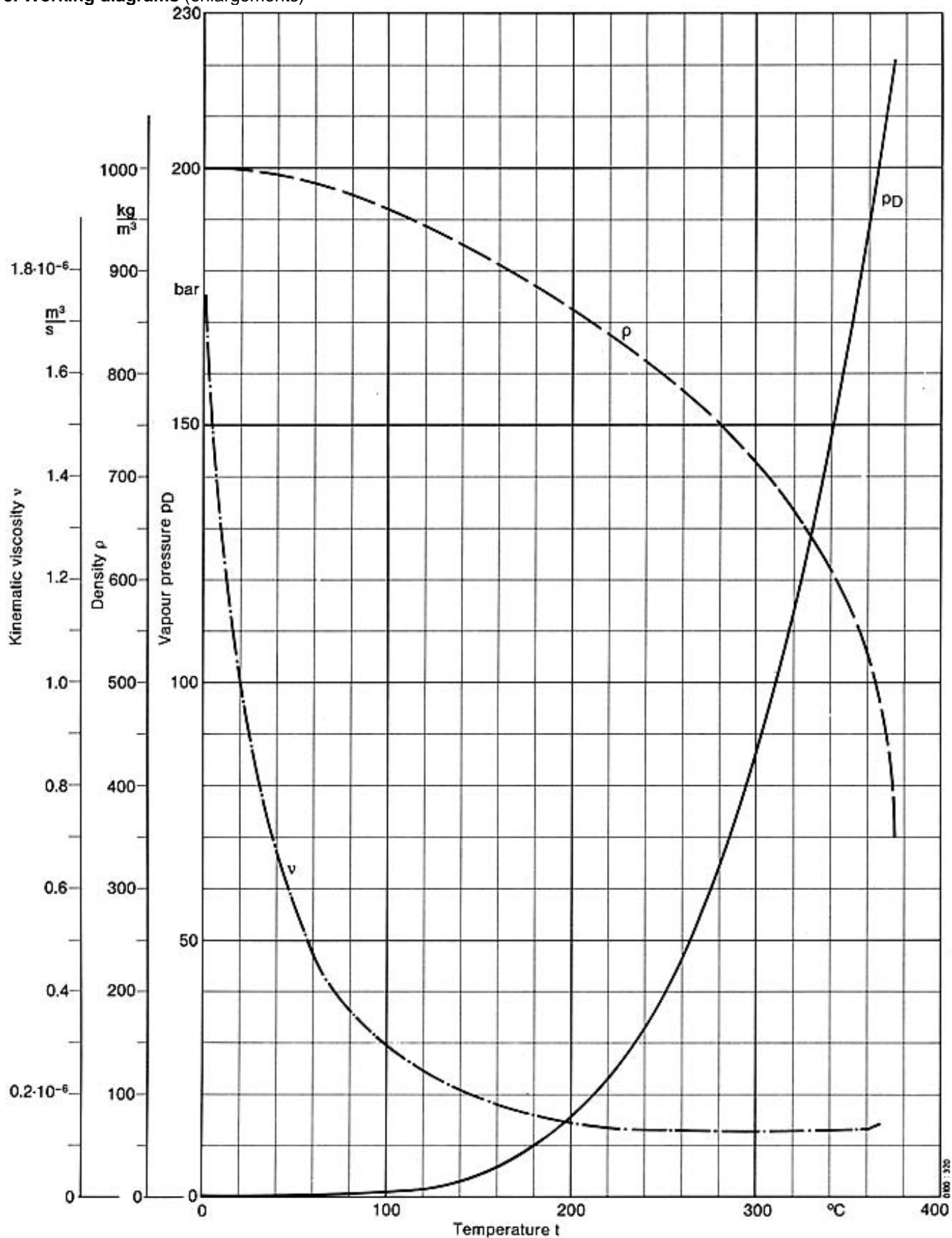


Diagram relating to Table 1 "Vapour Pressure" and to "Viscosity":
vapour pressure p_D and kinematic viscosity v in function of the temperature t

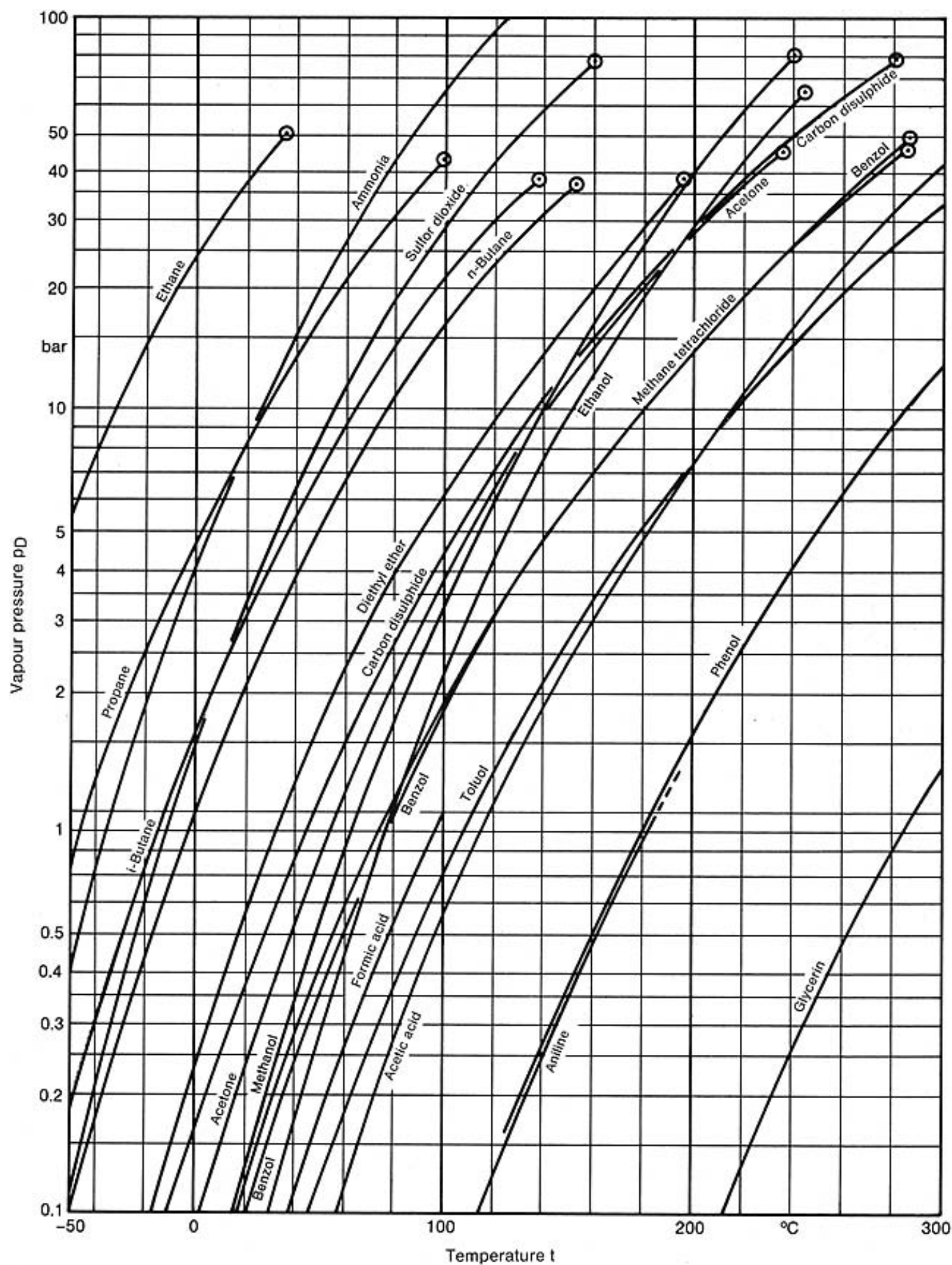


Diagram relating to Table 2 "Vapour Pressure":
vapour pressure p_D of various liquids in function of the temperature t

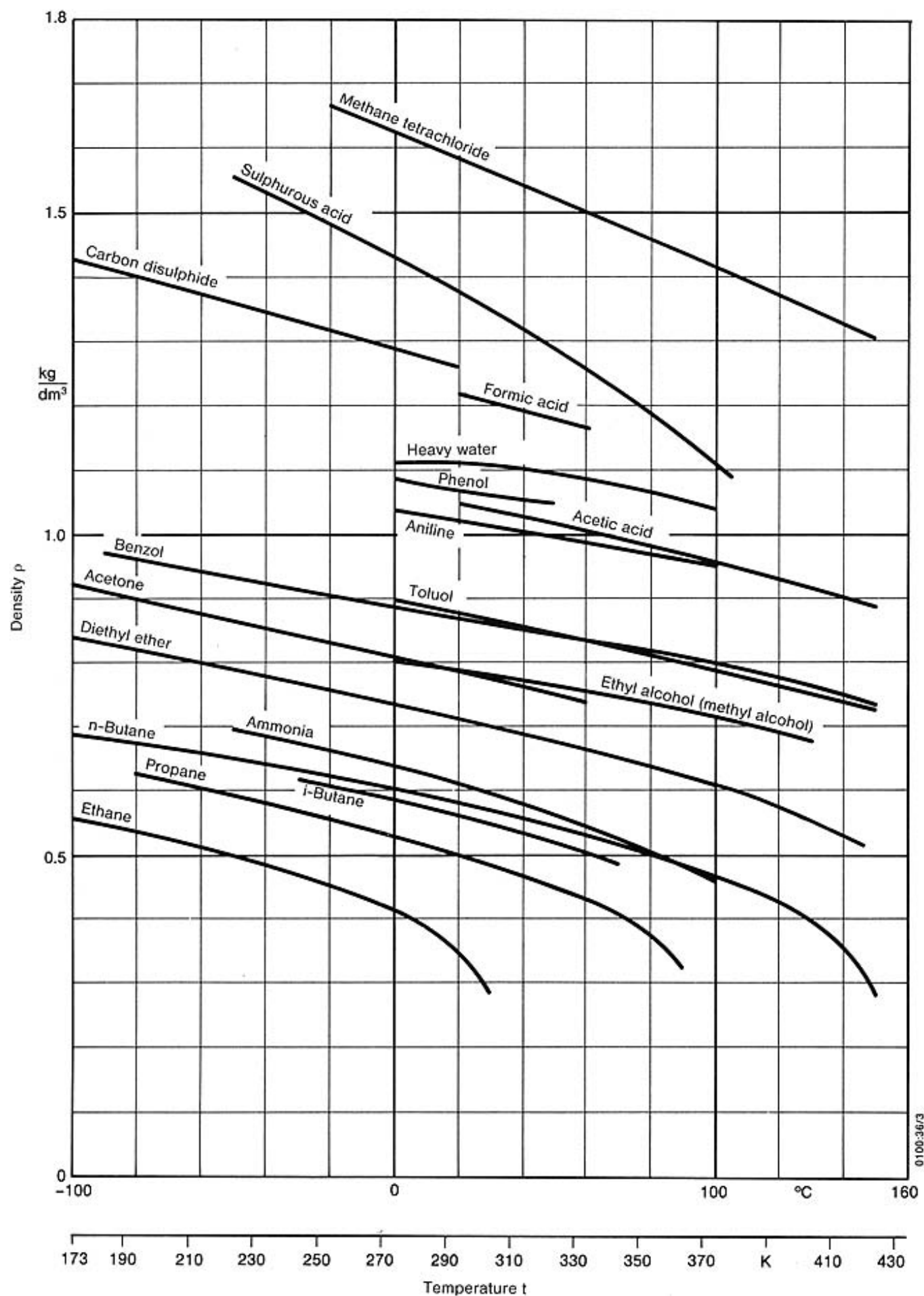


Diagram relating to "Density of Pumped Media"

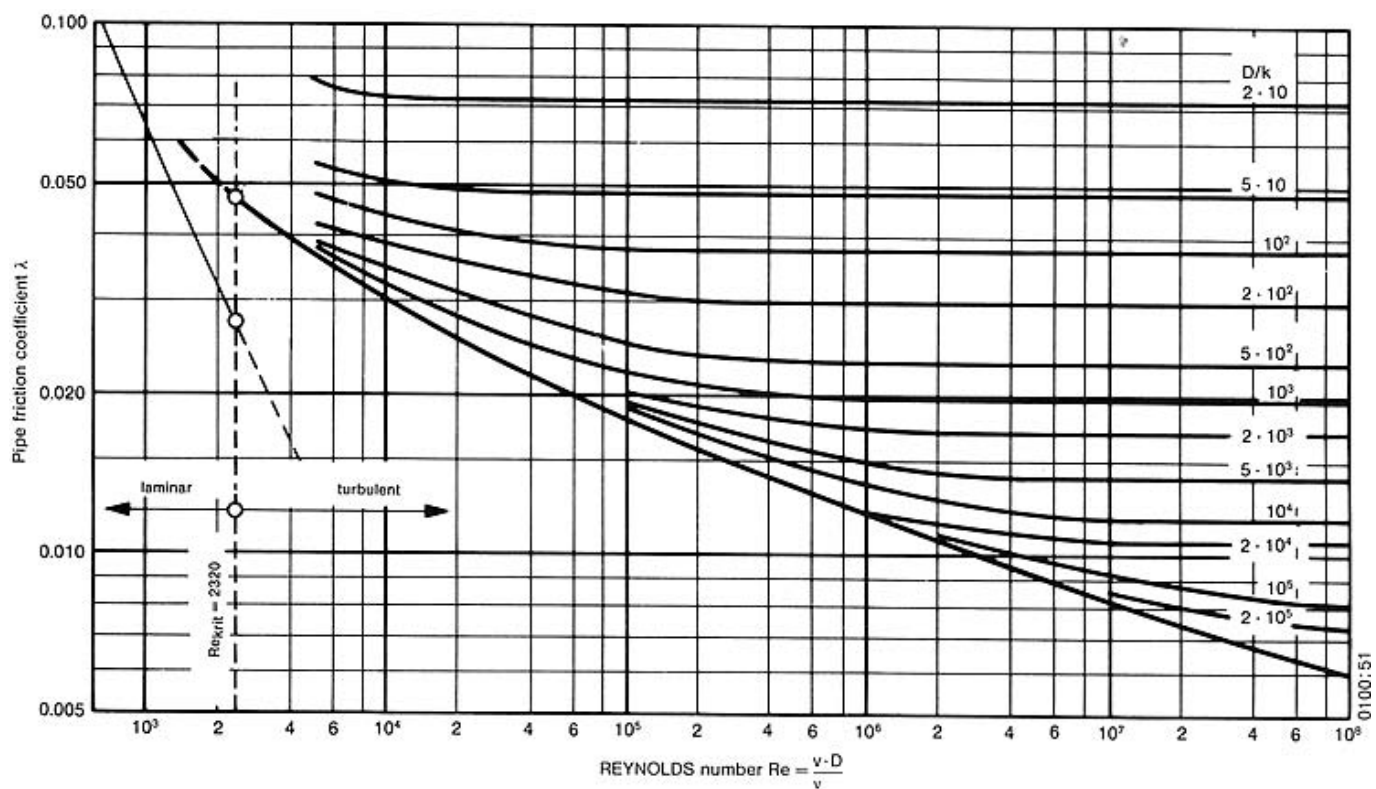


Fig. 1 relating to "Pressure Loss"

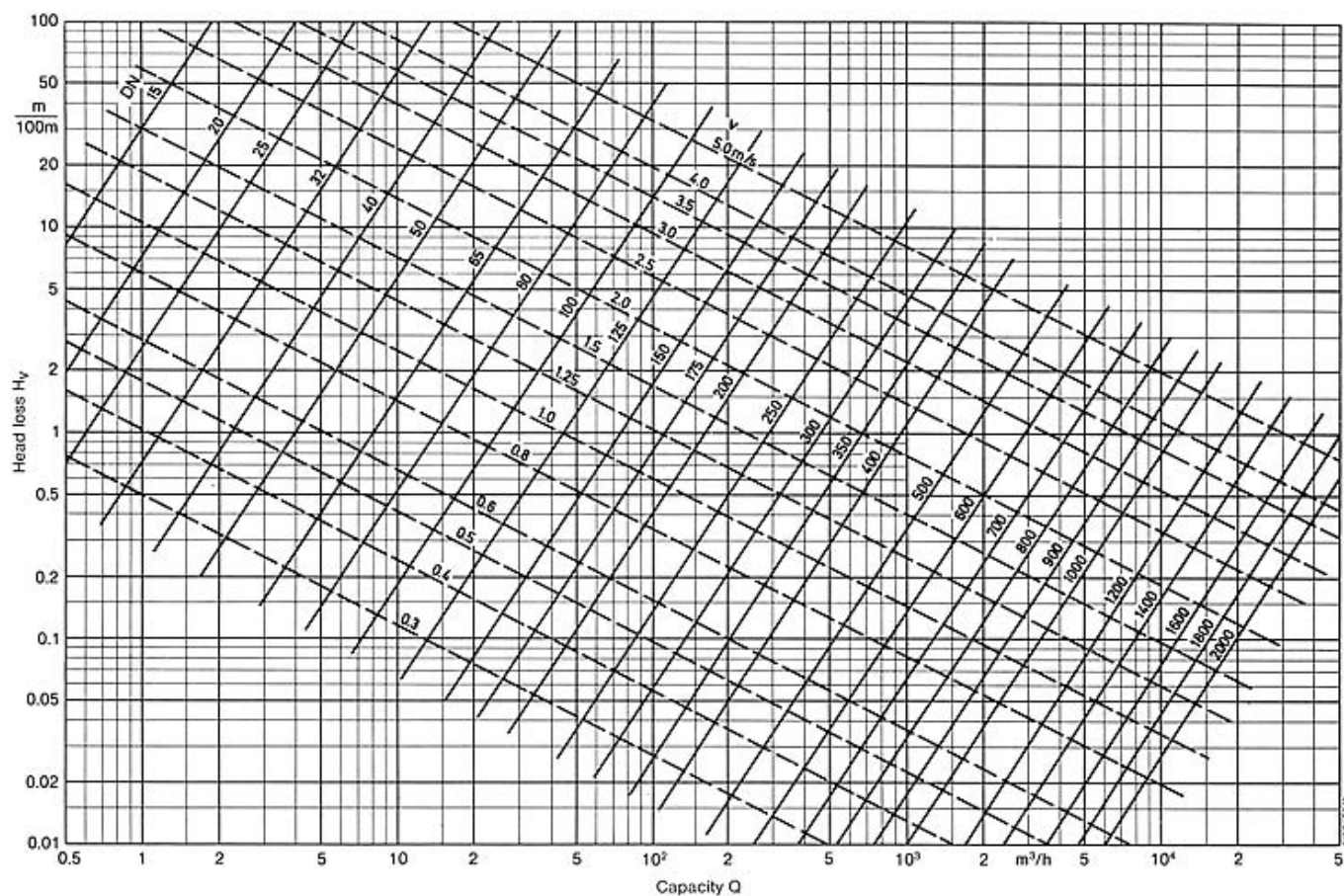


Fig. 2 relating to "Pressure Loss"

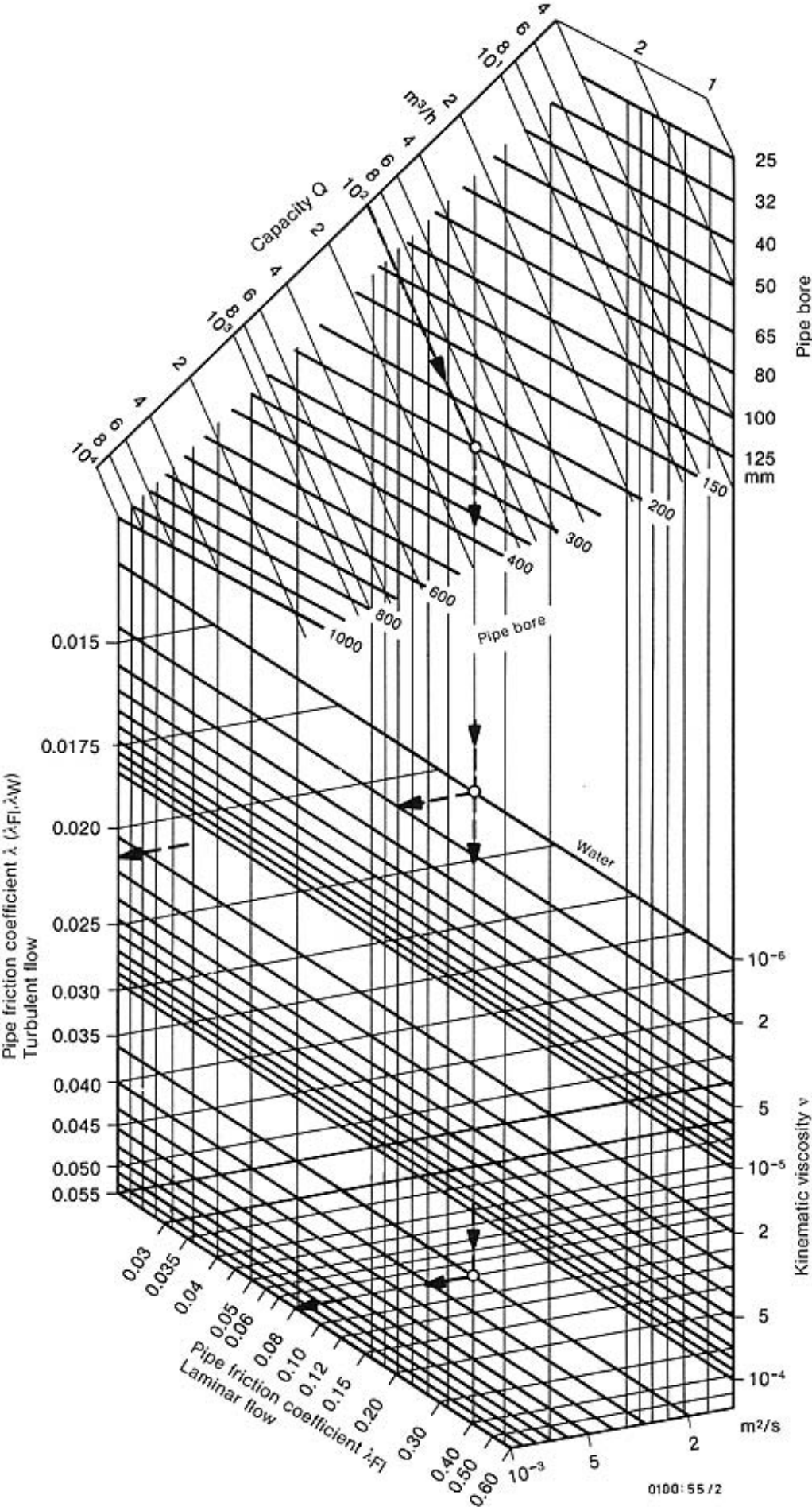


Fig. 5 relating to "Pressure Loss"

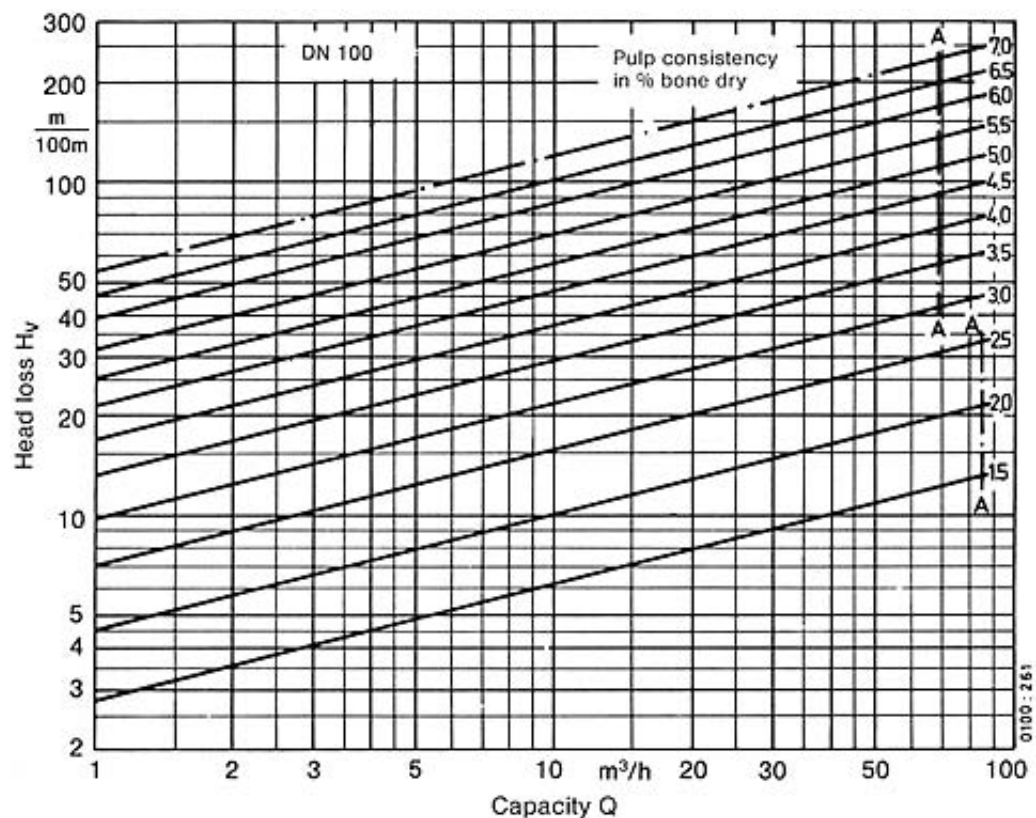


Fig. 6 a relating to "Pressure Loss"

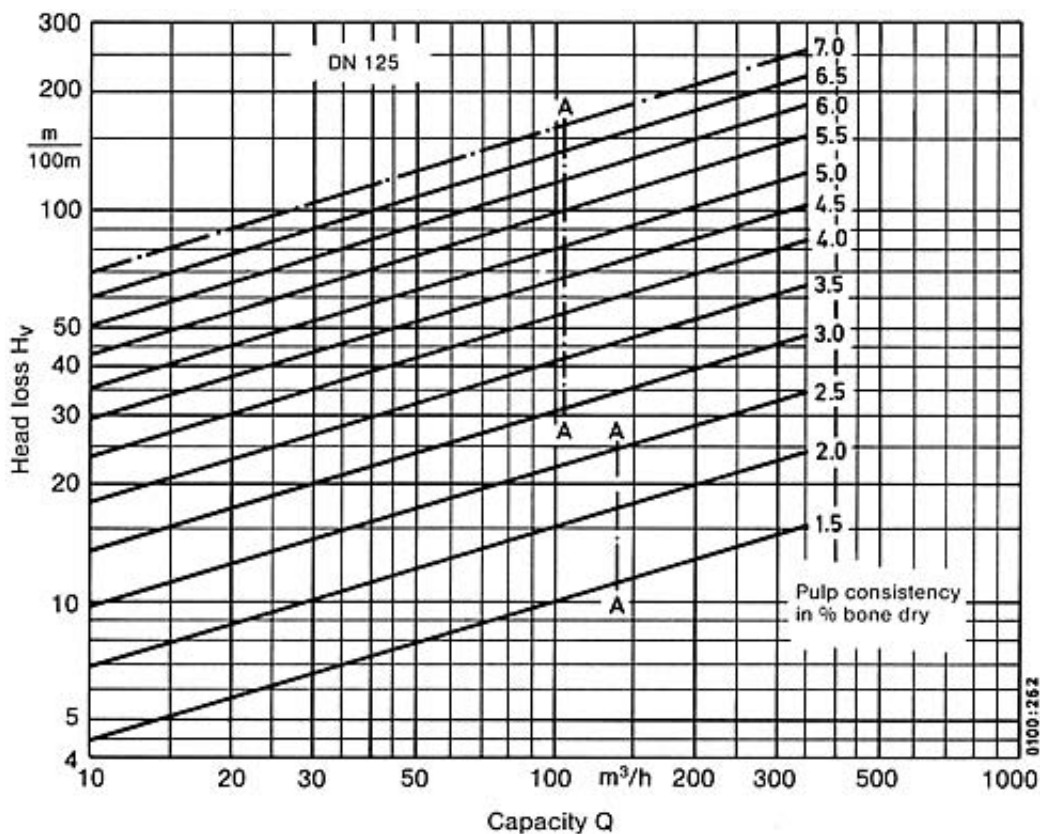


Fig. 6 b relating to "Pressure Loss"

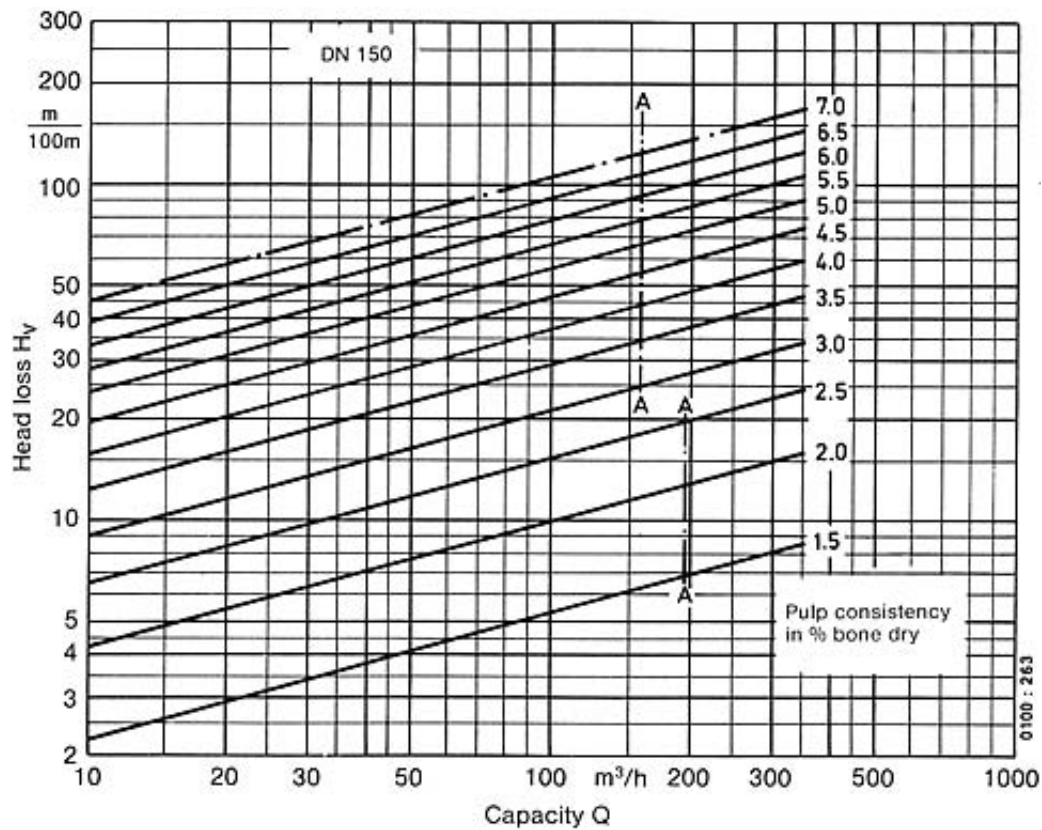


Fig. 6 c relating to "Pressure Loss"

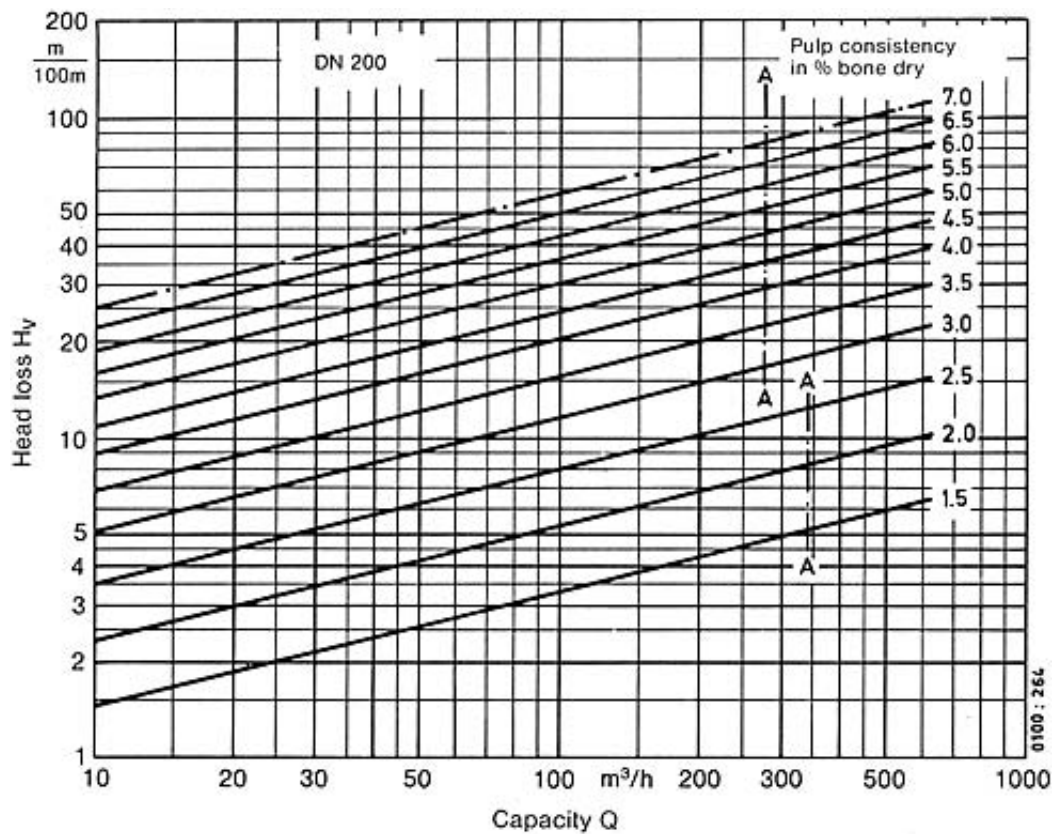


Fig. 6 d relating to "Pressure Loss"

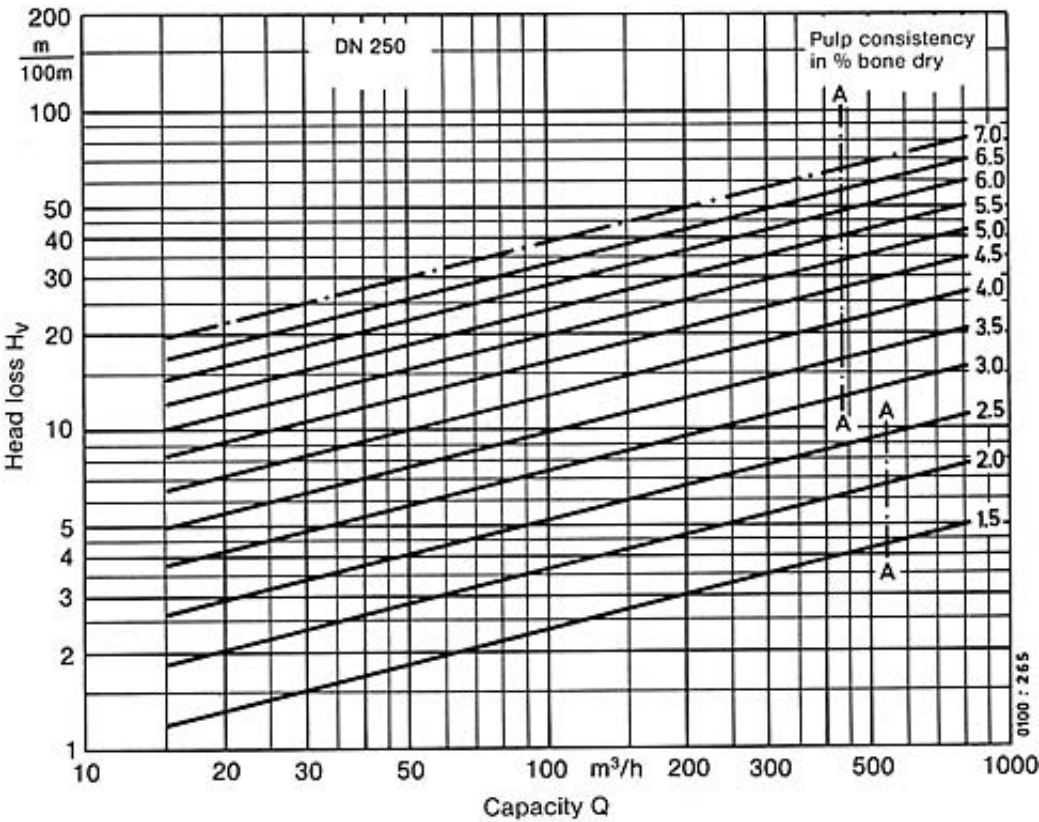


Fig. 6 e relating to "Pressure Loss"

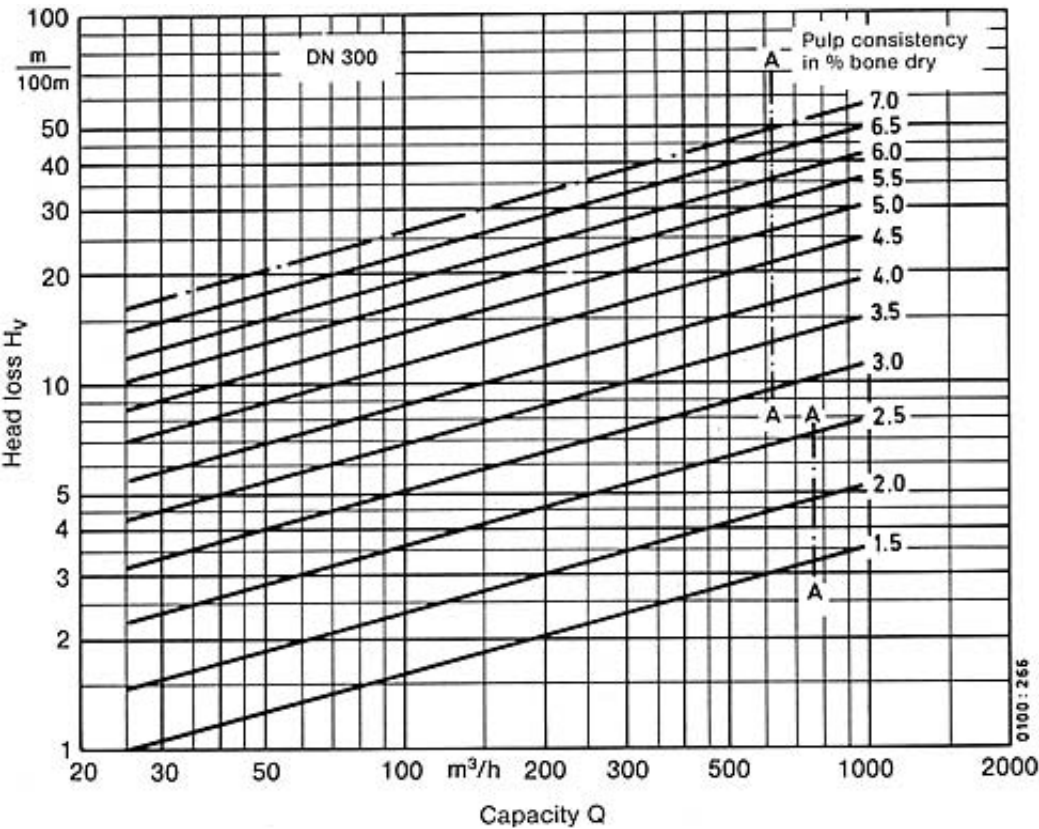


Fig. 6 f relating to "Pressure Loss"

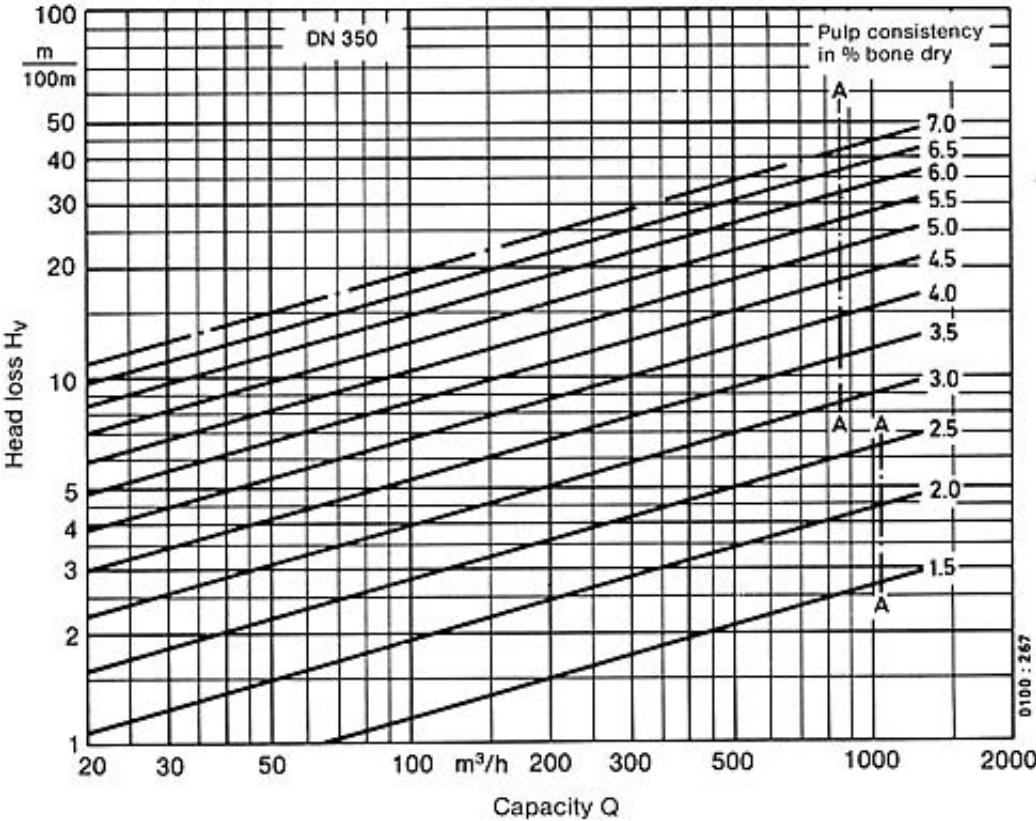


Fig. 6 g relating to "Pressure Loss"

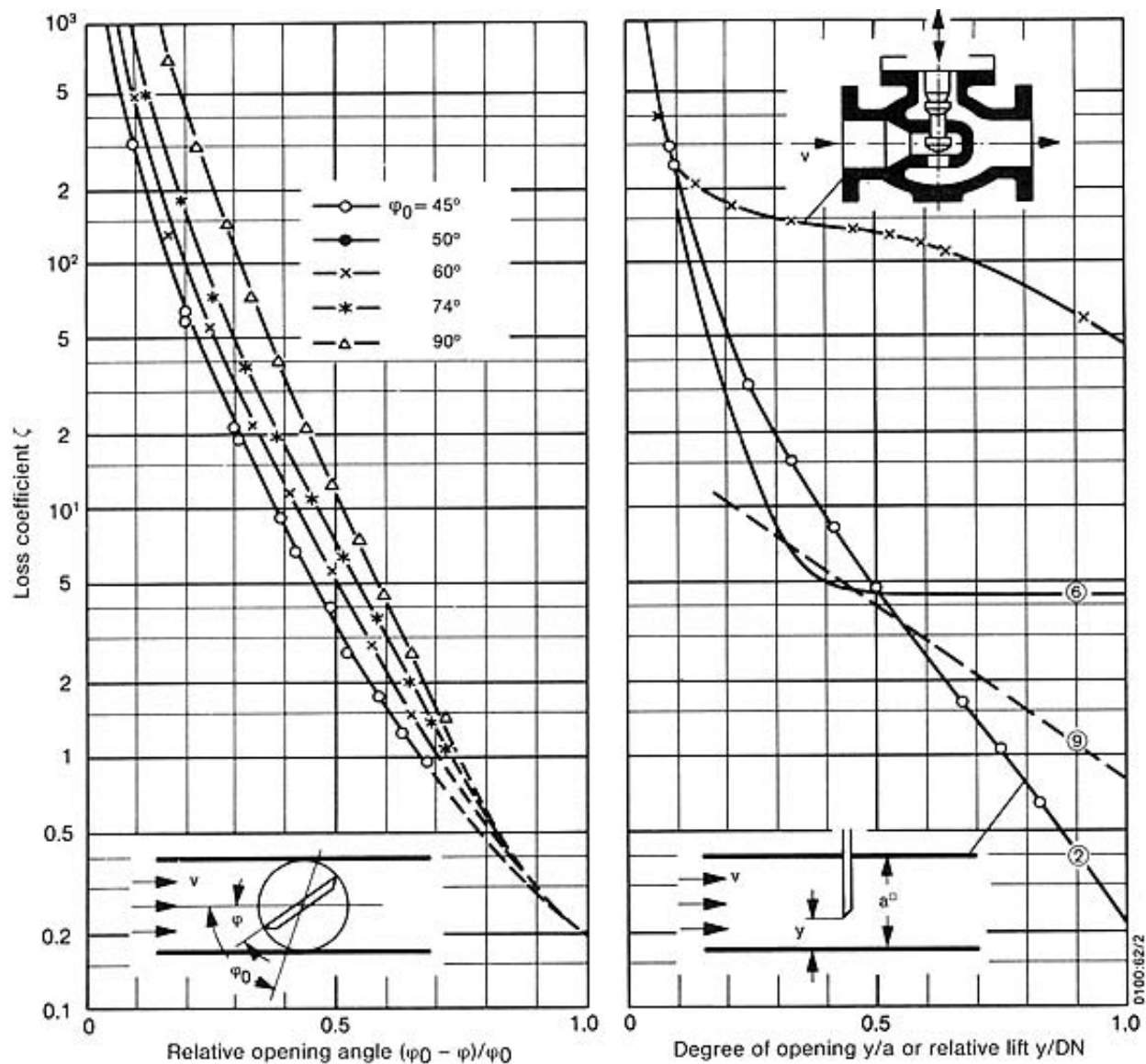


Fig. 10 relating to "Pressure Loss"

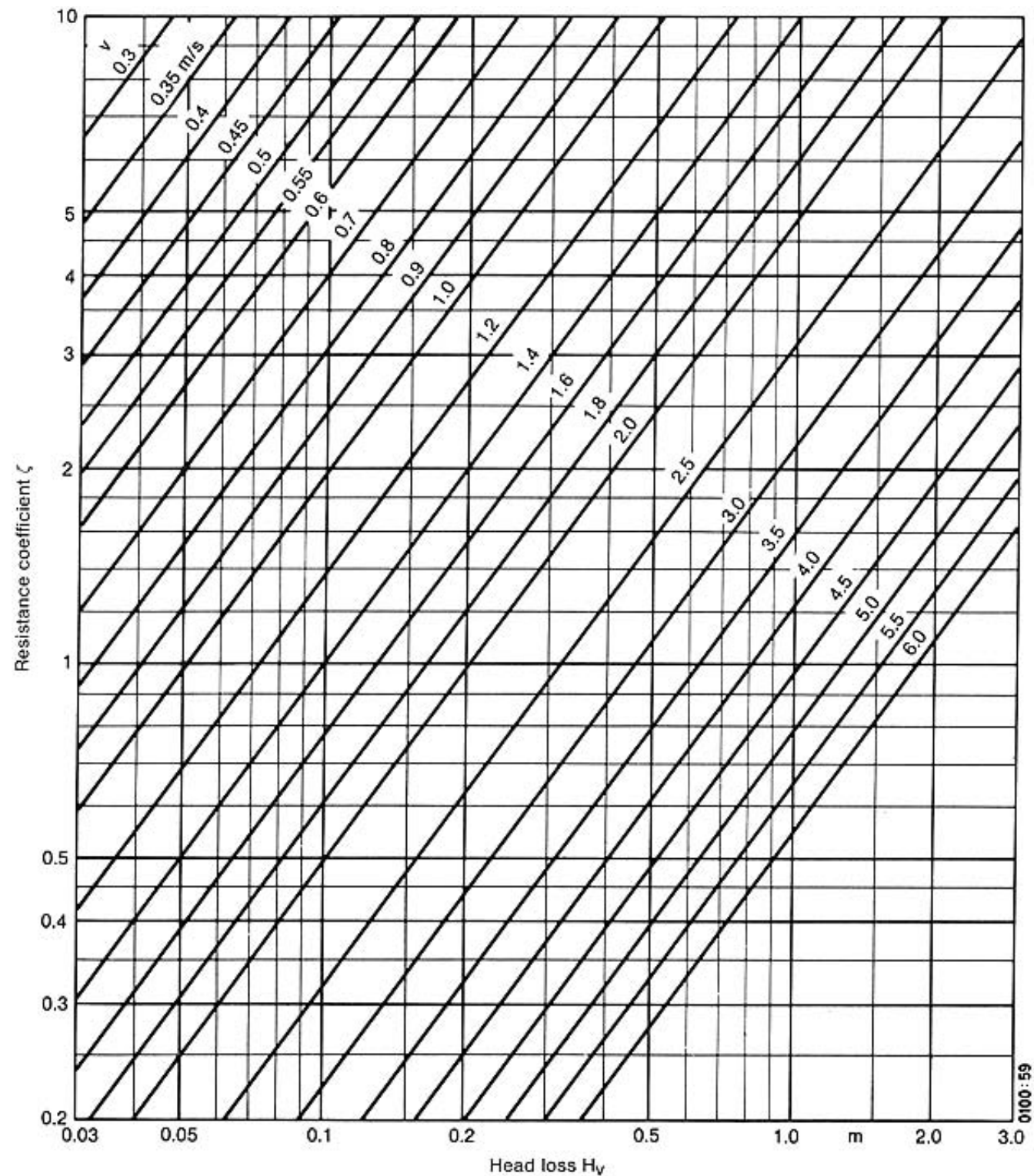


Fig. 7 relating to "Pressure Loss"

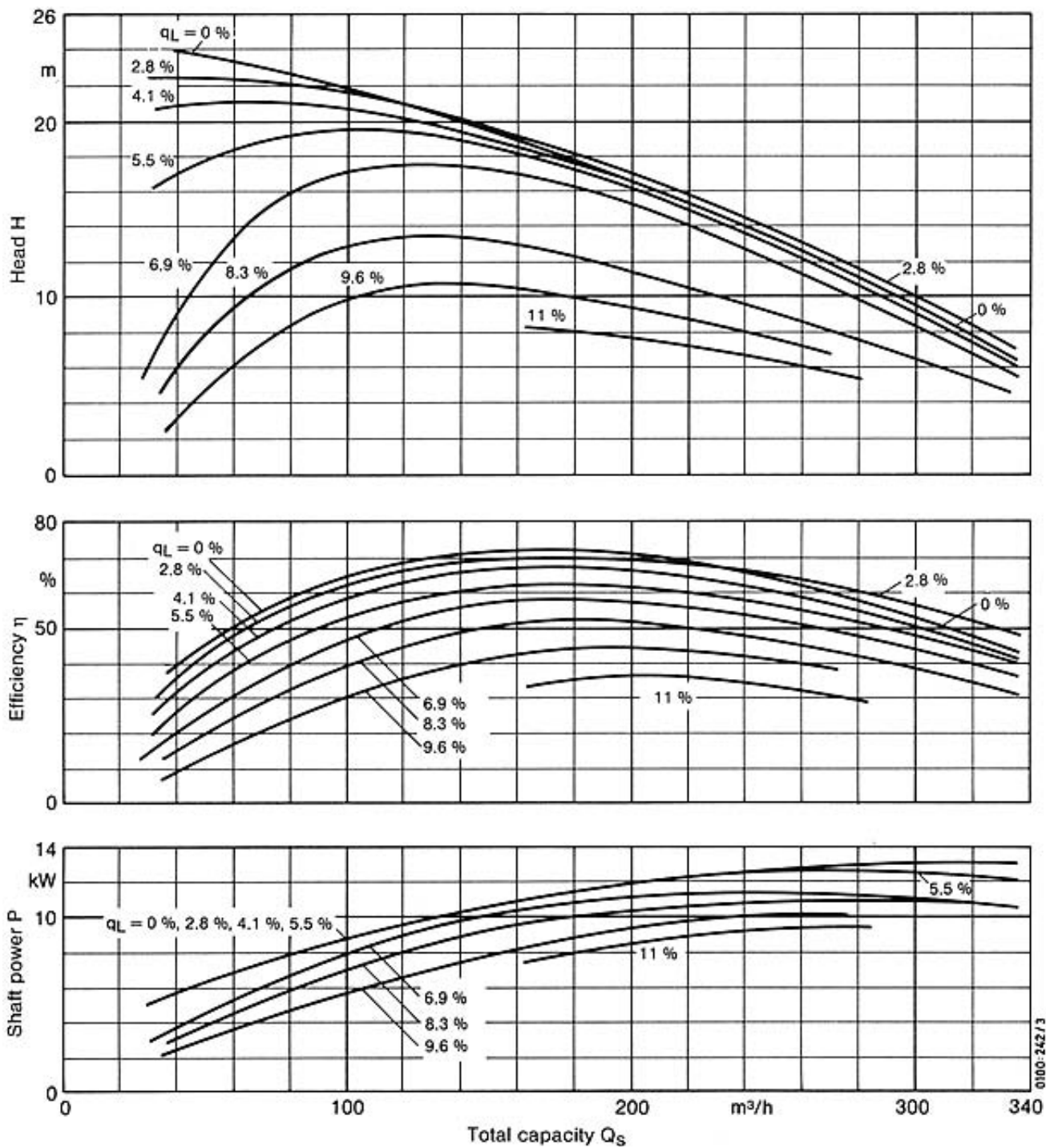


Fig. 2 relating to "Gas Content of Pumped Medium"

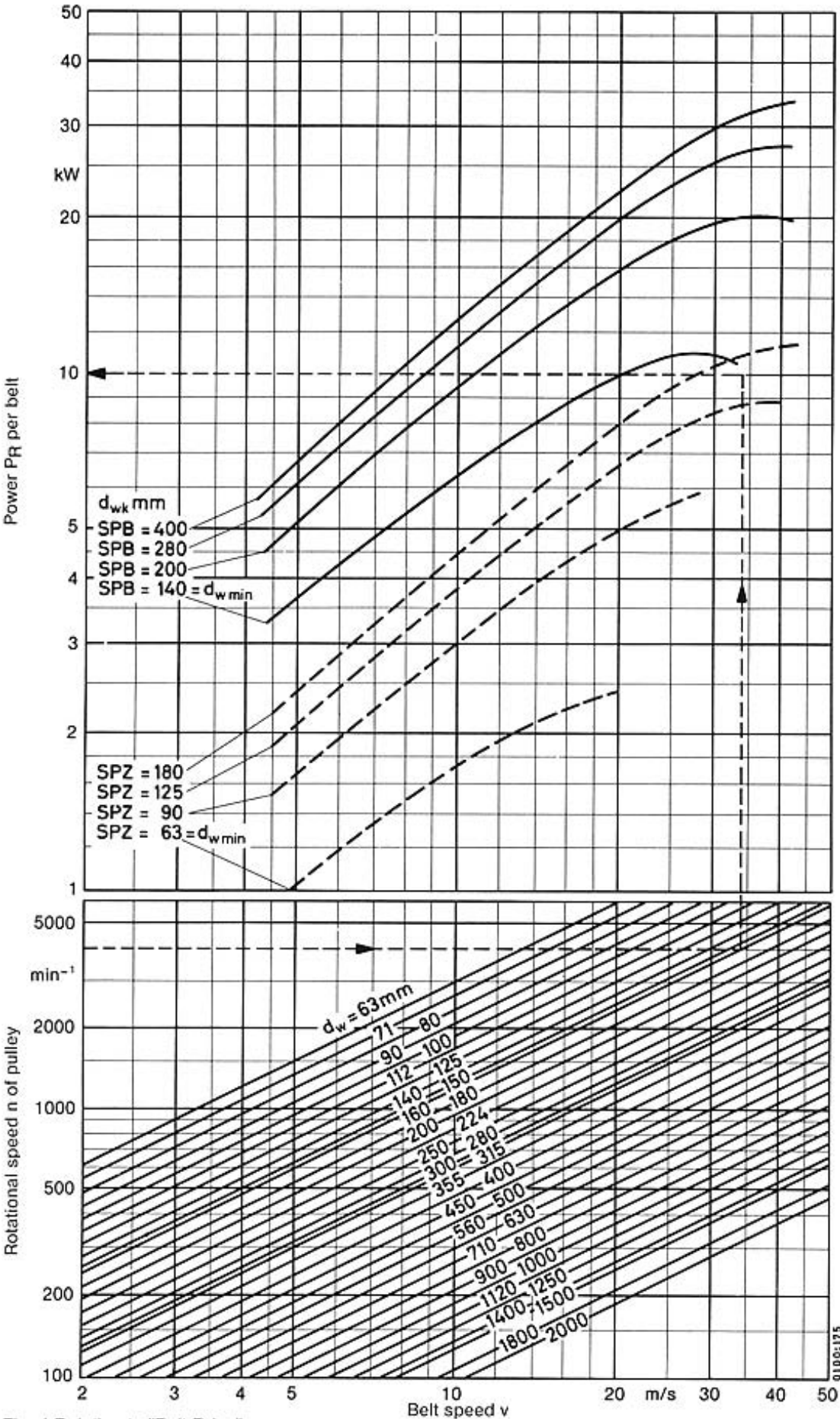


Fig. 4 Relating to "Belt Drive"

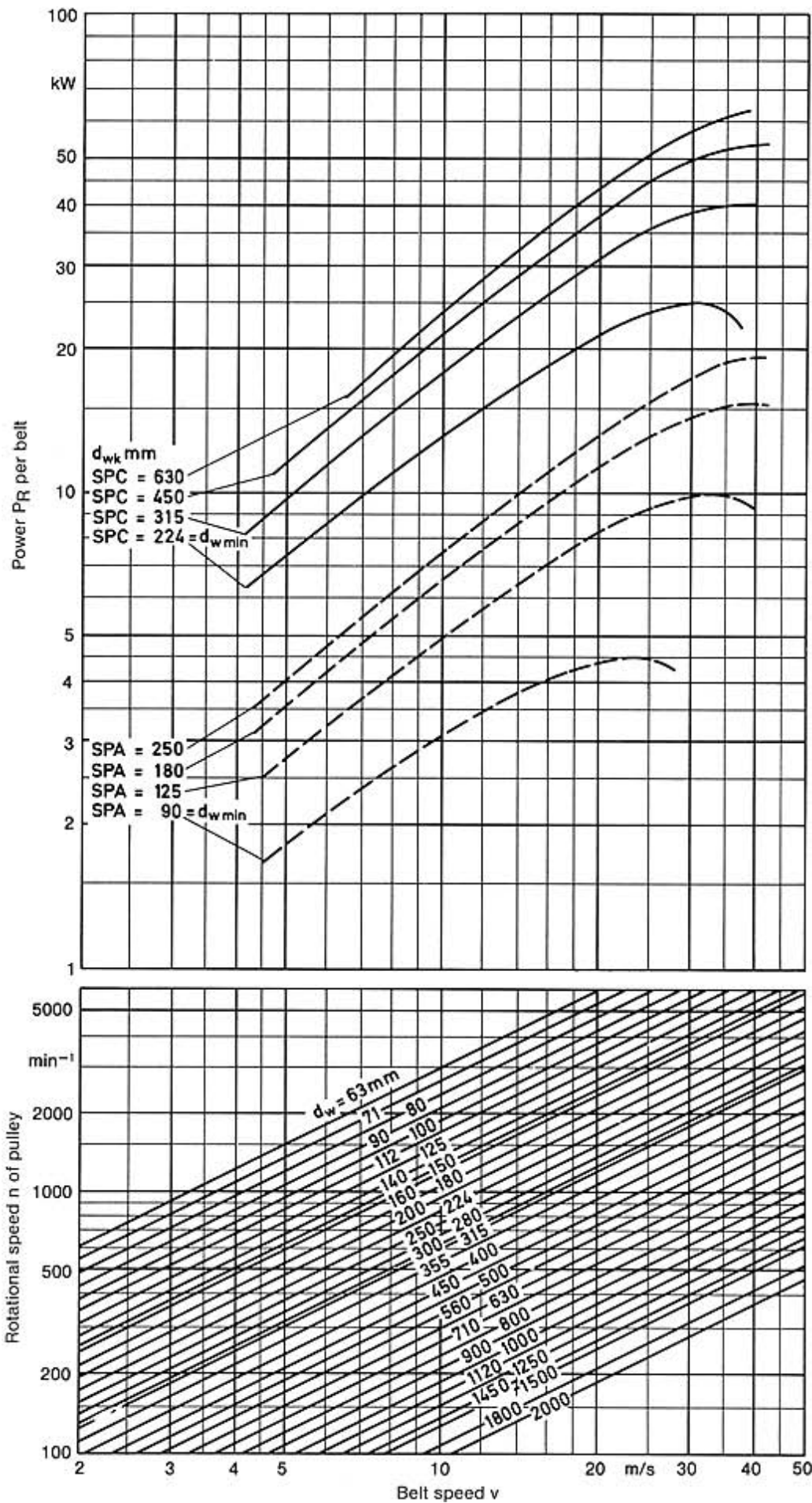


Fig. 5 relating to "Belt Drive"

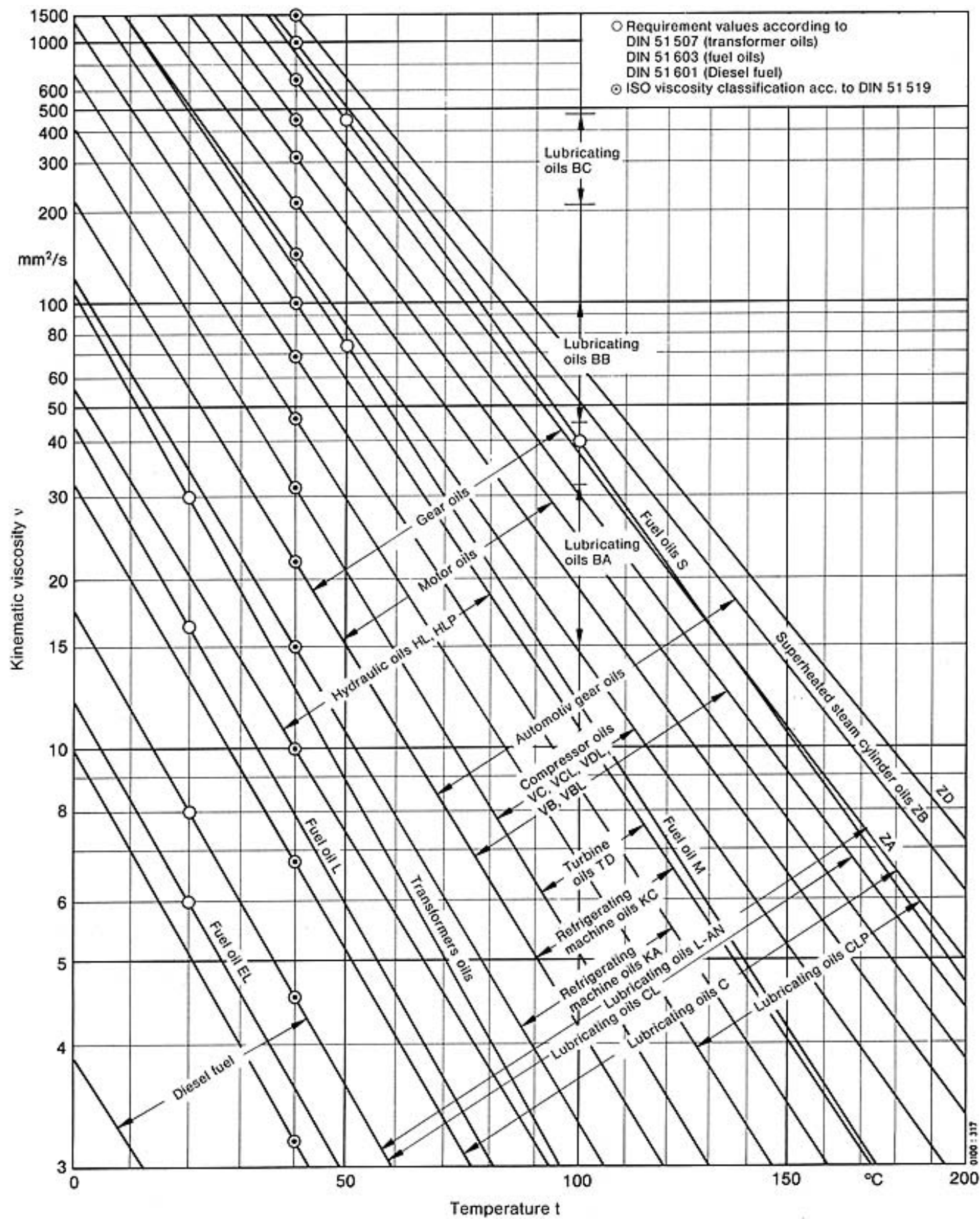


Fig. 1 relating to "Viscosity"

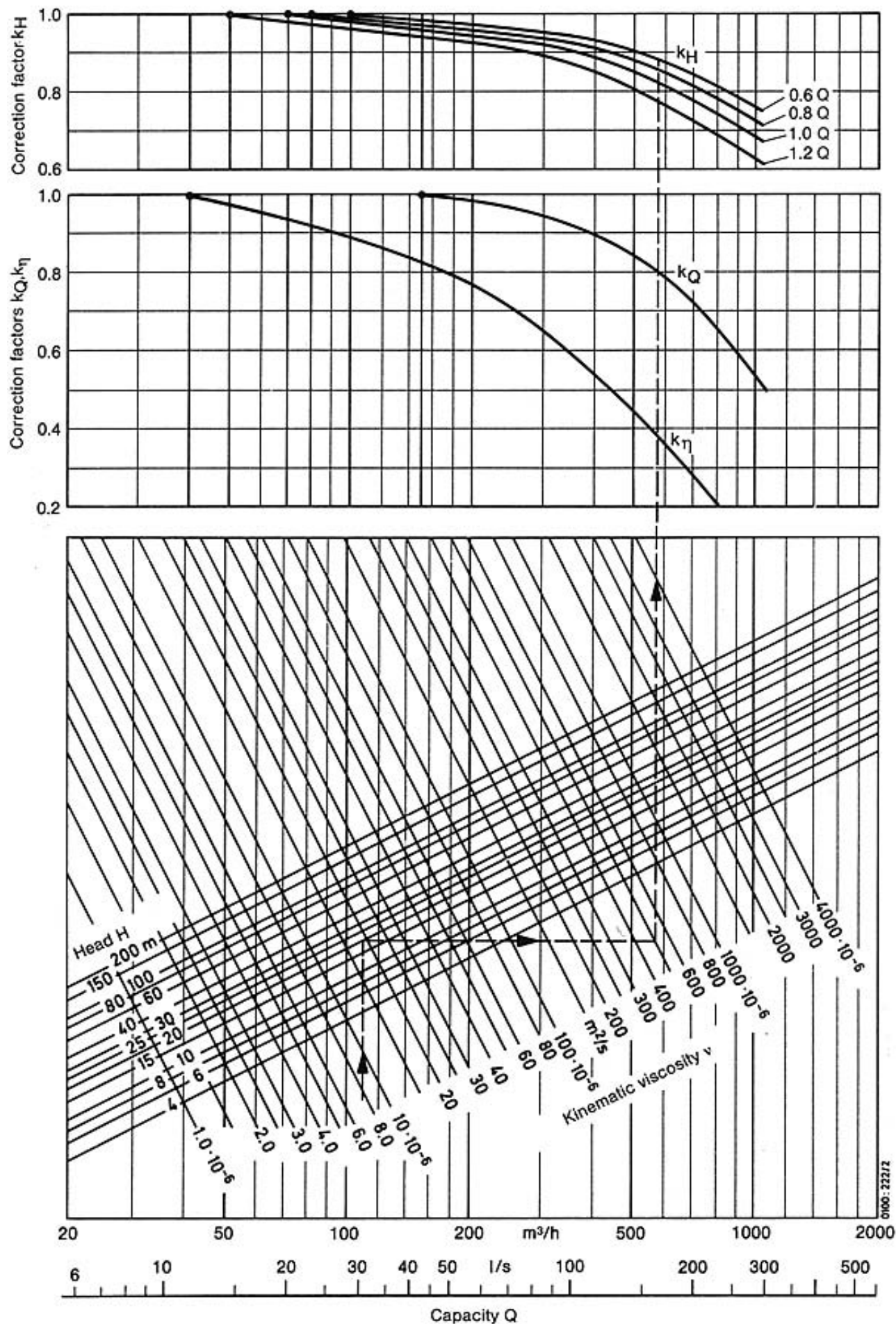


Fig. 3 relating to "Viscosity"

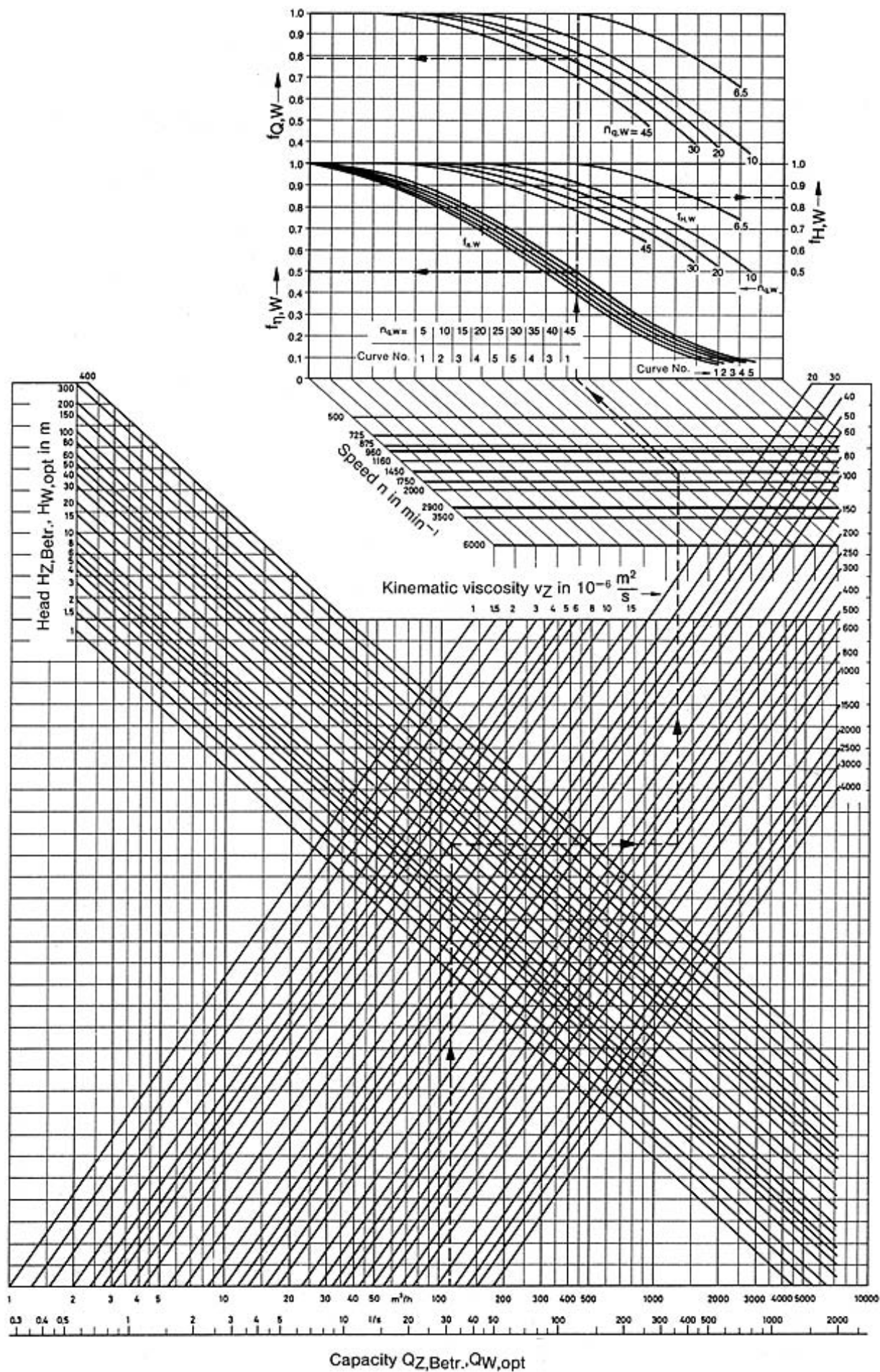


Fig. 4 relating to "Viscosity"



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