

Petroleum, chemical and gas service industries — Centrifugal compressors

The European Standard EN ISO 10439:2002 has the status of a
British Standard

ICS 75.180.20

National foreword

This British Standard is the official English language version of EN ISO 10439:2002. It is identical with ISO 10439:2002.

The UK participation in its preparation was entrusted to Technical Committee MCE/8, Compressors, pneumatic tools, pneumatic machines and vacuum technology, which has the responsibility to:

- aid enquirers to understand the text;
- present to the responsible international/European committee any enquiries on the interpretation, or proposals for change, and keep the UK interests informed;
- monitor related international and European developments and promulgate them in the UK.

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

Users are cautioned that this Standard is based on a now outdated edition of API Standard 617. In particular, improved and workable provisions to replace clause 4.9 on “Dynamics” have been developed and are incorporated in ISO/FDIS 10437:2003. The responsible ISO committee (TC118/SC1) has agreed to review this and other specific provisions at the next revision of the Standard.

Cross-references

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Summary of pages

This document comprises a front cover, an inside front cover, the EN ISO title page, the EN ISO foreword page, the ISO title page, pages ii to vi, pages 1 to 119, the Annex ZA page, an inside back cover and a back cover.

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Industries du pétrole, de la chimie et du gaz -
Compresseurs centrifuges (ISO 10439:2002)

This European Standard was approved by CEN on 7 October 2002.

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Foreword

This document (EN ISO 10439:2002) has been prepared by Technical Committee ISO/TC 118 "Compressors, pneumatic tools and pneumatic machines" in collaboration with Technical Committee CEN/TC 12 "Materials, equipment and offshore structures for petroleum and natural gas industries", 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of an identical text or by endorsement, at the latest by April 2003, and conflicting national standards shall be withdrawn at the latest by April 2003.

According to the 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, Malta, Netherlands, Norway, Portugal, Spain, Sweden, Switzerland and the United Kingdom.

Endorsement notice

The text of ISO 10439:2002 has been approved by CEN as EN ISO 10439:2002 without any modifications.

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

INTERNATIONAL STANDARD

ISO
10439

First edition
2002-10-15

Petroleum, chemical and gas service industries — Centrifugal compressors

*Industries du pétrole, de la chimie et des services du gaz — Compresseurs
centrifuges*



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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

Attention is drawn to the possibility that some of the elements of this International Standard may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 10439 was prepared by a Joint Working Group of Technical Committees ISO/TC 118, *Compressors, pneumatic tools and pneumatic machines*, and ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, Subcommittee SC 6, *Processing equipment and systems*.

Annexes C, D and G form a normative part of this International Standard. Annexes A, B, E, F, H, I and J are for information only.

Introduction

This International Standard is based on the sixth edition of the American Petroleum Institute standard API 617.

Users of this International Standard should be aware that further or differing requirements may be needed for individual applications. This International Standard is not intended to inhibit a vendor from offering, or the purchaser from accepting, alternative equipment or engineering solutions for the individual application. This may be particularly applicable where there is innovative or developing technology. Where an alternative is offered, the vendor should identify any variations from this International Standard and provide details.

Petroleum, chemical and gas service industries — Centrifugal compressors

1 Scope

This International Standard specifies requirements and gives recommendations for the design, materials, fabrication, inspection, testing and preparation for shipment of centrifugal compressors for use in the petroleum, chemical and gas service industries. It is not applicable to machines that develop less than 35 kPa above atmospheric pressure, nor is it applicable to packaged, integrally geared centrifugal air compressors, which are covered in ISO 10442.

NOTE In this International Standard, where practical, US customary units have been included in brackets for information.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

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

ISO 3744, *Acoustics — Determination of sound power levels of noise sources using sound pressure — Engineering method in an essentially free field over a reflecting plane*

ISO 3977-5, *Gas turbines — Procurement — Part 5: Applications for petroleum and natural gas industries*

ISO 5389, *Turbocompressors — Performance test code*

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

ISO 8821, *Mechanical vibration — Balancing — Shaft and fitment key convention*

ISO 9614 (both parts), *Acoustics — Determination of sound pressure levels of noise sources using sound intensity*

ISO 10437, *Petroleum and natural gas industries — Special-purpose steam turbines for refinery service*

ISO 10438 (all parts), *Petroleum and natural gas industries — Lubrication, shaft sealing and control-oil systems*

ISO 10441, *Petroleum and natural gas industries — Flexible couplings for mechanical power transmission — Special purpose applications*

ISO 13691, *Petroleum and natural gas industries — High-speed special-purpose gear units*

1) To be published. (Revision of ISO 1940-1:1986)

IEC 60079-10, *Electrical apparatus for explosive gas atmospheres — Part 10: Classification of hazardous areas*

API²⁾ RP 550, *Manual on installation of refinery instruments and control systems*

API Std 670, *Machinery protection systems, fourth edition*

ASME³⁾ PTC 10, *Test code on compressors and exhausters*

ASTM⁴⁾ A 388/A 388M, *Standard practice for ultrasonic examination of heavy steel forgings*

ASTM A 578/A 578M, *Standard specification for straight-beam ultrasonic examination of plain and clad steel plates for special applications*

ASTM A 609/A 609M, *Standard practice for casting, carbon, low-alloy, and martensitic stainless steel, ultrasonic examination thereof*

ASTM E 94, *Standard guide for radiographic examination*

ASTM E 165, *Standard test method for liquid penetrant examination*

ASTM E 709, *Standard guide for magnetic particle examination*

ISA⁵⁾ RP 12.4, *Pressurized enclosures*

NACE⁶⁾ MR 0175, *Sulfide stress cracking resistant metallic materials for oilfield equipment*

NFPA⁷⁾ 496, *Standard for purged and pressurized enclosures for electrical equipment*

3 Terms and definitions

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

3.1

alarm condition

preset value of a parameter at which an alarm is actuated to warn of a condition requiring corrective action

3.2

axially split

casing or other component in which the main joint is parallel to the axis of the shaft

3.3

compressor rated point

point on the 100 % speed curve at the highest capacity of any specified operating point

NOTE The use of the word “design” in any term (such as design power, design pressure, design temperature, or design speed) should be avoided in the purchaser's specification. This terminology should be used only by the equipment designer and manufacturer.

2) American Petroleum Institute.

3) American Society of Mechanical Engineers.

4) American Society for Testing and Materials.

5) Instrument Society of America.

6) US National Association of Corrosion Engineers.

7) US National Fire Protection Association.

3.4**head**

specific compression work

3.5**inlet volume flow**

volume flow rate determined at the conditions of pressure, temperature, compressibility and gas composition, including moisture, at the compressor inlet flange

3.6**maximum allowable temperature**

maximum continuous temperature for which the manufacturer has designed the equipment (or any part to which the term is referred) when handling the specified fluid at the specified pressure

3.7**maximum allowable working pressure**

maximum continuous pressure for which the manufacturer has designed the equipment (or any part to which the term is referred) when it is operating at the maximum allowable temperature

3.8**maximum continuous speed**

highest rotational speed at which the machine is capable of continuous operation

3.9**maximum sealing pressure**

highest pressure the seals are required to seal during any specified static or operating conditions and during start-up and shutdown

3.10**minimum allowable speed**

lowest speed at which the manufacturer's design will permit continuous operation

3.11**normal operating point**

point at which usual operation is expected and optimum efficiency is desired

NOTE This will usually be the point at which the vendor certifies that performance is within the tolerances stated in this International Standard.

3.12**normal speed**

speed corresponding to the requirements of the normal operating point

3.13**100 % speed**

highest speed required for any specified operating point

3.14**pressure design code**

recognized pressure vessel standard specified or agreed by the purchaser (e.g. ASME VIII)

3.15**radially split**

casing or other component in which the main joint is perpendicular to the axis of the shaft

3.16**stability**

difference in inlet volume flow (as percentage of rated inlet volume flow) between the rated inlet volume flow and the surge point at rated speed

3.17

settling out pressure

pressure of the compressor system when the compressor is shut down

3.18

shutdown condition

preset value of a parameter requiring automatic or manual shutdown of the system

3.19

trip speed

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

NOTE For constant speed motor drivers, this is the speed corresponding to the synchronous speed of the motor at the maximum frequency of the electrical supply.

3.20

turndown

percentage of change in inlet volume flow (referred to rated inlet volume flow) between the rated inlet volume flow and the surge point inlet volume flow at the rated head, when the unit is operating at rated suction temperature and gas composition

3.21

unit responsibility

responsibility for coordinating the technical aspects of the equipment train and all auxiliary systems

4 Basic design

4.1 General

4.1.1 A bullet (•) at the beginning of a clause indicates that the purchaser is required to make a decision or provide information. This information should be indicated on the data sheets (see annex A).

4.1.2 The equipment (including auxiliaries) covered by this International Standard shall be designed and constructed for a minimum service life of 20 years and at least 3 years of uninterrupted operation.

4.1.3 Unless otherwise specified, the compressor vendor shall assume unit responsibility.

4.1.4 The compressor shall be designed to deliver required head and capacity at the normal operating point without negative tolerance. The input power at the above condition shall not exceed 104 % of the predicted value for this point.

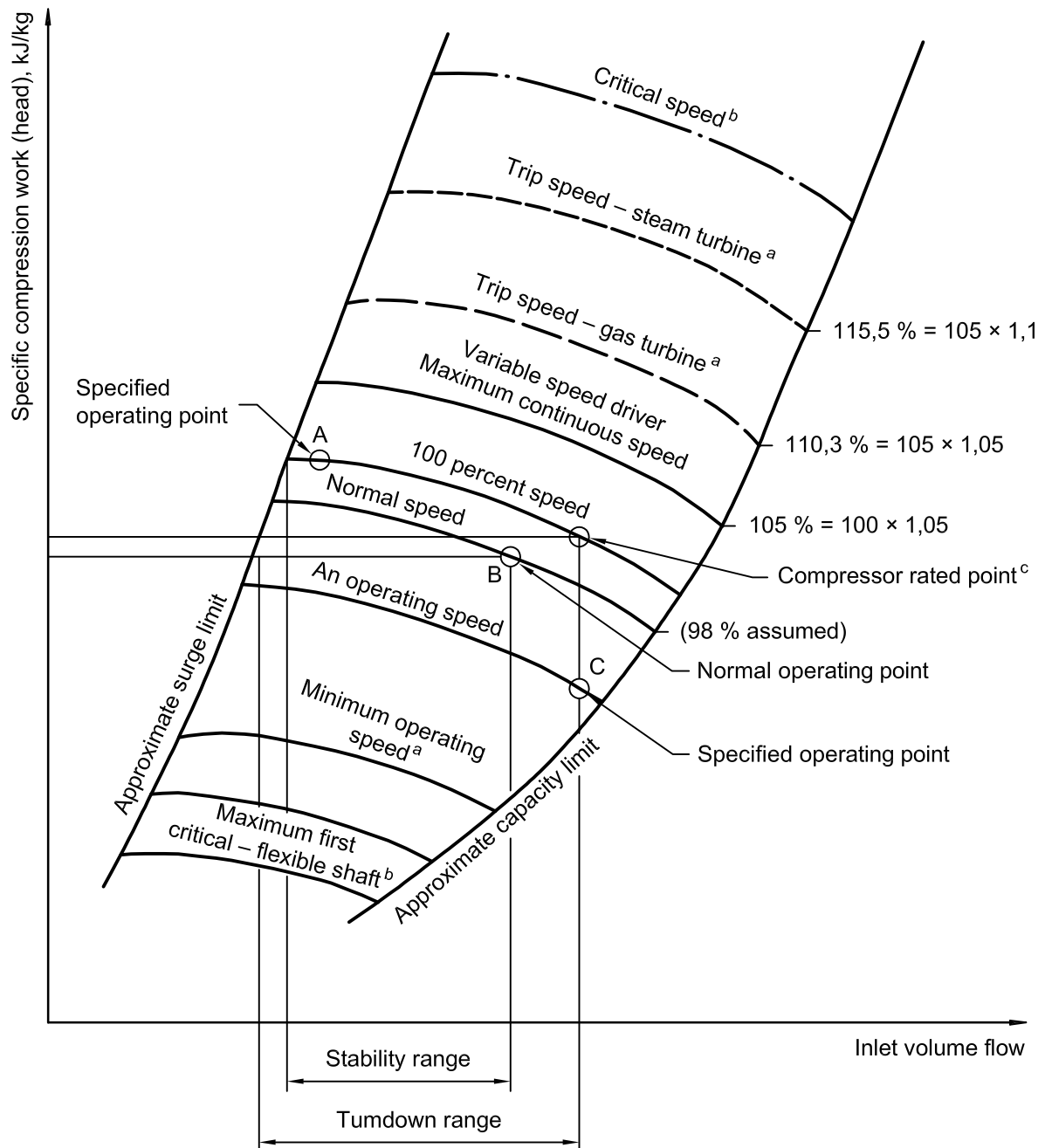
NOTE See the optional performance test criteria in 6.3.6.2 and handling of excess head for constant speed drivers.

4.1.5 The head versus capacity characteristic curve (see Figure 1) shall rise continuously from the rated point to the predicted surge. The compressor, without the use of a bypass, shall be suitable for continuous operation at any capacity at least 10 % greater than the predicted approximate surge capacity shown in the proposal.

4.1.6 Cooling water systems, if required, shall be designed for the conditions specified in Table 1 unless otherwise specified. Provision shall be made for complete venting and draining of the system.

The vendor shall notify the purchaser if the criteria for minimum temperature rise and velocity over heat exchange surfaces result in a conflict. The criterion for velocity overheat exchange surfaces is intended to minimize the use of cooling water. The purchaser shall approve the final selection.

4.1.7 The arrangement of the equipment, including piping and auxiliaries, shall be developed jointly by the purchaser and the vendor. The arrangement shall provide adequate clearance areas and safe access for operation and maintenance.



The head versus capacity curve at 100 % speed shall extend to at least 115 % capacity of the CRP. Head versus capacity curves at other speeds shall be extended to equivalent capacity at each speed. For example, the head versus capacity curve at 105 % speed shall be extended to 1,05 times 1,15 times capacity of the CRP; the head versus capacity curve at 90 % speed shall be extended to 0,9 times 1,15 times capacity at the CRP; and so on. These points define the "approximate capacity limit" curve.

Except where specific numerical relationships are stated, the relative values implied in this figure are assumed values for illustration only.

The 100 % speed is determined from the operating point requiring the highest head — point A in the illustration.

The compressor rated point (CRP) is the intersection on the 100 % speed line corresponding to the highest flow of any operating point — point C in the illustration.

^a Refer to the applicable standard for the compressor driver (e.g. ISO 10437 or ISO 3977-5) for trip speed and minimum operating speed limits.

^b See 4.9 for allowable margins of critical speeds to operating speeds.

^c The maximum continuous speed shall be 105 % for variable speed drivers. The maximum continuous speed shall be the speed corresponding to the synchronous speed of the motor.

Figure 1 — Illustration of terms

Table 1 — Cooling water systems — Design requirements

Velocity over heat exchange surfaces	1,5 m/s to 2,5 m/s (5 ft/s to 8 ft/s)
Maximum allowable gauge working pressure	≥ 500 kPa (75 psi)
Test gauge pressure	≥ 750 kPa (110 psi)
Maximum inlet temperature	30 °C (90 °F)
Maximum temperature rise	20 K (35 °F)
Fouling factor on water side	0,35 m ² K/kW (0,002 h.ft ² °F/Btu)
Maximum pressure drop	100 kPa (15 psi)
Maximum outlet temperature	50 °C (120 °F)
Minimum temperature rise	10 K (20 °F)
Shell corrosion allowance	3,0 mm (1/8 in)

4.1.8 All equipment shall be designed to permit rapid and economical maintenance. Major parts such as casing components and bearings housings shall be designed and manufactured to ensure accurate alignment on reassembly. This may be accomplished by the use of shouldering, cylindrical dowels or keys.

4.1.9 The inner casing of radially split barrel type compressors shall be designed for easy withdrawal from the outer shell and easy disassembly for inspection or replacement of parts.

- **4.1.10** The equipment, including all auxiliaries, shall be suitable for operation under the environmental conditions specified by the purchaser. These conditions shall include whether the installation is indoors (heated or unheated) or outdoors (with or without a roof), maximum and minimum temperatures, unusual humidity, and dusty or corrosive conditions. For the purchaser's guidance, the vendor shall list in the proposal any special protection that the purchaser is required to supply.

4.1.11 Control of the sound pressure level (SPL) of all equipment furnished shall be a joint effort of the purchaser and the vendor. The equipment furnished by the vendor shall conform to the maximum allowable sound pressure level specified by the purchaser.

- **4.1.12** The purchaser shall advise the vendor of any requirements for liquid injection.

4.1.13 Equipment shall be designed to run without damage to the trip speed and the maximum allowable working pressure.

4.1.14 The machine and its driver shall perform on the test stand and on their permanent foundation within the specified acceptance criteria. After installation, the performance of the combined units shall be the joint responsibility of the purchaser and the vendor having unit responsibility.

4.1.15 Many factors (such as piping loads, alignment at operating conditions, supporting structure, handling during shipment, and handling and assembly at site) may adversely affect site performance. To minimize the influence of these factors, the vendor shall review and comment on the purchaser's piping and foundation drawings. If specified, the vendor's representative shall

- observe a check of the piping performed by parting the flanges,
- check alignment at the operating temperature, and
- be present during the initial alignment check.

- **4.1.16** Motors and all other electrical components and installations shall be suitable for the area classification (zone) specified by the purchaser on the data sheets (see annex A), shall meet the requirements of IEC 60079-10 and shall comply with applicable local codes and regulations specified by the purchaser.

4.1.17 Spare parts for the compressor and all furnished auxiliaries shall meet all the criteria of this International Standard.

- **4.1.18** If specified, the compressor or compressors shall be suitable for field running on air. Performance parameters, including any required precautions, shall be mutually agreed upon by the purchaser and the vendor.

4.1.19 A guide to centrifugal compressor nomenclature is given in annex F.

- **4.1.20** The pressure design code shall be specified or agreed by the purchaser.

Pressure components shall comply with the pressure design code and the supplemental requirements given in this International Standard.

- **4.1.21** The purchaser and the vendor shall agree on the measures to be taken in order to comply with governmental regulations, ordinances or rules that are applicable to the equipment.

4.2 Casings

4.2.1 The thickness of the casing shall be suitable for the maximum allowable working and test pressures and shall include at least a 3 mm corrosion allowance. The thickness of the casing shall not be less than that calculated in accordance with the pressure design code.

4.2.2 The equipment feet shall be provided with vertical jackscrews and shall be drilled with pilot holes that are accessible for use in final doweling.

4.2.3 Supports and alignment bolts shall be rigid enough to permit the machine to be moved by the use of its lateral and axial jackscrews.

- **4.2.4** The maximum allowable working pressure of the casing shall be at least equal to the specified relief valve setting; if a relief valve setting is not specified or if a relief valve is not installed, the maximum allowable working pressure shall be at least 1,25 times the maximum specified discharge pressure.

NOTE System protection is normally provided by the purchaser.

4.2.5 Casings designed for more than one maximum allowable pressure level (split-pressure-level casings) are not permitted unless specifically approved by the purchaser, and if so, the vendor shall define the physical limits and the maximum allowable working pressure of each part of the casing.

4.2.6 Each axially split casing shall be sufficiently rigid to allow removal and replacement of its upper half without disturbing rotor-to-casing running clearances and bearing alignment.

4.2.7 Casings shall be made of steel for the following:

- air or non-flammable gas at a maximum allowable gauge working pressure above 2 500 kPa (360 psi);
- air or non-flammable gas at a calculated discharge temperature that is over 260 °C (500 °F) at maximum continuous speed at any point within the operating range;
- flammable or toxic gas.

4.2.8 Cast iron or other materials of construction may be offered for operating conditions other than those specified in 4.2.7.

4.2.9 Unless otherwise specified, casings shall be radially split if the partial pressure of hydrogen (at maximum allowable gauge working pressure) exceeds 1 400 kPa (200 psi).

NOTE The partial pressure of hydrogen is calculated by multiplying the highest specified mole (volume) percent of hydrogen by the maximum allowable working pressure.

4.2.10 Axially split casings shall use a metal-to-metal joint (with a suitable joint compound) that is tightly maintained by suitable bolting. Gaskets (including string type) shall not be used on the axial joint. O-rings with ring grooves machined into the flange facing of an axially split casing joint may be used with the purchaser's approval. If

gasketed joints are used between the end covers and the cylinder of radially split casings, they shall be securely maintained by confining the gaskets. Gasket materials shall be suitable for all specified service conditions.

4.2.11 Jackscrews, guide rods and cylindrical casing alignment dowels shall be provided to facilitate disassembly and reassembly. If jackscrews are used as a means of parting contacting faces, one of the faces shall be relieved (counter-bored or recessed) to prevent a leaking joint or an improper fit caused by marring of the face. Guide rods shall be of sufficient length to prevent damage to the internals or casing studs from the casing during disassembly and reassembly. Lifting lugs or eyebolts shall be provided for lifting only the top half of the casing. Methods of lifting the assembled machine shall be specified by the vendor.

4.2.12 The use of threaded holes in pressure parts shall be minimized. To prevent leakage in pressure sections of casings, metal equal in thickness to at least half the nominal bolt diameter, in addition to the allowance for corrosion, shall be left around and below the bottom of drilled and threaded holes. The depth of threaded holes shall be at least 1,5 times the stud diameter.

4.2.13 The sealing of stud clearance holes to prevent leakage is not permitted.

4.2.14 The machined finish of the compressor mounting surfaces shall be 3,2 μm to 6,4 μm (125 micro-inches to 250 micro-inches) arithmetical average roughness (R_a). Hold-down or foundation bolt holes shall be drilled perpendicular to the mounting surface or surfaces and spot faced to a diameter three times that of the hole.

4.2.15 Studded connections shall be furnished with studs installed. Blind stud holes should be drilled only deep enough to allow a preferred tap depth of 1,5 times the major diameter of the stud; the first 1,5 threads at both ends of each stud shall be removed.

4.2.16 External and internal bolting shall be furnished as follows.

- a) Bolting external to the casing shall be in accordance with the pressure design code. Internal bolting shall have the same thread form.
- b) Studs should be used instead of cap screws (external only).
- c) Adequate clearance shall be provided at bolting locations to permit the use of socket or box wrenches (external only).
- d) Socket, slotted-nut or spanner-type bolting shall not be used unless specifically approved by the purchaser (external only).

4.3 Interstage diaphragms and inlet guide vanes

- **4.3.1** Interstage diaphragms and inlet guide vanes shall be suitable for all specified operating conditions, start-up, shutdown, trip-out, settling out and momentary surge. If intermediate main process connections are used, the purchaser shall specify the maximum and minimum pressures at each connection. The vendor shall confirm that the diaphragms furnished are suitable for the maximum differential pressure.

4.3.2 Internal joints shall be designed to minimize leakage and permit easy disassembly.

4.3.3 Renewable labyrinths shall be provided at all internal close clearance points to minimize internal leakage. These shall be easily replaceable.

4.3.4 Diaphragms shall be axially split unless otherwise approved by the purchaser. The diaphragms shall be furnished with threaded holes for eyebolts or with another means to facilitate removal.

4.3.5 If diaphragm cooling is specified, the top and bottom halves of axially split diaphragms shall have independent cooling passages. Each coolant inlet and outlet connection shall be manifolded at both the top and bottom of each casing.

4.4 Casing connections

4.4.1 General

4.4.1.1 All process gas connections to the casing shall be suitable for the maximum allowable working pressure of the casing (see 4.2.4).

4.4.1.2 All of the purchaser's connections shall be accessible for maintenance without the machine being moved.

4.4.1.3 Connections, pipe, valves and fittings of nominal pipe size DN 32 (NPS 1¼), DN 65 (NPS 2½), DN 90 (NPS 3½) or DN 125 (NPS 5) shall not be used.

4.4.1.4 Connections welded to the casing shall meet the material requirements of the casing, including impact values, rather than the requirements of the connected piping.

4.4.1.5 All welding of connections shall be done before hydrostatic testing (see 6.3.2).

4.4.2 Main process connections

- **4.4.2.1** Inlet and outlet connections shall be flanged or machined and studded and oriented as specified in the data sheets (see annex A). Inlet and outlet connections for barrel type compressors shall be located in the outer casing, not in the end covers. On radially split overhung design compressors, the process inlet connection may be in the end cover.

- **4.4.2.2** Flanges shall be in accordance with the pressure design code. If specified, the vendor shall supply all mating flanges, including studs and nuts.

4.4.2.2.1 Flat-faced flanges with full raised-face thickness may be used on casings other than cast iron.

4.4.2.2.2 Unless otherwise specified, flanges that are thicker or have a larger outside diameter than that required by the pressure design code may be used.

4.4.2.3 Cast iron flanges shall be flat-faced and conform to the dimensional requirements of ISO 7005-2. Class 125 flanges shall have a minimum thickness equal to class 250 for sizes DN 200 and smaller.

4.4.2.4 The concentricity of the bolt circle and the bore of all casing flanges shall be such that the area of the machined gasket-seating surface is adequate to accommodate a complete standard gasket without protrusion of the gasket into the fluid flow.

4.4.2.5 The finish of all flanges and nozzles shall conform to the requirements of 4.4.2.2 as applicable to the material furnished, including flange finish roughness requirements.

4.4.3 Auxiliary connections

4.4.3.1 Auxiliary connections may include, but are not limited to, those for vents, liquid injection, drains (see 4.4.3.2) water cooling, "lube and seal" oil, flushing, buffer gas and the balance piston cavity.

- **4.4.3.2** For axially split casings, the vendor shall provide connections for complete drainage of all gas passages. For radially split casings, the drains shall be located at the lowest point of each inlet section, the lowest point of the section between the inner and outer casings and the lowest point of each discharge section. If specified, individual stage drains, including a drain for the balance piston cavity, shall be provided.

4.4.3.3 Flanges shall be in accordance with the pressure design code.

4.4.3.4 Auxiliary connections shall be at least nominal pipe size DN 20 (NPS ¾) (see 4.4.1.3) and shall be socket welded and flanged, or machined and studded. For socket welded construction, a 1,5 mm gap, as measured prior to welding, shall be left between the pipe end and the bottom of the socket in the casing.

4.4.3.5 If socket welded and flanged or machined and studded openings cannot be provided, threaded openings in sizes DN 20 (NPS $\frac{3}{4}$) to DN 40 (NPS $1\frac{1}{2}$) may be used if approved by the purchaser. These threaded openings shall be installed as follows.

- a) Threaded openings and bosses for pipe threads shall comply with the pressure design code.
- b) Pipe threads shall be taper threads (e.g. ISO 7 or ASME B 1.20.1) and shall comply with the pressure design code.
- c) Threaded connections shall not be used for flammable or toxic fluids. Where threaded joints are permitted, they shall not be seal welded.

4.4.3.6 A pipe nipple, which should not be more than 150 mm (6 in) long, shall be installed in a threaded or socket weld opening. Pipe nipples shall be a minimum of schedule 160 seamless for threaded connections and schedule 80 for socket welded connections. Each pipe nipple shall be provided with a welding neck, socket weld or slip-on flange.

NOTE The schedules are as specified in ASME B 36.10M.

4.4.3.7 Tapped openings not connected to piping shall be plugged with solid steel plugs. As a minimum these plugs shall meet the material requirements of the casing. Plugs that may later require removal shall be of corrosion-resistant material. Threads shall be lubricated. Tape shall not be applied to threads of plugs inserted into oil passages. Plastic plugs shall not be used.

4.5 External forces and moments

4.5.1 The compressor shall be designed to withstand external forces and moments on each nozzle calculated in accordance with Annex G. The vendor shall furnish the allowable forces and moments for each nozzle in tabular form together with the co-ordinates.

4.5.2 Casing and supports shall be designed to have sufficient strength and rigidity to limit coupling misalignment caused by imposing allowable forces and moments to 50 μm (0,002 in).

4.6 Rotating elements

4.6.1 Shafts shall be made of one-piece heat-treated steel, suitably machined. Shafts that have a finished diameter larger than 200 mm (8 in) shall be forged steel. Shafts that have a finished diameter of 200 mm (8 in) or less shall be forged steel or, with the purchaser's approval, hot rolled bar stock, providing that the bar stock meets all quality and heat treatment criteria established for shaft forgings.

4.6.2 Shaft ends for coupling fits shall be in accordance with ISO 10441.

4.6.3 Unless other shaft protection is approved by the purchaser, renewable shaft sleeves shall be furnished at all close clearance points unless rotating seals are used. These sleeves shall be made of a material that is corrosion-resistant in the specified service. The sleeves under close clearance bushing end seals shall be suitably treated to resist wear, and sealed to prevent leakage between the shaft and sleeve (see 4.11.1.7 for limitations).

4.6.4 The design of shaft-sleeve-impeller assemblies shall not create temporary or permanent distortions of the rotor assembly. The method of attaching the impeller shall adequately maintain concentricity and balance under all specified operating conditions, including overspeed to trip speed.

4.6.5 The rotor shaft sensing areas to be observed by radial vibration probes shall be concentric with the bearing journals. All shaft sensing areas (both radial vibration and axial position) shall be free from stencil and scribe marks or any other surface discontinuity, such as an oil hole or a keyway. These areas shall not be metallised, sleeved, or plated. The final surface finish shall be 0,4 μm to 0,8 μm (16 to 32 micro-inches) R_a , preferably obtained by honing or burnishing. These areas shall be properly demagnetized or otherwise treated so that the combined total electrical and mechanical runout does not exceed 25 % of the maximum allowed peak-to-peak vibration amplitude or the following value, whichever is greater:

- a) for areas to be observed by radial vibration probes, 6 μm (250 micro-inches);
- b) for areas to be observed by axial position probes, 13 μm (250 micro-inches).

If all reasonable efforts fail to achieve these limits, the vendor and the purchaser shall mutually agree on alternate acceptance criteria.

4.6.6 Each rotor shall be clearly marked with a unique identification number. This number shall be on the end of the shaft opposite the coupling or in an accessible area that is not prone to maintenance damage.

4.6.7 Impellers may be closed, consisting of a disk, vanes, and a cover, or they may be semi-open, consisting of a disk and vanes. Impellers shall be of welded, brazed, milled or cast construction. Other manufacturing methods, such as electroerosion and riveting, may be used if approved by the purchaser. Each impeller shall be marked with a unique identification number.

4.6.8 Welded, brazed and riveted impellers may consist of forged and cast components. Welds in the gas passageway shall be smooth and free from weld spatter. Impellers shall be heat-treated and stress-relieved after welding or brazing. Vane entrance and exits shall not have knife edges.

4.6.9 Cast impellers shall be finished all over except for gas passageways. Upgrade or repair welding may be permitted only with the purchaser's approval.

4.6.10 Welding as a means of balancing an impeller is not permitted.

4.6.11 The design of stressed parts shall include proper evaluation of the stress concentration factor (SCF) for the geometry. The design of stressed rotating parts shall include fillets that limit the SCF.

NOTE Areas of concern include the impeller vane-to-disk intersections, keyways and shaft section changes.

4.6.12 Integral thrust collars are preferred. Replacement thrust collars shall be furnished if required for removal of liquid film type, mechanical contact type or gas type shaft seals. If integral collars are furnished, they shall be provided with at least 3 mm ($\frac{1}{8}$ in) of additional stock to enable refinishing if the collar is damaged. If replaceable collars are furnished (for assembly and maintenance purposes), they shall be positively locked to the shaft to prevent fretting.

4.6.13 Both faces of thrust collars shall have a surface finish of not more than 0,4 μm (16 micro-inches) R_a and the axial total indicated runout of either face shall not exceed 13 μm (500 micro-inches).

4.6.14 Compressor designs that do not require a balance drum are acceptable.

4.6.15 If required, a balance drum, line and porting shall be provided to limit axial loads on the thrust bearings. A separate pressure tap connection or connections shall be provided to indicate the pressure in the balancing chamber, not in the balance line.

4.6.16 The balance line shall be flanged and sized to handle balance drum gas leakage at twice the initial design labyrinth clearance without exceeding the load rating of the thrust bearings (see 4.7.3.3). If the balance line requires a purchaser connection to this piping, then the connection sizes shall be indicated on the data sheets (see Annex A).

4.6.17 To prevent the build-up of potential voltages in the shaft, residual magnetism of the rotating element shall not exceed 0,000 5 T (5 gauss).

4.7 Bearings and bearing housings

4.7.1 General

4.7.1.1 Hydrodynamic radial and thrust bearings shall be provided unless specific approval to the contrary is obtained from the purchaser.

NOTE Annex J gives application considerations for use of active magnetic bearings, where specified. These bearings are new technology at this time.

4.7.1.2 Unless otherwise specified, thrust bearings and radial bearings shall be fitted with bearing metal temperature sensors installed in accordance with API 670.

4.7.2 Radial bearings

4.7.2.1 Radial bearings of the sleeve or pad type shall be used and shall be split for ease of assembly. The use of non-split designs requires the purchaser's approval. The bearings shall be precision bored with steel-backed, babbitted replaceable liners, pads or shells. The bearings shall be equipped with anti-rotation pins and shall be positively secured in the axial direction.

4.7.2.2 The bearing design shall suppress hydrodynamic instabilities and provide sufficient damping over the entire range of allowable bearings clearances to limit rotor vibration to the maximum specified amplitudes (see 4.9.5.6) while the equipment is operating loaded or unloaded at any speed (see 4.9.1.3) within the specified speed range.

4.7.2.3 The liners, pads or shells shall be in axially split housings 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 unit 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 unless approved by the purchaser.

4.7.2.4 Compressors equipped with sleeve type journal bearings shall be designed for field installation of tilting pad type radial bearings without remachining of the bearing bracket.

4.7.3 Thrust bearings

4.7.3.1 Hydrodynamic thrust bearings shall be of the steel-backed, babbitted multiple segment type, designed for equal thrust capacity in both directions and arranged for continuous pressurized lubrication to each side. Both sides shall be of the tilting pad type incorporating a self-levelling feature assuring that each pad carries an equal share of the thrust load, even with minor variations, in pad thickness.

4.7.3.2 Each pad shall be designed and manufactured with dimensional precision (thickness variation) such that interchangeability or replacement of the individual pads is allowed.

4.7.3.3 Thrust bearings shall be sized for continuous operation under the most adverse specified operating conditions. Calculation of the thrust force shall include, but shall not be limited to, the following factors:

- a) seal maximum design internal clearances and twice the maximum design internal clearances;
- b) pressurized rotor diameter step changes;
- c) stage maximum differential pressures;
- d) specified extreme variations in inlet, interstage, and discharge pressures;
- e) external thrust forces transmitted through the couplings;
- f) the maximum thrust force from the drive motor if the motor is directly connected.

4.7.3.4 For gear type coupling, the external thrust force shall be calculated from the following formula:

$$F = \frac{(0,25) \times (9\,550) P_r}{N_r \times D}$$

Or, in US customary units:

$$F = \frac{(0,25) \times (63\,300) P_r}{N_r \times D}$$

where

F is the external force, kilonewtons (pound force);

P_r is the rated power, in kilowatts (horsepower);

N_r is the rated speed, in revolutions per minute;

D is the shaft diameter of the coupling, in millimetres (inches).

4.7.3.5 Thrust forces for flexible element type couplings shall be calculated on the basis of the maximum allowable deflection permitted by the coupling manufacturer.

4.7.3.6 If two or more rotor thrust forces are to be carried by one thrust bearing (such as in a gear box) the resultant of the forces shall be used, provided the directions of the forces make them numerically additive; otherwise, the largest of the forces shall be used.

4.7.3.7 Hydrodynamic thrust bearings shall be selected such that, under any operating condition, the load does not exceed 50 % of the bearing manufacturer's ultimate load rating. The ultimate load rating shall be the load that will produce the minimum acceptable oil film thickness without inducing failure during continuous service, or the greatest load that does not exceed the creep initiation or yield strength of the babbitt at the location of maximum temperature on the pad, whichever load is the lesser of the two. In sizing thrust bearings, consideration shall be given to the following for each specific application:

- a) shaft speed;
- b) temperature of the bearing babbitt;
- c) deflection of the bearing pad;
- d) minimum oil film thickness;
- e) feed rate, viscosity, and supply temperature of the oil;
- f) design configuration of the bearing;
- g) babbitt alloy and pad material;
- h) turbulence of the oil film.

The basis for the sizing of thrust bearings shall be reviewed and approved by the purchaser.

4.7.3.8 Thrust bearings shall be arranged to allow axial positioning of each rotor relative to the casing and setting of the thrust bearings' clearance.

4.7.4 Bearing housings

4.7.4.1 Rotor support system parts (bearings, bearing housings, bearing shells, and bearing brackets) shall be axially split, non-pressurized (vented to atmosphere) and furnished with plugged connections for dry air or inert gas purge to any atmospheric labyrinth seals. Axial split bearing housings shall have a metal-to-metal split joint whose halves are located by means of dowels.

4.7.4.2 Compressors that use semi-enclosed coupling guards shall have bearing housings equipped with replaceable labyrinth type end seals and deflectors where the shaft passes through the housing; lip type seals shall not be used. The seals and deflectors shall be made of non-sparking materials. The design of the seals and deflectors shall effectively retain oil in the housing and prevent entry of foreign material into the housing.

4.7.4.3 Bearing housings for pressure-lubricated hydrodynamic bearings shall be arranged to minimize foaming. The drain system shall be adequate to maintain the oil and foam level below shaft end seals. Oil outlets from flooded thrust bearings shall be tangential and in the upper half of the control ring or, if control rings are not used, in the thrust bearing cartridge.

4.7.4.4 The rise in oil temperature through the bearing and housing shall not exceed 30 °C (50 °F) under the most adverse specified operating conditions. The bearing outlet oil temperature shall not exceed 85 °C (180 °F). If the inlet oil temperature exceeds 50 °C (120 °F), special consideration shall be given to bearing design, oil flow and allowable temperature rise.

4.7.4.5 Shaft support structures bolted to casings shall be of steel.

4.7.4.6 Oil connections on bearing housings shall be in accordance with 4.4.3.

4.7.4.7 Provision shall be made for mounting two radial vibration probes in each bearing housing, two axial position probes at the thrust end of each machine, and a one event per revolution probe in each machine shaft line. The probe installation shall be as specified in API 670.

4.8 Shaft seals

4.8.1 General

- **4.8.1.1** Shaft seals shall be provided to restrict or prevent process gas leakage to the atmosphere or seal fluid leakage into the process gas stream over the range of specified operating conditions, including start-up and shutdown. Seal operation shall be suitable for specified variations in suction conditions that may prevail during start-up, shutdown or settling out, or during any other special operation specified by the purchaser, such as slow roll or reverse rotation. The maximum sealing pressure shall be at least equal to the settling out pressure. The shaft seals and seal system shall be designed to permit safe compressor pressurization with the seal system in operation prior to process start-up.

The purchaser should establish a realistic value for the settling out pressure. The value should be shown on the data sheets.

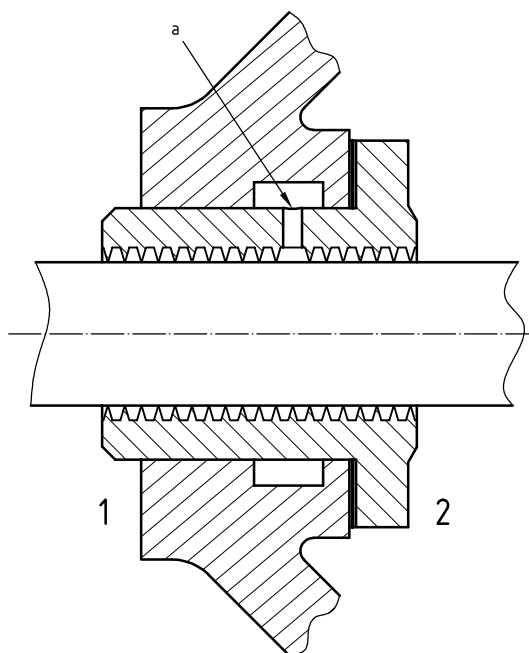
- **4.8.1.2** Shaft seals and, if specified, shaft sleeves shall be accessible for inspection and replacement without removing the top half of the casing of an axially split compressor or the heads of a radially split unit.

NOTE It is recognized that this requirement may not be feasible for overhung designs or shrink fit sleeves.

- **4.8.1.3** Shaft seals may be one or a combination of the following (4.8.2.1 to 4.8.2.5) types as specified by the purchaser on the data sheets. The materials for component parts shall be suitable for the service.

4.8.2 Requirements for types

4.8.2.1 The labyrinth seal (a typical seal is shown in Figure 2) may include carbon rings in addition to the labyrinths, if approved by the purchaser. Labyrinths may be stationary or rotating. Eductors or injection systems, if used, shall be furnished complete with piping regulation and control valves, pressure gauges, strainers, and so forth. Each item shall be piped and valved to permit its removal during operation of the compressor. Where gas from the compressor discharge is used for the motivating power of the eductor, provisions shall be made for sealing during start-up and shutdown (see 4.8.3.4 and 5.5.1.6).



Key

- 1 Internal gas pressure
- 2 Atmosphere

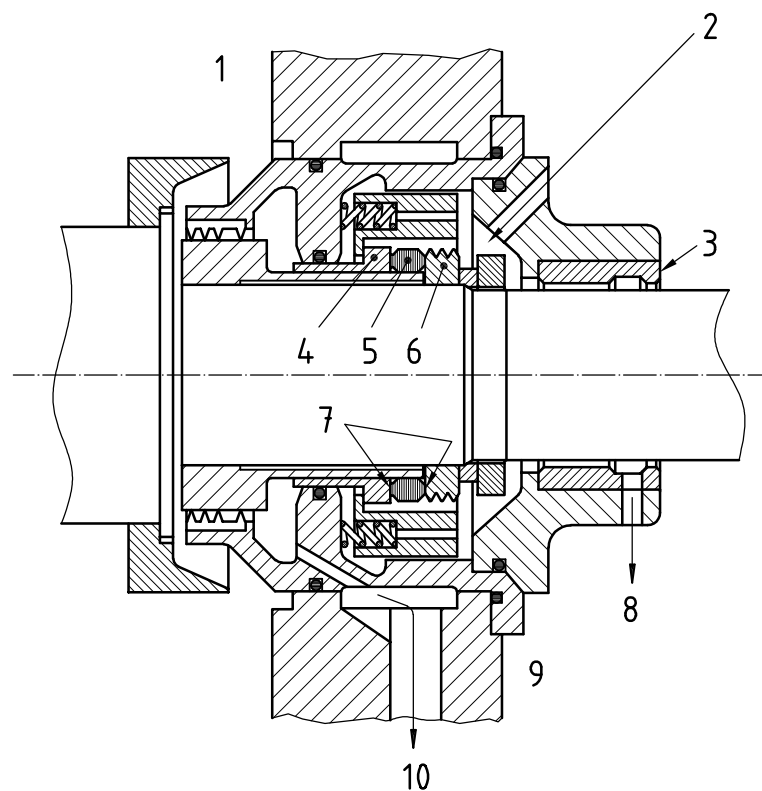
a Ports may be added for scavenging, inert-gas sealing or both.

Figure 2 — Labyrinth shaft seal

- **4.8.2.2** The mechanical (contact) seal (see Figure 3) shall be provided with labyrinths and slingers. Oil or another suitable liquid furnished under pressure to the rotating seal faces may be supplied from the lube oil system or from an independent seal system. Mechanical seals shall be designed to prevent gas leakage while the compressor is pressurized and being shut down, and after it is stopped in the event of seal oil failure. Various supplemental devices may be provided to ensure sealing when the compressor is pressurized but not running and the seal oil system is shut down. The purchaser shall specify whether such a device is to be provided. The final design shall be mutually agreed upon by the purchaser and the vendor.

4.8.2.3 The restrictive ring seal (see Figure 4) shall include rings of carbon or another suitable material mounted in retainers or in spacers. The seal may be operated dry, as in the labyrinth type, or with a sealing liquid, as in the mechanical type or with a buffer gas.

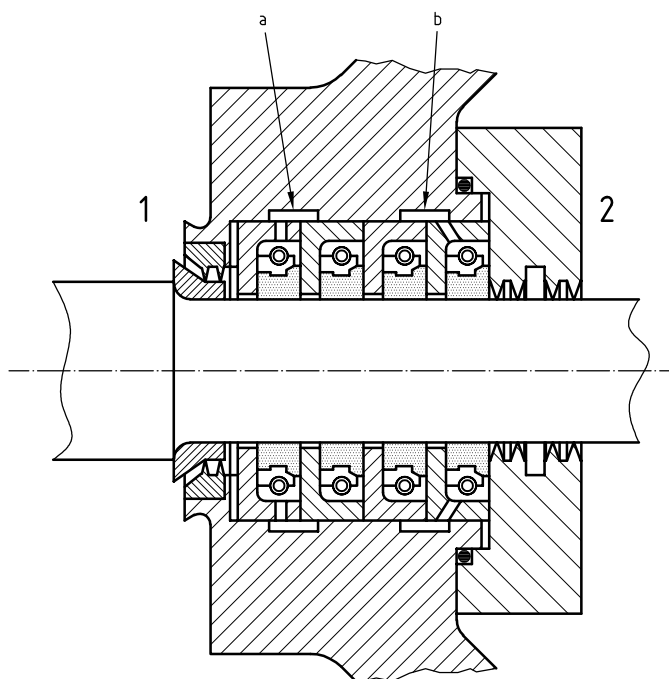
4.8.2.4 The liquid film seal (see Figures 5 and 6) shall be provided with sealing rings or bushings and labyrinths. A sealing liquid shall be supplied as in the mechanical type. Liquid film seals may be of the cylindrical bushing type shown in Figure 5 or the pumping type shown in Figure 6. An elevated tank to maintain static head in excess of the compressor sealing pressure shall be provided. The vendor shall state the height of the tank above the compressor centreline. Other means of maintaining this differential pressure and positive seal may be used with the purchaser's approval.



Key

- | | |
|-----------------------------|-------------------------|
| 1 Internal gas pressure | 6 Rotating seat |
| 2 Clean oil in | 7 Running faces |
| 3 Pressure breakdown sleeve | 8 Oil out |
| 4 Stationary seat | 9 Atmosphere |
| 5 Sealing ring | 10 Contaminated oil out |

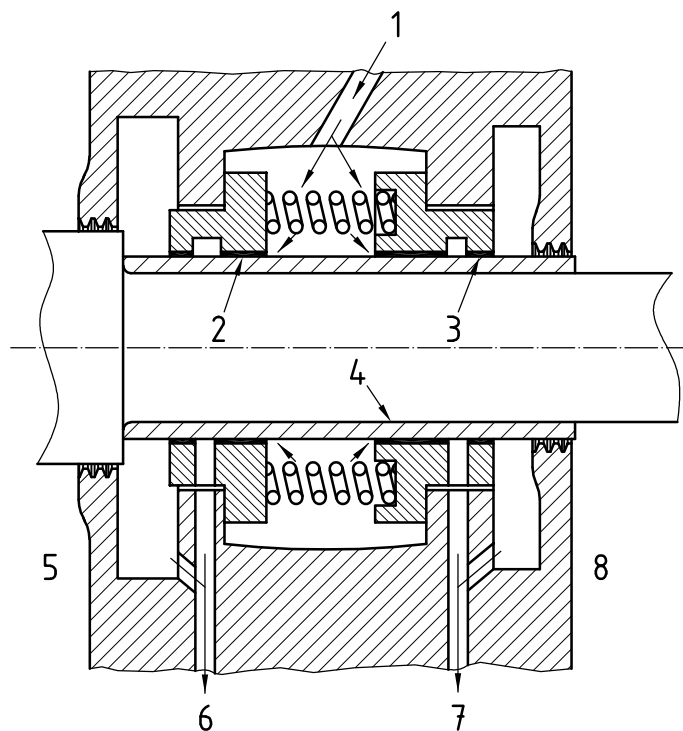
Figure 3 — Mechanical (contact) shaft seal

**Key**

- 1 Internal gas pressure
- 2 Atmosphere

- a Ports may be added for sealing.
- b Scavenging port may be added for vacuum application.

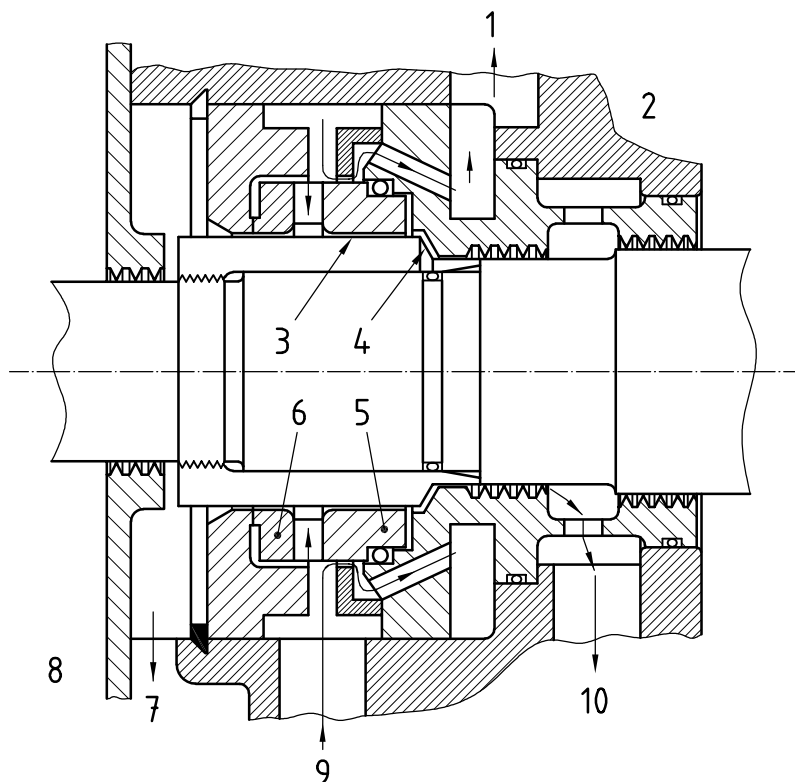
Figure 4 — Restrictive ring shaft seal



Key

- | | |
|-----------------|-------------------------|
| 1 Clean oil in | 5 Internal gas pressure |
| 2 Inner bushing | 6 Contaminated oil out |
| 3 Outer bushing | 7 Oil out |
| 4 Shaft sleeve | 8 Atmosphere |

Figure 5 — Liquid film shaft seal with cylindrical bushings



Key

- | | |
|---------------------------|-------------------------|
| 1 Clean oil recirculation | 6 Outer bushing |
| 2 Internal gas pressure | 7 Oil out |
| 3 Shaft sleeve | 8 Atmosphere |
| 4 Pumping area | 9 Clean oil in |
| 5 Inner bushing | 10 Contaminated oil out |

Figure 6 — Liquid film seal with pumping bushings

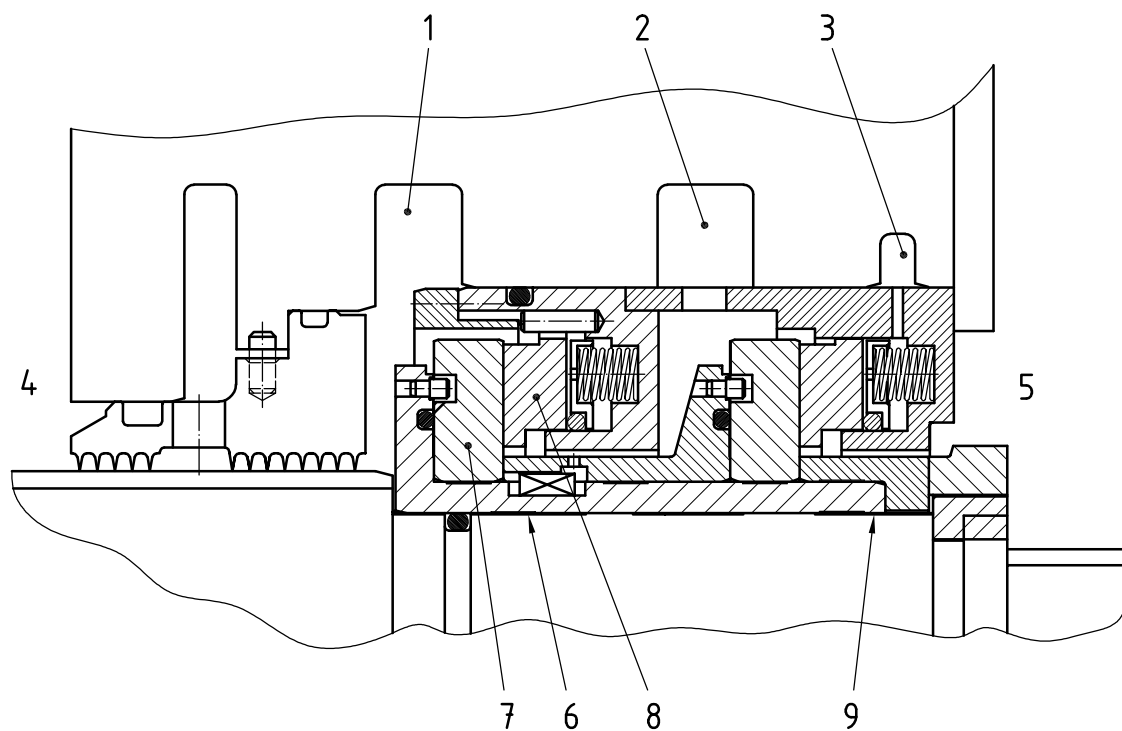
4.8.2.5 The self-acting gas seal may require a clean seal gas supply but does not require any liquid lubrication. The purchaser may specify the required seal configuration. Typical tandem seal configurations are shown in Figures 7 and 8. In these configurations, two identical seals are arranged in series as primary and back-up seals, one configuration without an internal labyrinth (Figure 7), and the other configuration with an internal labyrinth (Figure 8). Where there is danger of leakage of toxic or flammable gases to the atmosphere, then the arrangement in Figure 8 shall be used. A separation seal is generally required to prevent leakage to the atmosphere or to the bearing housing as well as oil leakage to the seal.

The seal gas shall be filtered and free from any contaminants that form residues. The seal gas may be taken from the compressor discharge or interstage point. An alternative seal gas source may be used and may be required during start-up and shutdown. Suitable measures shall be taken to protect the seal against reverse pressurization. The method of control shall be mutually agreed between purchaser and vendor. For testing considerations at the seal manufacturer's shop for this type of seal, see annex I and the data sheets (annex A).

Other configurations, for example, single, double (back-to-back) or triple, may be used, depending on the application.

NOTE 1 For certain applications external cooling of seals could be required.

NOTE 2 The seal will leak a small amount of seal gas.

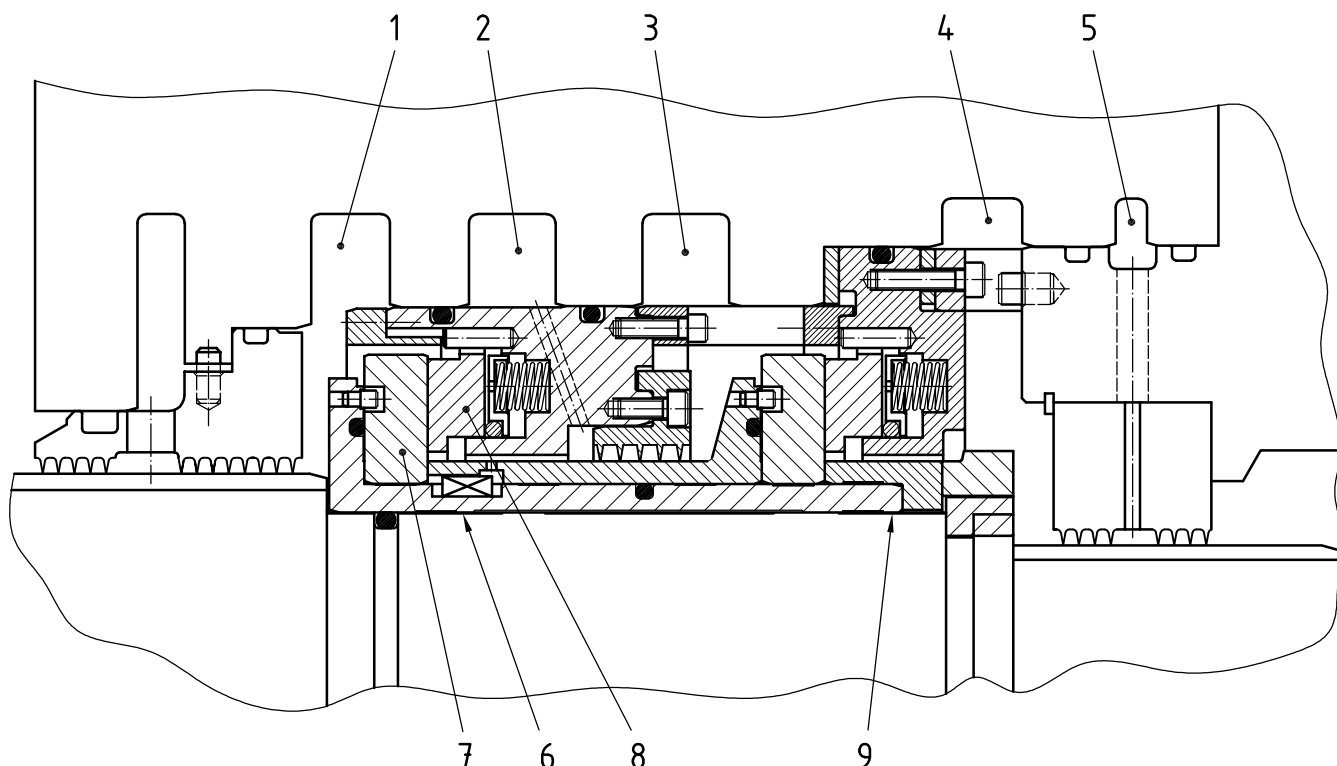


Key

- | | |
|---|--------------------------------|
| 1 | Seal gas in |
| 2 | Primary seal leakage |
| 3 | Buffer gas supply |
| 4 | Internal gas pressure |
| 5 | Atmospheric or bearing housing |

- | | |
|---|-----------------|
| 6 | Primary seal |
| 7 | Rotating seat |
| 8 | Stationary seat |
| 9 | Backup seal |

Figure 7 — Self-acting tandem gas seal



Key

- | | |
|------------------------|-------------------|
| 1 Seal gas in | 6 Primary seal |
| 2 Primary seal leakage | 7 Rotating seat |
| 3 Buffer gas | 8 Stationary seat |
| 4 Atmospheric vent | 9 Backup seal |
| 5 Separation gas | |

Figure 8 — Self-acting tandem gas seal with internal labyrinth

4.8.3 Other requirements

4.8.3.1 For any shaft end seals that use sealing liquid, the inward leakage from each seal shall be piped to an independent drain pot. The individual shaft end inward seal leakages shall be adequate to prevent outward migration of process gas. No individual shaft end seal shall have a leakage rate greater than 70 % of the total expected leakage from all shaft end seals in a single machine.

4.8.3.2 Oil contaminated by process gas that would damage components such as bearings, seal rings, O-rings and couplings shall be piped away separately to allow disposal or reconditioning.

4.8.3.3 Seal pressure equalizing lines and associated gas passages (including those for reference gas and axial thrust force balancing) shall be sized to maintain design shaft end seal performance at twice the maximum initial design clearances. The lines and passages shall also be sized to maintain substantially equal pressures at both shaft end seals during acceleration.

- **4.8.3.4** Unless otherwise specified, the seal design shall have provision for buffer gas injection to each seal. The purchaser shall specify whether buffer gas injection is to be used and, if so, the composition of that gas. In addition, the vendor shall state whether buffer gas injection is required for any specified operating conditions. If buffer gas injection is required (see Figure 8), the vendor shall state the gas requirements, and, if specified, furnish the complete control system schematic and bill of material. The method of control shall be mutually agreed between vendor and purchaser.
- **4.8.3.5** If specified for compressors with sub-atmospheric pressure at the shaft end seals, provision shall be made to pressurize these seals with gas at a pressure that is higher than atmospheric.

4.9 Dynamics

4.9.1 Critical speeds

4.9.1.1 If the frequency of a periodic forcing phenomenon (exciting frequency) applied to a rotor-bearing support system corresponds to a natural frequency of that system, the system may be in a state of resonance.

4.9.1.2 A rotor-bearing support system in resonance shall have its normal vibration displacement amplified. The magnitude of amplification and the rate of phase angle change are related to the amount of damping in the system and the mode shape taken by the rotor.

NOTE The mode shapes are commonly referred to as the first rigid (translatory or bouncing) mode, the second rigid (conical or rocking) mode and the (first, second, third*n*th) bending mode.

4.9.1.3 When the rotor amplification factor (see Figure 9) as measured on the test stand at the vibration probe is greater than or equal to 2,5, that frequency is called critical and the corresponding shaft rotational frequency is called a "critical speed". For the purposes of this International Standard, a critically damped system is one in which the amplification factor is less than 2,5.

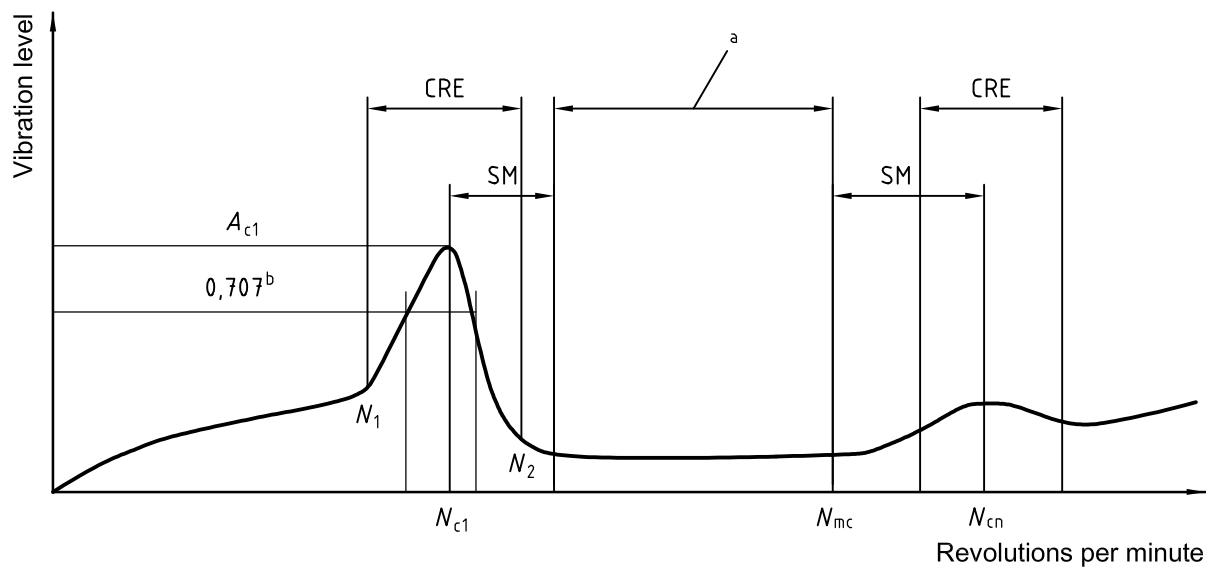
4.9.1.4 Critical speeds shall be determined analytically by means of a damped, unbalanced rotor response analysis and shall be confirmed by test stand data.

4.9.1.5 An exciting frequency may be less than, equal to, or greater than the rotational speed of the rotor. Potential forced and self-exciting frequencies considered in system design shall include, but are not limited to, the following sources:

- a) unbalance in the rotor system;
- b) oil film instabilities (whirl);
- c) internal rubs;
- d) blade, vane, nozzle, and diffuser passing frequencies;
- e) gear tooth meshing and side bands;
- f) coupling misalignment;
- g) loose rotor system components;
- h) hysteretic and friction whirl;
- i) boundary layer flow separation;
- j) acoustic and aerodynamic cross coupling forces;
- k) asynchronous whirl.

4.9.1.6 Resonances of support systems within the vendor's scope of supply shall not occur within the specified operating speed range or the specified separation margins, unless the resonances are critically damped.

4.9.1.7 The vendor having unit responsibility shall ensure the compatibility of drive-train critical speeds (rotor lateral, system torsional, blading modes, and the like) with the critical speeds of the machinery being supplied and that the combination is suitable for the specified operating speed range, including any starting speed detent (hold point) requirements of the train. A list of all undesirable speeds from zero to trip shall be submitted to the purchaser for review and included in the instruction manual for the guidance of the purchaser (see annex C).



Key

N_{c1}	rotor first critical, centre frequency, cycles per minute	μ	amplification factor $\mu = \frac{N_{c1}}{N_2 - N_1}$
N_{cn}	critical speed, n th	SM	separation margin
N_{mc}	maximum continuous speed, 105 %	CRE	critical response envelope
N_1	initial (lesser) speed at 0,707 times peak amplitude (critical)	A_{c1}	amplitude at N_{c1}
N_2	final (greater) speed at 0,707 times peak amplitude (critical)	A_{cn}	amplitude at N_{cn}
$N_2 - N_1$	peak width at the half-power point		

NOTE The curve shape is for illustration only and does not necessarily represent any actual rotor response plot.

- a Operating speeds.
- b Peak.

Figure 9 — Rotor response plot

4.9.2 Lateral analysis

4.9.2.1 The vendor shall provide a damped, unbalanced response analysis for each machine in order to ensure acceptable amplitudes of vibration at any speed from zero to trip. An example of the logic diagram of the lateral analysis and test procedures is provided in annex E.

4.9.2.2 The damped, unbalanced response analysis shall include, but shall not be limited to, the following considerations.

- a) Support (base, frame, and bearing housing) stiffness, mass and damping characteristics, including effects of rotational speed variation. The vendor shall state the assumed support system values.
- b) Bearing lubricant film stiffness and damping changes due to speed, load, preload, oil temperatures, accumulated assembly tolerances and maximum-to-minimum clearances.
- c) Rotational speed, including the various starting speed detents, operating speed and load ranges (including agreed upon test conditions if different from those specified), trip speed, and coast-down conditions.
- d) Rotor masses, including the mass moment of coupling halves, stiffness and damping effects (for example, accumulated fit tolerances, and frame and casing effects).
- e) The influence over the operative range of the calculated values for hydrodynamic stiffness and damping generated by the casing, seals and labyrinths.

- **4.9.2.3** If specified, the effects of other equipment in the train shall be included in the damped, unbalanced response analysis (i.e. a train lateral analysis shall be performed).

EXAMPLE A train lateral analysis should be specified for trains with a rigid coupling.

4.9.2.4 The damped unbalanced response analysis shall include items a) to e) as follows. (See annex C).

- a) A plot and identification of the mode shape at each resonant speed (critically damped or not) from zero to trip, as well as the next mode occurring above the trip speed.
- b) Frequency, phase and response amplitude data at the vibration probe locations through the range of each critical speed, using the following arrangement of unbalance for the particular mode. This unbalance shall be sufficient to raise the displacement of the rotor at the probe locations to the vibration limit defined by the following equation:

$$L_v = 25,4\sqrt{12\,000/N}$$

or in US customary units,

$$L_v = \sqrt{12\,000/N}$$

where

L_v is the vibration limit (amplitude of unfiltered vibration), in micrometres (mil), peak-to-peak;

N is the operating speed nearest the critical of concern, in revolutions per minute.

The unbalance shall be no less than two times the unbalance limit specified in 4.9.5.3. The unbalance mass or masses shall be placed at the location or locations within the bearing span that have been analytically determined as affecting the particular mode most adversely (e.g. at mid-span for transitory modes or near both ends and 180° out of phase for conical modes). For bending modes with maximum deflections at the shaft's ends, the amount of unbalance shall be based on the overhung mass rather than the static bearing loading.

- c) Modal diagrams for each response in b), indicating the phase and major axis amplitude at each coupling engagement plane, the centrelines of the bearings, the locations of the vibration probes, at each seal area throughout the machine. The minimum design diametrical running clearance of the seals shall also be indicated.
- d) For the purposes of the verification test (see 4.9.3), an additional plot of a test unbalance, as specified in b) (based on static bearing loading for rigid modes or based on overhung mass for bending modes). This test mass shall be at least two times and no more than eight times the unbalance limit specified in 4.9.5.3, and shall be placed at a location determined by the vendor.
- e) If specified, the generation of a stiffness map of the undamped rotor response from which the damped unbalance response analysis specified in c) has been derived. This plot shall show frequency versus support system stiffness with the calculated support system stiffness curves superimposed.

4.9.2.5 The damped unbalance response analysis shall indicate that the machine in the unbalanced condition 4.9.2.4, b), shall meet the following acceptance criteria (see Figure 9).

- a) If the amplification factor is less than 2,5, the response is considered critically damped and no separation margin is required.
- b) If the amplification factor is 2,5 to 3,55, a separation margin of 15 % above the maximum continuous speed and 5 % below the minimum operating speed is required.
- c) If the amplification factor, μ , is greater than 3,55 and the critical response peak is below the minimum operating speed, the required separation margin (a percentage of minimum speed) is calculated as follows:

$$SM = 100 - \left(84 + \frac{6}{\mu - 3} \right)$$

- d) If μ is greater than 3,55 and the critical response peak is above the trip speed, the required SM (a percentage of maximum continuous speed) is equal to the following:

$$SM = \left(126 - \frac{6}{\mu - 3} \right) - 100$$

4.9.2.6 The calculated unbalanced peak-to-peak rotor amplitudes [see 4.9.2.4.b)] at any speed from zero to trip shall not exceed 75 % of the minimum design diametrical running clearances throughout the machine (with the exception of floating ring and abradable seal locations).

4.9.2.7 If, after the purchaser and the vendor have agreed that all practical design efforts have been exhausted, the analysis indicates that the separation margins still cannot be met or that a critical response peak falls within the operating speed range, acceptable amplitudes shall be mutually agreed upon by the purchaser and the vendor, subject to the requirement of 4.9.2.6.

- **4.9.2.8** If specified, or when average gas density in a compressor casing exceeds 60 kg/m³ (3,75 lb/ft³), the vendor shall carry out a rotor stability analysis. This shall be made at rated speed for constant speed machines and over the speed range from minimum to maximum continuous speed for variable speed compressors. The analysis shall be performed without, and then with, destabilizing aerodynamic effects, taking into account the highest gas density. The results are to be provided as plots showing the damped critical speeds and the log decrement as a function of speed. The vendor should demonstrate the acceptability of the calculated value of log decrement by reference to similar machines in satisfactory operation. This should be over the speed range from minimum to maximum continuous speed. This stability analysis should also take into account all items listed in 4.9.2.2.

4.9.3 Shop verification of unbalanced response analysis

- **4.9.3.1** If specified, the vendor shall demonstrate the accuracy of the vendor's unbalanced response calculation by performing an unbalanced response test in accordance with 4.9.

4.9.3.2 The actual critical speed responses, as revealed on the test stand with a rotor unbalance magnitude in accordance with 4.9.2.4, d), placed at a location (usually the coupling) determined by the vendor, shall be the criteria for confirming the validity of the damped unbalanced response analysis.

NOTE It is recognized that the dynamic response of the machine on the test stand will be a function of the agreed upon test conditions and that unless the test stand results are obtained at the conditions of pressure, temperature, speed, and load expected in the field, they may not be the same as the results expected in the field.

4.9.3.3 The parameters to be measured during the test shall be speed and shaft vibration amplitudes with corresponding phase. The vibration amplitudes and phase from each pair of $x - y$ vibration probes shall be vectorially summed at each response peak to determine the maximum amplitude of vibration. The major axis amplitude of each response peak shall not exceed the limits specified in 4.9.2.6. The gain of the recording instrumentation used shall be predetermined and preset before the test so that the highest response peak is within 60 % to 100 % of the recorder's full scale on the test unit coast-down (deceleration).

NOTE 1 It is recognized that vectorial subtraction of slow roll (300 r/min to 600 r/min) total electrical and mechanical runout is always required for this verification and that vectorial subtraction of bearing housing motion is normally required.

NOTE 2 The phase on each vibration signal, x or y , is the angular measure, in degrees, of the phase difference (lag) between a phase reference signal (from a phase transducer sensing a once per revolution mark on the rotor, as described in API 670) and the next positive peak, in time, of the synchronous ($1x$) vibration signal. (If proximity probes are used, this is the lag angle between the vibration probe and the high spot on the rotor).

NOTE 3 The major axis amplitude is properly determined from a lissajous (orbit) display on an oscilloscope, oscillograph or equivalent. If the phase angle between the x and y signals is not 90° , the major axis amplitude can be approximated by $(x^2 + y^2)^{1/2}$. If the phase angle between the x and y signals is 90° , the major axis value is the greater of the two vibration signals.

4.9.3.4 Additional testing and correction of the original damped, unbalanced rotor response analysis shall be required if, from the test data described above, or from a phase or amplitude indication in the damped unbalanced response analysis [based on the unbalance conditions described in 4.9.2.4, b)], or both, it appears that either of the following conditions exists:

- a) any critical response fails to meet the separation margin requirements (see 4.9.2.5) or falls within the operating speed range;
- b) the requirement of 4.9.2.6 has not been met.

Unbalance masses shall be determined and placed as mutually agreed upon by the purchaser and the vendor (see 4.9.2.4, b) and d). Unbalance magnitudes shall be achieved by adjusting the residual unbalance that exists in the rotor from the initial run to raise the displacement of the rotor at the probe locations to the vibration limit defined by the equation in 4.9.2.4, b) at the maximum continuous speed. The measurements from this test, taken in accordance with 4.9.3.2, shall indicate the following acceptance criteria for the machine.

- a) at no speed shall the shaft deflections exceed 90 % of the minimum design running clearances.
- b) at no speed within the operating speed range shall the shaft deflections exceed 55 % of the minimum design running clearances or 150 % of the allowable vibration limit at the probes [see 4.9.2.4, b)].

The internal deflection limits specified in a) and b) immediately above shall be based on the calculated displacement ratios between the probe locations and the areas of concern identified in 4.9.2.4, c). Actual internal displacements for these tests shall be calculated by multiplying these ratios by the major axis amplitudes (see 4.9.3.2). Acceptance shall be based on these calculated displacements, not on inspection of seals after testing; however, damage to any portion of the machine as a result of this testing shall constitute failure of the test. Minor internal seal rubs that do not cause clearance changes outside the vendor's new part tolerance do not constitute damage.

4.9.4 Torsional analysis

4.9.4.1 Excitations of torsional natural frequency may come from many sources, which should be considered in the analysis. These sources may include, but are not limited to, the following:

- a) gear problems such as unbalance and pitch line runout;
- b) start-up conditions such as speed detents (under inertial impedances) and other torsional oscillations;
- c) torsional transients such as switch on and terminal short circuit of all kinds of electric motors, start-up, operation and worst-case transient of variable speed electric motors, and start-up of synchronous electric motors.

4.9.4.2 The torsional natural frequency of the complete train shall be at least 10 % above or 10 % below any possible excitation frequency within the specified operating speed range (from minimum to maximum continuous speed).

4.9.4.3 Torsional criticals at two times running speed as well as one and two times the supply frequency for motor driven systems shall preferably be avoided or, in systems in which corresponding excitation frequencies occur, shall be shown to have no adverse effect. In addition to multiples of running speeds, torsional excitations that are not a function of operating speeds or that are non-synchronous in nature shall be considered in the torsional analysis if applicable. Identification of these frequencies shall be the mutual responsibility of the purchaser and the vendor.

4.9.4.4 For the torsional analysis of variable-speed motor-driven compressors, the vendor, together with the variable speed motor supplier shall identify all excitation frequencies and their consequences on the train. These frequencies shall include but not be limited to

- non-speed dependant excitations such as ripple,
- integer harmonics,
- non-integer harmonics,
- carrier frequency harmonics, and
- switching harmonics between speed-control windows.

4.9.4.5 If torsional resonances are calculated to fall within the margin specified above, and the purchaser and vendor have agreed that all efforts to remove the critical from within the limiting frequency range have been exhausted, the vendor shall demonstrate that the resonances have no adverse effect on the complete train.

- **4.9.4.6** For motor-driven units and units including gears, or — if specified — for turbine driven units, the vendor shall perform a torsional vibration analysis of the complete train and shall be responsible for directing the modifications necessary to meet the requirements of 4.9.4.1 to 4.9.4.4.

4.9.4.7 In addition to the torsional analyses required in 4.9.4.2 to 4.9.4.5, the vendor shall perform a transient torsional vibration analysis for motor-driven units. The acceptance criteria for this analysis shall be mutually agreed upon by the purchaser and the vendor.

4.9.5 Vibration and balancing

4.9.5.1 Major parts of the rotating element, such as the shaft, balancing drum and impellers, shall be individually dynamically balanced before assembly to ISO 1940-1 grade G2,5 (with respect to the maximum continuous speed) or better. If a bare shaft with a single key way is dynamically balanced, the key way shall be filled with a fully crowned half-key in accordance with ISO 8821. A shaft with key ways 180° apart, but not in the same transverse plane, shall also be filled thus. The initial balance correction to the bare shaft shall be recorded.

4.9.5.2 The complete rotor shall be balanced either at low speed in accordance with 4.9.5.3 or at high speed in accordance with 4.9.5.5.

4.9.5.3 For low-speed balancing, the rotating element shall be multiplane dynamically balanced during assembly. This shall be accomplished after the addition of no more than two major components. Balancing correction shall be applied only to the elements added. Minor correction of other components may be required

during the final trim balancing of the rotor (see 4.9.5.1). In the sequential balancing process, any half-keys used in the balancing of the bare shaft shall continue to be used until they are replaced with the final key and mating element. The mass of all half-keys used during final balancing of the assembled element shall be recorded on the residual unbalance work sheet (see annex D). The maximum allowable residual unbalance per plane (journal) shall be calculated as follows:

$$U_{\max} = 6\,350\, W / N$$

or, in US customary units,

$$U_{\max} = 4\, W / N$$

where

$U_{\max}$ is the residual unbalance, in gram millimetres (ounce inches);

W is the journal static mass load, in kilograms (pounds);

N is the maximum continuous speed, in revolutions per minute.

If spare rotors are supplied they shall be dynamically balanced to the same tolerances as the main rotor.

4.9.5.4 After the final low-speed balancing of each assembled rotating element has been completed, a residual unbalanced check shall be performed and recorded as described in annex D.

4.9.5.5 High-speed balancing may be done (balancing in a high-speed balancing machine at the operating speed). The acceptance criteria for this balancing shall be mutually agreed upon by the purchaser and the vendor.

4.9.5.6 During the shop test of the machine, assembled with the balanced rotor, operating at its maximum continuous speed or at any other speed within the specified operating speed range, the peak-to-peak amplitude of unfiltered vibration in any plane, measured on the shaft adjacent and relative to each radial bearing, shall not exceed the following value or 50 µm (2,0 mil), whichever is less:

$$A = 25,4 \sqrt{12\,000 / N}$$

or, in US customary units,

$$A = \sqrt{12\,000 / N}$$

where

A is the amplitude of unfiltered vibration, in micrometres (mil), peak-to-peak;

N is the maximum continuous speed, in revolutions per minute

At any speed greater than the maximum continuous speed, up to and including the trip speed of the driver, the vibration shall not exceed 150 % of the maximum value recorded at the maximum continuous speed, unless the unbalance response analysis indicates a steeper rise of vibration levels, when the limit shall be that given by calculation or 150 % of those given by the equation above, whichever is the lower.

NOTE These limits are not to be confused with the limits specified in 4.9.3. for shop verification of unbalanced response.

4.9.5.7 Electrical and mechanical runout shall be determined and recorded.

4.9.5.8 If the vendor can demonstrate that electrical or mechanical runout is present, a maximum of 25 % of the test level calculated from the equation in 4.9.5.6 or 6,5 µm (0,25 mil), whichever is greater, may be vectorially subtracted from the vibration signal measured during the factory test.

4.10 “Lube” oil and seal oil systems

4.10.1 Unless otherwise specified, a pressurized oil system or systems shall be furnished to supply oil at a suitable pressure or pressures, as applicable, to the following:

- a) the bearings of the driver and of the driven equipment (including any gear);
- b) the continuously lubricated couplings;
- c) the governing and control oil system;
- d) the seal oil system;
- e) the purchaser's control system (if hydraulic).

4.10.2 Oil reservoirs and housings that enclose moving lubricated parts (such as bearings, shaft seals), highly polished parts, instruments and control elements shall be designed to minimize contamination by moisture, dust and other foreign matter during periods of operation or idleness.

- **4.10.3** The purchaser shall specify whether the seal oil and “lube” oil systems are to be separate or combined. If separate systems are specified, the means of preventing the interchange of oil between the two systems shall be described in the vendor's proposal.

4.10.4 Unless otherwise specified, bearings and bearing housings shall be arranged for hydrocarbon oil lubrication.

4.10.5 Unless otherwise specified, pressurized oil systems shall conform to the requirements of ISO 10438. If approved by the purchaser, a pressurized integral oil system may be provided for a closed loop, non-hydrocarbon refrigerant refrigeration system.

4.11 Materials

4.11.1 General

4.11.1.1 Construction materials shall be at the manufacturer's discretion for the specified operating conditions, except as required or prohibited by the data sheets or otherwise by this International Standard. Table B.1 in annex B lists material specifications that, if used with appropriate heat treatment or impact testing requirements or both, are generally considered acceptable for major component parts. Other international material specifications may, by agreement between purchaser and vendor, be used for major component parts. The metallurgy of all major components shall be clearly stated in the vendor's proposal. See 5.5 for requirements for auxiliary piping materials.

4.11.1.2 Materials and the material grade shall be identified in the proposal using established international, national or industry designations. If no such designation is available, the vendor's material specification, giving physical properties, chemical composition and test requirements, shall be included in the proposal.

- **4.11.1.3** Copper and copper alloys (excluding Monel or its equivalent, bearing babbitt, and precipitation hardening stainless steels) shall not be used for parts of compressors or auxiliaries in contact with corrosive gas or with gases capable of forming explosive copper compounds. The purchaser shall note such gas characteristics on the inquiry data sheets.

4.11.1.4 The vendor shall specify optional tests and inspection procedures necessary to ensure that materials are satisfactory for the service. Such tests and inspections shall be listed in the proposal. The purchaser should consider specifying additional tests and inspections, especially for materials in critical service.

4.11.1.5 Selection of casing material shall be restricted by the limits imposed by 4.2.7 and 4.2.8.

4.11.1.6 Material that is notch-sensitive and prone to brittle fracture at ambient temperatures (e.g. ASTM A 515) shall not be used.

- **4.11.1.7** Materials exposed to Hydrogen-sulfide (H₂S) gas service as defined by NACE MR 0175 shall be in accordance with the requirements of NACE MR 0175. Ferrous materials not covered by NACE MR 0175 shall be limited to a yield strength not exceeding 620 MPa (90 000 psi) and a Rockwell hardness not exceeding HRC 22. Components that are fabricated by welding shall be stress relieved, if required, so that both the welds and the heat-affected zones meet the yield strength and hardness requirements. The purchaser shall specify on the data sheets the presence of H₂S in the process gas.

NOTE Shafts in compressors of between-bearing design can exceed the stated limits of yield strength and hardness because of requirements for higher strength at reduced shaft sections near couplings and because of the low levels of working stress in the portion of the shaft between bearings.

- **4.11.1.8** The purchaser shall specify the presence of any corrosive agents in the process gas, the process stream and the environment, including constituents that may cause stress corrosion cracking.

4.11.1.9 If parts made from austenitic stainless steels are exposed to a process gas or environmental conditions that promote corrosion, and they could be fabricated, hard-surfaced, repaired or overlaid by welding, then stabilized or low-carbon grades shall be used.

4.11.1.10 Austenitic steels shall not be used in services where stress corrosion cracking is a possibility.

NOTE Overlays or hard surface that contain more than 0,10 % carbon can sensitize both low-carbon and stabilized grades of austenitic stainless steel unless a buffer layer that is not sensitive to inter-granular corrosion is applied.

4.11.1.11 In hydrogen gas service at partial gauge pressures greater than 700 kPa (100 psi) or at concentrations greater than a molal percentage of 90 % at any pressure, impeller materials that have a yield strength in excess of 830 MPa (120 000 psi) or a hardness in excess of Rockwell HRC 34 are prohibited.

4.11.1.12 Materials, casting factors and the quality of any welding shall be in accordance with the pressure design code.

4.11.1.13 External parts that are subject to rotary or sliding motions (such as control linkage joints and adjusting mechanisms) shall be of corrosion-resistant materials suitable for the site environment.

4.11.1.14 Minor parts that are not identified (e.g. nuts, springs, washers, gaskets and keys) shall have corrosion resistance at least equal to that of specified parts in the same environment.

4.11.1.15 Bolting material for pressure joints shall be in accordance with the pressure design code.

4.11.1.16 If mating parts such as studs and nuts of austenitic stainless steel or materials with similar galling tendencies are used, they shall be lubricated with a suitable anti-seizure compound compatible with the process conditions.

NOTE Torque loading values will differ considerably with and without anti-seizure compound.

4.11.1.17 O-rings shall be compatible with all specified services. For high-pressure services, special consideration shall be given to the selection of O-rings that will not be damaged by rapid depressurization of the compressor.

4.11.2 Pressure-containing parts

4.11.2.1 The vendor shall specify on the data sheets (see annex A) the material grade of castings.

4.11.2.2 Welding of piping and pressure-containing parts, as well as any dissimilar metal welds and weld repairs, shall be performed and inspected by operators and using procedures qualified in accordance with the pressure design code.

4.11.2.3 The vendor shall be responsible for establishing weld repair procedures that are in accordance with the pressure design code and for the implementation of repairs in accordance with these procedures as well as for defect description, including post-repair heat treatment, if required, and non-destructive examination of repairs. Such procedures are subject to review by the purchaser before any repair is made.