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ICS 27.040; 75.180.20

Erdöl-, petrochemische und Erdgasindustrie – Dampfturbinen – Spezial-Einsatzbereiche (ISO 10437:2003); Englische Fassung EN ISO 10437:2003

Petroleum, petrochemical and natural gas industries – Steam turbines – Special-purpose applications (ISO 10437:2003); English version EN ISO 10437:2003

Industries du pétrole, de la pétrochimie et du gaz naturel – Turbines à vapeur – Usage spécial (ISO 10437:2003); Version anglaise EN ISO 10437:2003

Gesamtumfang 121 Seiten

Normenausschuss Erdöl- und Erdgasgewinnung (NÖG) im DIN
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Nationales Vorwort

Diese Europäische Norm wurde vom Technischen Komitee CEN/TC 12 „Materialien, Ausrüstungen und Offshore-Bauwerke für die Erdöl-, petrochemische- und Erdgasindustrie“ (Sekretariat: Frankreich) erstellt. Es handelt sich dabei um die unveränderte Übernahme von ISO 10437:2003, erarbeitet vom ISO/TC 67 „Materials, equipment and offshore structures for petroleum, petrochemical and natural gas industries“, Subkomitee SC 6 „Processing equipment and systems“ (Sekretariat: Frankreich), zu welchem die nationale Zuarbeit beim NAM im DIN liegt.

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EUROPEAN STANDARD

NORME EUROPÉENNE

EUROPÄISCHE NORM

EN ISO 10437

July 2003

ICS 27.040; 75.180.20

English version

Petroleum, petrochemical and natural gas industries – Steam turbines – Special-purpose applications (ISO 10437:2003)

Industries du pétrole, de la pétrochimie et du gaz naturel – Turbines à vapeur – Usage spécial (ISO 10437:2003)

Erdöl-, petrochemische und Erdgasindustrie – Dampfturbinen – Spezial-Einsatzbereiche (ISO 10437:2003)

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Foreword

This document (EN ISO 10437:2003) has been prepared by Technical Committee ISO/TC 67 "Materials, equipment and offshore structures for petroleum and natural gas industries" 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of an identical text or by endorsement, at the latest by January 2004, and conflicting national standards shall be withdrawn at the latest by January 2004.

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, Hungary, Iceland, Ireland, Italy, Luxembourg, Malta, Netherlands, Norway, Portugal, Slovakia, Spain, Sweden, Switzerland and the United Kingdom.

Endorsement notice

The text of ISO 10437:2003 has been approved by CEN as EN ISO 10437:2003 without any modifications.

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Introduction

This International Standard is based on API Std 612, fourth edition, June 1995.

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 appropriate 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.

This International Standard requires the purchaser to specify certain details and features.

A bullet (●) at the beginning of a clause or subclause indicates that either a decision is required or further information is to be provided by the purchaser. This information or decision should be indicated on the data sheets; otherwise it should be stated in the quotation request (inquiry) or in the order.

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

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1 Scope

This International Standard specifies requirements and gives recommendations for the design, materials, fabrication, inspection, testing and preparation for shipment of steam turbines for special-purpose applications. It also covers the related lube-oil systems, instrumentation, control systems and auxiliary equipment. It is not applicable to general-purpose steam turbines, which are covered in ISO 10436.

NOTE For the purpose of this provision, API Std 611 is equivalent to ISO 10436.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 261, *ISO general-purpose metric screw threads — General plan*

ISO 262, *ISO general-purpose metric screw threads — Selected sizes for screws, bolts and nuts*

ISO 724, *ISO general-purpose metric screw threads — Basic dimensions*

ISO 965 (all parts), *ISO general-purpose metric screw threads — Tolerances*

ISO 1940-1, *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 7005-1, *Metallic flanges — Part 1: Steel flanges*

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

ISO 8068, *Petroleum products and lubricants — Petroleum lubricating oils for turbines (categories ISO-L-TSA and ISO-L-TGA) — Specifications*

ISO 8501-1, *Preparation of steel substrates before application of paints and related products — Visual assessment of surface cleanliness — Part 1: Rust grades and preparation grades of uncoated steel substrates and of steel substrates after overall removal of previous coatings*

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

ISO 10438 (all parts)[1)] , *Petroleum and natural gas industries — Lubrication, shaft sealing and control oil systems for special-purpose applications*.

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*

ISO 15649, *Petroleum and natural gas industries — Piping*

1) To be published.

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IEC 60045-1, *Steam turbines — Part 1: Specifications*

IEC 60072, *Dimensions and output series for rotating electrical machines*

IEC 60079, *Electrical apparatus for explosive atmospheres*

IEC 60953, *Rules for steam turbine thermal acceptance tests*

EN 287, *Approval testing of welders — Fusion welding*[2)]

EN 288, *Specification and approval of welding procedures for metallic materials*

API RP 520 PT I, S*izing, selection, and installation of pressure-relieving systems in refineries, Part I — Sizing and selection*.[3)]

API RP 520 PT II, *Sizing, selection, and installation of pressure-relieving systems in refineries, Part II — Installation*

API Std 526, *Flanged steel pressure relief valves*

API Std 613, *Special-purpose gear units for petroleum, chemical and gas industry services*

API Std 670, *Machine protection systems*

API Std 671, *Special-purpose couplings for petroleum, chemical and gas industry services*

API RP 686 (First edition, April 1996), *Recommended Practices for machinery installation and installation design*

ASME, *Boiler and pressure vessel code, Section V — Nondestructive examination*.[4)]

ASME, *Boiler and pressure vessel code, Section VIII — Pressure vessels*

ASME, *Boiler and pressure vessel code, Section IX — Qualification standard for welding and brazing procedures, welders, brazers, and welding and brazing operators*

ASME B1.1, *Unified screw threads (UN and UNR Thread Form)*

ASME B16.1, *Cast iron pipe flanges and flanged fittings, Class 25, 125 and 250*

ASME B16.5, *Pipe flanges and flanged fittings, NPS 1/2 through NPS 24*

ASME B16.11, *Forged fittings, socket-welding and threaded*

ASME B16.42, *Ductile iron pipe flanges and flanged fittings, classes 150 and 300*

ASME B16.47, *Large diameter steel flanges NPS 26 through NPS 60*

ASME B17.1, *Keys and keyseats*

ASME PTC 6, *Performance test code 6 on steam turbines*

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2) Comité Européen de Normalisation, 36, rue de Stassart, B-1050 Brussels, Belgium.

3) American Petroleum Institute, Publications and Distribution Section, 1220 L Street Northwest , Washington DC 20005, USA.

4) ASME International, 3 Park Avenue, New York, NY 10016-5990, USA.

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ASME PTC 20.2, *Overspeed trip systems for steam turbine-generator units*

ASTM A 194, *Standard specification for carbon and alloy steel nuts for bolts for high-pressure or high-temperature service, or both*[5)]

ASTM A 247, *Standard test method for evaluating the microstructure of graphite in iron castings*

ASTM A 278, *Standard specification for gray iron castings for pressure-containing parts for temperatures up to 650 °F (350 °C)*

ASTM A 307, *Standard specification for carbon steel bolts and studs, 60 000 psi tensile strength*

ASTM A 395, *Standard specification for ferritic ductile iron pressure-retaining castings for use at elevated temperatures*

ASTM A 418, *Standard test method for ultrasonic examination of turbine and generator steel rotor forgings*

ASTM A 472, *Standard test method for heat stability of steam turbine shafts and rotor forgings*

ASTM A 536, *Standard specification for ductile iron castings*

AWS D1.1, *Structural welding code — Steel*[6)]

NEMA SM 23, *Steam turbines for mechanical drive service*.[7)]

NFPA 70, *National electrical code*.[8)]

NACE MR0175, *Sulfide stress cracking resistant metallic materials for oilfield equipment*.[9)]

SSPC-SP6/NACE No. 3, *Commercial blast cleaning*.[10)]

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5) American Society for Testing and Materials, 1916 Race Street, Philadelphia, PA 19103-1187.

6) American Welding Society, 550 NW Le Jeune Road, PO Box 351040, Miami, FL 33130, USA.

7) National Electrical Manufacturers Association, 1300 N 17th Street; Suite 1847, Rosslyn, VA 22209, USA.

8) National Fire Protection Association, 1 Batterymarch Park, Quincy, MA 02269, USA.

9) NACE International, 1440 South Creek Drive, Houston, TX 77084, USA.

10) SSPC: The Society for Protective Coatings, 40 24th Street 6th floor, Pittsburgh, PA 15222-4656, USA.

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3 Terms and definitions

For the purposes of this document, the following terms and definitions apply. (See Annex B for a guide to steam turbine nomenclature.)

3.1
alarm point
preset value of a measured parameter at which an alarm is actuated to warn of a condition that requires corrective action

3.2
anchor bolts
bolts used to attach the mounting plate to the support structure or foundation

3.3
axially split
split with the principal joint parallel to the shaft centreline

3.4
control mechanism
all of the equipment between the speed governor and the governor-controlled valve(s) (such as linkages, pilot valves, power servos, and so forth)

3.5
critical speed
shaft rotational speed at which the rotor-bearing-support system is in a state of resonance

3.6
design
equipment manufacturer's description of various parameters relevant to the equipment

NOTE This terminology is for use only by the equipment manufacturer and is not intended to appear in the purchaser's specifications.

3.7
fail safe system
system which causes the equipment to revert to a permanently safe condition (shutdown and/or depressurized) in the event of a component failure or failure of the energy supply to the system

3.8
field changeable
design feature that permits alteration of a function after the equipment has been installed

NOTE The alteration can be accomplished by the following:

a) soldering jumper leads to terminal pins especially provided for this purpose;

b) employing circuit-board-mounted switches or potentiometers;

c) using a shorting or diode-pin-type matrix board;

d) using prewired shorting plugs;

e) using authorized controlled access.

3.9
gauge board
bracket or plate used to support and display gauges, switches, and other instruments

NOTE A gauge board is open and not enclosed.

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3.10
general-purpose turbines
horizontal or vertical turbines used to drive equipment that is usually spared, is relatively small in size (power), or is in non-critical service

NOTE General-purpose steam turbines are intended for applications where the inlet gauge pressure does not exceed 4 800 kPa (48 bar) (700 psi) and the inlet temperature does not exceed 400 °C (750 °F), and where the speed does not exceed 6 000 r/min.

3.11
governor-controlled valve
device that controls the flow of steam into or out of the turbine in response to the speed governor

3.12
hold-down bolts
mounting bolts
bolts that hold the equipment to the mounting plate or plates

3.13
hydrodynamic bearings
bearings that use the principles of hydrodynamic lubrication

NOTE The bearing surfaces are oriented so that relative motion forms an oil wedge, or wedges, to support the load without shaft-to-bearing contact.

3.14
local
⟨device⟩ mounted on or near the equipment or console

3.15
maximum allowable speed
highest speed at which the manufacturer's design permits continuous operation

3.16
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 any specified operating conditions

3.17
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 handling the specified fluid at the maximum allowable temperature

3.18
maximum continuous speed
highest speed at which the turbine, as built and tested, is capable of continuous operation, at any of the specified steam conditions

3.19
maximum exhaust casing pressure
highest exhaust steam pressure that the purchaser requires the casing to contain, with steam supplied at maximum inlet conditions

NOTE The turbine casing is subjected to the maximum temperature and pressure under these conditions.

3.20
maximum exhaust pressure
highest exhaust steam pressure at which the turbine is required to operate continuously

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3.21
maximum inlet pressure and temperature
highest inlet steam pressure and temperature conditions at which the turbine is required to operate continuously

3.22
maximum sealing pressure
highest pressure the seals are required to seal during any specified static or operating conditions and during startup and shutdown

3.23
minimum allowable speed
lowest speed at which the manufacturer's design permits continuous operation

3.24
minimum exhaust pressure
lowest exhaust steam pressure at which the turbine is required to operate continuously

3.25
minimum inlet pressure and temperature
lowest inlet steam pressure and temperature conditions at which the turbine is required to operate continuously

3.26
normal operating point
point at which usual operation is expected and optimum efficiency is desired

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

3.27
observed
test or inspection for which the purchaser requires notification of the timing and may wish to attend

NOTE This is not a hold point. The inspection or test is performed as scheduled, and if the purchaser or the purchaser's representative is not present, the vendor proceeds to the next step.

3.28
owner
final recipient of the equipment who may delegate another agent as the purchaser of the equipment

3.29
panel
enclosure used to mount, display and protect gauges, switches and other instruments

3.30
potential maximum power
approximate maximum power to which the turbine can be uprated at the specified normal speed and steam conditions when it is furnished with suitable (that is, larger or additional) nozzles and, possibly, with a larger governor-controlled valve or valves

3.31
pressure casing
composite of all stationary pressure-containing parts of the unit, including all nozzles and other attached parts

3.32
purchaser
agency that issues the order and specification to the vendor

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NOTE The purchaser may be the owner of the plant in which the equipment is to be installed or the owner's appointed agent.

3.33
radially split
split with the principal joint perpendicular to the shaft centreline

3.34
rated power
greatest turbine power specified and its corresponding speed; it includes all the margin required by the specifications of the driven equipment

3.35
rated speed
100 % speed
highest rotational speed required to meet any of the specified operating conditions

3.36
relief valve set pressure
pressure at which a relief valve starts to lift

3.37
remote
⟨device⟩ located away from the equipment or console, typically in a control room

3.38
separation margin
margin between a critical speed and the nearest required operating speed

3.39
shutdown set point
preset value of a measured parameter at which automatic or manual shutdown of the system or equipment is required

3.40
slow roll
speed recommended by the vendor (typically 400 r/min to 500 r/min) for warm-up and initial check of equipment integrity prior to full operation

3.41
special-purpose turbines
horizontal turbines used to drive equipment that are usually not spared and are used in uninterrupted continuous operation in critical service

NOTE This category is not limited by steam conditions, power or turbine speed.

3.42
special tool
tool which is not a commercially available catalogue item

3.43
standby service
normally idle, or idling, piece of equipment that is capable of immediate automatic or manual startup and continuous operation

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3.44
steam rate
quantity of steam required by the turbine per unit of power output measured at the output shaft of the turbine

3.45
turbine manufacturer
company that designs, manufactures, tests and provides service support for the turbine

3.46
trip speed
rotational speed at which the independent overspeed shutdown system operates to shut down a turbine

3.47
unit responsibility
responsibility for coordinating the technical aspects of the equipment and all auxiliary systems included in the scope of the order

NOTE The technical aspects to be considered include, but are not limited to, such factors as the power requirements, speed, rotation, general arrangement, couplings, dynamics, noise, lubricant selection, instrumentation, piping, testing of components, conformance to specifications and material test reports.

3.48
vendor
supplier
agency that supplies the equipment in response to the order

NOTE The vendor may be the manufacturer of the equipment or the manufacturer's agent and is normally responsible for service support.

3.49
witnessed
inspection or test that the purchaser attends

NOTE A hold is applied to the production schedule and the inspection or test is carried out with the purchaser or purchaser's representative in attendance

4 Dimensions

Drawings and maintenance dimensions shall be in SI units or United States Customary (USC) units. Use of an ISO datasheet indicates that SI units shall be used. Use of a USC datasheet indicates that USC units shall be used. Typical datasheets in both systems of units are given in Annex A.

5 Statutory requirements

The purchaser and vendor shall mutually determine the measures that must be taken to comply with any governmental codes, regulations, ordinances or rules that are applicable to the equipment.

6 Basic design

6.1 General

6.1.1 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 five years of uninterrupted operation.

NOTE It is recognized that achievement of this objective requires a joint effort of the purchaser, vendor and user.

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6.1.2 The vendor shall assume unit responsibility for all equipment and all auxiliary systems included in the scope of the order.

6.1.3 The vendor who has unit responsibility shall assure that all subvendors comply with the requirements of this International Standard.

- **6.1.4** The purchaser shall specify the equipment's normal operating point and any other required operating points, including the inlet and exhaust steam conditions and any extraction or induction steam quantities and pressures. The purchaser shall also specify the maximum and minimum values of the inlet, exhaust and extraction/induction steam conditions.

- **6.1.5** The purchaser shall specify the purity of steam available.

NOTE IEC 60045-1 and NEMA SM 23 contain recommendations for steam purity levels.

6.1.6 Turbines shall be capable of the following:

a) operation at normal power and speed with normal steam conditions, with the steam rate certified by the manufacturer at these conditions;

b) delivering rated power at its corresponding speed with coincident minimum inlet and maximum exhaust conditions specified.

NOTE To prevent oversizing or to obtain higher operating efficiency or both, it may be desirable to limit maximum turbine capability by specifying normal power or a selected percentage of rated power instead of rated power at the conditions specified.

c) continuous operation at maximum continuous speed and at any other speed within the range specified, with a maximum continuous speed of at least 105 % of rated speed;

d) continuous operation at rated power and speed with maximum inlet steam conditions and maximum or minimum exhaust steam conditions;

e) continuous operation at the lowest speed at which maximum torque is required with minimum inlet and maximum exhaust conditions, with the purchaser specifying both the speed and torque values required;

f) continuous operation at specified conditions for extraction and/or induction;

g) operation with variations from rated steam conditions and steam purity levels recommended in IEC 60045-1 or NEMA SM 23;

h) operation uncoupled with maximum inlet steam conditions.

Governing instability and high acceleration rates may occur and require action such as throttling of inlet pressure. Care should be taken when operating uncoupled or no load for generator sets. Consideration should be given to the high exhaust and extraction steam temperatures that would result during light or no-load operation.

6.1.7 Equipment shall be designed to run to the trip speed without damage. The turbine trip speed shall be 110 % of maximum continuous speed, normally 116 % of rated speed.

This requirement should not be construed to allow continuous operation above maximum continuous speed.

6.1.8 The turbine and accessories shall perform on the test stand and when installed on the 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 who has unit responsibility.

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6.1.9 Many factors may adversely affect site performance. These factors include such items as piping loads, alignment at operating conditions, supporting structure, handling during shipment, and handling and assembly at the site. 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 witness one or more of the following:

a) a check of the piping alignment performed by unfastening the major flanged connections of the equipment;

b) the initial shaft alignment check;

c) shaft alignment at operating temperature.

NOTE Further information on machinery installation and installation design is given in API RP 686.

6.1.10 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.

6.1.11 All equipment shall be designed to permit rapid and economical maintenance. Major parts, such as casing components and bearing 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.

6.1.12 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.

6.1.13 Unless otherwise specified, cooling water systems shall be designed for the conditions specified in Table 1. Provision shall be made for complete venting and draining of the system.

Table 1 — Design criteria and specifications for cooling water systems

Criteria	SI units	US Customary units
Water velocity over heat exchanger surfaces	1,5 m/s to 2,5 m/s	5 ft/s to 8 ft/s
Maximum allowable working pressure, gauge	700 kPa	100 psi (7 bar)
Test pressure (1,5 MAWP), gauge	1 050 kPa	150 psi (10,5 bar)
Maximum pressure drop	100 kPa	15 psi (1 bar)
Maximum inlet temperature	32 °C	90 °F
Maximum outlet temperature	49 °C	120 °F
Maximum temperature rise	17 K	30 °R
Minimum temperature rise	11 K	20 °R
Fouling factor on water side	0,32 $m^2 \cdot K/kW$	0,002 hr-ft^2-°F/Btu
Shell corrosion allowance	3 mm	1/8 in
NOTE The criterion for velocity over heat exchange surface is intended to minimize water side fouling; the criterion for minimum temperature rise is intended to minimize the use of cooling water.		

6.1.14 The vendor shall notify the purchaser if the criteria for minimum temperature rise and velocity over heat exchange surfaces result in a conflicting design. If such a conflict exists, the purchaser shall approve the final selection.

- **6.1.15** 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. In order to determine compliance, the vendor shall provide both maximum

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sound pressure and sound power level data per octave band for the equipment. The vendor shall provide details of any special measures taken to achieve the stated levels.

- **6.1.16** Motors, electrical components, and electrical installations shall be suitable for the area classification (class, group, and division or zone) specified by the purchaser and shall meet the requirements of the applicable sections of IEC 60079 or NFPA 70, Articles 500, 501, 502, and 504, as well as any local codes specified (and, if requested by the vendor, furnished) by the purchaser.

- **6.1.17** 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.

- **6.1.18** The equipment, including all auxiliaries, shall be suitable for operation, using the utility conditions specified by the purchaser.

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

6.2 Nameplates and rotation arrows

6.2.1 A nameplate shall be securely attached at a readily visible location on the equipment and on any other major piece of auxiliary equipment.

6.2.2 Rotation arrows shall be cast in, or attached to, each major item of rotating equipment at a readily visible location. Welding is not permitted. A rotation arrow shall be located on the thrust-bearing housing.

6.2.3 Nameplates and rotation arrows (if attached) shall be of austenitic stainless steel or of nickel-copper alloy. Attachment pins shall be of the same material.

6.2.4 Data shall be clearly stamped on the nameplate and shall include, but not be limited to, the following:

a) purchaser's equipment item number (may be on a separate nameplate if there is insufficient space on the rating nameplate);

b) vendor's name;

c) serial number;

d) size and type;

e) rated power and speed;

f) all lateral critical speeds less than trip speed;

g) the next lateral critical speed greater than trip speed;

h) maximum continuous speed;

i) trip speed (see 3.46);

j) normal and maximum inlet steam temperature and pressure;

k) normal and maximum exhaust steam pressure;

l) number of teeth in the multi-toothed surface provided for speed sensing (see 8.4).

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6.2.5 Any critical speeds determined from mechanical running tests shall be stamped on the nameplate followed by the word “Test”. Critical speeds, predicted by calculation, up to and including the first critical speed above trip speed, and not identifiable by test, shall be stamped on the nameplate followed by the abbreviation “Calc”.

● **6.2.6** The purchaser shall specify the units to be shown on the nameplate.

7 Casings

7.1 Pressure casings

7.1.1 All pressure parts shall be suitable for operation at the most severe conditions of coincident pressure and temperature expected with the specified steam conditions.

7.1.2 The tensile stress used in the design of the pressure casing for any material shall not exceed 0,25 times the minimum ultimate tensile strength for that material at the maximum specified operating temperature, and, for castings, multiplied by the appropriate casting factor as shown in Table 2. The manufacturer shall state the source of the material properties, such as ASTM, as well as the casting factors applied in his/her proposal.

NOTE For bolting, the allowable tensile stress is used to determine the total bolting area based on pressure load or gasket preload. It is recognized that to provide the initial load required to obtain a reliable bolted joint, the bolting will be tightened to produce a tensile stress higher than the design tensile stress. Values in the range of 0,7 times yield are common.

Table 2 — Casting factors

Type of NDE	Casting factor
Visual, magnetic particle and/or liquid penetrant	0,8
Spot radiography	0,9
Ultrasonic	0,9
Full radiography	1,0

7.1.3 The maximum allowable working pressure(s) of the casing shall be at least equal to the specified relief valve set point(s). For condensing turbines, the exhaust casing shall be designed for both full vacuum and for a maximum allowable working gauge pressure of at least 70 kPa (0,7 bar) (10 psi).

Normally, a full-capacity safety relief valve is required in the exhaust piping between each exhaust connection and exhaust block valve to prevent overpressure and possible rupture of the turbine casing.

7.1.4 The vendor shall define the physical limits and the maximum allowable working pressure of the turbine casing and of each part of turbine casings designed for more than one maximum allowable pressure level (split-pressure-level casings).

7.1.5 The turbine casing shall be axially split. Turbine casings may also be split radially between high-pressure and low-pressure sections.

7.1.6 The main joint of axially and radially split casings shall use a metal-to-metal joint that is tightly maintained by bolting. The joint shall be sealed with a compound that is compatible with the fluids to be handled. Gaskets (including string type) shall not be used.

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

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7.1.8 Casings and supports shall be designed to have sufficient strength and rigidity to limit any change in the relative position of the shaft ends caused by the worst combination of allowable pressure, torque, and piping forces and moments, to 50 µm (0,002 in) (see 7.4).

NOTE This clause does not apply to thermal growth.

7.1.9 Supports, and the design of jackscrews and their attachments, shall be rigid enough to permit the machine to be moved by the use of its lateral and axial jackscrews without damage.

7.1.10 Jackscrews, guide rods, cylindrical casing-alignment dowels and/or other appropriate devices shall be provided to facilitate disassembly and reassembly. Guide rods shall be of sufficient length to prevent damage to the internals or casing studs by the casing during disassembly and reassembly. Lifting lugs or eyebolts shall be provided for lifting only the top half of the casing.

7.1.11 When 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.

7.1.12 The steam chest and casing shall be provided with connections to ensure complete drainage. Drain connections shall be DN25 (NPS 1) minimum size.

7.1.13 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 any allowance for corrosion, shall be left around and below the bottom of drilled and threaded holes. The depth of the threaded holes shall be at least 1,5 times the stud diameter. Through bolting is preferred in areas of the casing where the temperature may exceed 410 °C (775 °F).

7.1.14 Studded connections shall be furnished with studs installed. The first 1,5 threads at both ends of each stud shall be removed.

7.1.15 Bolting shall be furnished as follows.

a) The details of threading shall conform to ISO 261, ISO 262, ISO 724 and ISO 965, or ASME B1.1.

b) Studs or through bolts shall be supplied on the main joint of axially split casings and end covers of radially split casing joint(s).

c) Studs or through bolts shall be used instead of cap screws on all other joints, except where hexagonal head cap screws are essential for assembly purposes and have been approved by the purchaser.

d) The manufacturer's marking, in accordance with the appropriate standard (e.g. ASTM), shall be located on all fasteners 6 mm (1/4 in) and larger (excluding washers and headless set screws). For studs, the marking shall be located on the nut end of the exposed stud.

e) Adequate clearance shall be provided at all bolting locations to permit the use of socket or box wrenches.

● f) If specified, the main casing joint studs and nuts shall be designed for the use of hydraulic bolt tensioning. Procedures and any special tooling required shall be provided by the vendor.

g) Internal socket-type, slotted-nut or spanner-type bolting shall not be used on the exterior of the turbine case unless specifically approved by the purchaser.

7.1.16 Mounting surfaces shall meet the following criteria.

a) The surface finish shall be 6 µm (250 µin) arithmetic average roughness, Ra, or better.

b) The mounting surfaces in one plane shall be machined such that no point on any surface deviates from the common plane by more than 50 µm (0,002 in).

c) The different mounting planes shall be parallel to each other within 1 in 1 000, in any direction.

d) The upper surfaces shall be machined or spot-faced (to a diameter three times that of the hole), parallel to the mounting surface.

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7.1.17 Hold-down bolt holes shall be drilled perpendicular to the mounting surface or surfaces and shall be 15 mm (0,6 in) larger in diameter than the hold-down bolt.

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

7.2 Casing connections

● **7.2.1** The purchaser shall specify the orientation of the main inlet and outlet steam connections. All connections shall be suitable for the maximum allowable working pressure(s) of the casing. Flanged connections shall be integral with the casing or, for casings of weldable material, may be formed by a socket-welded or butt-welded pipe nipple or transition piece, and shall terminate with a weld-neck or socket-weld flange.

7.2.2 Connections welded to the casing shall meet the material requirements of the casing, including impact values, rather than the requirements of the connected piping (see 11.3.4). All welding of connections shall be completed before the casing is hydrostatically tested (see 16.3.2).

7.2.3 Casing openings for piping connections shall be at least DN 20 (NPS 3/4) and shall be flanged or machined and studded. For casings of non-weldable materials, and where flanged or machined and studded openings are impractical, threaded openings in DN 20 (NPS 3/4) to DN 40 (NPS 1½) are permissible. Connections to these threaded openings shall be installed as follows.

a) Pipe nipples shall be provided with a weld-neck or socket-weld flange.

b) The nipple and flange materials shall meet the requirements of 7.2.2.

c) Threaded openings and bosses for tapered pipe threads shall conform to ASME B16.5.

d) Pipe threads shall be tapered threads conforming to ISO 7-1.

NOTE For the purpose of this provision, ASME B1.20.1 is equivalent to ISO 7-1.

e) A process-compatible thread lubricant of proper temperature specification shall be used on all threaded connections. Thread tape shall not be used.

7.2.4 Pipe nipples screwed or welded to the casing should not be more than 150 mm (6 in) long and shall be a minimum of Schedule 160 seamless for sizes DN 25 (NPS 1) and smaller, and a minimum of Schedule 80 for DN 40 (NPS 1½). Bracing or reinforcement is required when nipples longer than 150 mm (6 in) are necessary.

7.2.5 Sizes DN 32, DN 65, DN 90, DN 125, DN 175 and DN 225 (NPS 1 1/4, 2 1/2, 3 1/2, 5, 7, and 9) shall not be used.

7.2.6 For non-weldable casing materials, threaded openings not required to be connected to piping shall be plugged with solid, round-head steel plugs in accordance with ASME B16.11, unless otherwise specified. Plugs that may later require removal shall be of a corrosion-resistant material. Plastic plugs are not permitted.

7.2.7 Flanges shall conform to the following.

a) Cast iron flanges shall be flat faced and conform to the dimensional requirements of ISO 7005-2 or ASME B16.1 or ASME 16.42. Class 125 flanges shall have a minimum thickness equal to Class 250 for sizes DN 200 (8 NPS) and smaller. Gray cast iron shall not be used.

b) Steel flanges shall conform to dimensional requirements of ISO 7005-1 or ASME B16.5 or B16.47.

c) Flat face flanges with full raised face thickness are acceptable on casings of all materials. Flanges in all materials that are thicker or have a larger outside diameter than required by ISO or ANSI are acceptable. Non-standard (oversized) flanges shall be completely dimensioned on the arrangement drawing.

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d) Machined and studded connections and flanges not in accordance with ISO 7005-1, ISO 7005-2, ASME B16.1, ASME B16.5, ASME B16.42 or ASME B16.47 require purchaser's approval. Unless otherwise specified, the vendor shall supply mating flanges, studs nuts and gaskets for these non-standard connections.

e) 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.

f) Flanges shall be full faced or spot faced on the back and shall be designed for through bolting.

g) Vertical nozzles shall have flanges parallel with the horizontal plane within 0,5°. Horizontal nozzles shall have flanges parallel with the vertical plane within 0,5°. Studs or bolt holes shall straddle centrelines parallel to the main axes of the equipment.

7.2.8 Machined and studded connections shall conform to the facing and drilling requirements of ISO 7005-1 or ISO 7005-2, or ASME B16.1, ASME B16.5, ASME B16.42 or ASME B16.47.

7.2.9 All of the purchasers' connections shall be accessible for disassembly without requiring the steam turbine, any major part or attached auxiliary piping to be moved.

7.3 Internal stationary components

7.3.1 All control stage nozzle rings shall be replaceable. Nozzle rings welded to the outer casing are acceptable only when approved in advance by the purchaser.

7.3.2 All other stationary blading shall be mounted in replaceable diaphragms or blade carriers. Nozzles or blades welded to the diaphragm are preferred (see 11.3.1).

7.3.3 Any internal fasteners shall be positively retained to prevent them from entering the steam path.

7.3.4 The design shall ensure that water cannot be trapped in the wet region of a condensing turbine. The design of water drain holes in the wet region of a condensing turbine shall avoid the holes becoming blocked by corrosion or debris. In the case of drain holes in the diaphragms of an impulse type turbine, this shall be achieved by lining the drain holes with corrosion resistant material.

7.4 External forces and moments

Turbines shall be designed to withstand external forces and moments at least equal to the values calculated in accordance with NEMA SM 23.

8 Rotating elements

8.1 General

8.1.1 Rotors shall be capable of safe operation at maximum overshoot speed of 121 % of maximum continuous speed at any specified operating temperature. Following such an excursion, the rotor shall be capable of operation without immediate maintenance intervention.

NOTE Rubbing of seals and minor localized yielding of rotor components can occur and vibration levels can increase after such an event.

8.1.2 The purchaser’s approval is required for built-up rotors (disks shrunk on the shaft) when blade tip velocities exceed 250 m/s (825 ft/s) at maximum continuous speed or when stage inlet steam temperatures exceed 440 °C (825 °F).

8.1.3 Each rotor shall be permanently marked with a unique identification number. This number shall be on the shaft end, in an area that is not prone to damage.

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- **8.1.4** If specified, provisions shall be made for field balancing without disassembly of the turbine. The vendor shall describe these provisions and the method of use in the proposal.

8.2 Shafts

8.2.1 Shafts shall be accurately finished throughout their entire length. The surface of the shaft at shrink fit areas and bearing areas shall be finished to a roughness not exceeding 0,8 µm (32 µin) Ra.

8.2.2 The rotor shaft sensing areas to be observed by radial vibration probes shall be concentric with the bearing journals. All 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, for a minimum of one probe-tip diameter on each side of the probe. These areas shall not be metallized, sleeved or plated. The surface shall be finished to a roughness not exceeding 0,8 µm (32 µin) Ra. These areas shall be properly demagnetized to the levels specified in API Std 670 or otherwise treated so that the combined total electrical and mechanical runout does not exceed the following.

a) For areas to be observed by radial-vibration probes, 25 % of the allowed peak-to-peak vibration amplitude or 6,3 µm (0,25 mil), whichever is greater.

b) For areas to be observed by axial-position probes, 12,7 µm (0,5 mil).

8.2.3 All shaft keyways shall have fillet radii conforming to ASME B17.1.

8.2.4 Shafts shall be capable of transmitting torque at least equal to the torque determined by potential maximum power. The coupling shaft end design shall conform to the requirements of ISO 10441 or API Std 671.

8.2.5 To prevent the buildup of harmful potential voltage, magnetic flux density of any part of the rotating element shall not exceed 0,000 3 T (3 G).

8.3 Blading

8.3.1 For each blade row, the vendor shall verify by Campbell diagrams or their equivalent (corrected to actual operating temperatures and speeds), for both nozzle passing frequency and twice nozzle passing frequency, that excitation does not occur within the specified operating speed range for all multiples up to 15 times running speed. The following modes shall be verified:

— in-phase tangential;

— out-of-phase tangential;

— axial;

— torsional;

— any other high-response modes.

If this is not feasible, blade-stress levels developed in any specified driven-equipment operation shall be low enough to ensure trouble-free operation if resonant vibration occurs within the operating range. This shall be verified by Goodman diagrams or their equivalent. Copies of Campbell or Goodman diagrams or both shall be provided to the purchaser. Blades shall be designed to withstand operation at resonant frequencies during normal warm-up.

NOTE Excitation sources include — but are not limited to — fundamental and first harmonic passing frequencies of rotating blades and stationary vanes upstream and downstream of each blade row, steam passage splitters, irregularities in vane pitch at horizontal casing flanges, the first four turbine speed harmonics, casing openings (exhaust or extraction), partial arc diaphragms or nozzle plates, internal struts and structural members in the inlet and exhaust casing or horizontal joints, and meshing frequencies in gear units.

8.3.2 All blades shall be mechanically suitable for operation (including transient conditions) over the specified speed range. The vendor shall assume that torque varies as speed squared, unless otherwise notified by the purchaser.

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8.4 Speed-sensing element

A dedicated multi-toothed surface for speed sensing shall be provided integral with, or positively attached and locked to, the turbine shaft. This surface may be shared by other speed sensors but shall not be used as a gear for driving other mechanical components. The axial width of the multi-toothed surface (the width of the surface being viewed by radial probes) shall be a minimum of one and one-half times the diameter of the probe tip.

9 Rotor dynamics

9.1 General

9.1.1 In the design of rotor-bearing systems, consideration shall be given to all potential sources of periodic forcing phenomena (excitations), which shall include, but are not limited to, the following:

a) unbalance in the rotor system;

b) oil-film instabilities (whirl);

c) internal rubs;

d) blade and nozzle 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;

l) electrical line frequency.

NOTE 1 The frequency of a potential source of excitation can be less than, equal to, or greater than the rotational speed of the rotor.

NOTE 2 When the frequency of a periodic forcing phenomenon (excitation) applied to a rotor-bearing support system coincides with a natural frequency of that system, the system is in a state of resonance. A rotor-bearing-support-system in resonance can have the magnitude of its normal vibration amplified. The magnitude of amplification and, in the case of critical speeds, the rate of change of the phase-angle with respect to speed, are related to the amount of damping in the system.

NOTE 3 Publication 684 contains additional information on rotor dynamics not covered in this International Standard.

9.1.2 For the purposes of this International Standard, critical speeds and other resonant conditions of concern are those with an amplification factor greater than 2,5.

9.1.3 Resonances of structural support systems that are within the vendor's scope of supply and that affect the rotor vibration amplitude shall not occur within the specified operating speed range or the specified separation margins (see 9.2.9). The effective stiffness of the structural support shall be considered in the analysis of the dynamics of the rotor bearing support system [see 9.2.4 d)].

NOTE Resonances of structural support systems can adversely affect the rotor vibration amplitude.

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- **9.1.4** The vendor who is specified to have unit responsibility for the complete drive train shall communicate the existence of any undesirable running speeds in the range from zero to trip speed. This shall be illustrated by the use of Campbell (forced frequency) diagrams for individual machines or, if specified, for the complete train, or for both. These diagrams shall be submitted to the purchaser for review and included in the instruction manual.

NOTE Examples of undesirable speeds are those associated with the rotor lateral criticals of concern, system torsionals and blading modes.

9.2 Lateral analysis

9.2.1 Critical speeds and their associated amplification factors shall be determined by means of a damped unbalanced rotor response analysis.

9.2.2 The location of all critical speeds below the trip speed shall be confirmed on the test stand during the mechanical running test by unbalancing the rotor if necessary. The accuracy of the numerical model shall be demonstrated (see 9.3).

9.2.3 Before carrying out the damped unbalanced response analysis, the vendor shall conduct an undamped analysis to identify the undamped critical speeds and determine the mode shapes located in the range from 0 % to 125 % of trip speed. Unless otherwise specified, the results of the undamped analysis shall be provided to the purchaser. The presentation of the results shall include:

a) mode shape plots (relative amplitude vs. axial position on rotor), and

b) critical speed-support stiffness map (frequency versus support stiffness) with the calculated support stiffness; horizontal, K_{xx}, and vertical, K_{yy}, (see Figure 1) superimposed on the map.

For machinery with widely varying bearing loads and/or load direction, the vendor may propose to substitute mode shape plots listing the undamped critical speed for each of the identified modes in lieu of the undamped critical speed map.

9.2.4 The damped unbalanced response analysis shall include, but shall not be limited to, the following:

a) rotor masses, including the mass moment of coupling halves, stiffness, and damping effects (e.g. accumulated fit tolerances, fluid stiffening, dampening);

b) bearing lubricant-film stiffness and damping values, including changes due to speed, load, preload, range of oil temperatures, maximum-to-minimum clearances resulting from accumulated assembly tolerances, and the effect of asymmetrical loading caused by partial arc admission, side streams, eccentric clearances, etc.;

c) the pad pivot stiffness (for tilting pad bearings);

d) support[11)] stiffness, mass, and damping characteristics, including effects of frequency dependent variation and for which

   1) the vendor shall state the support stiffness values used in the analysis and the basis for these values (e.g. modal tests of similar rotor support systems, or calculated support stiffness values),

   2) for machines whose bearing support stiffness values are less than or equal to 3,5 times the bearing oil-film stiffness values, support stiffness values derived from modal testing or calculated frequency dependent support stiffness and damping values (impedances) shall be used, and

   3) the value used for support stiffness should, in most cases, not exceed $0{,}875 \times 10^6$ N/mm (5×10^6 lbf/in);

11) The term “support” includes the foundation or support structure, the base, the machine frame, and the bearing housing as appropriate.

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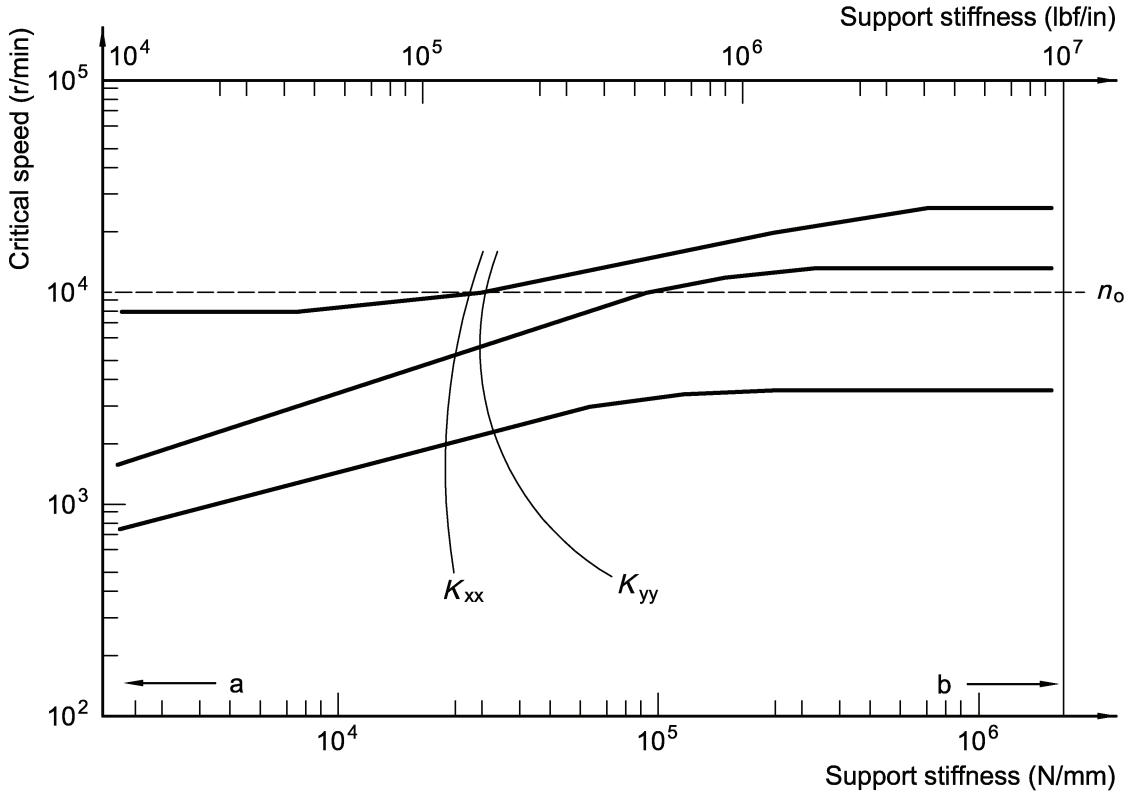

Key

a low amplification factors

b high amplification factors

K_{xx} calculated support stiffness, horizontal

K_{yy} calculated support stiffness, vertical

n_o operating speed

Figure 1 — Undamped critical speed map

e) 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;

f) the influence, over the operating range, of values for hydrodynamic stiffness and damping generated by the casing end seals;

g) location and orientation of the radial vibration probes which shall be the same in the analysis as in the machine.

● **9.2.5** 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). In particular, this analysis should be considered for machinery trains with rigid couplings.

9.2.6 A separate damped unbalanced response analysis shall be conducted for each critical speed within the speed range of 0 % to 125 % of trip speed. Unbalance shall analytically be placed at the locations that have been determined, by the undamped analysis, to most adversely affect the particular mode. For the translatory (symmetric) modes, the unbalance shall be based upon the sum of the journal static loads and shall be applied at the location of maximum displacement. For conical (asymmetric) modes, the unbalance shall be added at the location of maximum displacement nearest to each journal bearing. These unbalance masses shall be 180° out of phase and of magnitude based on the static load on the adjacent bearing. Figure 2 shows the typical mode shapes and indicates the locations and definitions of U for each of the shapes. The magnitude of the unbalance mass shall be four times the value of U as calculated by Equation (1).

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In SI units:

$$U = \frac{6\,350 \times m}{n} \tag{1}$$

In US Customary units:

$$U = \frac{4 \times m}{n}$$

where

U is the input unbalance to be used in the rotor dynamic response analysis, expressed in gram-millimetres (ounce-inches);

m is the mass, as follows, expressed in kilograms (pounds):

— the rotor mass, for symmetric modes [see Figure 2 a) and c)];

— that portion of the rotor mass supported by the adjacent bearing, for asymmetric modes [see Figure 2 b) and d)];

— the overhung mass, i.e. the mass of the rotor outboard of the bearing, for overhung cantilevered modes [see Figure 2 e)];

— the overhung mass, i.e. the mass of the rotor outboard of the bearing, or the mass load on the adjacent bearing, whichever is greater, for overhung rigid modes [see Figure 2 f)];

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

9.2.7 As a minimum, the unbalanced response analysis shall produce the following:

a) identification of the frequency of each critical speed in the range from 0 % to 125 % of the trip speed;

b) frequency, phase, and response amplitude data (Bode plots) at the vibration probe locations through the range of each critical speed resulting from the unbalance specified in 9.2.6;

c) the plot of the deflected rotor shape for each critical speed resulting from the unbalances specified in 9.2.6, showing the major-axis amplitude at each coupling plane of flexure, the centrelines of each bearing, the locations of each radial probe, and at each seal area throughout the machine as appropriate, while, in addition, the minimum design diametral running clearance of the seals shall be indicated;

d) additional Bode plots that compare absolute shaft motion with shaft motion relative to the bearing housing for machines where the support stiffness is less than 3,5 times the oil film thickness.

9.2.8 Additional analyses shall be made for use with the verification test specified in 9.3. The location and magnitude of the unbalance shall be determined by the vendor and approved by the purchaser. Any test stand parameters which influence the results of the analysis shall be included.

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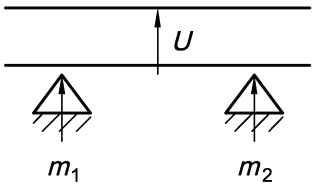
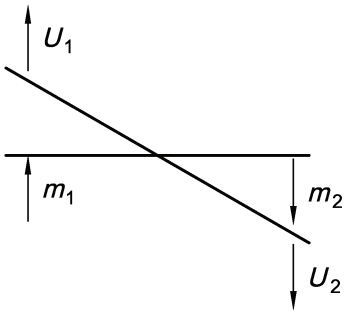


SI units: $U = 6\,350\ (m_1 + m_2) / n$

USC units: $U = 4\ (m_1 + m_2) / n$

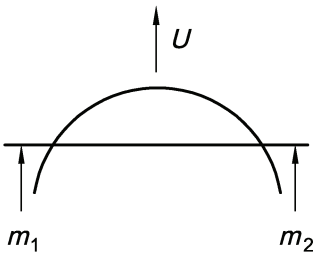
a) Translatory first rigid



SI units: $U_1 = 6\,350\ m_1 / n$
$U_2 = 6\,350\ m_2 / n$

USC units: $U_1 = 4\ m_1 / n$
$U_2 = 4\ m_2 / n$

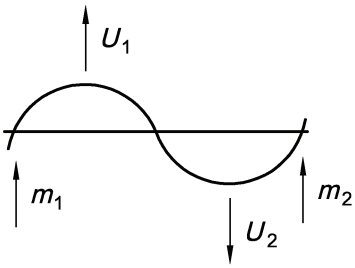
b) Conical, rocking second rigid



SI units: $U = 6\,350\ (m_1 + m_2) / n$

USC units: $U = 4\ (m_1 + m_2) / n$

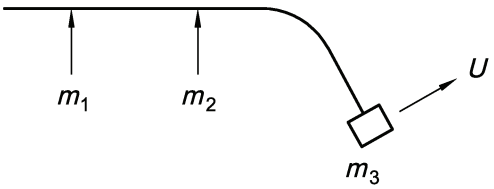
c) First bending



SI units: $U_1 = 6\,350\ m_1 / n$
$U_2 = 6\,350\ m_2 / n$

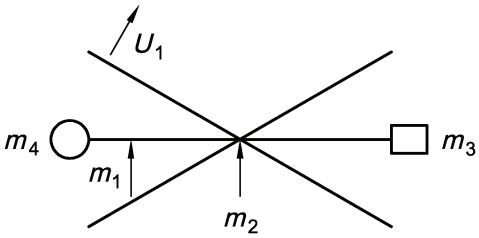
USC units: $U_1 = 4\ m_1 / n$
$U_2 = 4\ m_2 / n$

d) Second bending



SI units: $U = 6\,350\ m_3 / n$
USC units: $U = 4\ m_3 / n$

e) Overhung, cantilevered



SI units: $U_1 = 6\,350\ m / n$
USC units: $U_1 = 4\ m / n$
(where m is the larger of m_1 or m_4)

f) Overhung, rigid

Key

U unbalance mass, expressed in gram-millimetres (ounce-inches)

m see 9.2.6

n operating speed nearest speed of concern, expressed in revolutions per minute

The position of the arrows labelled U indicates the location of the applied unbalances.

Figure 2 — Typical mode shapes

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9.2.9 The damped unbalanced response analysis shall indicate that the machine meets the following separation margins (see Figure 3).

a) If the amplification factor at a particular critical speed is less than 2,5, the response is considered critically damped and no separation margin is required.

b) If the amplification factor at a particular critical speed is 2,5 or greater, and that critical speed is below the minimum speed, the separation margin (M_s), as a percentage of the minimum speed, shall not be less than the value from Equation (2) or the value of 16, whichever is less.

$$M_s = 17 \times \left[1 - \frac{1}{F_a - 1{,}5} \right] \tag{2}$$

where

M_s is the separation margin;

F_a is the amplification factor.

c) If the amplification factor at a particular critical speed is 2,5 or greater, and that critical speed is above the maximum continuous speed, the separation margin, as a percentage of the maximum continuous speed, shall not be less than the value from Equation (3) or the value of 26, whichever is less.

$$M_s = 10 + 17 \times \left[1 - \frac{1}{F_a - 1{,}5} \right] \tag{3}$$

where

M_s is the separation margin;

F_a is the amplification factor.

9.2.10 The calculated unbalanced peak-to-peak amplitudes (see 9.2.7) shall be multiplied using the factor calculated using Equation (4).

$$F_c = \frac{A_1}{A_{4x}} \tag{4}$$

where

F_c is the correction factor;

A_1 is the amplitude limit, calculated using Equation (5), in µm (mil);

A_{4x} is the calculated peak-to-peak amplitude at the probe location in accordance with 9.2.7.b in µm (mil) peak to peak.

In SI units:

$$A_1 = 25{,}4 \times \sqrt{\frac{12\ 000}{n}} \tag{5}$$

In USC units:

$$A_1 = \sqrt{\frac{12\ 000}{n}}$$

where

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

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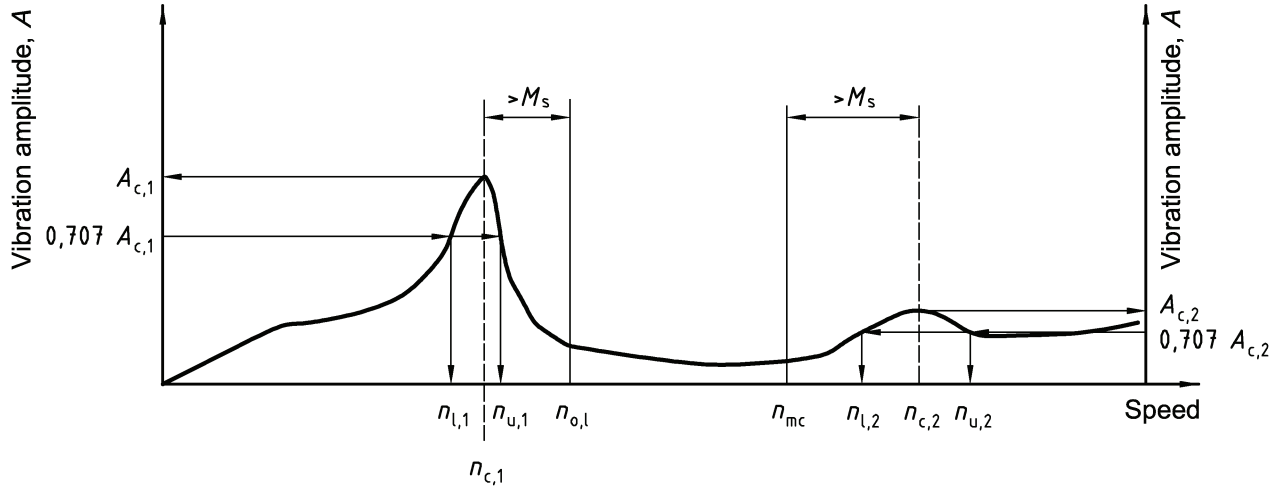

Key

$n_{c,n}$	critical speed n	n_{mc}	maximum continuous speed, 105 % rated speed
$n_{l,n}$	lower speed at 0,707 of amplitude at critical speed n	$A_{c,n}$	amplitude at critical speed n
$n_{u,n}$	upper speed at 0,707 of amplitude at critical speed n	M_s	separation margin, function of F_a
$n_{o,l}$	minimum operating speed	F_a	amplification factor = $n_{c,n}$ / ($n_{u,n} - n_{l,n}$)

NOTE1 Operating speed range is from $n_{o,l}$ to n_{mc}

NOTE2 The shape of the curve is for illustration only and does not necessarily represent any actual rotor response plot; multiple critical speeds can be present below and above operating speed range.

Figure 3 — Rotor response plot

9.2.11 The calculated major-axis, peak-to-peak, unbalanced rotor response amplitudes, corrected in accordance with 9.2.10, at any speed from zero to trip speed, shall not exceed 75 % of the minimum design diametral running clearances throughout the machine.

NOTE Running clearances can be different from the assembled clearances with the machine shut down.

9.3 Unbalanced rotor response verification test

9.3.1 An unbalanced rotor response test shall be performed as part of the mechanical running test (see 16.3.3), and the results shall be used to verify the analytical model. The actual response of the rotor on the test stand to the same arrangement of unbalance as was used in the analysis specified in 9.2.8 shall be the criterion for determining the validity of the damped unbalanced response analysis. To accomplish this, the procedure given in a) to f), shall be performed.

a) During the mechanical running test (see 16.3.3), record the amplitudes and phase angle of the shaft vibration, from zero to trip speed. Preset the gain of any analog recording instruments used 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 This set of readings is normally taken during coast-down, with convenient increments of speed such as 50 r/min. Since the rotor is fully balanced at this point, any vibration amplitude and phase detected is usually the result of residual unbalance and mechanical and electrical runout.

b) Establish the location of all critical speeds below the trip speed.

c) Add the unbalance agreed upon in accordance with 9.2.8 and which was used in the analysis performed in 9.2.8 to the rotor in the location used in the analysis. The unbalance shall not exceed eight times the value of U calculated in accordance with 9.2.6, Equation (1).

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d) Bring the machine up to the operating speed nearest to the critical speed in question and record the indicated vibration amplitudes and phase using the same procedure used for a) above.

e) Vectorially subtract the corresponding indicated vibration data taken in accordance with a) from the results of this test.

f) Compare the results of the mechanical run, including the unbalance response verification test, with those from the analytical model specified in 9.2.8.

9.3.2 The vendor shall correct the model if it fails to meet either of the criteria contained in a) and b) below.

a) The actual critical speeds determined on test shall not deviate from the corresponding critical speeds predicted by analysis by more than 5 %. Where the analysis predicts more than one critical speed in a particular mode (due, for example, to the bearing characteristics being significantly different horizontally and vertically or between the two ends of the machine), the test value shall not be lower than 5 % below the lowest predicted value nor higher than 5 % above the highest predicted value.

   It is possible, particularly on electric motors, that the vertical and horizontal stiffnesses are significantly different and the analysis predicts two differing critical speeds. Should the operating speed fall between these critical speeds, these two critical speeds should be treated separately, as if they resulted from separate modes.

b) The actual major axis amplitude of peak responses from test, including those critically damped, shall not exceed the predicted values. The predicted peak response amplitude range shall be determined from the computer model based on the four radial probe locations.

9.3.3 If the support stiffness is less than two times the bearing oil film stiffness, the absolute vibration of the bearing housing shall be measured and vectorially added to the relative shaft vibration, in both the balanced [9.3.1 a)] and in the unbalanced [9.3.1 d)] condition, before proceeding with the step specified in 9.3.1 e). In such a case, the measured response shall be compared with the predicted absolute shaft movement.

9.3.4 Unless otherwise specified, the verification test of the rotor unbalance shall be performed only on the first rotor tested, if multiple identical rotors are purchased.

9.3.5 The vibration amplitudes and phase from each pair of x-y vibration probes shall be vectorially summed at each vibration response peak after correcting the model, if required, to determine the maximum amplitude of vibration. The major axis amplitudes of each response peak shall not exceed the limits specified in 9.2.11.

9.4 Additional testing

9.4.1 Additional testing is required (see 9.4.2) if, from the corrected damped unbalance response analysis (9.2, 9.3.2), either of the following conditions exist:

a) any critical response fails to meet the separation margin requirements of 9.2.9;

b) the clearance requirements of 9.2.11 have not been met.

9.4.2 Unbalance masses shall be placed as described in 9.2.6; this may require disassembly of the machine. Unbalance magnitudes shall be achieved by adjusting the indicated 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 9.2.10, Equation (5) at the maximum continuous speed; however, the unbalance used shall be no less than two or greater than eight times the unbalance limit specified in 9.2.6, Equation (1). The measurements from this test, taken in accordance with 9.3.1 a) and 9.3.1 b) shall meet the following criteria:

a) at no speed outside the operating speed range, including the separation margins, shall the shaft deflections exceed 90 % of the minimum design running clearances;

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b) at no speed within the operating speed range, including the separation margins, shall the shaft deflections exceed 55 % of the minimum design running clearances or 150 % of the allowable vibration limit at the probes (see 9.2.10).

9.4.3 The internal deflection limits specified in 9.4.2 a) and 9.4.2 b) shall be based on the calculated displacement ratios between the probe locations and the areas of concern identified in 9.2.11, based on a corrected model if required. Actual internal displacements for these tests shall be calculated by multiplying these ratios by the peak readings from the probes. Acceptance shall be based on these calculated displacements or inspection of the seals if the machine is opened. 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.

9.5 Torsional analysis

- **9.5.1** For units including gears or generators, or for units comprising three or more coupled machines, or if specified, the vendor having unit responsibility shall ensure that a torsional vibration analysis of the complete coupled train is performed and shall be responsible for directing any modifications necessary to meet the requirements of 9.5.2 to 9.5.6.

9.5.2 Excitation of torsional natural frequencies could come from many sources which might or might not be a function of running speed and should be considered in the analysis. These sources to be considered shall include at least the following:

a) gear problems such as unbalance, pitchline runout, and cumulative pitch error;

b) cyclic process impulses;

c) torsional transients such as start-up of synchronous electric motors and generator phase-to-phase or phase-to-ground faults;

d) torsional excitation resulting from electric motors and rotary type positive displacement machines;

e) control loop resonances from hydraulic, electronic governors, and variable frequency drives;

f) one or two times line frequency;

g) running speed or speeds;

h) harmonic frequencies from variable frequency drivers when motor drivers are part of the drive train.

9.5.3 The torsional natural frequencies 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 governor speed to maximum continuous speed.

9.5.4 Torsional natural frequencies at two or more times running speeds shall be avoided or, in systems in which corresponding excitation frequencies occur, shall be shown to have no adverse effect.

9.5.5 When torsional resonances are calculated to fall within the margin specified in 9.5.3 (and the purchaser and vendor have agreed that all efforts to remove the critical from within the limiting frequency range have been exhausted), a stress analysis shall be performed to demonstrate that the resonances have no adverse effect on the complete train. The assumptions made in this analysis regarding the magnitude of excitation and the degree of damping shall be clearly stated. The acceptance criteria for this analysis shall be mutually agreed upon by the purchaser and vendor.

9.5.6 In addition to the torsional analysis required in 9.5.2 and 9.5.5, the vendor shall perform a transient torsional vibration analysis when synchronous motor or generator drives are part of the train. The acceptance criteria for this analysis shall be mutually agreed upon by the purchaser and vendor.

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9.6 Vibration and balancing

- **9.6.1** Major parts of the rotating element, such as the shaft, balancing drum, and discs, shall be individually dynamically balanced before assembly to ISO 1940 Grade G1 or better. When a bare shaft with a single keyway is dynamically balanced, the keyway shall be filled with a fully crowned half key in accordance with ISO 8821. Keyways 180° apart, but not in the same transverse plane, shall be filled. The initial balance correction to the bare shaft shall be recorded. The components to be mounted on the shaft (balance drum, disc, etc.) shall be balanced in accordance with the "half-key-convention" as described in ISO 8821.

9.6.2 The rotating element shall be multiplane dynamically balanced during assembly. This shall be accomplished after the addition of no more than two major components, except on integral disc rotating elements. Balancing correction shall only be applied to the elements added. Minor correction of other components may be required during the final trim balancing of the completely assembled element. In the sequential balancing process, any half-keys used in the balancing of the bare shaft (see 9.6.1) shall continue to be used until they are replaced with the final key and mating element. On rotors with single keyways, the keyway shall be filled with a fully crowned half-key. 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 C). The maximum allowable residual unbalance per plane (journal) shall be calculated as follows:

In SI units:

$$U_{\text{max}} = \frac{6\,350 \times m}{n} \tag{6}$$

In USC units:

$$U_{\text{max}} = \frac{4 \times m}{n}$$

where

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

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

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

- **9.6.3** If specified, completely assembled rotating elements shall be subject to high-speed (at rated speed) balancing in lieu of a sequential low speed balancing (see 9.6.2). When the vendor's standard balancing method is by high-speed balancing in lieu of a sequential low-speed balancing and high-speed balancing is not specified, it may be used with the purchaser's approval. The high-speed balance shall be in accordance with 9.6.4.

9.6.4 When the complete rotating element has been specified to be high-speed balanced (9.6.3), the rotor shall be supported in bearings of the same type and with similar dynamic characteristics as those in which it will be supported in service. The final check balance shall be performed at maximum continuous speed. Before making any corrections (unless it is necessary to improve the initial balance in order to be able to run the rotor at high speed), the rotor shall be run in the balancing machine at trip speed for at least 5 min to allow seating of any shrunk-on components.

9.6.5 Unless otherwise specified, the vibration acceptance criteria for high-speed balancing, with maximum pedestal stiffness at all speeds, as measured on the bearing cap, shall be as follows:

a) for speeds above 3 000 r/min, it shall not exceed the "greater" of 7 400/n mm/s (291/n in/s) or 1 mm/s (0,039 in/s), where n is the speed, in revolutions per minute;

b) for speeds at or less than 3 000 r/min, it shall not exceed 2,5 mm/s (0,098 in/s).

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- **9.6.6** A rotor that is to be high-speed balanced shall, if specified, first receive a sequential low-speed balance as specified in 9.6.2.

- **9.6.7** For a rotor that has been low-speed balanced (see 9.6.2), and if specified for rotors that are high–speed balanced (see 9.6.3), a low-speed residual unbalance check shall be performed in a low-speed balance machine in accordance with Annex C and recorded on the residual unbalance work sheet.

NOTE This procedure provides a reference of residual unbalance and phase for future use in a low-speed balance machine.

9.6.8 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 25 µm (1,0 mil), whichever is less.

In SI units:

$$A = 25{,}4 \times \sqrt{\frac{12\,000}{n}} \tag{7}$$

In USC units:

$$A = \sqrt{\frac{12\,000}{n}}$$

where

A is the amplitude of unfiltered vibration, measured in µm (mil) peak to peak;

n is the maximum continuous speed, measured 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.

IMPORTANT — Do not confuse these limits with the limits specified in 9.3 for shop verification of unbalanced response.

9.6.9 Electrical and mechanical runout shall be determined by rotating the rotor throughout the full 360° supported in V-blocks at the journal centres while measuring runout with a noncontacting vibration probe and a dial indicator at the centreline of each probe location and one probe-tip diameter to either side.

9.6.10 Accurate records of electrical and mechanical runout, for the full 360° at each probe location, shall be included in the mechanical test report.

10 Bearings, bearing housings, and seals

10.1 Radial bearings

10.1.1 Hydrodynamic radial bearings shall be provided. They shall be split for ease of assembly, precision bored, and sleeved or padded, with steel-backed, babbitted replaceable liners, pads or shells. These bearings shall be equipped with antirotation pins and shall be positively secured in the axial direction. The liners, pads or shells shall be in axially split housings and shall be replaceable without the removal of the top half of the casing of an axially split machine or the head of a radially split unit and without the removal of the coupling hub.

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10.1.2 Radial bearing design shall be such that load transfer to the bearing housing by the bearing shall not cause damage to critical bearing housing surfaces which are not considered normal wearing surfaces subject to repair or replacement.

10.2 Thrust bearings and collars

10.2.1 Thrust bearings shall be hydrodynamic and steel-backed, with babbitted multiple segments, designed for equal thrust capacity in both directions, and arranged for continuous pressurized lubrication to each side. Tilting pads shall be used on both sides and shall incorporate a self-levelling feature ensuring that each pad carries an equal share of the thrust load with minor variations in pad thickness.

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

a) fouling and variation in seal clearances up to twice the design internal clearances;

b) step thrust from all diameter changes;

c) stage reaction and stage differential pressure;

d) variations in inlet, extraction, induction, and exhaust pressure;

e) external loads from the driven equipment in accordance with 10.2.3 and 10.2.4.

10.2.3 External loads transmitted through flexible-element couplings shall be calculated on the basis of the maximum allowable deflection permitted by the coupling manufacturer.

10.2.4 If two or more rotor thrust forces are to be carried by one thrust bearing, 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.

10.2.5 Thrust bearings shall be selected at no more than 50 % of the bearing manufacturer's ultimate load rating. The ultimate load rating is the load that produces the minimum acceptable oil-film thickness without inducing failure during continuous service or the load that does not exceed the creep-initiation or yield strength of the babbitt at the maximum temperature location on the pad, whichever load is less. In sizing thrust bearings, consideration shall be given to, but shall not be limited to, the following for each specific application:

a) the shaft speed;

b) the temperature of the bearing babbitt;

c) the deflection of the bearing pad;

d) the minimum oil-film thickness;

e) the feed rate, viscosity and supply temperature of the oil;

f) the design configuration of the bearing;

g) the babbitt alloy;

h) the pad material;

i) the turbulence of the oil film.

The basis for sizing of thrust bearings shall be made available for review by the purchaser.

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

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10.2.7 Unless otherwise specified, integral thrust collars shall be furnished. They shall be provided with at least 3 mm (1/8 in) of additional stock on total thickness to enable refinishing if the collar is damaged. When replaceable collars are furnished, they shall be shrunk on, and positively locked to, the shaft to prevent fretting.

10.2.8 Both faces of thrust collars shall be finished to a surface roughness not exceeding 0,8 µm (32 µin) Ra. The axial total indicated runout of either face shall not exceed 13 µm (0,000 5 in).

10.3 Bearing housing

10.3.1 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. The rise in oil temperature through the bearing and housings shall not exceed 30 °C (50 °F) under the most adverse specified operating conditions. The bearing outlet oil temperature shall not exceed 80 °C (180 °F). When the inlet oil temperature exceeds 50 °C (120 °F), special consideration shall be given to bearing design, oil flow, and allowable temperature rise. 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.

10.3.2 Bearing housings shall be equipped with replaceable labyrinth end seals and deflectors where the shaft passes through the housing; lip-type end 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.

10.3.3 Cantilevered shaft support structures bolted to steel casings shall be steel.

10.3.4 Cast iron bearing housings or bearing housing supports shall not be used.

10.3.5 Provision shall be made for mounting two non-contacting 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. The probe installation shall be as specified in API Std 670.

● **10.3.6** When specified, provisions for mounting accelerometers on the bearing housings shall be made in accordance with API Std 670.

10.4 Grounding

Unless otherwise specified, condensing turbines shall be provided with at least two grounding brushes on the same end of the shaft. Unless otherwise specified, the arrangement of the brushes shall be designed to enable the brushes to be replaced with the turbine in operation. The brushes shall be a standard size and grade suitable for the service. The vendor shall include drawings with the proposal showing the number and location of the brushes.

10.5 Shaft seals

10.5.1 Unless otherwise specified, casing end seals shall be replaceable labyrinth seals.

10.5.2 Interstage sealing shall be by replaceable labyrinths.

10.5.3 Labyrinth casing end seals operating at less than atmospheric pressure shall be designed for admission of dry steam to seal against air ingress. Piping with pressure gauges, regulators and other necessary valves shall be provided to interconnect the end labyrinth seals. The piping shall have one common connection to the purchaser's sealing steam supply. The admission and the pressure of the sealing steam shall be automatically controlled. The normal operating sealing steam supply should preferably come from a positive pressure section of the turbine.

10.5.4 Unless otherwise specified, a separate vacuum system shall be furnished to reduce external leakage from the end labyrinth seals and possible contamination of the bearing oil (15.6). Unless otherwise specified, the system shall be supplied loose for mounting and connection by others. Annex G shows a typical labyrinth end seal vacuum system.

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