

Technical specifications for centrifugal pumps — Class I

ICS 23.080

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National foreword

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The start and finish of text introduced or altered by amendment is indicated in the text by tags. Tags indicating changes to ISO text carry the number of the ISO amendment. For example, text altered by ISO amendment 1 is indicated by A1 A1 tags.

The UK participation in its preparation was entrusted to Technical Committee MCE/6, Pumps and pump testing.

A list of organizations represented on this committee can be obtained on request to its secretary.

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English version

Technical specifications for centrifugal pumps — Class I

(ISO 9905:1994)

Spécifications techniques pour pompes
centrifuges — Classe I
(ISO 9905:1994)

Kreiselpumpen technische Anforderungen —
Klasse I
(ISO 9905:1994)

This European Standard was approved by CEN on 18 September 1997.

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CEN

European Committee for Standardization
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Foreword

The text of the International Standard from Technical Committee ISO/TC 115, Pumps, of the International Organization for Standardization (ISO) has been taken over as a European Standard by Technical Committee CEN/TC 197, Pumps, the secretariat of which is held by AFNOR.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by April 1998, and conflicting national standards shall be withdrawn at the latest by April 1998.

According to CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Czech Republic, Denmark, Finland, France, Germany, Greece, Iceland, Ireland, Italy, Luxembourg, Netherlands, Norway, Portugal, Spain, Sweden, Switzerland and the United Kingdom.

Endorsement notice

The text of the International Standard ISO 9905:1994 has been approved by CEN as a European Standard without any modification.

NOTE Normative references to International Standards are listed in Annex ZA (normative).

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Endorsement notice

The text of ISO 9905:1994/AMD 1:2011 has been approved by CEN as a EN ISO 9905:1997/A1:2011 without any modification.

Foreword to amendment A1

This document (EN ISO 9905:1997/A1:2011) has been prepared by Technical Committee ISO/TC 115 “Pumps” in collaboration with Technical Committee CEN/TC 197 “Pumps” the secretariat of which is held by AFNOR.

This Amendment to the European Standard EN ISO 9905:1997 shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by October 2011, and conflicting national standards shall be withdrawn at the latest by October 2011.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

1 Scope

1.1 This International Standard covers the Class I (most severe) requirements for centrifugal pumps used in various industries. It consists of a basic text covering general requirements. The technical requirements refer only to the pump unit.

Storage pumps are not included in this International Standard. A separate standard will be issued by IEC.

The requirements of this International Standard do not apply to centrifugal pumps used in petroleum, petrochemical and natural gas applications. The requirements for centrifugal pumps used in these applications are given in ISO 13709.

1.2 This International Standard includes design features concerned with installation, maintenance and safety of such pumps, including baseplate, coupling and auxiliary piping.

1.3 Where this International Standard specification has been called for:

- a) and requires a specific design feature, alternative designs may be offered which meet the intent of this International Standard, provided that the alternative is described in detail;
- b) pumps not complying with all requirements of this International Standard may be offered for consideration, provided that all deviations are stated.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 7-1:1982, *Pipe threads where pressure-tight joints are made on the threads — Part 1: Designation, dimensions and tolerances.*

ISO 76:1987, *Rolling bearings — Static load ratings.*

ISO 185:1988, *Grey cast iron — Classification.*

ISO 228-1:1982, *Pipe threads where pressure-tight joints are not made on the threads — Part 1: Designation, dimensions and tolerances.*

ISO 281:1990, *Rolling bearings — Dynamic load ratings and rating life.*

A1 Text deleted **A1**

ISO 544:1989, *Filler materials for manual welding — Size requirements.*

ISO 1940-1:1986, *Mechanical vibration — Balance quality requirements of rigid rotors — Part 1: Determination of permissible residual unbalance.*

A1 Text deleted **A1**

ISO 2858:1975, *End-suction centrifugal pumps (rating 16 bar) — Designation, nominal duty point and dimensions.*

ISO 3069:1974, *End suction centrifugal pumps — dimensions of cavities for mechanical seals and for soft packing.*

ISO 3274:1975, *Instruments for the measurement of surface roughness by the profile method — Contact (stylus) instruments of consecutive profile transformation — Contact profile meters, system M.*

ISO 3506:1979, *Corrosion-resistant stainless steel fasteners — Specifications.*

ISO 3744:1981, *Acoustics — Determination of sound power levels of noise sources — Engineering methods for free-field conditions over a reflecting plane.*

ISO 3746:1979, *Acoustics — Determination of sound power levels of noise sources — Survey method.*

ISO 3755:1991, *Cast carbon steels for general engineering purposes.*

ISO 4863:1984, *Resilient shaft couplings — Information to be supplied by users and manufacturers.*

ISO 7005-1:1992, *Metallic flanges — Part 1: Steel flanges.*

ISO 7005-2:1988, *Metallic flanges — Part 2: Cast iron flanges.*

ISO 7005-3:1988, *Metallic flanges — Part 3: Copper alloy and composite flanges.*

A1 ISO 9906:—¹⁾, *Rotodynamic pumps — Hydraulic performance acceptance tests — Grades 1, 2 and 3* **A1**

A1 ¹⁾ To be published. (Revision of ISO 9906:1999) **A1**

3 Definitions

For the purposes of this International Standard, the following definitions apply.

3.1

normal conditions

conditions at which usual operation is expected

3.2

rated conditions

specified guarantee-point operating conditions, including flowrate, head, power, efficiency, net positive suction head, suction pressure, temperature, density, viscosity and speed

3.3

operating conditions

all operating parameters (for example temperature, pressure) determined by a given application and pumped liquid

these parameters will influence the type of construction materials

3.4

allowable operating range

flow range, defined by the manufacturer/supplier, at the specified operating conditions using the impeller supplied, as limited by cavitation, heating, vibration, noise, shaft deflection and other similar criteria; range whose upper and lower limits are denoted by maximum and minimum continuous flow, respectively

3.5

maximum allowable casing working pressure

greatest outlet pressure at the specified operating temperature for which the pump casing is suitable

3.6

basic design pressure

pressure derived from the permitted stress at 20 °C of the material used for the pressure-containing parts

3.7

maximum outlet working pressure

sum of the maximum inlet pressure plus maximum differential pressure at rated conditions using the supplied impeller

3.8

rated outlet pressure

outlet pressure of the pump at the guarantee point with rated flow, rated speed, rated inlet pressure and density

3.9

maximum inlet pressure

highest inlet pressure to which the pump is subjected during operation

3.10

rated inlet pressure

inlet pressure for the operating conditions at the guarantee point

3.11

maximum allowable temperature

highest allowable continuous temperature for which the equipment (or any part to which the term refers) is suitable when handling the specified operating fluid at the specified operating pressure

3.12

rated power input

power required by the pump at the rated conditions

3.13

maximum dynamic sealing pressure

highest pressure expected at the shaft seals during any specified operating condition and during startup and shutdown

NOTE 1 In determining this pressure, consideration should be given to the maximum inlet pressure, circulation or injection (flush) pressure and the effect of internal clearance changes.

3.14

minimum permitted flow

- 1) for stable flow: Lowest flowrate at which the pump can operate without exceeding the noise and vibration limits imposed by this International Standard.
- 2) for thermal flow: Lowest flowrate at which the pump can operate and still maintain the temperature of the pumped liquid below that at which net positive suction head available equals net positive suction head required.

3.15

corrosion allowance

that portion of the wall thickness of the parts wetted by the pumped liquid in excess of the theoretical thickness required to withstand the pressure limits given in 4.4.2.2 and 4.4.2.4

3.16

maximum allowable continuous speed

highest speed at which the manufacturer will permit continuous operation

3.17

rated speed

number of revolutions of the pump per unit time required to meet the rated conditions

NOTE 2 Induction motors will operate at a speed that is a function of the load imposed.

3.18

trip speed

speed at which the independent emergency overspeed device operates to shut down a prime mover

3.19

first critical speed

speed of rotation at which the lowest lateral natural frequency of vibration of the rotating parts corresponds to the frequency of rotation

3.20

design radial load

maximum hydraulic radial forces on the largest impeller (diameter and width) operating within the manufacturer's specified range on its maximum speed curve using the design liquid (normally 1 000 kg/m³)

3.21

maximum radial load

maximum hydraulic radial forces on the largest impeller (diameter and width) operating at any point on its maximum speed curve with a maximum liquid density

3.22

shaft runout

total radial deviation indicated by a device measuring shaft position in relation to the bearing housing as the shaft is rotated manually in its bearings with the shaft in the horizontal position

3.23

face runout

total axial deviation indicated at the outer radial face of the stuffing box by a device attached to and rotated with the shaft when the shaft is rotated manually in its bearings in the horizontal position the radial face is that which determines the alignment of a seal component

3.24

shaft deflection

displacement of a shaft from its geometric centre in response to the radial hydraulic forces acting on the impeller

NOTE 3 Shaft deflection does not include shaft movement caused by tilting within the bearing clearances, bending caused by impeller unbalance or shaft runout.

3.25

circulation (flush)

return of pumped liquid from a high pressure area to seal cavity, by external piping or internal passage, to remove heat generated at the seal or to maintain positive pressure in the seal cavity or treated to improve the working environment for the seal

NOTE 4 In some cases it may be desirable to circulate from the seal cavity to a lower pressure area (for example, the inlet).

3.26

injection (flush)

introduction of an appropriate (clean, compatible, etc.) liquid into the seal cavity from an external source and then into the pumped liquid

3.27

quenching

continuous or intermittent introduction of an appropriate (clean, compatible, etc.) fluid on the atmospheric side of the main shaft seal to exclude air or moisture, to prevent or clear deposits (including ice), lubricate an auxiliary seal, snuff out fire, dilute, heat or cool leakage

3.28

barrier liquid (buffer)

an appropriate (clean, compatible, etc.) liquid inserted between two seals (mechanical seal and/or soft packing)

NOTE 5 The barrier liquid pressure depends on the seal arrangement. The barrier liquid may be used to prevent air entering the pump. The barrier liquid is normally easier to seal than the pumped liquid and/or creates less hazard if leakage occurs.

3.29

throttle bush (safety bush)

close-clearance restrictive bush around the shaft (or sleeve) at the outboard end of a mechanical seal to reduce leakage in case of seal failure

3.30

throat bush

close-clearance restrictive bush around the shaft (or sleeve) between the seal (or packing) and the impeller

3.31

pressure casing

composite of all stationary pressure-containing parts of the unit, including all branches and other attached parts

3.32

double casing

type of construction in which the pressure casing is separate and distinct from the pumping elements contained in it

3.33

barrel casing

refers specifically to a pump of the double casing type

3.34

vertical canned pump

vertical pump inserted in an outer casing (can or caisson) taking its suction from the liquid in the annular space

3.35

vertical canned motor pump

glandless pumping set in which the stator of the (electric) motor is sealed by a can against the rotor which runs in the pumped liquid or in any other liquid

3.36

hydraulic power recovery turbine

pump operated with reversed flow to deliver mechanical energy at the coupling obtained from the recovery of energy released by the reduction of fluid pressure (and sometimes from the additional energy released by vapour or gas evolution from the fluid)

NOTE 6 For hydraulic power recovery turbine branches, all references in this standard to suction and discharge apply to the outlet and inlet, respectively.

3.37

radial split

refers to casing joints that are transverse to the shaft centreline

3.38

axial split

refers to casing joints that are parallel to the shaft centreline

3.39

net positive suction head (NPSH)

absolute total inlet head above the head equivalent to the vapour pressure referred to the NPSH datum plane

NOTE 7 NPSH is referred to the datum plane, whereas inlet total head is referred to the reference plane. The NPSH datum plane is the horizontal plane through the centre of the circle described by the external points of the entrance edges of the impeller blades; in the case of double inlet pumps with vertical or inclined axis, it is the plane through the higher centre. The manufacturer/supplier should indicate the position of this plane with respect to precise reference points on the pump.

3.40

net positive suction head available (NPSHA)

NPSH determined by the conditions of the installation for a specified liquid, temperature and rate of flow

3.41

net positive suction head required (NPSHR)

minimum NPSH for a pump achieving a specified performance at the specified rate of flow and speed (occurrence of visible cavitation, increase of noise due to cavitation, appearance of head or efficiency drop, head or efficiency drop of a given amount, etc.)

3.42

suction specific speed

parameter relating the rotational speed, the flowrate and the NPSHR, determined at the best efficiency point

3.43

hydrodynamic bearing

bearing whose surface is oriented to another surface such that relative motion forms an oil wedge to support the load without metal-to-metal contact

3.44

hydrodynamic radial bearing

bearing of sleeve-journal or tilting-shoe type construction

3.45

hydrodynamic thrust bearing

bearing of multiple-segment or tilting-shoe type construction

3.46

design values

values used in the design of a pump for the purpose of determining the performance, the minimum permissible wall thickness and physical characteristics of the different parts of the pump

NOTE 8 Use of the word design in any term (such as design pressure, design power, design temperature or design speed) should be avoided in the purchaser's specifications. This terminology should be used only by the equipment designer and manufacturer/supplier.

3.47

coupling service factor

a factor k , by which is multiplied the nominal torque T_N of the driver in order to obtain the rated torque $T_K = kT_N$, which makes due allowance for cycle torque fluctuations from the pump and/or its driver, and therefore ensures satisfactory coupling life

4 Design

4.1 General

Whenever the documents include contradicting technical requirements, they shall apply in the following sequence:

- a) purchase order (or enquiry, if no order is placed) (see Annex C and Annex D);
- b) data sheet (see Annex A);
- c) this International Standard;
- d) other standards to which reference is made in the order (or enquiry, if no order is placed).

The applicability of any national and local codes, regulations, ordinances or rules shall be mutually agreed upon by the purchaser and the manufacturer/supplier.

4.1.1 Characteristic curve

4.1.1.1 The characteristic curve for the supplied impeller shall show the head, efficiency, NPSHR and the power input, plotted against flowrate. It shall also show the allowable operating range of the pump. Head/flowrate curves (on the basis of calculation or test) for the largest and smallest impeller diameters shall be plotted for single stage pumps, and for multistage pumps when requested.

4.1.1.2 Pumps that have stable head/flowrate curves which continuously rise to shutoff are preferred for most applications and are required when parallel operation is specified by the purchaser. Unstable head/flowrate curves or curves with dips (such as propeller pump curves) can be offered providing the application is suitable and the curve shape deviations are shown. When service conditions are such that a stable curve is technically impossible, other means of ensuring the desired flow(s) must be used. When parallel operation is specified, the rise of the head at rated flowrate shall have sufficient slope to avoid instability of flow.

4.1.1.3 The best efficiency point for the supplied impeller should preferably be between the rated point and the normal point (see 3.1).

4.1.1.4 When the pump design permits a constant-speed driver, the pump shall be capable of approximately a 5 % head increase at rated conditions by installing a new larger impeller or impellers.

4.1.1.5 Pumps that handle Newtonian liquids more viscous than water shall have their performance corrected in accordance with the conversion factors to be agreed between purchaser and manufacturer/supplier. Non-Newtonian liquids require special consideration.

4.1.2 Net positive suction head (NPSH)

[A1] NPSHR shall be based on cold water in accordance with ISO 9906 unless otherwise agreed. [A1]

The NPSHR curve as a function of flow for water shall be provided.

NPSHA must exceed NPSHR by a 10 % margin but in each case by not less than 0,5 m. The basis for use in performance curves is the NPSH corresponding to a drop of 3 % of the total head of the first stage of the pump (NPSH3).

Should the pump manufacturer/supplier consider that, because of the construction material and liquid pumped, a greater NPSH is required, this should be stated in the proposal and the appropriate curve provided.

The manufacturer/supplier shall specify on the data sheet the net positive suction head required (NPSHR) when the pump is operated with water at the rated flowrate and rated speed.

Hydrocarbon reduction or correction shall not be applied.

For NPSH tests, see 6.3.5.

4.1.3 Pump design

4.1.3.1 Pumping units may be of single-stage or multistage design. When the rated inlet gauge pressure is positive or the differential pressure is more than 3,5 bar, the pump should be designed to minimize the pressure on the shaft seals unless thrust balance requirements dictate otherwise. On single-stage overhang designs this can be accomplished with rings or pumping vanes on the back of the impeller. On multistage pumps this can be accomplished either by a back-to-back impeller arrangement combined with a close clearance throttle bush, or by an in-line impeller arrangement using balance drums or discs.

Other means can be used after agreement between purchaser and manufacturer/supplier.

4.1.3.2 High-energy pumps (head greater than 200 m per stage and power more than 225 kW per stage) require special consideration to ensure that the radial distance between the volute tongue (including double volute casing) or diffuser vane and the impeller periphery is so dimensioned to avoid undue vibrations and noise (blade-passing frequency and low frequency at reduced flowrates).

4.1.3.3 Vertical pumps with threaded line shaft coupling that could be damaged by reverse rotation shall be provided with a non-reverse ratchet or other approved means.

4.1.3.4 All equipment shall be designed to permit rapid and economical maintenance. Major parts such as casing components and bearing housings shall be designed (shouldered or dowelled) to ensure accurate alignment on reassembly.

[A1] Note deleted [A1]

4.1.3.5 Control of the sound level of all equipment supplied shall be a joint effort of the purchaser and the manufacturer/supplier. Unless otherwise specified, the equipment supplied by the manufacturer/supplier shall conform to the requirements of local regulations and to the maximum allowable sound level specified by the purchaser.

NOTE 9 The scope of this International Standard excludes the driver, but a contribution of the driver to the sound level should be taken into account.

4.1.4 Outdoor installation

The purchaser shall specify whether the installation is indoors (heated or unheated) or outdoors (with or without a roof) and the local ambient conditions in which the equipment must operate (including maximum and minimum temperatures, unusual humidity, corrosive air or dust problems). The unit and its auxiliaries shall be suitable for operation in these specified conditions. For the purchaser's guidance, the manufacturer/supplier shall list in the proposal any special protection that the purchaser is required to supply.

4.2 Drivers

4.2.1 General

4.2.1.1 Requirements for determining rated drive performance

The following shall be considered when determining the rated performance of the drive:

- a) application and method of operation of the pump. For instance in the case of parallel operation, the possible performance range with only one pump in operation, taking into account the system characteristics, shall be considered;
- b) position of the operating point on the characteristic curve of the pump;
- c) shaft seal friction loss;
- d) circulation flow for the mechanical seal (especially for pumps with low rate of flow);
- e) properties of the pumped medium (viscosity, solids content, density);
- f) power loss and slip through the transmission;
- g) atmospheric conditions at the pump site.

Drivers for any pumps covered by this International Standard shall have power output ratings at least equal to the percentage of rated pump power input given in Figure 1 but not less than 1 kW. Where it appears that this will lead to unnecessary oversizing of the driver, an alternative proposal shall be submitted for the purchaser's approval.

4.2.1.2 Thrust load

When the thrust bearing is not part of the pump, and unless otherwise approved by the purchaser, motor, turbine or gear drivers for vertical pumps, including in-line vertical pumps, shall be designed to carry the maximum thrust the pump may develop while starting, stopping or operating at any flowrate. The maximum thrust load shall be determined at double the initial internal clearances. If the driver is not supplied by the manufacturer/supplier he shall notify the purchaser of such requirements.

4.2.2 Turbine-driven pumps

4.2.2.1 Steam turbines

The steam turbines selected shall be capable of carrying the pump rated power input required for the rated conditions based on the guaranteed pump efficiency, or alternatively the maximum power input required for the full operating range of the pump. The steam turbine power rating shall be based on the minimum inlet and maximum exhaust steam conditions specified.

4.2.2.2 Turbine-driven pump speed

The turbine-driven pumps shall be designed to operate continuously at 105 % of rated speed and to operate briefly, under emergency conditions, at up to 110 % of rated speed (turbine overspeed trip setting).

For steam turbines and reciprocating engines, the trip speed shall be at least 110 % of the maximum allowable continuous speed. For gas turbines the trip speed shall be at least 105 % of the maximum allowable continuous speed.

4.3 Critical speed, balance and vibration

4.3.1 Critical speed

4.3.1.1 Critical speeds correspond to resonant frequencies of the rotor-bearing support system. The basic identification of critical speeds is made from the natural frequencies of the system and of the forcing phenomena. If the frequency of any harmonic component of a periodic forcing phenomenon is equal to or approximates the frequency of any mode of rotor vibration, a condition of resonance may exist. If resonance exists at a finite speed, that speed is called a critical speed. This specification is concerned with actual critical speeds rather than various calculated values both in lateral vibration and in torsional oscillation.

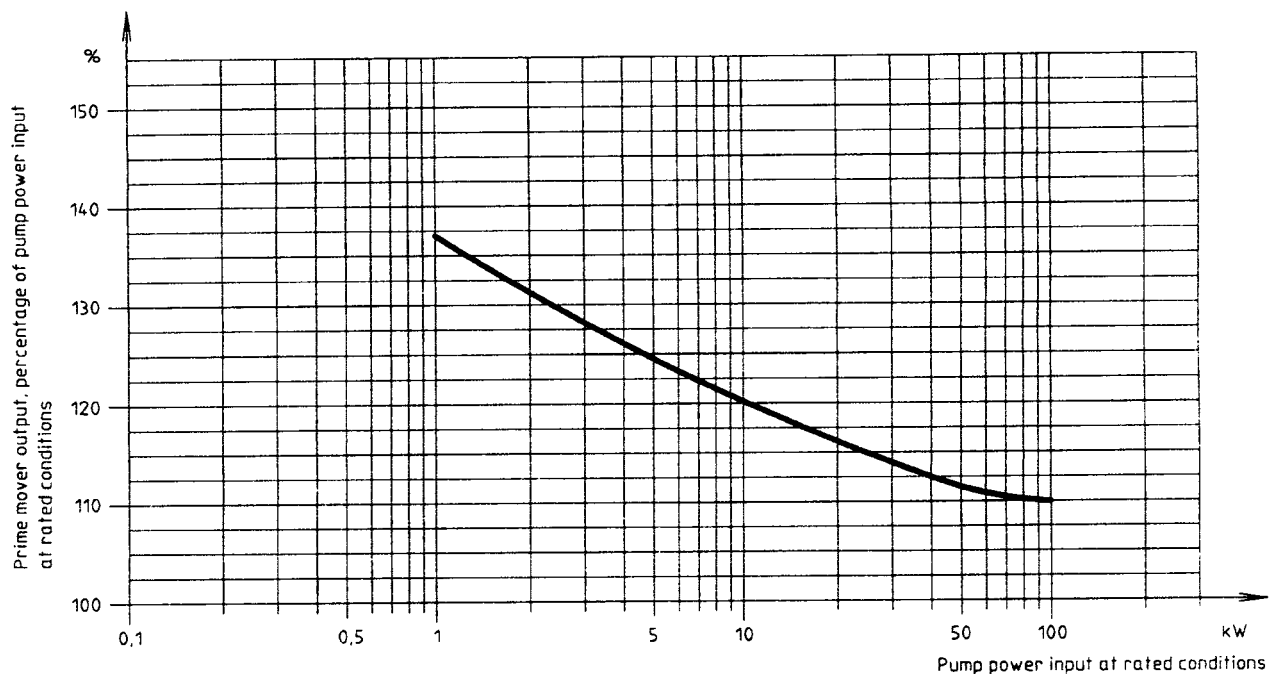


Figure 1 — Driver output percentage of rated pump power required in the range of 1 kW to 100 kW

4.3.1.2 A forcing phenomenon or exciting frequency may be less than, equal to, or greater than the synchronous frequency of the rotor. Such forcing frequencies may include but are not limited to the following phenomena:

- a) unbalance in the rotor system,
- b) oil film effects,
- c) internal rub frequencies,
- d) blade, vane, nozzle or diffuser passing frequencies,
- e) gear meshing and side band frequencies,
- f) coupling misalignment frequencies,
- g) loose rotor system component frequencies,
- h) hysteresis and friction whirl frequencies,
- i) boundary layer (vortex shedding),
- j) acoustic or aerodynamic effects,
- k) start-up conditions, for example, speed detents (under inertial impedance) or torsional deflections contributing to torsional resonances,
- l) number of cylinders, angle between banks, and whether two-or four-stroke in the case of internal combustion engines.

4.3.1.3 Actual critical speeds shall not encroach upon specified speed ranges.

Torsional modes of the complete unit shall be at least 10 % below any operating speed or

The first critical speed (in bending) shall preferably be at least 20 % above the maximum operating speed, except when it is not possible to design a stiff shaft pump, and shall have the purchaser's agreement.

For vertical shaft pumps, this applies particularly when the liquid handled contains an appreciable proportion of solid particles.

When it is not possible to design a stiff shaft pump and with the purchaser's agreement

- the first critical speed N_{c1} shall not exceed 0,37 ($= 1/2,7$) times the minimum operating speed N_{min} ,
- the second critical speed N_{c2} shall not be less than 1,2 times the maximum continuous speed N_{max} .

This can be illustrated as in Figure 2.

4.3.1.4 The separation margin of encroachment from all lateral modes (including rigid and bending) shall be at least

- a) 20 % over the maximum continuous speed for rigid rotor systems, or
- b) 15 % below any operating speed and 20 % above the maximum continuous speed for flexible-shaft rotor systems.

at least 10 % above the trip speed.

The separation margins specified are intended to prevent the overlapping of the critical response envelope into the operating speed range.

4.3.1.5 Slow roll, start-up and shutdown of rotating equipment shall not cause any damage as critical speeds are passed.

4.3.1.6 Support and bearing housing resonances of the driver and driven equipment shall not occur within the specified operating speed range or the specified separation margins.

4.3.1.7 When specified by purchaser, critical speeds shall be confirmed by test stand data, or if above test speeds, they shall be

- a) calculated damped values, or
- b) values determined by externally applied rotor excitations.

4.3.1.8 When specified by the purchaser, the calculations detailed in items a) and b) below shall be provided by the manufacturer/supplier. If the purchaser supplies the driving equipment, it shall be his responsibility to provide the data for these calculations:

- a) a lateral critical speed analysis to determine that the critical speeds of the driver are compatible with the critical speeds of the pump and that the combination is suitable for the specified operating speed range;
- b) a torsional vibration analysis of the pump-driver system and a transient torsional vibration analysis for synchronous motor-driven systems. The manufacturer/supplier shall be responsible for the satisfactory performance of the system.

In case of drive by internal combustion engine, the manufacturer/supplier of the latter is responsible for the analysis.

4.3.2 Balance and vibration

4.3.2.1 General

4.3.2.1.1 All major rotating components shall be balanced. When specified by the purchaser, the assembled rotors shall be balanced.

4.3.2.1.2 When specified by the purchaser, the manufacturer/supplier shall demonstrate that the pump can operate at the quoted minimum continuous stable flow without exceeding the vibration limits given in 4.3.2.2.

4.3.2.1.3 Pumps shall operate smoothly throughout their speed range in reaching rated speed, and to the overspeed limit in the case of turbine-driven units.

4.3.2.1.4 The smooth running of the pump (and its driver) after installation shall be the joint responsibility of the manufacturer/supplier and the purchaser. The units shall perform as well on their permanent foundation as they do on the manufacturer/supplier's test stand.

4.3.2.2 Horizontal pumps

Unfiltered vibration shall not exceed the vibration severity limits given in Table 1 when measured on the manufacturer/supplier's test facilities. These values are measured radially at the bearing housing at a single operating point at rated speed ($\pm 5\%$) and rated flow ($\pm 5\%$) when operating without cavitation. This can normally be achieved by balancing in accordance with grade G 6.3 of ISO 1940-1; for further information see ISO 5343 and ISO 8821.

Pumps with a special impeller, for example a single channel impeller, may exceed the limits given in Table 1. In such case the pump manufacturer/supplier should indicate this in his offer.

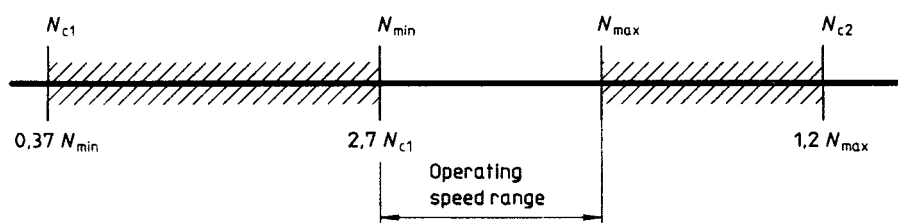


Figure 2 — Conditions of critical speed (see 4.3.1.3)

Table 1 — Limits of vibration severity for horizontal pumps with multivane impellers

Speed of rotation, N min^{-1}	Maximum rms values, in mm/s, of the vibration velocity for the shaft centreline height h_1^a	
	$h_1 \leq 225 \text{ mm}$	$h_1 > 225 \text{ mm}$
$N \leq 1\,800$	2,8	4,5
$1\,800 < N \leq 4\,500$	4,5	7,1

^a For horizontal foot-mounted pumps h_1 is the distance between baseplate area in contact with pump feet (including supports) and pump shaft centreline.

4.3.2.3 Vertical pumps

4.3.2.3.1 Vibration readings shall be taken on the top flange of the driver mount on vertical pumps with rigid couplings and near to the top pump bearing on vertical pumps with flexible couplings.

4.3.2.3.2 Vibration limits for both rolling and sleeve bearing pumps shall not exceed a velocity of 7,1 mm/s during shop test at rated speed ($\pm 5\%$), and rated flow ($\pm 5\%$) operating without cavitation.

4.4 Pressure-containing parts (see also 5.1)

4.4.1 Pressure-temperature rating

The maximum allowable working pressure of the pump at the most severe operating conditions must be clearly defined by the manufacturer/supplier. In no case shall the maximum allowable working pressure of the pump (casing and cover, including shaft seal housing and gland follower/end plate) exceed that of the pump flanges.

4.4.2 Pump casing

4.4.2.1 Pumps with radially split cases are required if any of the following operating conditions are specified:

- pumping temperature of 200 °C or higher (a lower temperature limit should be considered when thermal shock is probable);
- toxic pumped liquid or flammable liquid with a density of less than 0,7 kg/dm³ at the specified pumping temperature;
- flammable pumped liquid at a rated discharge gauge pressure above 70 bar.

NOTE 10 Pumps with axially split cases may be supplied for the conditions specified above with the specific approval of the purchaser. (it is recommended that the purchaser consider design details and previous manufacturer/supplier operating experience before approving pumps with axially split cases for these conditions. Maximum hydrostatic test, horizontal joint sealing technique, pump location, and the skill of field maintenance personnel should be factors in the decision.)

4.4.2.2 The thickness of the pressure casing shall be suitable for the maximum outlet working pressure plus allowances for head and speed increases at pumping temperature, and for hydrostatic test pressure at ambient temperature.

The maximum allowable casing working pressure shall be equal to or greater than the maximum outlet pressure.

Areas of double-casing, horizontal multistage (three or more stages) and axially split casing pumps normally subjected to inlet pressure need not be designed for discharge pressure. (The purchaser should consider installation of relief valves on the suction side of such installations.) **The purchaser shall specify if the vertical canned pump inlet can is to be suitable for maximum discharge pressure.** (This is advisable when two or more pumps are connected to a common discharge system.) The stress used in design for any given material shall not be in excess of the values given in specified material standards. The calculation methods for pressure-containing parts and the safety factors for the selected materials shall be in accordance with the relevant national rules.

The pressure-containing parts shall have a corrosion allowance of 3 mm unless a lower corrosion allowance can be accepted (e.g. for titanium).

4.4.2.3 The maximum discharge pressure shall apply to all parts referred to in the definition of pressure casing (see 3.31), except in the case of double-casing pumps, horizontal multistage (three or more stages) and axially split casing pumps.

4.4.2.4 The inner casing of double-casing pumps shall be designed to withstand the maximum internal differential pressure or 3,5 bar, whichever is greater.

4.4.2.5 If there is a risk of misalignment between pump and driver due to temperature differences or any other cause, precautions shall be taken to minimize this, for example centreline support, cooled pedestals, pre-alignment.

4.4.3 Materials

The materials used for pressure-containing parts shall depend on the liquid pumped, the pump configuration and the application of the pump (see clause 5).

4.4.4 Mechanical features

4.4.4.1 Dismantling

With the exception of vertical lineshaft pumps and ring-section type multistage pumps, the pump shall be designed to permit removal of the impeller, shaft, shaft seal and bearing assembly without disturbing the inlet and outlet flange connections.

For axially split pumps, lifting lugs or eyebolts shall be provided for lifting only the top half of the casing. Methods of lifting the assembled pump shall be specified by the manufacturer/supplier.

4.4.4.2 *Jackscrew and casing alignment dowels*

Jackscrew and casing alignment dowels shall be provided to facilitate dismantling and reassembly. When jackscrews are used as a means of parting contacting faces, one of the faces shall be relieved (counterbored or recessed) to prevent a leaking joint or improper fit caused by marring.

4.4.4.3 *Jackets*

Jackets for heating or cooling the casing or stuffing box, or both, are optional. Jackets shall be designed for an operating pressure of at least 6 bar at a temperature of 170 °C.

Jacket cooling systems shall be designed to positively prevent leakage of pumped liquid into the coolant. Coolant passage shall not open into casing joints.

4.4.4.4 *Casing gaskets*

Casing gaskets shall be of a design suitable for the working conditions and for hydrostatic test conditions at ambient temperature.

For radial split casings the casing-cover gaskets shall be confined on the atmospheric side to prevent blow-out.

Radially split casings (including mechanical seal end plate gaskets) shall have metal-to-metal fits with confined controlled compression gaskets.

4.4.4.5 *External bolting*

4.4.4.5.1 Bolts or studs that connect parts of the pressure casing, including shaft seal housing, shall be at least 12 mm diameter (ISO metric series).

The use of bolts or studs smaller than 12 mm diameter, if necessary due to space limitations, shall be agreed upon between purchaser and manufacturer/supplier. In such a case the bolting torque should be specified by the manufacturer/supplier.

4.4.4.5.2 The bolting selected (property class according to information in Annex L) shall be adequate for the maximum allowable working pressure and temperature and for normal tightening procedures. If at some points it is necessary to use a fastener of special quality, interchangeable fasteners for other joints shall be of the same special quality.

4.4.4.5.3 Tapped holes in pressure parts shall be held to a minimum. Sufficient metal in addition to the metal allowance for corrosion shall be left around and below the bottom of drilled and tapped holes in pressure sections of casings to prevent leakage.

4.4.4.5.4 To facilitate dismantling, internal bolting for vertical pumps shall be of a material fully resistant to corrosive attack by the fluid pumped.

4.4.4.5.5 Studded connections shall be supplied with studs installed. Blind stud holes should be drilled only deep enough to allow a preferred tap depth 1,5 times the major diameter of the stud.

4.4.4.5.6 Studs are preferred to cap screws.

4.4.4.5.7 A clearance shall be provided at bolting locations to permit the use of socket or box-type wrenches. The manufacturer/supplier shall supply any required special tools and fixtures.

4.5 **Branches (nozzles) and miscellaneous connections**

4.5.1 *General*

For the purpose of this International Standard, the terms branch and nozzle are synonymous.

This subclause is concerned with all fluid connections to the pump, whether for operation or maintenance.

4.5.2 *Vent, pressure-gauge and drain*

4.5.2.1 All pumps shall be provided with a vent connection unless the pump is made self-venting by the arrangement of the nozzles.

4.5.2.2 Preferably, no tapped openings shall be supplied in the suction or discharge passages or in other high-velocity areas of the pump unless they are essential for pump operation. **If drain, vent, or pressure gauge connections are required, they shall be specified by the purchaser in the enquiry and order.**

4.5.3 *Closures*

The material for the closures (plugs, blank/blind flanges etc.) shall be appropriate for the pumped liquid. Attention shall be paid to the suitability of material combinations to resist corrosion and to minimize the risk of seizure or galling of screw threads.

All openings exposed to the pumped liquid under pressure, including all shaft seal openings, shall be fitted with removable closures adequate for containing pressure.

4.5.4 *Auxiliary pipe connections*

4.5.4.1 All auxiliary pipe connections shall be of adequate material, size and thickness for the intended duty (see also 4.14).

4.5.4.2 Connections shall be at least 15 mm (outside diameter) for pumps with discharge openings of 50 mm and smaller. Connections shall be at least 20 mm (outside diameter) for pumps with discharge openings of 80 mm and larger, except that connections for seal flush piping and lantern rings may be 15 mm (outside diameter) without regard to pump size. When, because of space limitations, smaller connections must be used, all precautions shall be taken to protect them from damage and ensure their reliability.

4.5.5 Connection identification

All connections shall be identified on the installation drawing in accordance with their duty and function. If possible it is recommended that this identification also be applied on the pump, particularly for mechanical seals and for bearing lubrication and cooling (see Annex H).

4.6 External forces and moments on branches (inlet and outlet)

The method given in Annex B shall be used for pumps with flexible coupling unless another method is agreed upon between the purchaser and the manufacturer/supplier.

The purchaser shall calculate the forces and moments exerted by the piping on the pump.

The manufacturer/supplier shall verify that these loads are permissible for the pump under consideration. **If the loads are higher than those given in Annex B, the solution to the problem shall be agreed upon between purchaser and manufacturer/supplier.**

4.7 Inlet and outlet flanges and facings

4.7.1 Flanges shall conform to ISO 7005 except as specified in items a) to c) below:

- a) cast iron flanges shall be flat-faced;
- b) flat-faced flanges on casings other than cast iron are acceptable only with full raised-face thickness;
- c) flanges that are thicker or have a larger outside diameter than required by the specified standard are acceptable but shall be faced and drilled as specified.

4.7.2 Good seating of the bolt head and/or nut on the back face of cast flanges shall be ensured, if necessary by machining.

Bolt holes shall straddle the flange centreline.

4.8 Impellers

4.8.1 Impeller design

4.8.1.1 Impellers of closed, semi-open or open design may be selected according to the application.

4.8.1.2 The impeller, excluding wear rings, shall consist of one piece (such as cast or welded manufacture).

Impellers fabricated by other means are permissible in special cases, i.e. for small impeller outlet widths or if comprised of special material. This, however, requires agreement with the purchaser.

4.8.1.3 Impellers shall preferably have solid hubs.

4.8.1.4 If the pump shaft is wetted by the pumped fluid such that danger can occur or the product may become contaminated should the fluid become trapped in a confined space, then the design of the impeller and its securement shall be such that, when assembled onto the shaft, any enclosed spaces shall be freely drained by passageways of not less than 10 mm² cross-sectional area.

4.8.2 Securing of impellers

4.8.2.1 Impellers shall be secured against circumferential and axial movement when rotating in the intended direction. Pinning of impellers is not acceptable.

4.8.2.2 Overhung impellers shall be secured to the shaft by a cap screw or cap nut that does not expose shaft threads. Either securing device shall be threaded to tighten by liquid drag on the impeller during normal rotation, and a positive mechanical locking method (for example, a staked and corrosion-resistant set screw or a tongue-type washer) is required. Cap screws shall have fillets and a reduced diameter shank to decrease stress concentrations.

4.8.3 Axial adjustment

If field adjustment of impeller axial clearance is required, external means of adjustment shall be provided. If adjustment is achieved by axial movement of the rotor, attention must be paid to the possibly dangerous effect on the mechanical seal(s) (see also 4.11.6).

4.9 Wear rings

4.9.1 Wear rings should be fitted where appropriate. When wear rings are fitted they shall be renewable and securely locked to prevent rotation.

4.9.2 Mating wear surfaces of hardenable materials shall have a difference in Brinell hardness number of at least 50, unless both the stationary and rotating wear surfaces have a Brinell hardness number of at least 400 or if this difference in hardness is impossible to obtain with the material specified.

4.9.3 Renewable wear rings shall be held in place by a press fit with locking pins or threaded dowels (axial or radial) or by flanged and screwed methods. Other methods, including tack welding at three or more points, require the purchaser's approval.

4.10 Running clearances

4.10.1 When establishing running clearances between wear rings and between other moving parts, consideration shall be given to pumping temperatures, suction conditions, the character of the fluid handled, the expansion and galling characteristics of the materials, and hydraulic efficiency.

Clearances shall be sufficient to assure dependability of operation and freedom from seizure under normal operating conditions.

4.10.2 For cast iron, bronze, chromium hardened 11 % to 13 % and materials of similar low galling tendencies, the minimum clearances given in Table 2 shall be used. For diameters greater than 150 mm the minimum diametral clearance shall be 0,43 mm + 0,025 mm for each additional 25 mm of diameter or fraction thereof. For materials with greater galling tendencies and/or for operating temperatures above 260 °C, 0,125 mm shall be added to these diametral clearances.

When materials such as cast iron and/or bronze are used with cold and clean fluids such as water at temperatures below 50 °C, then the manufacturer/supplier may use clearances below those of Table 2.

Table 2 — Minimum running clearances

Dimensions in millimetres

Diameter of rotating member at clearance	Minimum diametral clearance
50	0,25
50 to 64,99	0,28
65 to 79,99	0,30
80 to 89,99	0,35
90 to 99,99	0,40
100 to 114,99	0,40
115 to 124,99	0,40
125 to 149,99	0,43

4.10.3 Interstage bushes on multistage pumps may have clearances to the manufacturer/supplier's standard, provided the clearances are stated in the proposal.

4.10.4 For vertical pumps, the running clearances specified in 4.10.2 shall not apply to the clearances of steady bearings or interstage bushings if materials of low galling tendencies are used. The clearances used shall be stated in the proposal.

4.11 Shaft and shaft sleeves

4.11.1 General

4.11.1.1 Shafts shall be of ample size and stiffness to:

- transmit the prime mover rated power,
- ensure satisfactory packing or seal performance,
- minimize wear and the risk of seizure,
- take due consideration of method of starting and inertia loading involved,
- take due consideration of radial thrust (static and dynamic).

4.11.1.2 The pump shaft of vertical pumps shall be where possible of one piece, unless otherwise approved by the purchaser (because of total shaft length or shipping restrictions).

4.11.2 Surface roughness

The surface roughness of the shaft or sleeve at the stuffing box, mechanical seal, and lubricant seal, if provided, shall be not greater than 0,8 µm R_a unless otherwise required for the seal. Measurement shall be in accordance with ISO 3274 (see also 4.11.7.1).

4.11.3 Shaft deflection

In order to have satisfactory packing or seal performance, avoid shaft breakage and prevent internal wear or seizure, the shaft stiffness for one- and two-stage horizontal and vertical in-line pumps shall limit the total shaft deflection under the most severe dynamic conditions over the complete head-capacity curve — with a maximum-diameter impeller and the specified speed and fluid — to a maximum of 50 µm at the face of the stuffing box (or at the mechanical seal face for built-in seal pumps) and to less than one-half the minimum diametral clearance at all bushes and wear rings. On in-line pumps, the stiffness of the total shaft system, including coupling and motor, shall be included in the calculations.

The required degree of shaft stiffness can be achieved by a combination of shaft diameter, shaft span or overhang and casing design (including use of dual volutes or diffusers). Support by conventional packing shall not be considered when determining shaft deflection.

4.11.4 Diameter

The dimensions of shaft ends should be in accordance with ISO/R 775, and the dimensions of key for shaft ends should be in accordance with ISO/R 773 where practical.

4.11.5 Shaft runout

4.11.5.1 Shafts shall be machined and properly finished throughout their length.

4.11.5.2 Manufacture and assembly of the shaft and sleeve, if fitted, should ensure that the runout (see 3.23) at a radial plane through the outer face of the stuffing box is not greater than 50 µm for nominal diameters smaller than 50 mm, not greater than 80 µm for nominal diameters 50 mm to 100 mm and not greater than 100 µm for nominal diameters greater than 100 mm.

4.11.6 Axial movement

Axial movement of the rotor permitted by the bearings shall not adversely affect the performance of the mechanical seal.

4.11.7 Shaft sleeves

4.11.7.1 Shaft sleeves when provided shall be locked or clamped to the shaft. The sleeves shall be of wear resistant and when necessary of corrosion- and erosion-resistant material. The outside surface of sleeves shall be suitable for the actual application (see also 4.11.2).

4.11.7.2 For shafts that require gaskets to pass over threads, the threads shall be at least 1,5 mm less than the internal diameter of the gasket, and the diameter transition shall be chamfered 15° to 20° to avoid damage to the gasket.

4.11.7.3 With the purchaser's approval, sleeves may be omitted for in-line pumps and for small horizontal pumps, provided the proposal so states and the shaft is constructed of a material that has equal wear and corrosion resistance and equal finish to that of a sleeve. If sleeves are not supplied, shafts or stub shafts shall include centres to permit refinishing.

4.11.7.4 On a pump arranged for packing, the end of the shaft sleeve assembly, if fitted, shall extend beyond the outer face of the packing gland follower. On a pump arranged for mechanical seals, the shaft sleeve shall extend beyond the seal end plate. On pumps employing an auxiliary seal or throttle bush, the shaft sleeve shall extend beyond the seal end plate. Leakage between the shaft and sleeve thus cannot be confused with leakage through box packing or mechanical seal faces.

4.11.7.5 On horizontal pumps, removable casing bushes and interstage shaft sleeves, or the equivalent, shall be provided at all interstage points.

4.11.7.6 On vertical pumps, renewable bushes shall be provided at all interstage and steady bearing points. However, the character of the fluid handled (for example, dirty or non-lubricating) should influence the need for corresponding shaft sleeves.

4.12 Bearings, bearing housings and lubrication

4.12.1 Bearings and bearing housings

4.12.1.1 Radial bearings shall be of the standard available design (ball, roller, sleeve or pivoted shoe) unless otherwise specified by the purchaser. Thrust bearings shall be either rolling or hydrodynamic as required.

4.12.1.2 Rolling bearings shall be selected and rated in accordance with ISO 76 and ISO 281. The minimum basic rating life L_{10} shall be 3 years (25 000 h), in continuous operation at rated pump conditions but not less than 16 000 h at maximum axial and radial loads and rated speed, within the allowed operating range.

4.12.1.3 Rolling bearings shall be retained on the shaft and fitted into housings according to the bearing manufacturer/supplier's instructions. Snap rings in direct contact with the bearings shall not be used for transmitting the thrust from the shaft to the inner face of the thrust bearing. Locknuts and lockwashers are preferred.

4.12.1.4 Hydrodynamic radial and/or thrust bearings shall be required under the following conditions:

- a) when DN factors are 300 000 or greater [the DN factor is the product of bearing size (bore) in mm and the rated speed in revolutions per minute];
- b) when the product of rated pump power input (in kilowatts) and rated speed (in revolutions per minute) is 2×10^6 or greater;
- c) when standard rolling bearings fail to meet the basic rating life L_{10} given in 4.12.1.2.

4.12.1.5 When allowed by the pump design and justified by the operating conditions, hydrodynamic radial bearings should be split for ease of assembly and shall be of the precision-bored sleeve or pad type, with babbited replaceable liners, shells or pads. Bearings shall be equipped with antirotation pins and shall be positively secured in the axial direction. The bearing design shall suppress hydrodynamic instabilities and provide sufficient damping to limit pump vibrations to the maximum specified amplitudes (see 4.3.2.2 and 4.3.2.3) while operating loaded or unloaded at specified operating speed, including operation at any critical frequency. The liners, pads or shells shall be in axially split housing and shall be replaceable. The removal of the top half of the casing of an axially split machine or the head of a radially split machine shall not be required for replacement of these elements. The bearing design shall not require removal of the coupling hub to permit replacement of the bearing liners, pads or shells.

4.12.1.6 Thrust bearings shall be sized for continuous operation under all specified conditions, including conditions such as maximum internal differential pressure. All loads shall be determined at design internal clearances. As a guide, hydrodynamic thrust bearings should be selected at no more than 50 % of the bearing manufacturer/supplier's rating, and shall be adequate for the pump design and application.

In addition to thrust from the rotor and any internal gear reactions due to the most extreme allowable conditions, the axial force transmitted through the flexible coupling shall be considered a part of the duty of any thrust bearing.

Thrust bearings must provide full load capabilities if the normal direction of rotation of the pump is reversed. Consideration should be given to type of driver, coupling and possible misalignment.

4.12.1.7 Hydrodynamic thrust bearings shall be designed for equal thrust capacity in both directions and arranged for continuous pressurized lubrication to each side. The thrust collar shall be replaceable when specified by the purchaser and shall be positively locked to the shaft to prevent fretting. When integral thrust collars are supplied, they shall be provided with a minimum of 3 mm of additional stock for refinishing should the collar be damaged. Both faces of the collar shall have a surface finish not exceeding $0,4 \mu\text{m } R_a$ and the axial total indicated runout of either face shall not exceed 13 μm .

4.12.1.8 Housings for oil-lubricated non-pressure-fed bearings shall be provided with 15 mm minimum thread tapped and plugged fill and drain openings. They shall be equipped with constant level sight-feed oilers connected to transparent containers (not subject to sunlight- or heat-induced opacity or deterioration), which shall be installed at an appropriate position on the bearing housing and positively locked in the operating position. **These oilers shall meet the purchaser's preference when specified.** A permanent indication of proper oil level shall be accurately located and clearly marked on the outside of the bearing housing with permanent metal tags, marks inscribed in the castings or other durable means, and shall state whether that level represents stationary or running conditions.

4.12.1.9 Bearing housings for hydrodynamic bearings designed for pressure lubrication shall be arranged to minimize foaming. The drain system shall be adequate to maintain the oil and foam levels below the shaft end seals. Oil temperature rise through the bearing and housings shall not exceed 30 °C under the most adverse specified operating conditions when oil inlet temperature is 40 °C. When oil inlet temperatures exceed 50 °C, special consideration shall be given to bearing design, oil flows and allowable temperature rise. Oil outlets from thrust bearings shall be tangential in the control ring, or in the thrust bearing cartridge if oil control rings are not used.

4.12.1.10 In order to prevent loss or contamination, gaskets or threaded connections shall not be used to separate cooling or heating fluids from lubricants.

4.12.1.11 All openings in the bearing housing, particularly the sealing between the bearing housing and the shaft, shall be designed to prevent the ingress of contaminants and the escape of the lubricant under normal operating conditions.

4.12.1.12 In hazardous areas, any device for sealing the bearing housing shall be designed not to be a source of ignition.

4.12.1.13 Housing for ring-oil-lubricated bearings should be provided with means to allow visual inspection of the oil rings while the pump is running.

4.12.1.14 If specified by the purchaser, the manufacturer/supplier shall supply oil heaters when the ambient or operating temperature dictates.

4.12.1.15 Bearing housings shall preferably be arranged so that bearings can be replaced without disturbing pump drives or mountings.

4.12.1.16 Sufficient cooling, including allowance for fouling, shall be provided to maintain oil temperature below 70 °C for drain oil in pressurized systems and below 80 °C for ring-oiled or splash systems based on specified operating conditions and 40 °C ambient temperature. Cooling coils (including fittings), if used, shall be of non-ferrous material and shall have no internal pressure joints or fittings. The coils shall be 1 mm minimum material thickness and 12 mm minimum tube outside diameter.

4.12.2 Guide bushes and bearings for vertical lineshaft pumps

4.12.2.1 The maximum spacing between shaft guide bushes shall be in accordance with Figure 3, except for cantilever-type pumps. If these bushes are product-lubricated, they must be suitably corrosion and abrasion resistant for the specified product and temperature.

4.12.2.2 Thrust bearings that are integral with the driver are covered in 4.2.1.2. Where the thrust bearing is integral with a vertical lineshaft pump, the paragraphs under 4.12.1 that relate to thrust bearings and housings shall apply.

4.12.3 Lubrication

4.12.3.1 Bearings and bearing housings shall be arranged for hydrocarbon oil or grease lubrication unless otherwise specified.

4.12.3.2 A pressure lubrication or oil mist lubrication system shall be supplied if specified by the purchaser or if recommended by the manufacturer/supplier and approved by the purchaser. Oil lubrication shall be ensured at constant level by internal arrangement.

4.12.3.2.1 If an external pressure lubrication system is required, it shall consist, as a minimum, of an oil pump with a suction strainer and/or filter, a supply-and-return system, an oil cooler when required, an oil reservoir, a full-flow filter and means for lubrication before the pump unit is running, and all necessary controls and instruments, including a low-oil-pressure alarm and shutdown (see G.6).

4.12.3.2.2 When specified by the purchaser, a removable steam-heating element external to the oil reservoir or a thermostatically controlled electrical immersion heater shall be provided for heating the charge capacity of oil prior to start-up in cold weather. The heating device shall have sufficient capacity to heat the oil in the reservoir from the specified minimum site ambient temperature to the manufacturer/supplier's required start-up temperature within 4 h when the lubrication oil system is running.

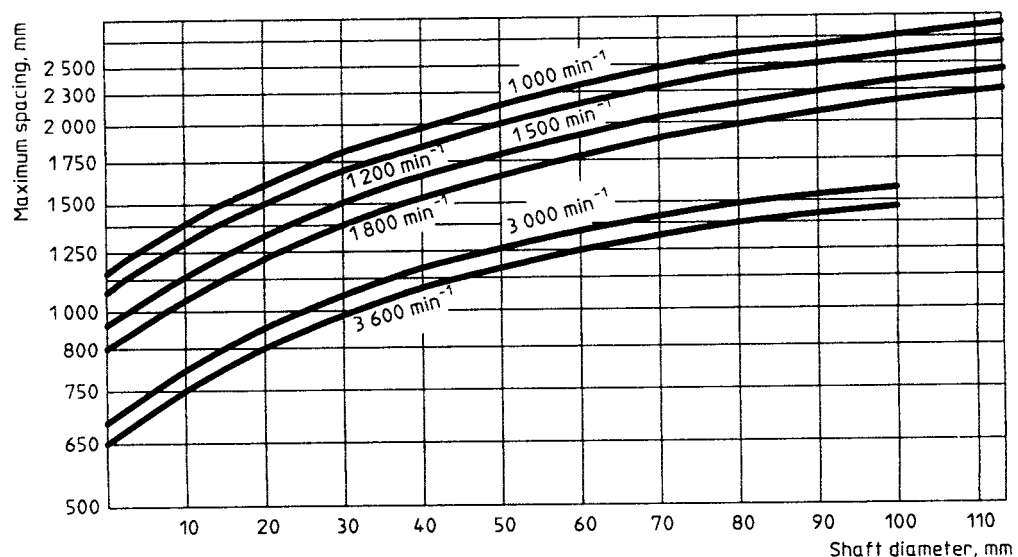


Figure 3 — Maximum spacing between shaft guide bushes (vertical lineshaft pumps)

4.12.3.2.3 An oil reservoir, with the characteristics specified in items a) through f) below, shall be supplied:

- a) the ability to provide a 3-minute minimum retention time, to avoid frequent refilling, and to provide adequate allowance for the contents of the system when drained;
- b) provision to eliminate air and to minimize flotation of foreign matter to pump suction;
- c) fill connections, level indicators and breathers suitable for outdoor use;
- d) sloped bottoms and connections for complete drainage;
- e) cleanout openings of practicable size;
- f) descaling, rust protection and permanent surface coating of the interior according to the manufacturer/supplier's standard procedure, unless otherwise specified.

4.12.3.2.4 Flingers or oil rings shall have a submergence above the lower edge of a flinger or above the lower edge of the bore of an oil ring. Oil flingers shall have mounting hubs to maintain concentricity and shall be positively secured to the shaft.

4.12.3.2.5 The manufacturer/supplier shall state in the operating manual the amount of and the specification for the lubricating oil required, taking into consideration the ambient and service conditions.

4.12.3.2.6 Refer to 4.14.3 for lubricating-oil piping requirements.

4.12.3.3 Where regreasable bearings are used, grease relief shall be provided.

4.13 Shaft sealing

4.13.1 General

When the pump shaft has to be sealed, the pump design shall permit the use of one or more of the following alternatives:

- soft packing (P),
- single mechanical seal (S),
- multiple mechanical seals (D),

as shown in Annex F. **If it is necessary to interchange one or the other alternative, this shall be specified by the purchaser. Other means of sealing devices (e.g. labyrinth seals, hydrodynamic seals, magnetic couplings) shall be mutually agreed upon.** Quench arrangements (Q), which in certain cases can become necessary, are also shown in Annex F. Devices shall be provided for limiting, collecting and draining all liquid leakage from the sealed area, in particular if the mechanical seals have controlled leakage.

The following information shall be given in the data sheet (see Annex A):

- shaft seal arrangement (as specified in Annex F);
- for mechanical seals:
 - type: balanced (B)
unbalanced (U)
bellows (Z)
 - size: nominal shaft or sleeve diameter, in millimetres, based upon shaft diameter passing through stationary ring (see ISO 3069);
- for stuffing box:
 - size: diameter of seal cavity as specified in ISO 3069.

4.13.2 Operating criteria for selection

The principal operating criteria for selection of mechanical seals and soft packings are:

- chemical and physical properties and the nature of the pumped liquid;
 - minimum and maximum expected sealing pressures;
 - temperature of the liquid at the seal;
 - special operating conditions (including start-up, shutdown, thermal and mechanical shocks, etc.);
 - shaft diameter and speed;
- and as a supplementary criterion for mechanical seals:
- direction of rotation of the pump.

4.13.3 Mechanical seals

4.13.3.1 Type and arrangement

Mechanical seals shall be of the balanced type. **Unbalanced seals shall only be supplied when specified or approved by the purchaser.**

This International Standard does not cover the design of the components of the mechanical seal; however, the components shall be suitable to withstand operating conditions specified in the data sheet (see Annex A).

Mechanical seal design shall take account of axial adjustment of shaft and shaft movement during normal operation.