

Special-purpose Gears for Petroleum, Chemical, and Gas Industry Services

API STANDARD 613
SIXTH EDITION, JULY 2021



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Should: As used in a standard, “should” denotes a recommendation or that which is advised but not required in order to conform to the standard.

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Introduction

Users of this standard should be aware that further or differing requirements may be needed for individual applications. This standard is not intended to inhibit a gear 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 gear vendor should identify any variations from this standard and provide details.

ANNEX A—Special-purpose Gear Unit Datasheets (informative).

ANNEX B—Bearing Temperature Limits (informative).

ANNEX C—Material Specifications for Special-purpose Gear Units (informative).

ANNEX D—Contract Documents and Engineering Design Data (informative).

ANNEX E—Vendor Drawing and Data Requirements (normative).

ANNEX F—Procedure for the Verification of Residual Unbalance (normative).

ANNEX G—Gear Tooth Quality Inspection (informative).

ANNEX H—Inspector's Checklist (informative).

ANNEX I—Special Considerations in Very High Speed Gears (informative).

This standard requires the purchaser to specify certain details and features. A bullet (•) at the beginning of a subsection or paragraph indicates that either a decision by, or further information from, the purchaser is required. Further information should be shown on the datasheets (see example in Annex A) or stated in the quotation request and purchase order.

In this standard, U.S. customary (USC) units are included in brackets for information.

Special-purpose Gears for Petroleum, Chemical, and Gas Industry Services

1 Scope

This standard covers the minimum requirements for special-purpose, enclosed, precision involute gear single- and double-helical one- and two-stage speed increasers and reducers of parallel-shaft design for use in petroleum, chemical, and gas industry services. This standard is intended for gear units that are in continuous service without installed spare equipment. Gear sets furnished to this standard are considered matched sets.

NOTE The purchase of a spare set of gear rotors or a complete gear unit does not mean that the gear unit in continuous service has an installed spare.

This standard includes related lubricating systems, controls, instrumentation, and other auxiliary equipment. This standard is not intended to apply to gear units in general-purpose service, which are covered by API 677; to gears integral with other equipment, such as integrally geared compressors and expanders covered by API 617 or API 672; or to gears other than involute helical.

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. Standards referenced in the document are undated but refer to the specific editions referenced in this section.

ANSI ¹/API Standard 614, *Lubrication, Shaft-sealing and Oil-control Systems and Auxiliaries*

API Standard 670, *Machinery Protection Systems*

API Standard 671, *Special-purpose Couplings for Petroleum, Chemical, and Gas Industry Services*

API Recommended Practice 684, *Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals and Rotor Balancing*

API Recommended Practice 686, *Machinery Installation and Installation Design*, Second Edition, 2009

AGMA 927 ², *Load Distribution Factors—Analytical Methods for Cylindrical Gears*

ANSI/AGMA/ISO 1328-1 ³, *Cylindrical gears—ISO system of flank tolerance classification—Part 1: Definitions and allowable values of deviations relevant to flanks of gear teeth*

ANSI/AGMA 2001-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth (U.S. Units)*

ANSI/AGMA 2101-D04, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth (Metric)*

ANSI/AGMA 6001, *Design and Selection of Components for Enclosed Gear Drives*

ANSI/AGMA 6011-J14, *Specification for High Speed Helical Gear Units*

ANSI/AGMA 6025, *Sound for Enclosed Helical, Herringbone and Spiral Bevel Gear Drives*

¹ American National Standards Institute, 25 West 43rd Street, 4th Floor, New York, NY 10036, www.ansi.org.

² American Gear Manufacturers Association, 1001 N. Fairfax Street, Suite 500, Alexandria, VA 22314-1587, www.agma.org.

³ International Organization for Standardization, Chemin de Blandonnet 8, CP 401, 1214 Vernier, Geneva, Switzerland, www.iso.org.

ANSI/ASME B1.1 ⁴, *Unified Inch Screw Threads (UN, UNR, and UNJ Thread Forms)*

ASME B1.13M, *Metric Screw Threads: M Profile*

ANSI/ASME B1.20.1, *Pipe Threads, General Purpose (Inch)*

ANSI/ASME B16.1, *Gray Iron Pipe Flanges and Flanged Fittings, Classes 25, 125, and 250*

ANSI/ASME B16.5, *Pipe Flanges and Flanged Fittings: NPS 1/2 through NPS 24 Metric/Inch Standard*

ANSI/ASME B16.11, *Forged Fittings, Socket-Welding and Threaded*

ANSI/ASME B16.42, *Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*

ANSI/ASME B17.1, *Keyways and Keyseats*

ASME B31.3, *Process Piping*

ASME Boiler and Pressure Vessel Code, Section VIII: *Rules for Construction of Pressure Vessels; Division 1*

ASME Boiler and Pressure Vessel Code, Section IX: *Welding, Brazing, and Fusing Qualifications*

ASTM E94, *Standard Guide for Radiographic Examination*

ASTM E165/A165M ⁵, *Standard Practice for Liquid Penetrant Testing for General Industry*

ASTM A269/A269M, *Standard Specification for Seamless and Welded Austenitic Stainless Steel Tubing for General Service*

ASTM A275/A275M, *Standard Practice for Magnetic Particle Examination of Steel Forgings*

ASTM A312/A312M, *Standard Specification for Seamless, Welded, and Heavily Cold Worked Austenitic Stainless Steel Pipes*

ASTM A388/A388M, *Standard Practice for Ultrasonic Examination of Steel Forgings*

ASTM A578/A578M, *Standard Specification for Straight-Beam Ultrasonic Examination of Rolled Steel Plates for Special Applications*

ASTM A609/A609M, *Standard Practice for Castings, Carbon, Low-alloy, and Martensitic Stainless Steel, Ultrasonic Examination Thereof*

ASTM D5445 *Standard Practice for Pictorial Markings for Handling of Goods*

ASTM E125, *Standard Reference Photographs for Magnetic Particle indications on Ferrous Castings*

ASTM E709, *Standard Guide for Magnetic Particle Testing*

ASTM E1417/E1417M, *Standard Practice for Liquid Penetrant Testing*

EN 1092-1 ⁶, *Flanges and their joints—Circular flanges for pipes, valves, fittings and accessories, PN designated Part 1: Steel flanges*

EN 1092-2, *Flanges and their joints—Circular flanges for pipes, valves, fittings and accessories, PN designated—Part 2: Cast iron flanges*

⁴ ASME International, 2 Park Avenue, New York, NY 10016-5990, www.asme.org.

⁵ ASTM International, PO Box C700, 100 Barr Harbor Drive, West Conshohocken, PA 19428, www.astm.org.

⁶ European Committee for Standardization (CEN-CENELEC), Rue de la Science 23, B-1040 Brussels, Belgium, www.cen.eu.

ANSI/IEC ⁷ 60529, *Degrees of protection provided by enclosures (IP Code)*

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 3448, *Industrial liquid lubricants—ISO viscosity classification*

ISO 3452 (all parts), *Non-destructive testing—Penetrant testing*

ISO 4386-1, *Plain bearings—Metallic multilayer plain bearings—Part 1: Non-destructive ultrasonic testing of bond of thickness greater than or equal to 0,5 mm*

ISO 6149-1, *Connections for hydraulic fluid power and general use—Ports and stud ends with ISO 261 metric threads and O-ring sealing—Part 1: Ports with truncated housing for O-ring seal*

ISO 6336-5, *Calculation of load capacity of spur and helical gears—Part 5: Strength and quality of materials*

ISO 6336-22, *Calculation of load capacity of spur and helical gears—Part 22: Calculation of micropitting load capacity*

ISO 6708, *Pipework components—Definition and selection of DN (nominal size)*

ISO 8068, *Lubricants, industrial oils and related products (class L)—Family T (turbines)—Specification for lubricating oils for turbines*

ISO 9606-1, *Qualification testing of welders—Fusion welding—Part 1: Steels*

ISO 9614-2, *Acoustics—Determination of sound power levels of noise sources using sound intensity—Part 2: Measurement by scanning*

ISO 9934 (all parts), *Non-destructive testing—Magnetic particle testing*

ISO/TR 10064-4, *Code of inspection practice—Part 4: Recommendations relative to surface texture and tooth contact pattern checking*

ISO 14635-1, *Gears—FZG test procedures—Part 1: FZG test method A/8,3/90 for relative scuffing load-carrying capacity of oils*

ISO 15607, *Specification and qualification of welding procedures for metallic materials—General rules*

ISO 16810, *Non-destructive testing—Ultrasonic testing—General principles*

ISO 17638, *Non-destructive testing of welds—Magnetic particle testing*

ISO 21940-11, *Mechanical vibration—Rotor balancing—Part 11: Procedures and tolerances for rotors with rigid behaviour*

ISO 21940-32, *Mechanical vibration—Rotor balancing—Part 32: Shaft and fitment key convention*

ISO 23277, *Non-destructive testing of welds—Penetrant testing—Acceptance levels*

ISPM 15 ⁸, *Regulation of Wood Packaging Material in International Trade*

⁷ International Electrotechnical Commission, 3 rue de Varembé, 1st Floor, PO Box 131, CH-1211 Geneva 20, Switzerland, www.iec.ch.

⁸ International Standards for Phytosanitary Measures, International Plant Protection Convention Secretariat, Food and Agriculture Organization of the United Nations, AGDI, Viale delle Terme di Caracalla, 00153 Rome, Italy, www.ippc.int/en.

NEMA 250⁹, *Enclosures for Electrical Equipment (1000 Volts Maximum)*

SAE J518¹⁰, *Hydraulic Flanged Tube, Pipe, and Hose Connections*

SAE J1926-1, *Connections for General Use and Fluid Power—Ports and Stud Ends with ASME B1.1 Threads and O-Ring Sealing—Part 1: Threaded Port with O-Ring Seal in Truncated Housing*

3 Terms, Definitions, Acronyms, Abbreviations, and Symbols

3.1 Terms and Definitions

For the purposes of this document, the following definitions apply.

3.1.1

alloy steel

Steel that is alloyed with a variety of elements in total amounts between 1.0 % and 50 % by weight.

3.1.2

anchor bolts

Bolts used to attach the equipment to the support structure (concrete foundation or steel structure).

3.1.3

approve

Written documentation confirming an agreement.

3.1.4

bending stress number

σ_{FP}

The bending stress number corresponds to the likelihood of fatigue cracking at the tooth root fillet.

NOTE If the tooth root stress is excessive, it can lead to failure of the gear teeth.

3.1.5

close clearance location

A close clearance location in a gear unit is any location on the rotor where the diametral clearance is less than the bearing clearance, with the exception of the bearings themselves, floating rings, and abradable or compliant seals.

3.1.6

contact stress number

σ_{HP}

The contact stress number corresponds to the likelihood of developing macropitting on the tooth load flank.

3.1.7

critical speed

Shaft rotational speed at which the rotor-bearing-support system is in a state of resonance.

3.1.8

design

Gear vendor's calculated parameter.

NOTE This is a term used by the gear vendor to describe various parameters such as design power, design pressure, design temperature, or design speed. It is not intended for the purchaser to use this term.

⁹ National Electrical Manufacturers Association, 1300 17th Street N, Suite 900, Rosslyn, VA 22209, www.nema.org.

¹⁰ SAE International, 400 Commonwealth Drive, Warrendale, PA 15096, www.sae.org.

3.1.9**equipment**

Machinery and components intended for use in the plant.

3.1.10**gauss levels**

Magnetic field level of a component measured with a “Hall effect” probe, with no interference from an adjacent magnetic part or structure.

3.1.11**gear**

Refers to either the pinion or gear wheel.

3.1.12**gear tooth service factor****SF**

Factor that is applied in the tooth power rating calculations to account for differences in potential overload, shock load, and/or continuous oscillatory torque characteristics.

NOTE 1 The appropriate gear tooth service factor depends on the characteristics of both the driver and the driven equipment.

NOTE 2 The gear tooth service factor is not to be confused with the fatigue safety factor of other gear unit components such as shafts.

3.1.13**gear unit**

Complete power transmission assembly including the housing, gears, and bearings.

3.1.14**gear unit rated power**

The power that is specified by the purchaser and stamped on the gear unit nameplate.

3.1.15**gear unit responsibility**

Obligation for coordinating the documentation, delivery, and technical aspects of the gear unit and all auxiliary systems included in the scope of the order to the gear vendor.

3.1.16**gear vendor**

Manufacturer or manufacturer's agent that supplies the gear unit.

3.1.17**gear wheel**

bull gear

Lowest speed rotor.

3.1.18**horizontally split joint**

Joint split that is horizontal and parallel to the shaft centerline.

3.1.19**hunting tooth combination**

Design combination for a gear pair with tooth numbers such that a tooth on the pinion does not repeat contact with a tooth on the gear wheel until it has contacted all the other gear wheel teeth.

3.1.20**informative**

Information only.

cf. **normative** (3.1.24).

NOTE An informative reference or annex provides advisory or explanatory information. It is intended to assist the understanding or use of the document.

3.1.21**lead modification**

helix modification

intentional deviation of the pinion and/or the gear wheel tooth flank from the theoretical helix form.

NOTE It is intended to improve uniformity of the load distribution across the face width of the mating rotors when the gears are operating at the normal operating point.

3.1.22**maximum continuous speed**

n_{mc}

Highest rotational speed (revolutions per minute) at which the gear unit, as built and tested, is capable of continuous operation.

3.1.23**normal operating point**

Power and speed at which operation is expected and optimum efficiency is desired.

NOTE The power at the normal operating point can be equal to or less than the gear unit rated power.

3.1.24**normative**

required

cf. **informative** (3.1.20).

NOTE A normative reference or annex enumerates a requirement or mandate of the specification.

3.1.25**observed**

cf. **witnessed** (3.1.41).

Inspection or test where the purchaser is notified of the timing of the inspection or test and the inspection or test is performed as scheduled even if the purchaser or their representative is not present.

3.1.26**owner**

Final recipient of the equipment.

NOTE The owner often delegates another agent as the purchaser of the equipment.

3.1.27**pinion**

Higher speed rotor in a meshed gear pair.

3.1.28**pitch line velocity**

The linear velocity of a point on the gear at the radius of the operating pitch circle.

3.1.29**profile modification**

Intentional deviation of the pinion and/or the gear wheel tooth flank profile from the theoretical involute tooth form.

NOTE It is intended to obtain a tooth load transfer distributed evenly along the path of contact in a transverse section when the gears are operating at the normal operating point.

3.1.30**purchaser**

Agency that issues the order and specification to the gear vendor.

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

3.1.31**rated input speed**

Specified rated speed of the gear unit driver (in revolutions per minute), as designated by the purchaser.

3.1.32**rated output speed**

Specified rated speed of the gear unit driven equipment (in revolutions per minute), as designated by the purchaser.

3.1.33**scuffing**

Form of gear tooth surface damage which refers to welding and tearing of the tooth surface by the flank of the mating gear.

NOTE Scuffing occurs when the oil film thickness is small enough to allow asperities on the flanks of the gear teeth to contact and slide against each other.

3.1.34**scuffing risk**

Probability of damage from scuffing represented by a number that is inversely proportional to the oil film's ability to prevent metal-to-metal contact of the gear teeth in the mesh during operation.

3.1.35**special purpose application**

Application for which the equipment is designed for uninterrupted, continuous operation in critical service and for which there is usually no installed spare equipment.

3.1.36**special tool**

Tool that is not a commercially available catalog item.

3.1.37**total indicator reading****total indicated runout****TIR**

Difference between the maximum and minimum readings of a dial indicator or similar device, monitoring a face or cylindrical surface during one complete revolution of the monitored surface.

NOTE For a cylindrical surface, the indicated runout implies an eccentricity equal to half the reading. For a flat face, the indicated runout implies an out-of-squareness equal to the reading. If the diameter in question is not cylindrical or flat, the interpretation of the meaning of TIR is more complex and can represent ovality or surface irregularities.

3.1.38**train responsibility**

Obligation for coordinating the documentation, delivery, and technical aspects of all the equipment and all auxiliary systems included in the machinery train for which the gear unit is intended.

3.1.39**trip**

Automated shutdown to ensure personnel safety (safety critical).

3.1.40**trip speed**

For a fixed speed electric motor driver: the speed corresponding to the synchronous speed of the motor at maximum supply frequency at the motor terminals.

For a variable speed driver such as turbine, engine, expander, hydrostatic driver, or variable speed electric motor: speed at which the independent emergency overspeed device operates to shut down the driver.

3.1.41**witnessed**

Inspection or test where the purchaser is notified of the timing of the inspection or test and a hold is placed on the inspection or test until the purchaser or the purchasers' representative is in attendance.

cf. **observed** (3.1.25).

3.1.42**written**

Communication by letter, facsimile (fax), email or similar method that enables the communication to be archived.

3.2 Acronyms and Abbreviations

AF	amplification factor
CW	clockwise (rotation)
CCW	counterclockwise/anticlockwise (rotation)
DN	nominal diameter
HBW	Brinell hardness number
HRC	Rockwell hardness number (regular hardness scale C)
HR15N	Rockwell hardness number (Rockwell Superficial hardness scale 15N)
ITP	inspection and testing plan
NPS	nominal pipe size
SI	System International (International System of Units)
USC	U.S. customary (units)
VDDR	vendor drawing and data requirements

3.3 Symbols

For the purposes of this document, the following symbols in Table 1 apply.

Table 1—Symbols

Symbol	Term	SI Units	USC Units
a	center distance	mm	in.
b	Face width, net active	mm	in.
C_{SF}	minimum specified service factor, contact stress	—	—
$d_{1,2}$	reference diameter of pinion, wheel	mm	in.
D	internal drain diameter	mm	in.
F_{df}	allowable lube oil drain flow	l/min	gals/min
k_{FP}	tooth root bending derating factor	—	—
k_{HP}	tooth flank pitting derating factor	—	—
K_{SF}	minimum specified service factor, bending	—	—
L	total face width including gap width	mm	in.
m_n	normal module	mm	—
m_t	transverse module	mm	—
$n_{1,2}$	rotational speed of pinion, wheel, nominal	min ⁻¹	rpm
P	gear rated power	kW	HP
R_a	arithmetic average roughness value	μm	μin.
SM_a	actual separation margin, %	—	—
SM_r	required separation margin, %	—	—
u	gear ratio z_2/z_1	—	—
U_r	residual rotor unbalance	g-mm	oz-in.
W_{st}	journal static load	kg	lbm
$z_{1,2}$	number of teeth of pinion, wheel	—	—
u	helix angle at referenced diameter	degrees	degrees
σ_{FP}	allowable bending stress number	N/mm ²	lb/in. ²
σ_{HP}	allowable contact stress number	N/mm ²	lb/in. ²

4 Unit Responsibility

The gear vendor shall assume gear unit responsibility and shall ensure that all sub-vendors comply with the requirements of this standard and all normative reference documents. The technical aspects to be considered by the gear vendor include, but are not limited to, such factors as the power requirements, speed, direction of rotation, general arrangement, couplings, dynamics, lubrication, sealing system, instrumentation, piping, inspection, testing, material test reports, and conformance to specifications.

5 Requirements

● 5.1 Units of Measure

The purchaser shall specify whether data and drawings for equipment supplied to this standard are to use the metric (SI) or U.S. customary (USC) system of measurements. Thread standards on customer connections and instrumentation requirements shall be as specified in the respective sections.

NOTE Dedicated datasheets for SI units and for USC units are provided in Annex A.

5.2 Statutory Requirements

The purchaser and the gear vendor shall determine the measures to be taken to comply with any governmental codes, regulations, ordinances, directives, or rules that are applicable to the equipment, its packaging, and any preservatives used.

● 5.3 Documentation Requirements

The hierarchy of documents shall be specified by the purchaser.

NOTE Typical documents include company and industry specifications, meeting notes, and modifications to these documents.

6 Basic Design

6.1 General

6.1.1 Equipment Reliability

6.1.1.1 Only equipment that is field proven, as defined by the purchaser, is acceptable.

NOTE Purchasers can use their engineering judgment in determining what equipment is field proven. API 691 can provide guidance.

- **6.1.1.2** If specified, the gear vendor shall provide the documentation to demonstrate that all equipment proposed qualifies as field proven.

6.1.1.3 In the event no such equipment is available, the gear vendor shall submit an explanation of how their proposed equipment can be considered field proven.

NOTE A possible explanation can be that all components comprising the assembled machine satisfy the field proven definition.

6.1.2 Gear units purchased according to this standard shall conform to ANSI/AGMA 6011-J14 and to related AGMA standards referenced therein, except as modified or supplemented by this standard.

- **6.1.3** The purchaser shall specify the period of uninterrupted continuous operation. Shutting down the equipment to perform required maintenance or inspection during the specified uninterrupted operation period is not acceptable.

NOTE It is realized that there are some services where this objective is easily attainable and others where it is difficult.

6.1.4 The gear vendor shall advise in the proposal any component designed for a finite life.

NOTE It is recognized that these are design criteria.

6.1.5 The gear unit shall be capable of operating at all specified operating and transient conditions.

- **6.1.6** The purchaser shall indicate if the input or output speed is specified (i.e. shall be exactly adhered to by the gear vendor) and which is nominal (that is, permits some variation). An *S* shall be used to indicate the specified speed, and an *N* shall be used to indicate the nominal speed. The purchaser shall specify the allowable percentage of variation in the designated gear ratio.

NOTE In selecting the number of teeth for the pinion and gear wheel, it is often impractical for the gear vendor to match exactly both the rated input and the rated output speed designated on the datasheets.

6.1.7 Sound Pressure Level

6.1.7.1 Control of the sound pressure level of all equipment furnished shall be a joint effort of the purchaser and the entity having train responsibility.

6.1.7.2 The equipment furnished by the gear vendor shall conform to the maximum allowable sound pressure level specified.

6.1.7.3 The gear vendor shall provide the expected sound pressure level and relative sound spectrum at full-speed/full-load conditions.

- **6.1.7.4** If specified, the gear vendor shall provide a map of points where sound pressure is measured during testing, the measured sound pressure at each point, and the relative spectrum per octave band, along with a record of the operating condition during measurements.

6.1.7.5 During testing, the gear vendor may measure the sound intensity level rather than the sound pressure level and relative sound spectrum.

NOTE 1 The entity having train responsibility can calculate the sound power level from this information.

NOTE 2 ISO 3740, ISO 3744, and ASME PTC 36 or ISO 3746, can be consulted for guidance.

NOTE 3 The sound intensity method allows separation of the disturbing effects from sound emission by other equipment in the vicinity of the gear unit from the sound emission of the gear unit itself. For sound intensity measurement method, ASME PTC 36 or ISO 9614-2 can be consulted for guidance.

6.1.8 Unless otherwise specified, the equipment's maximum continuous speed shall not be less than 105 % of the rated speed for variable speed machines (including adjustable frequency drive controlled electric motors) and shall be equal to the synchronous speed for constant speed electrical machines.

6.1.9 Trip Speed

- **6.1.9.1** The gear unit shall be designed to operate safely at the trip speed and the corresponding torque specified by the purchaser.

The equipment's trip speed shall not be less than the values in Table 2.

Table 2—Driver Trip Speeds

Driver Type	Trip Speed (% of Maximum Continuous Speed)
Steam turbine	
NEMA class A ^a	115 %
NEMA class B, C, D ^a	110 %
Gas turbine	105 %
Variable-speed motor	110 %
Constant-speed motor	100 %
Reciprocating engine	110 %
^a Indicates governor class as specified in NEMA SM 23.	

- **6.1.9.2** For turbine-driven and expander-driven gear units, the purchaser shall specify the momentary speed above trip speed. If the purchaser does not specify the momentary speed above trip speed, a value of 110 % of trip speed shall be used.

NOTE This is intended to address the scenario of sudden full load rejection and subsequent acceleration above trip speed.

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

- **6.1.11** All 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 the specified codes.

6.1.12 Oil reservoirs and housings that enclose moving lubricated parts such as bearings, shaft seals, highly polished parts, instruments, and control elements should be designed to minimize contamination by moisture, dust, and other foreign matter, both during periods of operation and idle time.

6.1.13 The gear unit shall perform on the test stand and on its 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 entity having train responsibility.

- **6.1.14** If specified, the gear vendor shall review and comment on the purchaser's baseplate and foundation drawings.

NOTE Many factors can 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.

- **6.1.15** If specified, the gear vendor's representative shall witness:

- a) the initial shaft alignment check at ambient conditions;
- b) shaft alignment at operating temperature, i.e. hot alignment check.

- **6.1.16** 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.

6.1.17 Gear units shall not require a break-in period at reduced speed and load in the field. Specific break-in requirements shall be covered in the Installation and Operating Manual.

NOTE It is recognized that under certain conditions a break-in period can be requested. If a break-in period is required, the gear vendor can specify in the proposal the required load, speed, and duration of the period. The gear vendor can also specify in the proposal any additional field inspection and commissioning required during the break-in period.

6.1.18 All equipment shall be designed to permit rapid maintenance. Major parts such as casing components and bearing housings shall be designed and manufactured to ensure accurate alignment on reassembly.

NOTE This can be accomplished using shouldering, cylindrical dowels, or keys.

6.1.19 All components that are specific as to rotational direction, top or bottom casing half location, or axial location in the casing shall be designed to prevent incorrect installation.

6.1.20 Spare and replacement parts for the gear unit and all furnished auxiliaries shall meet all the criteria of this standard.

- **6.1.21** The purchaser shall specify the appropriate shaft assembly designation selected from those listed in Table 3 and illustrated in Figure 1.

Table 3—Shaft Assembly Combinations

High-speed Shaft	Low-speed Shaft
L	R
R	L
L	L
R	R
R	LR
L	LR
LR	L
LR	R
LR	LR
L = Left; R = Right. The letters refer to the number and direction of shaft extensions (see Figure 1). The material for this table was extracted from ANSI/AGMA 6113-A06 (6013-A06) with permission from the publisher.	

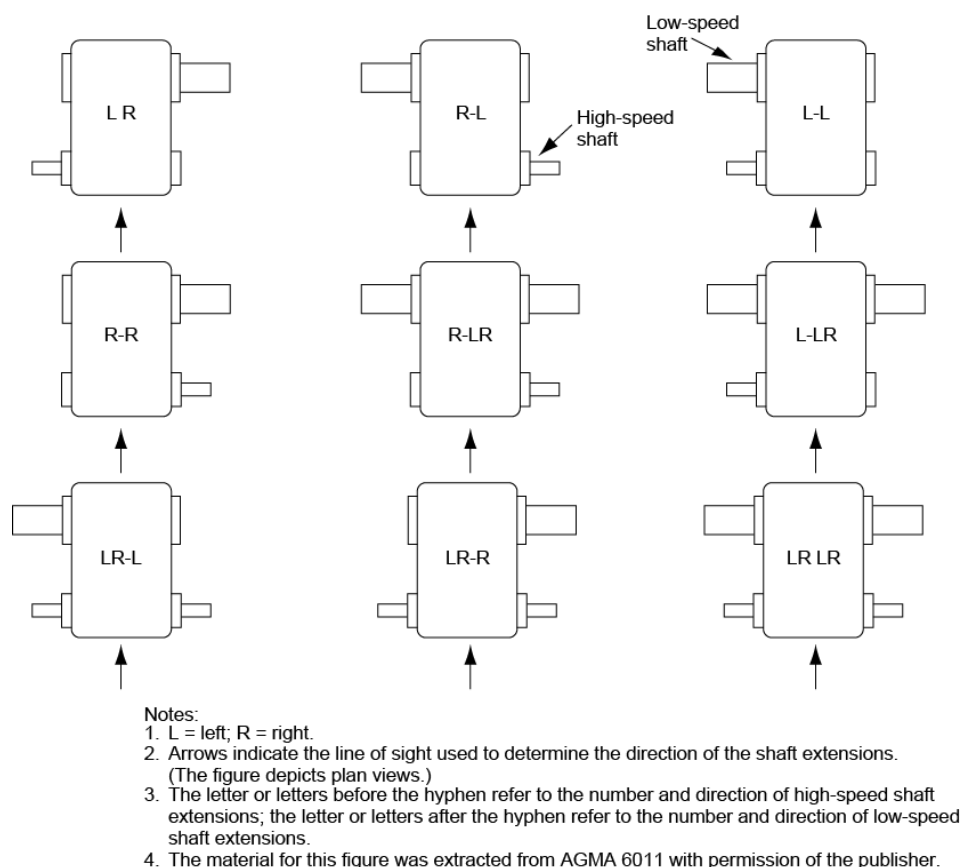


Figure 1—Shaft Assembly Designations (for Parallel-shaft, Single- and Double-helical One- and Two-stage Speed Increasers and Reducers)

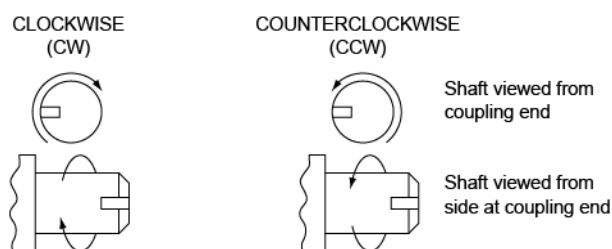
6.1.22 Direction of Rotation

6.1.22.1 General

The rotational direction of high-speed and low-speed shafts shall be defined as clockwise (CW) or counterclockwise (CCW) as viewed from the coupling ends of the respective shafts.

6.1.22.2 Designation on Drawings

6.1.22.2.1 The shaft rotational direction shall be designated by the abbreviations CW or CCW, as indicated by the circular arrows in Figure 2.



Note: The material for this figure was extracted from AGMA 6010 with permission of the publisher.

Figure 2—Shaft Rotation Designations

6.1.22.2.2 For increased rotordynamic stability, the configuration and shaft rotational direction should be selected in a way that achieves a downmeshed bull gear. In order to achieve this, speed increasers should be oriented with an upmesh rotation and speed decreaseers should be oriented with a downmesh rotation.

- **6.1.22.2.3** The purchaser shall specify the rotational direction of both the high-speed and the low-speed shaft. If either or both shafts have an extension at each end, the purchaser may alternatively indicate the rotational directions on the appropriate assembly designation (see Figure 1) and submit a copy of the figure with the quotation request.

6.1.22.2.4 In finalizing the data for purchase, the purchaser shall prepare a sketch that shows the direction of rotation of each item in the machinery train.

6.2 Rating

● **6.2.1 Gear Unit Rated Power**

The gear unit rated power shall be specified by the purchaser. For gear units connected directly to the driver, the minimum gear unit rated power shall be the maximum installed power of the driver unless the purchaser can demonstrate that the maximum installed power of the driver will never (including start-up and transient conditions) be transmitted through the gear. All modes of normal and transient operation shall be examined. For gear units between two items of driven equipment, the power rating of such gear units shall not be less than Item a) or b), whichever is greater.

- a) 110 % of the maximum power required by the equipment driven by the gear unit.
- b) The maximum power of the driver prorated between the driven equipment, based on normal power demands. If the maximum transmitted torque occurs at an operating speed other than the maximum continuous speed, this torque and its corresponding speed shall be specified by the purchaser and shall be the basis for sizing the gear unit.

NOTE The gear unit rated power is normally the maximum installed power of the driver. In some cases, a higher rated power is specified in order to plan for future upgrades. In other applications, the maximum installed power of the driver can be significantly greater than the power at the normal operating point, resulting in oversizing of the gear.

● **6.2.2 Normal Operating Point**

The purchaser shall specify the normal operating point and any special operating conditions (such as load reversals), in addition to the gear unit rated power.

● **6.2.3 Minimum Gear Tooth Service Factor**

The minimum gear tooth service factors (C_{SF} and K_{SF}) shall be specified by the purchaser based on the application, as listed in Table 4. The minimum gear tooth service factors apply to the gear unit rated power at rated speed.

6.2.4 Minimum Tooth Hardness

The minimum tooth hardness shall be as follows.

- a) For through hardened: 300 HBW.
- b) For nitrided: 83.5 HR15N.
- c) For carburized: 55 HRC.

Table 4—Minimum Gear Tooth Service Factors

Driven Equipment	Induction Motors	Synchronous and Variable Speed Motors	Steam and Gas Turbines	Reciprocating Engines
Centrifugal blowers	1.4	1.6	1.6	1.7
Compressors				
Centrifugal	1.4	1.6	1.6	1.7
Axial	1.4	1.6	1.6	1.7
Rotary lobe	1.7	1.8	1.7	2.0
Reciprocating	2.0	2.1	2.0	2.3
Fans				
Centrifugal	1.4	1.5	1.6	1.7
Forced draft	1.4	1.5	1.6	1.7
Induced draft	1.7	1.8	2.0	2.2
Generators and exciters				
Base load continuous	1.1	1.1	1.1	1.3
Peak-duty-cycle	1.3	1.3	1.3	1.7
Pumps				
Centrifugal (all services not listed below)	1.3	1.5	1.5	1.7
Centrifugal, boiler feed	1.7	1.8	2.0	—
Centrifugal, hot oil	1.7	1.8	2.0	—
Centrifugal, high speed (over 3600 rpm)	1.7	—	2.0	—
Centrifugal, water supply	1.5	1.6	1.7	2.0
Rotary, axial flow/all types	1.5	1.6	1.5	1.8
Rotary, gear	1.5	1.6	1.5	1.8
Reciprocating	2.0	2.1	2.0	2.3

6.2.5 Tooth Pitting Resistance Power Rating

6.2.5.1 The tooth pitting resistance power rating shall be calculated separately for the pinion and gear wheel according to Equation (1).

In SI units:

$$P_{az} = k_{HP} \left(\frac{\left(\frac{\pi n_1}{60} \right) b}{10^6} \right) \left(\frac{Z_I}{K_V K_H C_{SF}} \right) \left(\frac{d_{w1} \sigma_{HP} Z_N}{Z_E} \right)^2 \quad (1a)$$

In USC units:

$$P_{az} = k_{HP} \left(\frac{\left(\frac{\pi n_1}{60} \right) b}{6600} \right) \left(\frac{Z_I}{K_V K_H C_{SF}} \right) \left(\frac{d_{w1} \sigma_{HP} Z_N}{Z_E} \right)^2 \quad (1b)$$

where

n_1 is the pinion speed, rpm;

b is the net active face width, mm (in.);

d_{w1} is the operating pitch diameter of pinion = $2a/(u + 1)$, mm (in.);

Z_E is the elastic coefficient;

NOTE $190 \text{ [N/mm}^2\text{]}^{0.5}$ ($2300 \text{ [lb/in.}^2\text{]}^{0.5}$) for steel pinion and gear;

Z_I is the geometry factor (reference AGMA 908);

Z_N is the contact stress cycle factor = 0.68;

K_V is the dynamic factor, for ANSI/AGMA ISO 1328-1 class 4, $K_V = 1.12$; for class 3 or better, $K_V = 1.1$;

K_H is the load distribution factor, see 6.2.5.3;

k_{HP} is the tooth flank pitting derating factor, see 6.2.5.2;

C_{SF} is the pitting gear tooth service factor, see Table 4;

σ_{HP} is the allowable contact stress number; see 6.2.5.4 to 6.2.5.6.

NOTE Equation (1) comes from ANSI/AGMA 2101-D04, but it has an additional factor of k_{HP} to give a lower rating.

● 6.2.5.2 Tooth Flank Pitting Derating Factor, k_{HP}

Unless otherwise specified, the tooth flank pitting derating factor shall be in accordance with Table 5.

Table 5—Tooth Flank Pitting Derating Factor

Material Heat Treatment	Tooth Flank Pitting Derating Factor
Carburized	0.6
Nitrided	0.85
Through hardened	0.75

NOTE The values in Table 5, with $K_H = 1.3$, will result in a gearbox rating like that given by early editions of this standard. See Annex I for additional information.

6.2.5.3 Load Distribution Factor, K_H

The load distribution factor may be calculated either with an analytical method such as that described in AGMA 927, or by Equations (2) through (4). However, if the calculated value of the load distribution factor K_H is less than 1.1, then 1.1 shall be used as the value of K_H . Gear designs shall not result in a calculated value of K_H over 1.50 without approval from the purchaser.

In SI units:

$$K_H = 1.0 + 0.8(K_{Hpf} + 0.054 + 0.4032 \times 10^{-3}b - 1.152 \times 10^{-7}b^2) \quad (2a)$$

$$\begin{aligned} K_{Hpf} &= \text{bod} - 0.025, \text{ when } b \leq 25 \text{ mm} \\ &= \text{bod} - 0.0375 + 0.000492b, \text{ when } 25 < b \leq 432 \text{ mm} \\ &= \text{bod} - 0.1109 + 0.000815b - 3.53 \times 10^{-7}b^2, \text{ when } 432 < b \leq 1020 \text{ mm} \end{aligned} \quad (3a)$$

In USC units:

$$K_H = 1.0 + 0.8(K_{Hpf} + 0.054 + 0.01024b - 0.7408 \times 10^{-4}b^2) \quad (2b)$$

$$\begin{aligned} K_{Hpf} &= \text{bod} - 0.025, \text{ when } b \leq 1 \text{ in.} \\ &= \text{bod} - 0.0375 + 0.0125b, \text{ when } 1 < b \leq 17 \text{ in.} \\ &= \text{bod} - 0.1109 + 0.0207b - 0.000228b^2, \text{ when } 17 < b < 40 \text{ in.} \end{aligned} \quad (3b)$$

where

b is the net active face width, mm (in.);

$\text{bod} = \text{the greater of } 0.05 \text{ or } b/(10d_{w1}) \quad (4)$

Equation (2) only applies when:

- the distance from the center of the gear mesh to the center of the bearing span divided by the bearing span is less than 0.175 and
- the tooth contact is checked at assembly, with contact adjustments as required.

If these conditions are not met, or for wide face gears, an analytical approach may be used to determine the load distribution factor.

NOTE The value of K_H is used for both the pitting and bending power ratings.

6.2.5.4 For through-hardened gears, the allowable contact stress number σ_{HP} from Figure 8, grade 2 in ANSI/AGMA 2101-D04 (or ANSI/AGMA 2001-D04 when using USC units) shall be used.

6.2.5.5 For nitrided gears, the allowable contact stress number σ_{HP} from Table 4, grade 2 in ANSI/AGMA 2101-D04 (or ANSI/AGMA 2001-D04 when using USC units) shall be used.

6.2.5.6 For carburized gears with a hardness between 55 and 58 HRC, the allowable contact stress number σ_{HP} , is given in Figure 3. For carburized gears with a hardness of 58 to 64 HRC, the allowable contact stress number σ_{HP} is 1,550 N/mm² (225,000 lb/in.²).

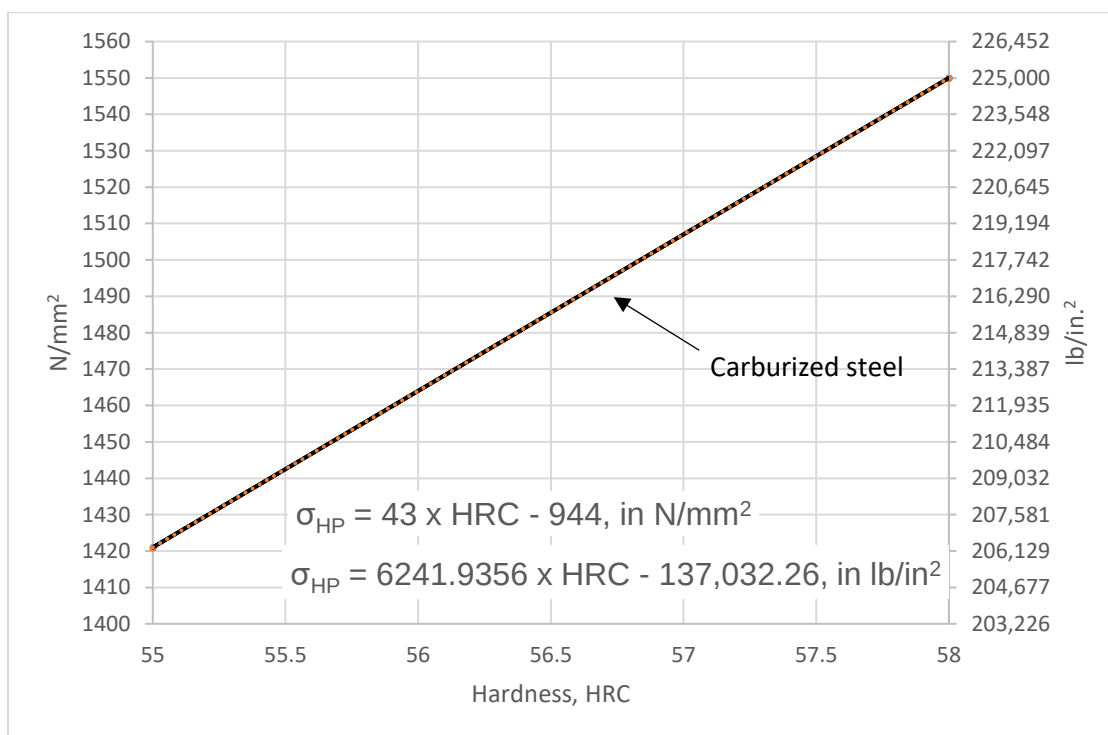


Figure 3—Contact Stress Number, σ_{HP} , Carburized Steel

6.2.6 Bending Strength Power Rating

6.2.6.1 The material hardness shall be selected by the gear vendor, but the minimum hardness shall not be less than the values given in 6.2.4. The allowable bending stress number shall be based on the specified minimum hardness for the element, pinion, or gear wheel. After manufacture, if the achieved hardness is below the specified minimum hardness, the allowable bending stress number shall be recalculated based on actual hardness, and this corrected allowable bending stress number shall be used to recalculate the bending strength power rating.

NOTE Normal heat-treating practice requires a tolerance range on hardness.

6.2.6.2 The bending strength allowable transmitted power, P_{ay} shall be calculated separately for the pinion and gear according to Equation (5):

In SI units:

$$P_{ay} = k_{FP} \left(\frac{\left(\frac{\pi n_1}{60} \right) d_{w1}}{10^6} \right) \left(\frac{b m_t Y_J Y_L \sigma_{FP} Y_N}{K_V K_H K_{SF}} \right) \quad (5a)$$

In USC units:

$$P_{ay} = k_{FP} \left(\frac{\left(\frac{\pi n_1}{60} \right) d_{w1}}{6600} \right) \left(\frac{b Y_J Y_L \sigma_{FP} Y_N}{P_d K_V K_H K_{SF}} \right) \quad (5b)$$

where

m_t is the transverse module (equal to normal module m_n divided by the cosine of the helix angle at the standard pitch diameter), mm;

P_d is the transverse diametral pitch, in.⁻¹;

Y_J is the geometry factor (reference AGMA 908);

σ_{FP} is the allowable bending stress number; see 6.2.6.5;

Y_N is the bending stress cycle factor = 0.8;

K_{SF} is the bending gear tooth service factor, see Table 5;

k_{FP} is the tooth root bending derating factor, see 6.2.6.3;

Y_L is the reverse loading factor, see 6.2.6.4.

NOTE Equation (5) comes from ANSI/AGMA 2101-D04, but it has an additional factor k_{FP} to give a lower rating.

● 6.2.6.3 Tooth Root Bending Derating Factor, k_{FP}

Unless otherwise specified, the tooth root bending derating factor shall be 0.60.

NOTE The value of 0.6, with $K_H = 1.3$, will result in a gearbox rating like that given by early editions of this standard. See Annex I for additional information.

6.2.6.4 Reverse Loading Factor, Y_L

The reverse loading factor is 1.0, except a reverse loading factor of 0.7 shall be used for idler gears and other gears where the teeth are completely reverse loaded on every cycle.

6.2.6.5 For through-hardened gears, the allowable bending stress number σ_{FP} from Figure 9, grade 2 in ANSI/AGMA 2101-D04 (or ANSI/AGMA 2001-D04 when using USC units) shall be used.

6.2.6.6 For nitrided gears, the allowable bending stress number σ_{FP} from Table 5, grade 2 in ANSI/AGMA 2101-D04 (or ANSI/AGMA 2001-D04 when using USC units) shall be used.

6.2.6.7 For carburized gears with a hardness between 55 and 58 HRC, the allowable bending stress number σ_{FP} , is given in Figure 4. For carburized gears with a hardness of 58 to 64 HRC, the allowable bending stress number σ_{FP} is 450 N/mm² (65,000 lb/in.²).

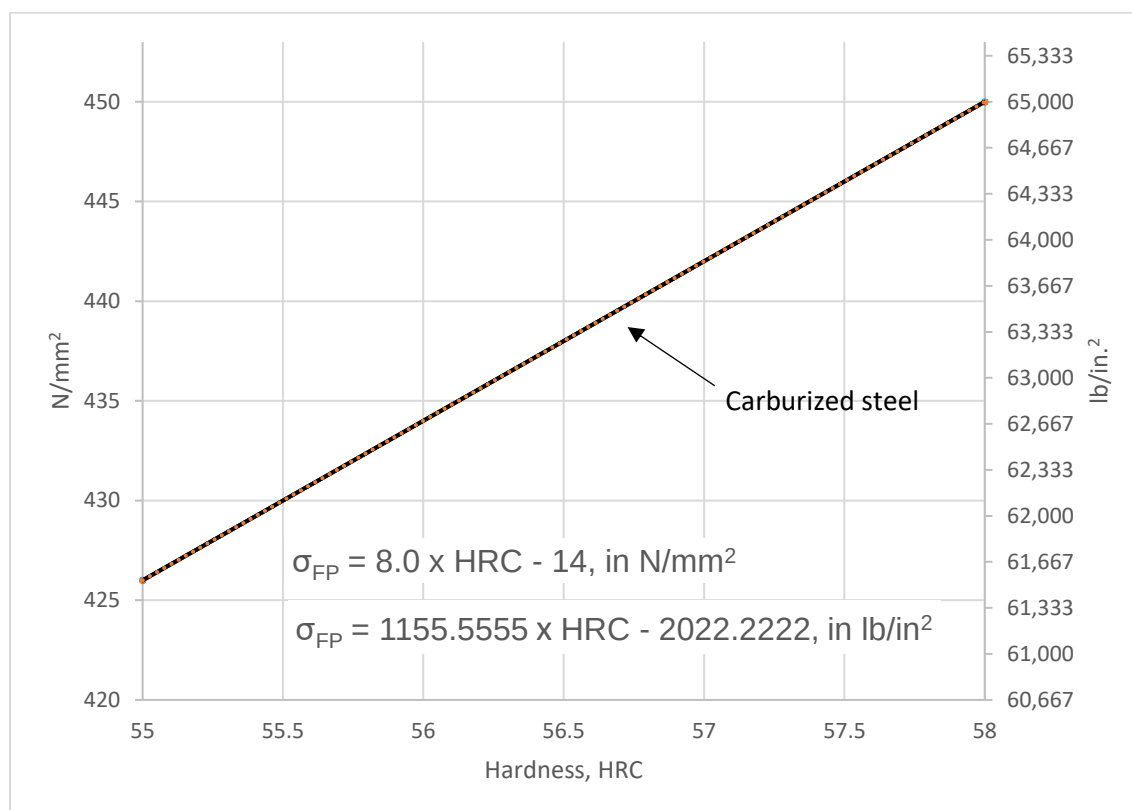


Figure 4—Allowable Bending Stress Numbers, σ_{FP} , Carburized Steel

6.2.7 Scuffing

To avoid scuffing, the gear unit shall meet the requirements of ANSI/AGMA 6011-J14, Annex B. The gear unit shall be designed for use with a load stage 6 oil according to FZG test A/8.3/90, as described in ISO 14635-1. If an alternative oil is requested, the gear vendor shall provide calculations and an experience list to support a request for an alternate oil selection. Alternate calculating methods may be used based on gear vendor experience.

NOTE Refer to AGMA 925 and/or ISO 6336-20 for additional information.

6.2.8 Pinion Length to Diameter Ratio

6.2.8.1 The length-to-diameter (L/d_1) ratio of the pinion shall not exceed 2.2 for double-helical and 1.6 for single-helical gears without purchaser approval,

where

L is the face width (b) for single-helical gears;

L is the full face width (b) plus the gap width for double-helical gears;

b is the net active face width (less gap), mm (in.);

d_1 is the pinion pitch diameter, mm (in.).

6.2.8.2 Gears with unmodified leads shall not have a total lead mismatch of the rotors across the gear face width greater than 0.015 mm (0.00059 in.) at the normal operating point.

6.2.8.3 If a higher L/d_1 ratio than that established in para 6.2.8.1 is proposed, the gear vendor shall submit justification in the proposal for using the higher L/d_1 ratio. If operating conditions other than the gear unit rated power are specified by the purchaser, such as the normal operating point, the gear vendor shall consider in the analysis the length of time and load range at which the gear unit will operate at each condition so that the correct lead modification can be determined.

6.2.8.4 If the lead mismatch exceeds the values limited by 6.2.8.2, regardless of the L/d_1 , an analytically determined lead modification shall be applied in order to reduce the total actual mismatch to a magnitude below the limiting values in 6.2.8.2. See Annex G for further discussion.

6.2.9 High-speed Gear Considerations

It is recognized that the rating rules specified in 6.2.1 through 6.2.8 can create design problems for some high-speed gears. See Annex I for additional information.

6.3 Casings

6.3.1 Design Parameters

6.3.1.1 Gear casings shall be designed and constructed to maintain rotor alignment under all load conditions.

NOTE Casings can be either cast or fabricated.

6.3.1.2 Provision shall be made to permit doweling or keying the casing to the soleplate or baseplate at two points as close as possible to the vertical plane of the pinion's centerline (to minimize misalignment of the high-speed pinion with connected equipment). Casings not doweled or keyed by the gear vendor shall be provided with dowel starter holes. Drilling interferences in the field installations should be considered in determining the location and angle of the starter holes.

6.3.1.3 Mounting surfaces shall meet the following criteria:

- a) they shall be machined to a finish of 6 μm (250 $\mu\text{in.}$) arithmetic average roughness (Ra) or better;
- b) to prevent a soft foot, they shall be in the same horizontal plane within 25 μm (0.001 in.);
- c) each mounting surface shall be machined within a flatness of 42 μm per linear m (0.0005 in. per linear ft) of mounting surface;
- d) different mounting planes shall be parallel to each other within 50 μm (0.002 in.);
- e) the upper machined or spot faced surface shall be parallel to the mounting surface.

6.3.1.4 Anchor bolt holes shall be drilled perpendicular to the mounting surface or surfaces, machined or spot faced to a diameter three times that of the anchor bolt hole. Diametrical clearance between anchor bolts and the anchor bolt holes in the mounting plates shall be a minimum of 6 mm ($1/4$ in.).

6.3.1.5 The equipment feet shall be provided with vertical jackscrews.

6.3.1.6 Casings shall be designed to prevent damage due to distortion caused by temperature, torque, and allowable external forces and moments, under specified operating conditions (see 6.2.2).



6.3.1.7 Oil piping external to the casing should be minimized by the use of internal oil passages.

6.3.1.8 All internal piping shall be ASTM A312/A312M welded or flanged. This piping may be connected to the casing by welding (only for steel casings) or flanges. Where internal space, design, or manufacturing constraints do not permit the use of DN 15, DN 20, or DN 25 ($1/2$, $3/4$, or 1 in.) pipe, stainless steel tubing conforming to ASTM A269/A269M with stainless steel fittings may be furnished. Tubing thicknesses shall meet the requirements of Table 6.

6.3.1.9 Internal piping shall be designed to prevent damage from vibration or from shipment, operation, and maintenance. Cantilevered piping shall include reinforcing gussets in two planes at all pipe-to-flange connections.

Table 6—Minimum Tubing Wall Thickness

	Nominal Tubing Size		Minimum Wall Thickness	
	mm	in.	mm	in.
DN 15	12.7	$1/2$	1.65	0.065
DN 20	19.05	$3/4$	2.40	0.095
DN 25	25.4	1	2.75	0.109

6.3.1.10 Prior to gear unit assembly, all internal piping and oil passages shall be cleaned to remove all foreign material.

6.3.1.11 Casings shall be designed to permit rapid drainage of lube oil and to minimize oil foaming, which could lead to excessive rise of the oil temperature. For gears with pitch line velocities exceeding 125 m/s (24,600 ft/min.), consideration should be given to special designs such as the following:

- a) a false bottom in the gear unit,
- b) a sump depth at least 610 mm (2 ft) below the bottom of the gear wheel,
- c) rotation and meshing directions,
- d) an oil drain line to the reservoir that is independent from all other drain systems,
- e) a second full-sized drain connection,
- f) two oil drains, one on each end, or one downward oil drain (particularly for offshore operations) to achieve proper oil draining with vessel movement,
- g) larger side and circumferential clearances between gears and pinions and the casing,
- h) windage baffles.

6.3.1.12 A flanged vent connection shall be provided. The connection shall be provided with an installed blind flange if a filter breather is not provided. The connection shall have a size of at least DN 40 (NPS $1\frac{1}{2}$).

NOTE The location generally recommended by gear manufacturers for the vent connection is on the top of the gear casing.

6.3.1.13 Filter Breather

- **6.3.1.13.1** If specified, a filter breather shall be provided by the gear vendor.

6.3.1.13.2 The filter breather shall be constructed of Series 300 stainless steel with Series 300 stainless steel or copper-nickel alloy UNS 04400 internals, designed and located to prevent: entrainment or discharge of oil to the atmosphere, pressure buildup in the casing, entrance of water during violent rainstorms, and entrance of dirt entrained in the air.

6.3.1.13.3 The filter breather shall be at least DN 40 (NPS 1½), and its construction shall permit easy disassembly for inspection and cleaning. The filter breather may be located on the casing or the drain.

6.3.1.14 A removable, gasketed inspection cover or covers shall be provided in the gear casing to permit direct visual inspection of the full face width of the pinion and gear and of the oil spray pattern. The inspection opening, or openings, shall be at least one-half of the full face width of the gear. Casing openings included only for verification of the oil spray pattern on the underside of the gear elements (not for tooth inspection) may be tapped holes with removable plugs suitable for use of a bore scope.

6.3.1.15 Permanent coatings or paint shall not be applied to the interior of the casing unless the purchaser approves in advance the material and method of application. If interior paint is approved by the purchaser, the gear vendor shall ensure compatibility between the coating and the specified lubricant.

6.3.2 Joints

Split casings shall use a metal-to-metal joint (with a suitable joint compound) that is tightly maintained by suitable bolting. All metal to metal joints shall be sufficiently flat to prevent any leak. Joint compound shall not influence bearing crush/clearance. Gaskets shall not be used on the split joint. O-ring sealing is permitted only with the purchaser's approval.

6.3.3 Bolting

6.3.3.1 Case bolting shall use through-bolting, studs, or cap screws.

6.3.3.2 The threading shall conform to ASME B1.1, ASME B1.13M, or ISO 261 as specified.

NOTE 1 ISO 261 covers general metric screw threads and ASME B1.1 covers general inch series screw threads.

NOTE 2 For the purposes of this provision, ISO 261 is equivalent to ASME B1.13M.

NOTE 3 Glossary of terms for screw threads can be found in ASME B18.12.

6.3.3.3 Slotted-nut or spanner-type bolting shall not be used unless approved by the purchaser.

NOTE For limited space locations, integrally flanged fasteners can be required.

6.3.3.4 Studded connections shall be furnished with studs installed. The first 1½ threads at both ends of each stud shall be removed.

6.3.3.5 Blind stud holes should be drilled only deep enough to allow a preferred tap depth of 1½ times the major diameter of the stud.

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

6.3.4 Assembly and Disassembly

6.3.4.1 It shall be possible to lift the upper half of the casing without disturbing the piping of the main oil supply to the lower half of the casing.

6.3.4.2 Jackscrews, guide rods, and 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

(counterbored or recessed) to prevent a leaking joint or improper fit caused by marring. Guide rods shall prevent damage to the internals or casing studs by the casing during disassembly and reassembly.

6.3.4.3 Lifting Provisions

6.3.4.3.1 Methods of lifting the assembled gear unit shall be specified by the gear vendor.

6.3.4.3.2 Lifting lugs or eyebolts shall be provided for lifting the top half of the casing. Lifting lugs or eyebolts not capable of lifting the entire gear unit shall be clearly and permanently marked on the casing.

6.4 Casing Connections

6.4.1 A single lube oil supply connection is required.

6.4.2 Oil Drain Connection

6.4.2.1 A single lube oil drain connection from the gear casing shall be provided unless purchaser approves an additional drain due to special circumstances as mentioned in 6.3.1.11.

6.4.2.2 The minimum drain connection size shall be based on the total inlet flow to the gear casing. Connections smaller than shown in Table 7 shall not be used.

The drain connection and drain pipe shall be sized and installed to maintain an oil drain velocity based on the drain line running no more than half full. The flows listed in Table 7 are based on theory of flow through a weir.

Table 7—Drain Connection Sizes

Maximum Inlet Flow Rate		Minimum Drain Size ^a	
l/min	gpm	mm	in.
26	7	75	3
56	15	100	4
170	45	150	6
380	100	200	8
585	155	250	10
830	220	300	12
^a Nominal.			

6.4.2.3 For drains with a nominal internal diameter greater than 300 mm (12 in.), the design velocity shall not exceed 0.38 m/s (1.25 ft/s). The allowable flow can be calculated as:

In SI units:

$$F_{df} = 0.009D^2 \quad (6a)$$

In USC units:

$$F_{df} = 1.53D^2 \quad (6b)$$

where

F_{df} is the allowable flow in drain line with D over 300 mm, l/min (gpm);

D is the internal drain line diameter, mm (in.).

NOTE Sizing and sloping of the drain piping is specified in API 614. If the drain piping is not designed to ensure that it is never more than half full at any point, the gear unit can have problems with drainage. Gear units with insufficient drainage can experience overheating, oil leakage, and potential damage.

- **6.4.3** If specified, casings shall be provided with an inlet purge connection which is at least DN 15 (NPS $1\frac{1}{2}$ in.), located to assure a sweep of purge gas across the casing to the filter breather.

6.4.4 Openings for oil piping connections shall be DN 20 ($3\frac{1}{4}$ NPS) or larger and in accordance with ISO 6708. Sizes DN 32, DN 65, DN 90, DN 125, DN 175, and DN 225 ($1\frac{1}{4}$, $2\frac{1}{2}$, $3\frac{1}{2}$, 5, 7, and 9 NPS) shall not be used.

6.4.5 All of the purchaser's connections shall be accessible for disassembly without requiring the gear unit, or any major part of the gear unit, to be moved.

6.4.6 Customer Connections

- **6.4.6.1** Customer connections, oriented as specified, shall be at least DN 20 ($3\frac{1}{4}$ NPS) and shall be flanged or machined and studded. Dimensions for these connections shall conform to ASME B16.5 or to EN 1092-1 or EN 1092-2, as specified. The connection face surface shall be machined flat with a nominal surface finish of $R_a = 3.2 \mu\text{m}$. Purchaser connections other than those covered by ASME B16.1, ASME B16.5, ASME B16.42, or EN 1092-1 or EN 1092-2 require the purchaser's approval. If specified, mating parts shall be furnished by the gear vendor.

6.4.6.2 Where flanged or machined and studded openings are impractical, threaded openings in sizes DN 20 (NPS $3\frac{1}{4}$) through DN 40 (NPS $1\frac{1}{2}$) are permissible.

6.4.6.3 Threaded Customer Connections

6.4.6.3.1 A pipe nipple, preferably not more than 150 mm (6 in.) long, shall be screwed into the threaded opening.

6.4.6.3.2 Pipe nipples shall be a minimum of Schedule 160 seamless for DN 25 (1 NPS) and smaller and a minimum of Schedule 80 for DN 40 ($1\frac{1}{2}$ NPS).

6.4.6.3.3 The pipe nipple shall be provided with a welding-neck flange.

6.4.6.3.4 The threaded connection shall be seal welded; however, seal welding is not permitted on cast iron equipment, for instrument connections, or where disassembly is required for maintenance. Seal-welded joints shall be in accordance with ASME B31.3.

6.4.7 Flange Specifications

6.4.7.1 Unless otherwise specified, the CLASS system applies and all flanges shall conform to ASME B16.5.

- **6.4.7.2** If the PN system is specified, all flanges shall conform to EN 1092-1 or EN 1092-2 as applicable.

6.4.7.3 EN flanges other than PN 6, 10, 16, 25, and 40 require purchaser approval.

NOTE EN 1092-1 flanges cover PN 6, 10, 16, 25, 40, 63, 100, 160, 250, 320, and 400.

6.4.7.4 Use of SAE J518 type flanges requires purchaser approval.

6.4.7.5 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 ASME or ISO are acceptable. Non-standard (oversized) flanges shall be completely dimensioned on the arrangement drawing. If oversized flanges require studs or bolts of non-standard length, this requirement shall be identified on the arrangement drawing.

6.4.8 Threaded connections shall not be larger than DN 40 (1½ NPS). Tapped opening bosses shall conform to ASME B16.5. Pipe threads shall be taper threads conforming to ASME B1.20.1 or ISO 7-1 as applicable. Where tubing connectors are used, they may be straight thread connectors according to SAE J1926-1 or ISO 6149-1.

6.4.9 Threaded Plugs

6.4.9.1 Tapped openings not connected to piping shall be plugged with solid plugs furnished in accordance with ASME B16.11, or with plugs in accordance with SAE J1926-1, or ISO 6149-1. Plastic plugs are not permitted.

6.4.9.2 Plugs that later require removal shall be of corrosion-resistant material.

6.4.9.3 Threads shall be lubricated.

6.4.9.4 Tape shall not be applied to threads of plugs inserted into oil passages.

6.4.10 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.

6.4.11 For socket-welded construction, there shall be a 1.5 mm (¹/₁₆ in.) gap between the pipe end and bottom of the socket before welding.

NOTE See ASME B31.3, Figure 328.5.2B (3) and 328.5.2C.

6.4.12 Slip-on flanges are permitted only with the purchaser's written approval.

6.5 Gear Elements

6.5.1 General

6.5.1.1 All gear teeth shall be finish cut or finish ground on the assembled gear and shaft. One or more of the following processes shall be used in finishing the gear teeth:

- a) grinding,
- b) shaving,
- c) precision hobbing,
- d) all gear teeth finished by shaving shall have been generated by hobbing. Shaving cutters shall have a hunting tooth combination with the workpiece.

6.5.1.2 The tooth surface on loaded faces of completed gears shall have a finish, in all directions, of 0.8 µm (32 µin.) Ra or better. See also ISO/TR 10064-4 on measuring methods.

6.5.1.3 The design of single-helical gears shall be such that the effects of the moments on the gear elements, resulting from axial tooth reaction at the gear mesh, will not impair the expected performance of the gear unit.

6.5.1.4 Hunting tooth combinations are required. To achieve this requirement, it can be necessary for the purchaser to adjust the exact gear ratio. If such adjustment is impractical, the purchaser and the gear vendor shall negotiate a solution.

NOTE A hunting tooth combination is required because the intent is for every tooth on a pinion to mesh with as many teeth as possible on the mating gear wheel before the same teeth mesh again or repeat.

6.5.1.5 Each gear wheel and each pinion shall be supported on two bearings. Overhung designs are not acceptable.

6.5.2 Quality Assurance

6.5.2.1 After the gear teeth are finish machined, the gear elements shall be checked per ANSI/AGMA ISO 1328-1. The gear elements shall meet tolerance class 4 or better.

6.5.2.2 Contact Checking

6.5.2.2.1 Each pair of mating gears shall be considered as a matched set and shall be checked for contact in the job casing at the gear vendor's shop. The check per 6.5.2.1 cannot be used as a replacement for this check.

6.5.2.2.2 A thin coating of color transfer material (such as Prussian blue) shall be applied to four or more teeth of the dry, degreased gear wheel at each of three locations, 120° apart. The coated teeth shall be rotated through the mesh with a moderate drag torque applied in a direction that will cause the teeth to contact on the normally loaded faces. The color transfer shall show evidence of contact distributed across each helix as prescribed by the gear vendor.

6.5.2.2.3 Prior to the contact tests, the gear vendor shall make available to the purchaser a contact drawing or gear vendor engineering specification that defines the acceptable contact.

6.5.2.2.4 The drawing or specification and the results of the job-casing contact checks shall be preserved for at least 20 years and shall be available to the purchaser on request. The results of the contact check shall be preserved by lifting the contrasting colors from a tooth by applying and peeling off a strip of clear adhesive tape and then applying the tape to an annotated sheet of white paper. If the results are transferred to a digital format for storage, they shall be high-resolution color images.

6.5.2.3 The gear vendor shall demonstrate the axial stability of each meshing pair of double-helical gears by either:

- a) measuring the unfiltered peak-to-peak shaft axial vibration, which shall not exceed 50 µm (0.002 in.) during testing, or
- b) using indicators to make a slow rotation check. The preferred method for this slow rotation check is to hold one member (usually the gear wheel) firmly in a fixed axial position and indicate the axial movement of the other member (usually the pinion) as the parts are rotated through at least one full revolution of the gear wheel while applying a drag torque in a direction that will force the normally loaded tooth faces into contact. The axial motion of the pinion relative to the gear wheel shall not exceed 38 µm (0.0015 in.) (see 8.3.2.2.10).

6.5.2.4 If quality assurance records include color documents, then the electronic records shall be in color as well.

6.5.3 Fabrication

6.5.3.1 Unless otherwise approved by the purchaser, pinions shall be integrally machined with their shafts.

NOTE At low pitch line velocity and large pinion diameter, integrally machined pinions may not be needed.

6.5.3.2 For pitch line velocities above 150 m/s (29,528 ft/min), gear wheels shall be integral with their shafts. For pitch line velocities of 150 m/s (29,528 ft/min) and less, gear wheels may be integral with or separate from their shafts. Separate gear wheels shall be forgings and shall be assembled on their shafts with an interference fit. See Annex I for additional information on special considerations for high-speed gears.

NOTE For gear ratios greater than 3:1, and pitch line velocity above 150 m/s (29,528 ft/min), separate gear wheel and shaft construction is often considered.

6.5.3.3 For gear wheels integral with their shafts, operating at pitch line velocities above 150 m/s (29,528 ft/min), the purchaser and gear vendor will agree to material suitability and additional inspection and testing necessary to ensure quality. This can include but is not limited to:

- a) material cleanliness in excess of ISO 6336-5 grade ME requirements;

- b) material mechanical properties at center of largest section (hardness, strength, fracture toughness);
- c) ultrasonic inspection of 100 % part volume after all heat treatment is completed;
- d) wet magnetic particle inspection of part;
- e) placing hole in center area of part (hollow shaft);
- f) forging reduction ratio in excess of normal requirements.

NOTE Large changes in diameter from gear outside diameter to shaft/bearing diameters can create difficulties in forging and heat treating that can lead to unexpected stresses and material flaws that are difficult to detect. When operating at high pitch line velocities, these defects can lead to sudden failure.

6.5.4 Shafts

6.5.4.1 Shafts shall be designed to meet the following:

- a) sized to transmit the maximum torque, taking into consideration the gear unit rated power and any other condition specified by the purchaser/entity having train responsibility (such as minimum speed and starting conditions), within the stress limits of ANSI/AGMA 6001 or other equivalent standards;
- b) shafts shall be machined from one-piece, heat-treated steel;
- c) shafts with finished diameters 200 mm (8 in.) and larger shall be forged;
- d) shafts with finished diameter less than 200 mm (8 in.) are allowed to be hot-rolled bar-stock manufactured to the same quality and heat treatment criteria as shaft forgings.

NOTE The fatigue safety factor of the shaft is not to be confused with the gear tooth service factor (SF).

6.5.4.2 Shaft Ends

6.5.4.2.1 Shafts shall be provided with integral flanges for couplings. Tapered or cylindrical keyed shaft ends are allowable with purchaser approval.

6.5.4.2.2 If shaft end keyways are specified, or keyed fits are used in the gear assembly, shaft keyways shall have fillet radii conforming to ANSI/ASME B17.1, ISO 21940-32, or other equivalent standards.

6.5.4.2.3 Shaft ends for couplings shall conform to the requirements of API 671.

6.5.4.3 Rotor Shaft Sensing Areas

6.5.4.3.1 The rotor shaft sensing areas to be observed by the radial vibration probes shall be concentric with the bearing journals.

6.5.4.3.2 All sensing areas (both radial vibration and axial position) shall be free from stencil marks or any other surface discontinuities, such as oil holes or keyways, for a minimum of one probe tip diameter on each side of the probe tip.

6.5.4.3.3 For gear units with total axial floats that exceed half of a probe tip diameter, the probe sensing area shall be long enough to cover the entire float, plus one probe tip diameter on each side.

6.5.4.3.4 The sensing areas shall not be metallized, sleeved, or plated.

6.5.4.3.5 The final surface finish of sensing areas observed by radial vibration probes should be a maximum of 0.8 μm (32 $\mu\text{in.}$) Ra. These areas shall be properly demagnetized to the levels specified in API 670 or otherwise treated so that the required combined total electrical and mechanical runout can be met.



6.5.4.3.6 For areas observed by radial vibration probes, the combined total electrical and mechanical runout shall not exceed the value from Equation (7) or 6.35 μm (0.25 mils), whichever is greater.

In SI units:

$$R = \frac{25.4}{4} \sqrt{\frac{12,000}{n_{mc}}} \quad (7a)$$

In USC units:

$$R = \frac{1}{4} \sqrt{\frac{12,000}{n_{mc}}} \quad (7b)$$

where

n_{mc} is the maximum continuous operating speed, rpm;

R is the allowable runout, μm (mils).

6.5.4.3.7 For areas to be observed by axial-position probes, the combined total electrical and mechanical runout shall not exceed 13 μm (0.0005 in.).

Steels used in manufacturing gears sometimes develop electrical runout that is difficult to reduce using conventional means such as degaussing, burnishing, and electro peening. If it becomes evident that further attempts to lower the electrical runout will be unproductive, the purchaser and the gear vendor should agree on the acceptability of the runout.

6.5.4.3.8 Electrical and mechanical runout shall be determined by one of the following methods:

- a) rotating the rotor through the full 360° supported in V blocks at the journal centers while continuously recording the combined runout with a non-contacting vibration probe and measuring the mechanical runout with a dial indicator or other contacting probe at the centerline of each probe location and one probe tip diameter to either side;
- b) rotating the rotor and measuring the mechanical runout on the shaft journal centers to determine the true centerline of rotation while simultaneously measuring the total runout at the probe location.

NOTE 1 The true centerline of rotation of the shaft is calculated from the mechanical runout measured on the journal centers and then the mechanical and total runout at the probe location referred to the true centerline of rotation is calculated. This method to determine the true centerline of rotation is applicable to any Coordinate Measuring Machine (CMM) and gear accuracy checking machine.

NOTE 2 Rotor runout determined in V blocks can differ slightly from that seen when the rotor is installed in a machine with hydrodynamic bearings. This is due to pad orientation on tilt pad bearings and effect of lubrication in all journal bearings. The rotor will assume a unique position in the bearings due to the slow roll speed and rotor weight.

6.5.4.3.9 Records of electrical and mechanical runout, for the full 360° at each probe location, shall be included in the manufacturing data book.

- **6.5.4.4** If specified, to enable measurement of torsional response, one DN 25 (1 NPS) tapped opening with a plug shall be provided in the upper half of the casing for each gear element. This will permit mounting of detectors for use with a frequency-modulating torsional analyzer. The tapped openings shall be oriented in a plane that is transverse (orthogonal) to the centerline axis of both the gear wheel shaft and the pinion shaft. The openings shall be positioned so that detectors screwed into them will sense the passing of the teeth.

6.5.4.5 Each rotor shall be clearly marked with a unique identification number. This number shall be on the non-drive end of the shaft or in another accessible area that is not prone to maintenance damage.

- **6.5.4.6** If specified, gear elements shall have provisions for vertical storage. This provision shall be capable of supporting 1.5 times the rotor weight. See 7.6.2 for provision of hanging fixture.

6.5.4.7 To prevent buildup of potential voltages in the shaft, residual magnetism (free air gauss levels) of the rotating element shall not exceed the limits specified in 8.2.2.4.1.

6.5.5 Balancing

6.5.5.1 All gear wheel and pinion assemblies shall be multiplane dynamically balanced to ISO 21940-11 grade 0.67 or better. The maximum allowable residual unbalance shall be calculated from one of the following equations:

In SI units (g-mm):

$$U_r = 6350 \frac{W_{st}}{n_{mc}} \quad (\text{for } n_{mc} < 25,000 \text{ rpm}) \quad (8a)$$

$$U_r = \frac{W_{st}}{3.937} \quad (\text{for } n_{mc} \geq 25,000 \text{ rpm}) \quad (8b)$$

In mixed units (g-in.):

$$U_r = 113.4 \frac{W_{st}}{n_{mc}} \quad (\text{for } n_{mc} < 25,000 \text{ rpm}) \quad (9a)$$

$$U_r = \frac{W_{st}}{220.46} \quad (\text{for } n_{mc} \geq 25,000 \text{ rpm}) \quad (9b)$$

In USC units (oz-in.):

$$U_r = 4 \frac{W_{st}}{n_{mc}} \quad (\text{for } n_{mc} < 25,000 \text{ rpm}) \quad (10a)$$

$$U_r = \frac{W_{st}}{6250} \quad (\text{for } n_{mc} \geq 25,000 \text{ rpm}) \quad (10b)$$

where

U_r is the maximum allowable residual rotor unbalance, g-mm (g-in., oz-in.);

W_{st} is the journal static load, kg (lbm);

n_{mc} is the maximum continuous speed, rpm.

NOTE 1 ISO 21940-11 grade 0.67 balance grade corresponds exactly to the $4W/n$ balance quality.

NOTE 2 ISO 21940-11 replaces ISO 1940-1.

NOTE 3 Above 25,000 rpm, the unbalance limit is based on $0.254 \mu\text{m}$ ($10 \mu\text{in.}$) mass displacement, which is in general agreement with the capabilities of conventional balance machines and is necessary to invoke for small rotors running at speeds above 25,000 rpm.