Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services

ANSI/API STANDARD 613
FIFTH EDITION, FEBRUARY 2003

ERRATA, DECEMBER 5, 2005

REAFFIRMED, AUGUST 2007

AMERICAN PETROLEUM INSTITUTE

Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services

Downstream Segment

ANSI/API STANDARD 613
FIFTH EDITION, FEBRUARY 2003

ERRATA, DECEMBER 5, 2005

REAFFIRMED, AUGUST 2007

AMERICAN PETROLEUM INSTITUTE

SPECIAL NOTES

API publications necessarily address problems of a general nature. With respect to particular circumstances, local, state, and federal laws and regulations should be reviewed.

API is not undertaking to meet the duties of employers, manufacturers, or suppliers to warn and properly train and equip their employees, and others exposed, concerning health and safety risks and precautions, nor undertaking their obligations under local, state, or federal laws.

Information concerning safety and health risks and proper precautions, with respect to particular materials and conditions, should be obtained from the employer, the manufacturer or supplier of that material, or the material safety data sheet.

Nothing contained in any API publication is to be construed as granting any right, by implication or otherwise, for the manufacture, sale, or use of any method, apparatus, or product covered by letters patent. Neither should anything contained in the publication be construed as insuring anyone against liability for infringement of letters patent.

Generally, API standards are reviewed and revised, reaffirmed, or withdrawn at least every five years. Sometimes a one-time extension of up to two years will be added to this review cycle. This publication will no longer be in effect five years after its publication date as an operative API standard or, where an extension has been granted, upon republication. Status of the publication can be ascertained from the API Downstream Segment [telephone (202) 682-8000]. A catalog of API publications and materials is published annually and updated quarterly by API, 1220 L Street, N.W., Washington, D.C. 20005, www.api.org.

This document was produced under API standardization procedures that ensure appropriate notification and participation in the developmental process and is designated as an API standard. Questions concerning the interpretation of the content of this standard or comments and questions concerning the procedures under which this standard was developed should be directed in writing to the Director, Standards Department, American Petroleum Institute, 1220 L Street, N.W., Washington, D.C. 20005, standards@api.org. Requests for permission to reproduce or translate all or any part of the material published herein should also be addressed to the general manager.

API standards are published to facilitate the broad availability of proven, sound engineering and operating practices. These standards are not intended to obviate the need for applying sound engineering judgment regarding when and where these standards should be utilized. The formulation and publication of API standards is not intended in any way to inhibit anyone from using any other practices.

Any manufacturer marking equipment or materials in conformance with the marking requirements of an API standard is solely responsible for complying with all the applicable requirements of that standard. API does not represent, warrant, or guarantee that such products do in fact conform to the applicable API standard.

FOREWORD

API publications may be used by anyone desiring to do so. Every effort has been made by the Institute to assure the accuracy and reliability of the data contained in them; however, the Institute makes no representation, warranty, or guarantee in connection with this publication and hereby expressly disclaims any liability or responsibility for loss or damage resulting from its use or for the violation of any federal, state, or municipal regulation with which this publication may conflict.

Suggested revisions are invited and should be submitted to the standardization manager, American Petroleum Institute, 1220 L Street, N.W., Washington, D.C. 20005.

CONTENTS

Tables

Figures

Page

Special Purpose Gear Units for Petroleum, Chemical, and Gas Industry Services

1 General

1.1 SCOPE

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

Note: The purchase of a spare set of gear rotors or a complete gear unit does not mean that the equipment is spared.

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 Std 677; to gears integral with other equipment, such as integrally geared compressors covered by Std 617 or Std 672; or to gears other than helical.

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

1.2 APPLICATIONS

The following gear-driven applications may be covered by this standard:

a. Speed increasers, including those for centrifugal compressors, axial compressors, blowers, rotary positive displacement compressors, separators, and centrifugal pumps.

b. Speed reducers, including those for reciprocating compressors, rotary positive displacement compressors, centrifugal compressors, centrifugal pumps, extruders, generators, and fans.

1.3 ALTERNATIVE DESIGNS

The vendor may offer alternative designs. Any exceptions to the standard including, alternate design differences from this standard, shall be clearly stated in the proposal as required by 5.2.1.

1.4 CONFLICTING REQUIREMENTS

In case of conflict between this standard and the inquiry, the inquiry shall govern. At the time of the order, the order shall govern.

1.5 DEFINITION OF TERMS

Terms used in this standard are defined in 1.5.1 through 1.5.38.

1.5.1 axially (horizontally) split refers to casing joints that are parallel to the shaft centerline.

1.5.2 The **bending stress number (S)** corresponds to the likelihood of fatigue cracking at the tooth root filet. If this stress is excessive, it may lead to failure of the gear teeth.

1.5.3 critical speed is a shaft rotational speed at which the rotor bearing support system is in a state of resonance (see 2.6.1).

1.5.4 The use of the word **design** in any term (such as design horsepower, design pressure, design temperature, or design speed) should be avoided in the purchaser's specifications. This terminology should be used only by the equipment designer and the manufacturer.

1.5.5 gauss levels refers to the magnetic field level of a component measured with a "Hall effect" probe, with no interference from adjacent magnetic part or structures.

1.5.6 gear refers to either the pinion or the gear wheel.

1.5.7 The **gear service factor (SF)** is the factor that is applied to the tooth pitting index and the bending stress number, depending on the characteristics of the driver and the driven equipment, to account for differences in potential overload, shock load, and/or continuous oscillatory torque characteristics.

1.5.8 gear unit is the complete power transmission assembly including the housing, gears, and bearings.

1.5.9 gear unit rated power is the maximum power specified by the purchaser on the data sheet and stamped on the nameplate (see 2.2.1).

1.5.10 gear wheel (bull gear) refers to the lowest speed rotor.

1.5.11 A **hunting tooth combination** exists for mating gears when 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.

1.5.12 informative element: Describes part of the standard which is provided for information and to assist in the understanding or use of the standard. Compliance with an informative part of the standard is not mandated.

Note: An appendix may be informative or normative as indicated.

1.5.13 lead modification: A calculated machined deviation of the pinion and/or the gear wheel tooth flank from the theoretical tooth form intended to improve uniformity of the load distribution across the face width of the mating rotors when the gears are operating at normal load.

1.5.14 maximum allowable speed (in revolutions per minute) is the highest speed at which the manufacturer's design will permit continuous operation.

1.5.15 maximum continuous speed (in revolutions per minute) is the speed at least equal to 105% of the rated pinion speed for variable-speed units and is the rated pinion speed for constant-speed units.

1.5.16 mechanical rating is the gear unit rated power (see 1.5.9) multiplied by the specified gear service factor (see 1.5.7).

1.5.17 minimum allowable speed (in revolutions per minute) is the lowest speed at which the manufacturer's design will permit continuous operation.

1.5.18 normal transmitted power is the power at which usual operation is expected and optimum efficiency is desired. The normal transmitted power may be equal to or less than the gear unit rated power.

1.5.19 normative: A requirement of the standard.

1.5.20 observed: An 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 if the purchaser or his representative is not present.

1.5.21 owner: The final recipient of the equipment who may delegate another agent as the purchaser of the equipment.

1.5.22 pinion refers to the highest speed rotor.

1.5.23 profile modification: A calculated machined deviation of the pinion and/or the gear wheel tooth flank from the theoretical tooth form. It is intended to obtain a trapeziform tooth load transfer distributed evenly along the path of contact in a transverse section when the gears are operating at normal load.

1.5.24 purchaser: The agency that issues the purchase order and specification to the vendor.

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

1.5.25 The **rated input speed** of the gear unit (in revolutions per minute) is the specified (or nominal) rated speed of its driver, as designated by the purchaser on the data sheets.

1.5.26 The **rated output speed** of the gear unit (in revolutions per minute) is the specified (or nominal) rated speed of its driven equipment, as designated by the purchaser on the data sheets.

1.5.27 scuffing: A form of gear tooth surface damage which refers to welding and tearing of the tooth surface by the flank of the mating gear. Scuffing occurs when the oil film thickness is small enough to allow the flanks of the gear teeth to contact and slide against each other.

1.5.28 scuffing resistance: A measure of the oil films ability to prevent metal to metal contact of the gear set teeth in the mesh during operation.

1.5.29 shall: Used to state a mandatory requirement.

1.5.30 should: Used to state a recommendation.

1.5.31 special purpose application: An application for which the equipment is designed for uninterrupted, continuous operation in critical service, and for which there is usually no spare equipment.

1.5.32 special tool: A tool which is not a commercially available catalog item.

1.5.33 The **tooth pitting index (*K*)** corresponds to a contact surface stress number. The tooth pitting index is used to determine a load rating at which progressive pitting of the teeth does not occur during their design life.

1.5.34 total indicator reading (TIR), also known as total indicated runout: The 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 perfectly cylindrical surface, the indicator reading implies an eccentricity equal to half the reading. For a perfectly flat face, the indicator reading gives an out-of-squareness equal to the reading. If the diameter in question is not perfectly cylindrical or flat interpretation of the meaning of TIR is more complex and may represent ovality or lobing.

1.5.35 trip speed (revolutions per minute) is the speed at which the independent emergency overspeed device operates to shut down a variable-speed prime mover. For the purpose of this standard, the trip speed of alternating current electric motors, except variable frequency drives, is the speed (revolutions per minute) corresponding to the synchronous speed at maximum supply frequency.

1.5.36 unit responsibility refers to the responsibility for coordinating the delivery and technical aspects of the equipment and all auxiliary systems included in the scope of the order. The technical aspects to be considered include but are not limited to such factors as power requirements, speed, rotation, general arrangement, couplings, dynamics, noise, lubrication, sealing system, material test reports, instrumentation, piping, conformance to specifications and testing of components.

1.5.37 vendor (also known as supplier): The agency that supplies the equipment.

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

1.5.38 witnessed: An 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 his representative are in attendance.

1.6 REFERENCE PUBLICATIONS

1.6.1 Referenced publications are listed in Appendix B.

1.6.2 The purchaser and the 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.

1.6.3 The vendor who has unit responsibility shall assure that all subvendors comply with the requirements of this standard and all reference standards.

1.6.4 All referenced standards, to the extent specified in the text, are normative.

1.6.5 Notes following a paragraph are informative.

● 1.7 STANDARDS

The purchaser shall specify whether the equipment supplied to this standard shall comply with the applicable ISO standards or applicable U.S. standards.

● 1.8 UNITS OF MEASURE

The purchaser shall specify whether data, drawings, gear unit name plate, hardware (including fasteners) and equipment supplied to this standard shall use the SI or U.S. customary or both systems of units.

2 Basic Design

2.1 GENERAL

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

2.1.2 The equipment (including auxiliaries) covered by this standard shall be designed and constructed for a minimum service life of 20 years and at least 5 years of uninterrupted operation.

Note: It is recognized that this is a design criterion.

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

● **2.1.4** The purchaser shall specify the equipment's normal operating point.

● **2.1.5** The purchaser shall indicate if the input or output speed is specified (that is, must be exactly adhered to by the 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 also indicate on the data sheets 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 impracticable for the vendor to match exactly both the rated input and the rated output speed designated on the data sheets.

● **2.1.6** Control of the sound pressure level (SPL) of all equipment furnished shall be a joint effort of the purchaser and the vendor having unit responsibility. 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 sound pressure and sound power level data per octave band for the equipment.

2.1.7 Equipment shall be designed to run safely to the trip speed setting indicated in Table 1. Unless otherwise specified, rotors for turbine-driven gear units shall be designed to operate safely at momentary speeds up to 130% of the rated speed.

2.1.8 Equipment driven by induction motors shall be rated at the actual motor speed for the rated condition.

2.1.9 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.

● **2.1.10** Motor, 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 60079-0 (NFPA 70, Articles 500, 501, 502, and 504) as well as local codes specified and furnished on request by the purchaser.

Note: Refer to Appendix B table of standards for a listing of typical electric codes.

2.1.11 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

Table 1—Driver Trip Speeds

Driver Type	Trip Speed (% of Max 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%

Note: [a] Indicates governor class, as specified in NEMA SM23.

minimize contamination by moisture, dust, and other foreign matter, both during periods of operation and idleness.

2.1.12 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 vendor who has unit responsibility.

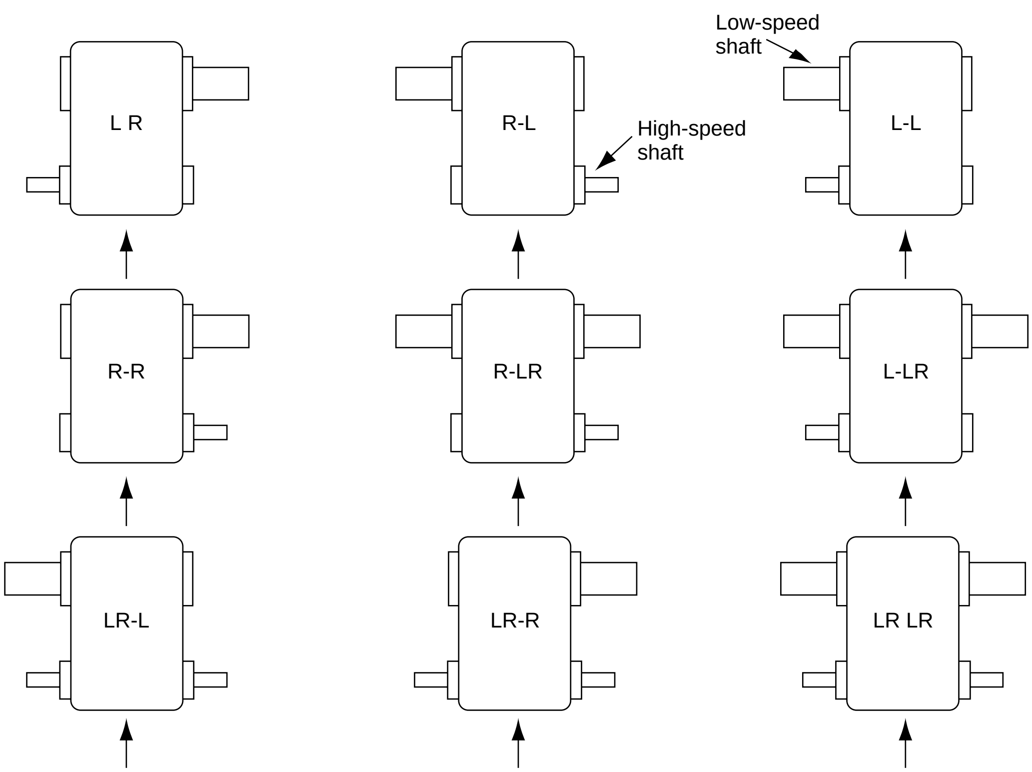
● **2.1.13** Many factors (such as piping loads, alignment at operating conditions, supporting structure, handling during shipment, and handling and assembly at the site) may adversely affect site performance. To minimize the influence of these factors, the vendor shall review and comment on the purchaser's baseplate and foundation drawings. In addition, the vendor's representative may be requested to check alignment at the operating temperature and may be requested to be present during the initial alignment check and the tooth contact check.

● **2.1.14** The equipment, including all auxiliaries, shall be suitable for operating 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.

2.1.15 Gear units shall not require a break-in period at reduced speed and load in the field.

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

● **2.1.16** The gear unit shall be designed to withstand all internal and external loads inherent to geared, rotating machinery systems. The gear unit shall be capable of withstanding the specified external loads (thrust, lube-oil piping, and so forth) transmitted across the gear mesh while the gear unit is operating at the rated power specified by the purchaser.



Notes:
1. L = left; R = right.
2. Arrows indicate the line of sight used to determine the direction of the shaft extensions. (The figure depicts plan views.)
3. The letter or letters before the hyphen refer to the number and direction of high-speed shaft extensions; the letter or letters after the hyphen refer to the number and direction of low-speed shaft extensions.
4. The material for this figure was extracted from AGMA 6011 with permission of the publisher.

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

Table 2—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

Note: 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 AGMA 6010 with permission from the publisher.

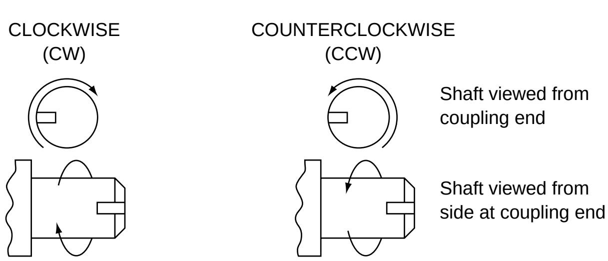
2.1.17 All equipment shall be designed to permit rapid and economical maintenance. Major parts such as casing components and bearing housings shall be designed and manufactured to ensure accurate alignment on reassembly. This may be accomplished by the use of shouldering, cylindrical dowels or keys.

2.1.18 Spare and replacement parts for the machine and all furnished auxiliaries shall meet all the criteria of the standard.

● **2.1.19** The purchaser shall specify the appropriate shaft assembly designation selected from the combinations listed in Table 2 and illustrated in Figure 1. The purchaser may alternatively circle one or more of the assembly designations on a copy of Figure 1 and submit the copy with the quotation request. If the shaft arrangement has not been finalized at the time of the quotation request, the purchaser shall designate all of the combinations under consideration.

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

2.1.20.1 On the data sheets and in drawings and tables, 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

● **2.1.20.2** The purchaser shall specify the rotational direction of both the high-speed and the low-speed shaft. When 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.

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

2.2 RATING

● 2.2.1 Gear Unit Rated Power

The gear unit rated power shall be specified by the purchaser. For gear units located next to the driver, the minimum gear unit rated power shall be the maximum installed power of the driver. For electric motor drivers, the gear unit rated power shall be the motor nameplate rating multiplied by the motor service factor. All modes of normal and abnormal operation shall be examined. Modes of operation may include the number of starts per unit of time, reduced load, removal, reversed load, reduced speed, and overload and overspeed conditions. For gear units between two items of driven equipment, the power rating of such units should normally be not 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 will be specified by the purchaser and shall be the basis for sizing the gear unit.

● 2.2.2 Normal Transmitted Power

For optimal gear unit design (rotordynamics, lead modification), the purchaser should specify the normal transmitted power and any special operating conditions (such as load reversals), in addition to the gear unit rated power.

2.2.3 Gear Service Factor

● **2.2.3.1** The minimum gear service factor (SF) shall be specified by the purchaser on the data sheets based on the application, as listed in Table 3.

2.2.3.2 In addition to specifying the service factor, the purchaser may specify the hardnesses of the pinion and gear wheel to establish power and frame sizes. The purchaser should consider the following factors when determining these hardnesses:

a. The ductility of the gear teeth.

b. Noise.

Table 3—Minimum Gear Service Factors

Driven Equipment	Synchronous & Variable Speed Motors	Induction Motors	Steam & Gas Turbines	Reciprocating Engines
Centrifugal blowers	1.6	1.4	1.6	1.7
Compressors				
Centrifugal	1.6	1.4	1.6	1.7
Axial	1.6	1.4	1.6	1.7
Rotary lobe	1.8	1.7	1.7	2.0
Reciprocating	2.1	2.0	2.0	2.3
Extruders	1.8	1.7	1.7	–
Fans				
Centrifugal	1.5	1.4	1.6	1.7
Forced draft	1.5	1.4	1.6	1.7
Induced draft	1.8	1.7	2.0	2.2
Generators & 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, except those listed below)	1.5	1.3	1.5	1.7
Centrifugal, boiler feed	1.8	1.7	2.0	–
Centrifugal, hot oil	1.8	1.7	2.0	–
Centrifugal, high speed (over 3600 rpm)	–	1.7	2.0	–
Centrifugal, water supply	1.6	1.5	1.7	2.0
Rotary, axial flow/all types	1.6	1.5	1.5	1.8
Rotary, gear	1.6	1.5	1.5	1.8
Reciprocating	2.1	2.0	2.0	2.3

c. Future upgrading.

d. Conventional versus special machine finish requirements.

2.2.4 Tooth Pitting Index

2.2.4.1 Gear units shall be sized on the basis of a tooth pitting index called a *K* factor. This method includes factors to account for such considerations as the radii of curvature of the contacting tooth surfaces, extended life, high reliability, dynamic load effects, maldistribution of tooth loading across the face, and the strength of the materials in terms of pitting resistance.

Note: This simplified system for sizing the gear unit is consistent with ANSI/AGMA 2101-D02 (2001-D02), with conservative assumptions for each variable in the more complex basic equations contained in that document.

2.2.4.2 The allowable *K* factor is defined as follows:

$$K = [W_t/df_w][(R+1)/R] \quad (1)$$

In SI units:

$$W_t = [(1.91 \times 10^7)\, P_g]/N_p d \quad (2)$$

In U.S. customary units, W_t can be expressed as follows:

$$W_t = (126{,}000\, P_g) / N_p d$$

where

K = tooth pitting index in megapascals (pounds per square in.),

W_t = transmitted tangential load at the operating pitch Diameter, in newtons (pounds),

F_w = net face width, in millimeters (in.),

d = pinion pitch diameter, in millimeters (in.),

R = number of teeth in the gear wheel divided by number of teeth in the pinion,

P_g = gear unit rated power, in kilowatts (horsepower),

N_p = pinion speed, in revolutions per minute.

2.2.4.3 The allowable *K* factor at the gear unit rated power will vary with the materials selected for the gear teeth, the tooth hardening processes used, and the service factor. The allowable K factor is calculated as follows:

$$K_a = I_m/(SF) \quad (3)$$

where

K_a = allowable K factor,

I_m = material index number (from Table 4 and Figure 3),

SF = minimum gear service factor (from Table 3).

2.2.4.4 Table 4 presents material index numbers and maximum length-to-diameter (*L/d*) ratios for several combinations of materials in current use. The minimum material hardness is selected for the element, pinion or gear wheel, with the lowest minimum hardness. Do not extrapolate the curve specified in Table 4 (see Figure 3 and note below) beyond the limits shown.

Note: Figure 3 uses the minimum specified hardness. Normal heat treating practice requires a tolerance range on hardness. The upper end of the hardness range can fall outside the limits shown in Figure 3.

2.2.4.5 Deflection of gears with unmodified leads shall not have a total lead mismatch (combined bending, torsional deflection & thermal distortion) of the rotors across the gear face width that is greater than 0.025 µm (0.0010 in.) for through hardened gears or greater than 0.018 µm (0.0007 in.) for case hardened gears. The determination of rotor deformation is to be based on the rated power.

2.2.4.6 When a higher *L/d* ratio than tabulated in Table 4 is proposed, the gear unit vendor shall submit justification in the proposal for using the higher *L/d* ratio. Purchaser's approval is required when *L/d* ratios exceed those of Table 4. When operating conditions other than the gear unit rated power are specified by the purchaser, such as the normal transmitted power, the gear unit 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. When modified leads are to be furnished, purchaser and vendor shall agree on the tooth contact patterns obtained in the checking stand, housing, and test stand.

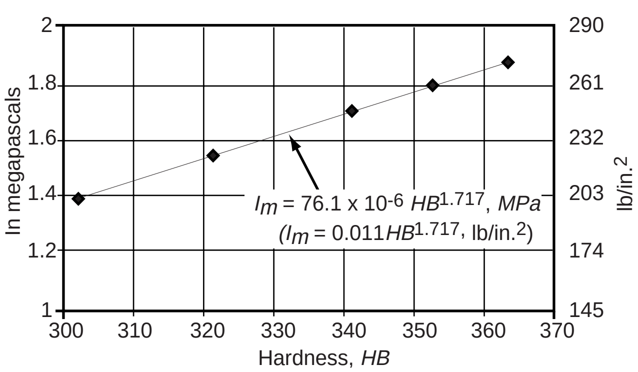


Figure 3—Material Index Number, Through Hardened

Table 4—Material Index Numbers and Maximum *L/d* Ratios

Hardening Method	Minimum Hardness	Material Index Number, Im: Megapascals	Material Index Number, Im: (lb/in.^{2})	Maximum Pinion L/d Ratio: Double Helical	Maximum Pinion L/d Ratio: Single Helical
Through Hardened	See Figure 3	See Figure 3	See Figure 3	2.2	1.6
Nitrided	90 HR15N	2.07	(300)	2.0	1.6
Carburized	58 Rc	3.03	(440)	2.0	1.6

Table 5—Allowable Bending Stress Number, S_a

Hardening Method	Minimum Hardness	Allowable Bending Stress Number, S_a: Megapascals	Allowable Bending Stress Number, S_a: (lb/in.^{2})
Through Hardened	See Figure 4	See Figure 4	See Figure 4
Nitrided	90 HR15N	190	(27500)
Carburized	58 Rc	266	(38500)

If the rotor deformation exceeds the values limited by 2.2.4.5, regardless of the *L/d*, an analytically determined lead modification shall be applied in order to reduce the total actual mismatch to a magnitude below the limiting values in para. 2.2.4.5. This will facilitate a more uniform load distribution across the entire face width. Successful application of lead modifications are dependent on high gear tooth accuracy of ANSI/AGMA/ISO-1328 accuracy Grade 4 or better (AGMA Grade 13).

Notes:

1. Below 20,000 FPM, the thermal distortion portion of the total deformation is generally not significant.
2. See Appendix H for further discussion.

2.2.5 Tooth Size and Geometry

2.2.5.1 The size and geometry of the gear teeth shall be selected so that the bending stress number as calculated using Equation 4 does not exceed the values in Table 5. Do not extrapolate the curve specified in Table 5 (see Figure 4 and note below) beyond the limits shown. This method includes factors similar to those used to determine the allowable K factor. This simplified system for sizing gear teeth is consistent with ANSI/AGMA 2101 (ANSI/AGMA 2001).

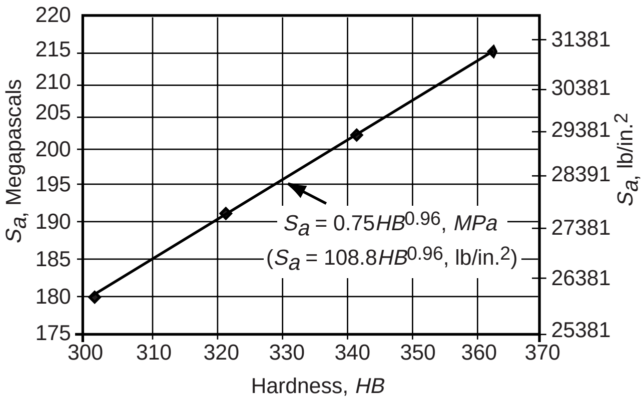


Figure 4—Allowable Bending Stress Number, S_a

Note: Figure 4 uses the minimum specified hardness. Normal heat treating practice requires a tolerance range on hardness. The upper end of the hardness range can fall outside the limits shown in Figure 4.

2.2.5.2 The vendor shall calculate the bending stress number for both the pinion and the gear wheel. Where idlers are used, the calculated stress shall be limited to 70% of the value given in Figure 4. The bending stress number is calculated as follows:

In SI units:

$$S = [W_t/(m_n F_w)](SF)[(1.8\cos\gamma)/J] \quad (4)$$

In U.S. customary units:

$$S = [W_t P_{nd})/F_w](SF)[(1.8\cos\gamma)/J]$$

where

S = bending stress number,

S_a = allowable bending stress number,

P_{nd} = normal diametral pitch, in 1/in.,

γ = helix angle,

J = geometry factor (from AGMA 908),

m_n = module number, in millimeters.

2.2.6 Scuffing

To avoid scuffing, the gear shall meet the requirements of ANSI/AGMA 6011-C98, Annex B. To avoid dependency on extreme pressure additives, unless otherwise specified, the gear unit shall be designed for use with an oil meeting ISO 14365-1 load stage 5. When an alternative oil is requested, the vendor shall provide calculations and an experience list to support a request for an alternate oil selection. Alternate calculating methods can be used based on supplier experience.

Notes:
1. Refer to AGMA 925 for additional information.

2. When high speed gears are subject to highly loaded conditions and high sliding velocities the lubricant film may not adequately separate the surfaces. This localized damage to the tooth surface is referred to as "scuffing" (sometimes incorrectly called scoring). Scuffing will exhibit itself as a dull matte or rough finish usually at the extreme end regions of the contact path or near the points of a single pair of teeth contact resulting in severe adhesive wear.

Scuffing is not a fatigue phenomenon and may occur instantaneously. The risk of scuffing damage varies with the material of the gear, the lubricant being used, the viscosity of the lubricant, the surface roughness of the tooth flanks, the sliding velocity of the mating gear set under load and the geometry of the gear teeth.

Changes in any or all of these factors can reduce scuffing risk.

2.2.7 Deviations

It is recognized that special cases will exist in which it may be desirable to deviate from the rating rules specified in 2.2.1 through 2.2.5 The vendor shall describe and justify such deviations in the proposal.

2.3 CASINGS

2.3.1 Design Parameters

2.3.1.1 Gear casings shall be either cast or fabricated and shall be designed and constructed to maintain rotor alignment under all load conditions.

2.3.1.2 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 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.

2.3.1.3 Mounting surfaces shall meet the following criteria:

a. They shall be machined to a finish of 6 µm (0.00025 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 meter (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.

Hold-down bolts holes shall be drilled perpendicular to the mounting surface or surfaces, machined or spot faced to a diameter three times that of the hole and to allow for equipment alignment, be 15 mm ($^1/_2$ in.) larger in diameter than the hold-down bolt.

2.3.1.4 The equipment feet shall be provided with vertical jackscrews.

2.3.1.5 Bores shall be machined to a sufficient degree of accuracy so that a gear set that contacts correctly on true centers on a rotor checking stand will also contact correctly in its own casing.

2.3.1.6 Casings shall be designed to prevent damage due to distortion caused by temperature, torque, and allowable external forces and moments.

2.3.1.7 To the maximum extent practicable, casings shall be designed with internal oil passages to minimize external piping. All internal piping should preferably be welded and should preferably use flanges for all connections. Any threaded piping shall be a minimum of Schedule 80 and shall be seal welded at flanges (see 2.4.6.4).

2.3.1.8 Where internal space does not permit the use of DN15, DN20, or DN25 ($^1/_2$-, $^3/_4$-, or 1-in.) ASTM A 312 pipe, seamless steel tubing conforming to ASTM A 192 with stainless steel fittings, or stainless steel tubing conforming to ASTM A 269 with stainless steel fittings may be furnished. Tubing thicknesses shall meet the requirements of Table 6. The make and model of fittings shall be subject to the purchaser's approval.

2.3.1.9 The design of internal piping shall achieve proper support and protection 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.

2.3.1.10 Internal piping and oil passages shall be cleaned to remove all foreign material.

2.3.1.11 Casings shall be designed to permit rapid drainage of lube oil and to minimize oil foaming, which could lead to excessive heat rise of the oil. For gears with pitch line velocities of more than 125 m/sec (25,000 ft/min.), consideration shall 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.
c. Meshing direction.
d. An oil drain line to the reservoir that is independent from all other drain systems.
e. A second full-sized drain connection.
f. Larger side and circumferential clearances between gears and pinions and the casing.
g. Windage baffles.

2.3.1.12 A filter-breather shall be provided by the vendor. The filter-breather shall be constructed of Series 300 stainless steel with Series 300 stainless steel or copper-nickel alloy (e.g., Monel) 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. The filter-breather shall be at least NPS 1$^1/_2$ and its construction shall permit easy disassembly for inspection and cleaning.

Table 6—Minimum Tubing Wall Thickness

Nominal Tubing Size		Minimum Wall Thickness	
Milimeters	Inches	Millimeters	Inches
12.7	$^1/_2$	1.65	0.065
19.05	$^3/_4$	2.40	0.095
25.4	1	2.75	0.109

Note: The location generally recommended by gear manufacturers for the filter-breather is on the drain pipe immediately adjacent to the casing, with no breather on the casing itself. The filter-breather may be located on the casing or the drain. Location of the filter-breather shall be shown on the outline drawing.

2.3.1.13 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. The inspection opening or openings shall be at least one-half the width of the gear face.

2.3.1.14 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.

2.3.2 Joints

Axially split casings shall use a metal-to-metal joint (with a suitable joint compound) that is tightly maintained by suitable bolting. Gaskets (including string type) shall not be used on the axial joint.

2.3.3 Bolting

2.3.3.1 Case bolting should preferably be of the through-bolt type. Where this is impractical, studs shall be used unless assembly or disassembly prevents their use. Cap screws are acceptable in such instances.

2.3.3.2 The details of threading shall conform to ISO 261, ISO 262, ISO 724, ISO 965 (ASME B1.1).

2.3.3.3 Studded connections shall be furnished with studs installed. Blind stud holes should be drilled only deep enough to allow a preferred tap depth of 1$^1/_2$ times the major diameter of the stud; the first 1$^1/_2$ threads at both ends of each stud shall be removed.

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

2.3.4 Assembly and Disassembly

2.3.4.1 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.

2.3.4.2 Jackscrews, guide rods, and casing alignment dowels shall be provided to facilitate disassembly and reassembly. When jackscrews are used as a means of parting contacting faces, one of the faces shall be relieved (counterbored or recessed) to prevent a leaking joint or improper fit caused by marring. Guide rods shall be of sufficient length to prevent damage to the internals or casing studs by the casing during disassembly and reassembly.

2.3.4.3 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 casing shall be clearly and permanently marked on the casing. Methods of lifting the assembled machine shall be specified by the vendor.

2.4 CASING CONNECTIONS

2.4.1 A single lube-oil supply connection is preferred.

2.4.2 A single lube-oil drain connection from the gear casing is preferred. The minimum drain pipe size shall be based on the total inlet flow to the gear casing, as shown in Table 7. Drains smaller than shown in the table 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.

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

In SI units:

$$F_{df} = 0.009D^2 \qquad (5)$$

In U.S. customary units:

$$F_{df} = 1.53D^2$$

where

F_{df} = allowable flow in drain line with D over 300mm, l/m (gpm),

D = internal drain line diameter, mm (in.).

● **2.4.3** When specified, casings shall be provided with an inlet purge connection at least NPS $^1/_2$ in., located to assure a sweep of purge gas across the casing to the filter-breather.

2.4.4 Openings for piping connections shall be DN 20 ($^3/_4$ NPS) or larger and in accordance with ISO 6708. Sizes DN 32, DN 65, DN 90, DN 125, DN 175, and DN 225, ($1^1/_4$, $2^1/_2$, $3^1/_2$, 5, 7, and 9 NPS) shall not be used.

Table 7—Drain Pipe Sizes

Inlet Flow Rate		Minimum Drain Size[a]	
Liters Per Minute	Gallons Per Minute	Millimeters	Inches
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

2.4.5 All of the purchaser's connections on the gear case shall be accessible for disassembly without the gear unit being moved.

● **2.4.6** Oil inlet and drain connections shall be flanged or machined and studded, oriented as specified, and not less than DN 20 (NPS $^3/_4$). Where flanged or machined and studded openings are impractical, threaded openings in sizes DN 20 (NPS $^3/_4$) through DN 40 (NPS $1^1/_2$) are permissible. These threaded openings shall be installed as specified in 2.4.6.1 through 2.4.6.4.

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

2.4.6.2 Pipe nipples shall be a minimum of Schedule 160 seamless for DN 25 (NPS 1) and smaller and a minimum of Schedule 80 for DN 40 (NPS $1^1/_2$).

2.4.6.3 The pipe nipple shall be provided with a welding-neck or slip-on flange.

2.4.6.4 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.

2.4.7 Flanges shall conform to ISO 7005-1, or ISO 7005-2 (ASME B16.1, B16.5, B16.42, or SAE J518) as applicable, except as specified in 2.4.7.1 through 2.4.7.4 as follows:

2.4.7.1 Cast iron flanges shall be flat faced and conform to the dimensional requrements of ISO 7005-2 (ANSI/ AMSE B 16.1, B16.42). Class 125 flanges shall have a minimum thickness equal to Class 250 for sizes DN 200 (8 NPS) and smaller.

2.4.7.2 Flat-faced flanges with full raised-face thickness are acceptable on cases other than cast iron.

2.4.7.3 Flanges that are thicker or have a larger outside diameter than that required by ISO 7005-1 (ASME B16.5) are acceptable.

2.4.7.4 Purchaser connections other than those covered by ISO 7005-1 or ISO 7005-2 (ASME B16.5) require the purchaser's approval. When specified, mating parts shall be furnished by the vendor.

2.4.8 Machined and studded connections shall conform to the facing and drilling requirements of ISO 7005-1 or ISO 7005-2 (ASME B16.1, B16.5, or B16.42). Studs and nuts shall be furnished installed, the first 1.5 threads at both ends of each stud shall be removed.

2.4.9 Threaded connections shall not be more than DN 40 (NPS $1^1/2$). Tapped openings and bosses for pipe threads shall conform to ISO 7-1 (ASME B16.5). Pipe threads shall be taper threads conforming to ASME B 1.20.1.

2.4.10 Tapped openings not connected to piping shall be plugged with solid plugs furnished in accordance with ASME B 16.11. Plugs that may later require removal shall be of corrosion-resistant material. Threads shall be lubricated. Tape shall not be applied to threads of plugs inserted into oil passages. Plastic plugs are not permitted.

2.5 GEAR ELEMENTS

2.5.1 General

2.5.1.1 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. Honing.
d. Precision hobbing.

All gear teeth finished by shaving or honing shall have been generated by hobbing. Shaving cutters and rotary hones shall have a hunting tooth combination with the workpiece.

2.5.1.2 The tooth surface on loaded faces of completed gears shall have a finish, as measured along the pitch line, of 0.8 μm (32 microinch) Ra or better. See also ISO TR 10064-4 on measuring methods.

2.5.1.3 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.

2.5.1.4 Hunting tooth combinations are required. To achieve this requirement, it may be necessary for the purchaser to adjust the exact gear ratio. If such adjustment is impractical, the purchaser and the 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.

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

2.5.2 Quality Assurance

2.5.2.1 After the gear teeth are finish-cut, shaved, or ground on a hobbing, shaving, or grinding machine, the gear elements will be checked per ANSI/AGMA ISO 1328-1 for gear tooth accuracy. The gear elements shall have an accuracy of Grade 4 or better. The records of the gear accuracy check shall be maintained by the vendor for a period of not less than 20 years and shall be available to the purchaser on request.

Note: Previous journal runout chart requirement is replaced with gear charts requirement. ANSI/AGMA ISO 1328-1 references the journal in the measurement of the gear teeth. See also ISO/TR 10064-1 on measuring methods.

2.5.2.2 Each pair of mating gears shall be considered as a matched set and shall be checked for contact on a contact checking stand and in the job casing at the vendor's shop. The check per 2.5.2.1 cannot be used as a replacement for this check. A thin coating of color transfer material (such as Prussian blue) shall be applied at three locations 120 degrees apart to 4 or more teeth of the dry degreased gear wheel. (Layout dye shall not be used for the contact check on the checking stand.) With the gear elements held firmly on the correct center distance and with the shaft centerlines parallel within 42 μm per meter (0.0005 in./ft) with a total misalignment of not more than 25 μm (0.001 in.), 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 vendor. Prior to the contact tests, the vendor shall make available to the purchaser a contact drawing or vendor engineering specification that defines the acceptable contact. Unmodified leads generally show a minimum of 80% contact across the tooth length. The drawing or specification and the results of the checking-stand and 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 a annotated sheet of white paper.

Notes:
1. When used to support the gear elements during contact checking, runout of shaft centers or rollers may require mechanical compensation to demonstrate the true contact pattern.
2. A thin coat of cuprous oxide paste (made with mineral spirits) may be placed on the pinion to enhance visibility of the transfer.

2.5.2.3 The 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 (2.0 mils) 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 4.3.2.2.10).

2.5.3 Fabrication

2.5.3.1 Unless otherwise specifically approved by the purchaser, pinions shall be integrally forged with their shafts.

2.5.3.2 For pitch line velocities above 150 m/sec (30,000 ft/min.), gears shall be integrally forged with their shafts. For pitch line velocities of 150 m/sec (30,000 ft/min.) and less, gears may be integrally forged with, or separate from their shafts. Separate gears shall be forgings or shall be of fabricated construction using forged steel rims and shall be assembled on their shafts with an interference fit. The pitch line velocities listed in Table 8 shall not be exceeded without the purchaser's specific approval.

2.5.3.3 For gears integrally forged with their shafts operating at pitch line velocities above 150 m/sec (30,000 ft/min.), the purchaser and vendor will agree to material suitability and additional inspection and testing necessary to ensure quality. This may include but is not limited to:

a. Material cleanliness in excess of ANSI/AGMA ISO 6336-5 Grade ME and MX 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 (magnaglo) 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.

Table 8—Pitch Line Velocities

	Maximum Pitch Line Velocity	
Gear Manufacturing Method	Meters Per Second	Feet Per Minute
Shrunk on Forged Rims	60	12,000
Welded with Forged Rims	127	25,000
Shrunk-on Forged Gears	150	30,000

2.5.4 Shafts

2.5.4.1 Shafts shall be sized to transmit the gear unit rated power within the stress limits of ANSI/AGMA6001-D97. Shafts shall be machined from one-piece, heat-treated steel. Shafts with finished diameters 200 mm (8 in.) and larger shall be forged. Shafts with finished diameter less than 200 mm (8 in.) may be hot-rolled barstock purchased to the same quality and heat treatment criteria as shaft forgings.

2.5.4.2 Unless otherwise specified, shafts shall be provided with integral flanges for couplings.

2.5.4.3 All shaft keyways shall have fillet radii conforming to ANSI/ASME B17.1.

2.5.4.4 The rotor shaft sensing areas to be observed by the 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. For gear units with 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. These areas shall not be metallized, sleeved, or plated. The final surface finish shall be to a maximum of 0.8 µm (32 microinch) Ra, preferably obtained by honing or burnishing. 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:

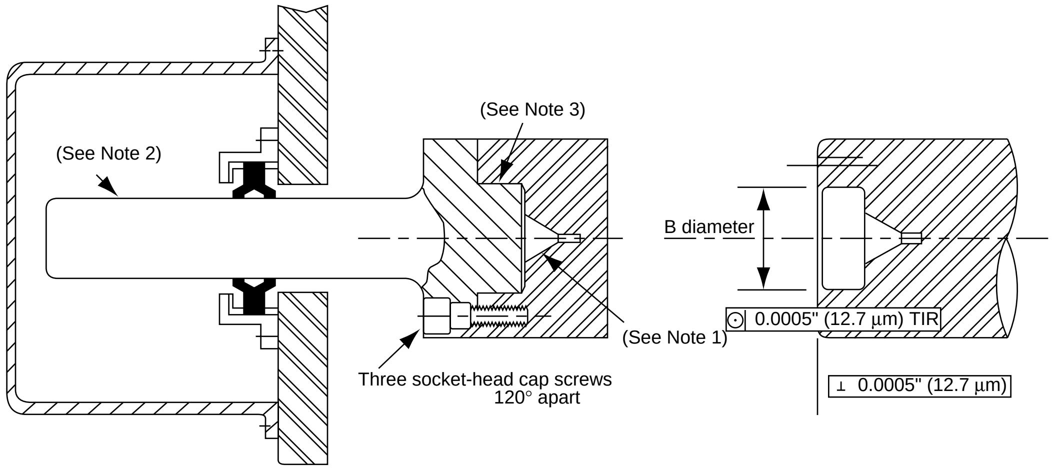
a. For areas to be observed by radial-vibration probes:
   1. For shaft journals less than 305 mm (12 in.) diameter, 25% of the allowable peak-to-peak vibration amplitude or 6.5 µm (0.00025 in.), whichever is greater.
   2. For shaft journals 305 mm (12 in.) and greater diameter, 25% of the allowable peak-to-peak vibration amplitude or 10 µm (0.0004 in.), whichever is greater.

b. For areas to be observed by axial-position probes, 13 µm (0.0005 in.).

Note: Steels used in manufacturing gears are prone to develop higher levels of electrical runout which is difficult to reduce using conventional means such as degaussing, burnishing, and electro peening. When it has become evident that further attempts to lower the electrical runout will be unproductive, other reduction methods may be implemented with purchaser's approval.

● **2.5.4.5** If specified, the vendor shall make one of the provisions specified in 2.5.4.5.1 through 2.5.4.5.3 to measure torsional response.

2.5.4.5.1 As agreed upon by the purchaser, the shaft end and stub shaft shall have a counterbore and bolting arrangement similar to that shown in Figure 5. The shaft end and mating stub faces shall be perpendicular to the shaft centerline within 13 µm (0.0005 in.) total indicated runout. The



Notes:
1. It is very important that the centering hole for the gear (pinion) shaft be permanent in the counter bore.
2. The stub shaft extension is to be machined true with the journals.
3. The tolerance for the B diameter is to be from 0.0 to 25.4 µm (0.000 to 0.001 in.) tight.

Figure 5—Typical Torsiograph Modification

counterbore and mating stubs shall be concentric with and parallel to the shaft centerline within 13 µm (0.0005 in.).

2.5.4.5.2 One DN 25 (NPS 1) tapped opening with a plug shall be provided in the upper half of the casing for each gear element to 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 also be positioned so that detectors screwed into them will sense the passing of the teeth.

2.5.4.5.3 The purchaser and the vendor shall agree upon any other method.

2.5.4.6 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.

2.5.4.7 Shaft ends where coupling hubs will be mounted shall conform to API Std 671 (see 3.2).

● **2.5.4.8** If specified, gear elements shall have provisions for vertical storage. This provision shall be capable of supporting 1.5 times the rotor weight. See 3.6.2 for provision of hanging fixture.

2.5.4.9 To prevent buildup of potential voltages in the shaft, residual magnetism (Free Air Gauss Levels) of the rotating element shall not exceed 5 Gauss (.0005 Tesla).

2.6 DYNAMICS

2.6.1 General

Note: Refer to API Publ 684 *Tutorial on the API Standard Paragraphs Covering Rotor Dynamics and Balancing: An Introduction to Lateral Critical and Train Torsional Analysis and Rotor Balancing* for more information on rotor dynamics.

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

a. Unbalance in the rotor system.
b. Oil-film instabilities (whirl).
c. Internal rubs.
d. Gear-tooth meshing and side band frequencies, as well as other frequencies produced by inaccuracies in the generation of the gear teeth.
e. Coupling misalignment.
f. Loose rotor-system components.
g. Hysteretic and friction whirl.
h. Asynchronous whirl.
i. Electrical line frequency.

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

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 will be in a state of resonance. A rotor-bearing-support-system in resonance may 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.

2.6.1.2 For the purpose of this standard a resonant conditions of concern, such as lateral and torsional critical speeds, are those with an amplification factor (AF) equal to or greater than 2.5.

2.6.1.3 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 2.6.2.4.1 and 2.6.2.4.2). The effective stiffness of the structural support shall be considered in the analysis of the dynamics of the rotor-bearing support system (see 2.6.2.5d).

Note: Resonances of structural support systems may adversely affect the rotor vibration amplitude.

2.6.2 Lateral Analysis

2.6.2.1 The vendor shall conduct an undamped analysis to identify the undamped critical speeds and determine their mode shapes located in the range from 0% – 125% of trip speed.

2.6.2.2 At least three power levels shall be analyzed for the undamped analyses, and shall include the following:

a. The bearing-oil film, bearing structure at 10%, 50% and 100% gear unit rated power, and gear unit casing support structure stiffnesses.
b. The coupling weight to be supported by each gearbox shaft.

2.6.2.3 Unless otherwise specified, the results of the undamped analysis shall be furnished to the purchaser. The presentation of the results shall include:

a. Mode shape plots for all stiffness values specified in 2.6.2.2a (relative amplitude vs. axial position on the rotor).
b. Critical speed-support stiffness map (frequency vs. support stiffness). Superimposed on this map shall be the calculated system support stiffness; horizontal (*kxx*) and vertical (*kyy*) (see Figure 6).

2.6.2.4 Three lateral critical speed modes are generally of concern. The frequency at which these modes occur will vary as a function of the transmitted load, primarily due to the resulting stiffness change of the bearing-oil film (see Figure 6). The gear rotors shall meet the requirements given in 2.6.2.4.1 through 2.6.2.4.3.

2.6.2.4.1 When operating at the maximum torque, the three defined critical speeds of each rotor shall not be less than 20% above the maximum continuous speed of that rotor.

Notes:
1. Sometimes it is not possible to operate the gear rotor below the first critical speed. When a gear rotor must operate above the first critical speed, the rotor should be designed with the first critical speed about 60% of operating speed. First critical speeds below 60% of running speed are prone to create bearing instability or oil whip and if running much above 60% are too close to running speed.
2. Gear wheels that operate between 40% – 50% of pinion speed (2:1 to 2.5:1 gear ratio) can excite instability in the pinion bearings. Therefore, if gear ratios between 2:1 and 2.5:1 are required special consideration should be given to performing a pinion stability analysis.

2.6.2.4.2 When the operating torque is in the range of 50% – 100% of the maximum torque the separation margin above the maximum continuous speed of each rotor shall be a minimum of 10% – 20% in linear proportion to the transmitted torque.

2.6.2.4.3 When the specified minimum operating conditions are less than 40% of the gear unit rated power or less than 70% of the maximum continuous speed or the undamped analysis indicates the first critical is less than 120% of the the maximum continuous speed, the vendor shall perform a damped unbalanced response analysis in addition to the undamped analysis.

2.6.2.5 When the damped unbalanced response analysis is required it shall include but shall not be limited to the following:

Note: The following is a list of items the analyst is to consider. It does not address the details and product of the analysis which is covered in 2.6.2.7 and 2.6.2.8.

a. Rotor masses, including the mass moment of coupling halves, stiffness, and damping effects (for example, accumulated fit tolerances).
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 which may be caused by gear forces.
c. For tilt-pad bearings, the pad pivot stiffness and the direction of loading relative to the pivot.
d. Support stiffness, mass, and damping characteristics, including effects of frequency dependent variation. The term "support" includes the foundation or support structure, the base, the machine frame and the bearing housing as appropriate. For machines whose bearing support system stiffness values are less than or equal to 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. The vendor shall state the support stiffness values used in the analysis and the basis for these values (for example, modal tests of similar rotor support systems, or calculated support stiffness values).

Note: The support stiffness should in most cases be no more than 12.26×10^6 N/mm (7×10^6 lbs/in.).

e. Rotational speed, including the various starting-speed detents (speeds at which the driver dwells during the start cycle), operating speed and load ranges (including agreed-upon test conditions if different from those specified), trip speed, and coast-down conditions.
f. The location and orientation of the radial vibration probes which shall be the same in the analysis as in the machine.

● **2.6.2.6** When a damped unbalanced response analysis is required per 2.6.2.4.3, or the critical speed does not meet the

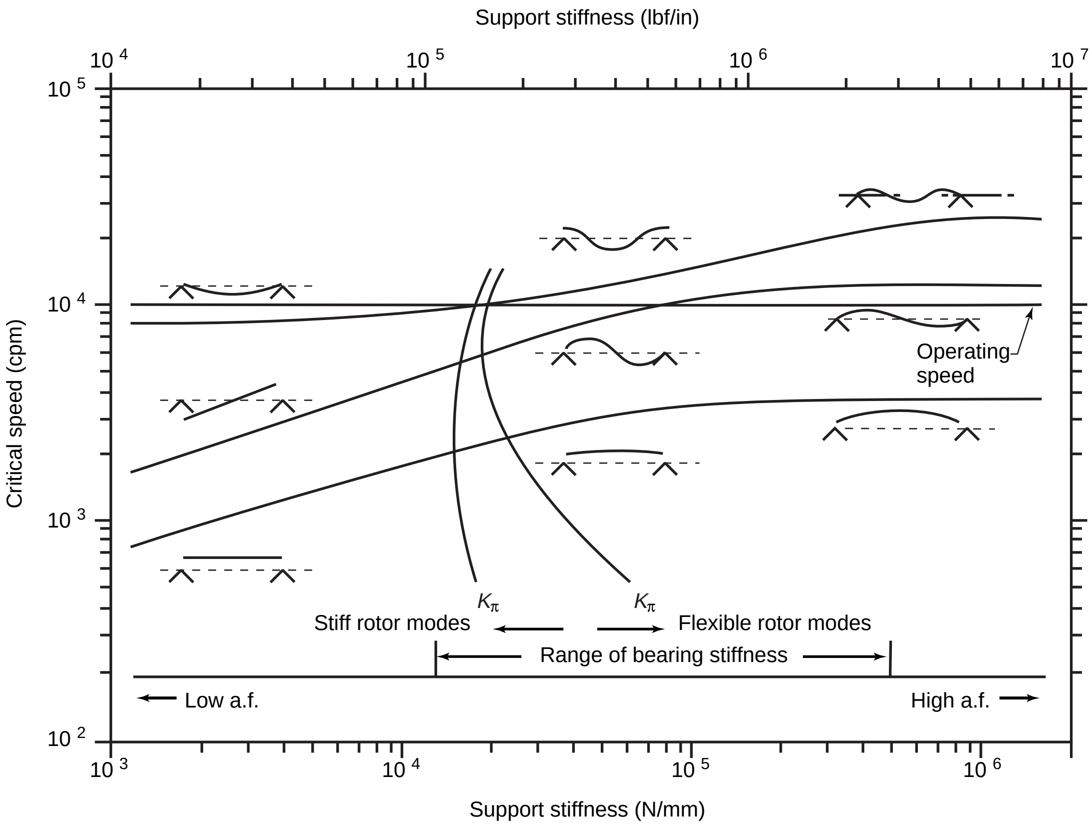


Figure 6—Undamped Critical Speed Map

requirements of 2.6.2.4.1 or 2.6.2.4.2, or if specified, it shall be conducted for each critical speed within the speed range of 0% – 125% of trip speed. Unbalance shall analytically be placed at the locations that have been determined by the undamped analysis to affect the particular mode most adversely. Unbalance representing the rotor shall be placed at the journal bearing centerline. The magnitude of the unbalance shall be 4 times the value of U as calculated by Equation 6. On a pinion, unbalance, representing the potential coupling unbalance for a non-integral coupling, shall be added at the coupling half center of gravity, see Figures 7a and 7b. The magnitude of unbalance shall be equal to 4 times the value of U as calculated at the overhung, cantilevered mode shape in Figures 7a and 7b.

In SI units:

$$U = 6350\ W/N \tag{6}$$

In Customary units:

$$U = 4\ W/N$$

where

U = input unbalance for the rotor dynamic response analysis in g-mm (oz-in.),

N = operating speed nearest to the critical speed of concern, in rpm,

W = journal static load in kg (lb.), or for bending modes where the maximum deflection occurs at the shaft ends, the overhung mass (that is the mass of the rotor outboard of the bearing) in kg (lb.) (see Figures 7a and 7b).

2.6.2.7 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% – 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 2.6.2.6.

c. The plot of deflected rotor shape for each critical speed resulting from the unbalances specified in 2.6.2.6, showing the major-axis amplitude at each coupling plane of flexure, the centerlines of each bearing, the locations of each radial probe, and at each seal throughout the machine as appropriate. The minimum design diametral running clearance of the seals shall also be indicated.

d. Additional Bode plots that compare absolute shaft motion with shaft motion relative to the bearing housing for

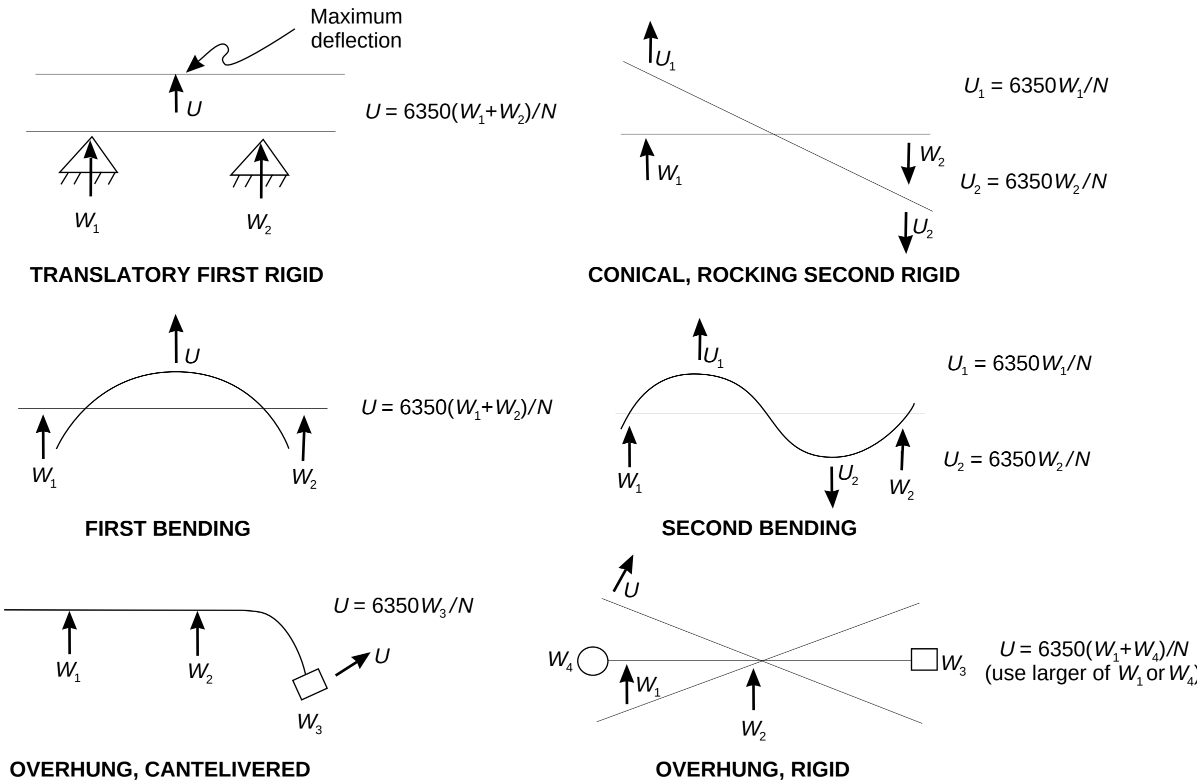


Figure 7A—Typical Mode Shapes—SI Units

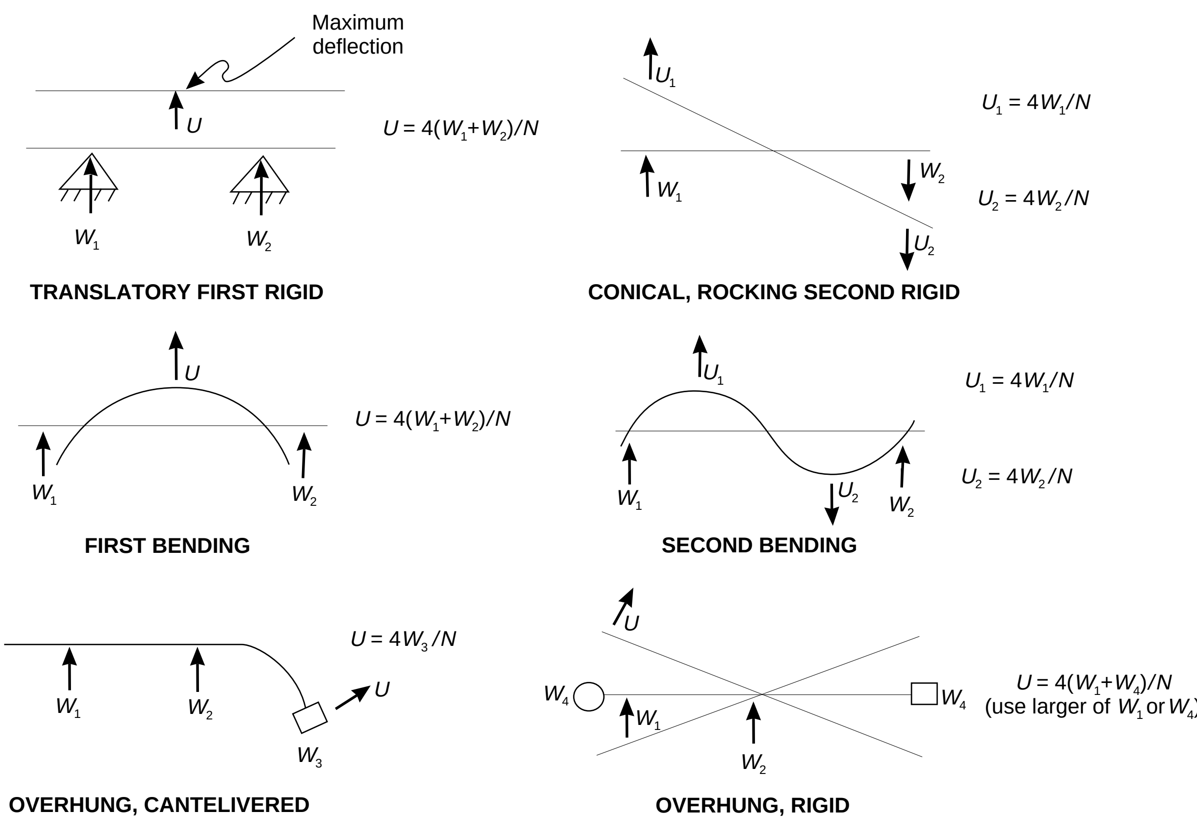


Figure 7B—Typical Mode Shapes—Customary Units machines where the support stiffness is less than 3.5 times the oil-film stiffness.

2.6.2.8 Additional analyses shall be made for use with the verification test specified in 2.6.3. The location of the unbalance shall be determined by the vendor. Any test stand parameters which influence the results of the analysis shall be included.

Note: For gears, there will only be one plane readily accessible for the placement of an unbalance, for example the coupling flange on each shaft. However, there is the possibility that more planes are available. When this occurs and there is the possibility of exciting other criticals, multiple runs may be required.

2.6.2.9 The calculated unbalanced peak to peak amplitudes (see 2.6.2.7, item b) shall be multiplied using the correction factor calculated from Equation 7.

$$CF = \frac{A_1}{A_{4x}} \tag{7}$$

The correlation factor shall have a value greater than 0.5.

where

CF = correction factor,

A_l = amplitude limit, calculated using Equation 8 in. microns (mils) peak to peak,

A_{4X} = peak to peak amplitude at the probe location per requirements of 2.6.2.7, item c in microns (mils peak to peak).

In SI units:

$$A_1 = 25\sqrt{\frac{12000}{N}} \tag{8}$$

In Customary units:

$$A_1 = \sqrt{\frac{12000}{N}}$$

where

N = operating speed nearest to the critical speed of concern, in rpm.

2.6.2.10 The calculated major-axis, peak-to-peak, unbalanced rotor response amplitudes, corrected in accordance with 2.6.2.9 at any speed from zero to trip speed shall not exceed 75% of the minimum design diametral running clearances throughout the gearbox.

Notes:
1. Running clearances may be different than the assembled clearances with the machine shutdown.
2. At this point the analysis is complete and acceptance is based strictly on the predicted values from the computer model. If a potential problem is discovered, the earlier it is addressed the more rational the solution. While the final outcome is not completely known until the unit is tested, it is reasonable to expect the vendor to have the expertise to properly model the unit and avoid any problems at test time.

2.6.2.11 If the analysis indicates that the separation margins still cannot be met or that a non-critically damped response peak falls within the operating speed range and the purchaser and vendor have agreed that all practical design efforts have been exhausted, then acceptable amplitudes shall be agreed upon by the purchaser and the vendor, subject to the requirements of 2.6.3.3

2.6.3 Unbalanced Rotor Response Verification Test

2.6.3.1 When an unbalanced rotor response analysis is required, an unbalance rotor response test shall be performed as part of the mechanical running test (see 4.3.2), 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 2.6.2.8 shall be the criterion for determining the validity of the damped unbalanced response analysis. To accomplish this, the requirements of 2.6.3.1.1 through 2.6.3.1.6 shall be followed:

2.6.3.1.1 During the mechanical running test (see 4.3.2), the amplitudes and phase angle of the shaft vibration from 0% – 120% of maximum continuous speed, shall be recorded. The gain of any analog recording instruments used shall be preset before the test so that the highest response peak is within 60% – 100% of the recorder's full scale on the test-unit coast-down (deceleration).

Note: This set of readings is normally taken during a coastdown, with convenient increments of speed such as 50 rpm. Since at this point the rotor is balanced, any vibration amplitude and phase detected should be the result of residual unbalance and mechanical and electrical runout.

2.6.3.1.2 The location of critical speeds below the trip speed, shall be established.

2.6.3.1.3 The unbalance which was used in the analysis performed in 2.6.2.8, shall be added to the rotor in the location used in the analysis. The unbalance shall not exceed 8 times the value from Equation 6.

2.6.3.1.4 The gear shall then be brought up to the operating speed nearest the critical and the indicated vibration amplitudes and phase shall be recorded using the same procedure used for 2.6.3.1.1.

2.6.3.1.5 The corresponding indicated vibration data taken in accordance with 2.6.3.1.1 shall be vectorially subtracted from the results of this test.

Note: It is practical to store the residual unbalance vibration measurements recorded in the step at 2.6.3.1.1 and by use of computer code perform the vectorial subtraction called for in this paragraph at each appropriate speed. This makes the comparison of the test results with the computer analysis of 2.6.2.8 quite practical. It is necessary for probe orientation be the same for the analysis and the machine for the vectorial subtraction to be valid.

2.6.3.1.6 The results of the mechanical run, including the unbalance response verification test, shall be compared with those from the analytical model specified at 2.6.2.8.

2.6.3.2 The vendor shall correct the model if it fails to meet either of the following criteria:

a. The actual critical speeds determined on test (within the range of test speeds) 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 (e.g., as a result of 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.

Note: It is possible that the vertical and horizontal stiffness are significantly different and the analysis will predict 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.

2.6.3.3 If the support stiffness is less than 2 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 (see 2.6.3.1.1) and in the unbalanced (see 2.6.3.1.4) condition before proceeding with the step specified in 2.6.3.1.5. In such a case, the measured response shall be compared with the predicted absolute shaft movement.

2.6.3.4 Unless otherwise specified, the verification test of the rotor unbalance shall be performed only on the first gear set tested, if multiple identical gear sets are purchased.

2.6.3.5 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 2.6.2.10.

2.6.4 Additional Testing

2.6.4.1 Additional testing is required (see 2.6.4.2) if, from the shop verification test data (see 2.6.3) or from the damped, corrected unbalanced response analysis (see 2.6.3.2), it appears that either of the following conditions exists.

Note: When the analysis or test data does not meet the requirements of the standard, additional more stringent testing is required. The purpose of this additional testing is to determine on the test stand that the machine will operate successfully.

a. Any critical response will fail to meet the separation margin requirements (see 2.6.2.4.1 and 2.6.2.4.2) or will fall within the operating speed range.

b. The clearance requirements of 2.6.2.10 have not been met.

2.6.4.2 Unbalance weights shall be placed as described in 2.6.2.6; 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 Equation 8 (see 2.6.2.9) at the maximum continuous speed; however, the unbalance used shall be no less than twice or greater than 8 times the unbalance limit specified in 2.6.2.6 Equation 6. The measurements from this test, taken in accordance with 2.6.3.1.1 and 2.6.3.1.2, 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.

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 2.6.2.9).

2.6.4.3 The internal deflection limits specified in 2.6.4.2 items a and b, shall be based on the calculated displacement ratios between the probe locations and the areas of concern identified in 2.6.2.10, 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 will 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.

2.6.5 Torsional Analysis

2.6.5.1 The vendor having unit responsibility shall ensure that a torsional vibration analysis of the complete coupled train is carried out. The gear vendor shall provide the party having train responsibility the gear mass-elastic data required for the torsional analysis. The vendor having train responsibility shall be responsible for directing any modifications necessary to meet the requirements of 2.6.5.3 through 2.6.5.7.

● **2.6.5.2** If specified, the gear vendor shall conduct the torsional vibration analysis of the complete coupled train.

2.6.5.3 Excitation of torsional natural frequencies may come from many sources which may or may not be a function of running speed and should be considered in the analysis. These sources shall include but are not limited to the following:

a. Gear characteristics such as unbalance, pitch line 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 reciprocating engines, and rotary-type positive displacement machines.
e. Control loop resonances from hydraulic, electronic governors, and variable frequency drives.
f. One and two times line frequency.
g. Running speed or speeds.
h. Harmonic frequencies from variable frequency drives.
i. Variation in electric power line frequency.

2.6.5.4 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 to maximum continuous speed).

2.6.5.5 Torsional natural frequencies at integral multiples of running speeds shall preferably be avoided or, in systems in which corresponding excitation frequencies occur, shall be shown to have no adverse effect.

2.6.5.6 When torsional resonances are calculated to fall within the margin specified in 2.6.5.4 (and the purchaser and the 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 agreed upon by the purchaser and the vendor.

2.6.5.7 In addition to the torsional analyses required in 2.6.5.3 through 2.6.5.6, the vendor shall perform a transient torsional vibration analysis for synchronous motor driven units, variable frequency motors, and turbine generators sets. The acceptance criteria for this analysis shall be agreed upon by the purchaser and the vendor.

Note: For more information on transient torsional analysis, consult API Std 617.

2.6.6 Vibration and Balancing

2.6.6.1 All gear wheel and pinion assemblies shall be multiplane dynamically balanced. On rotors with single keyways, the keyway shall be filled with a fully crowned half-key. The weight of all half-keys used during final balancing of the assembled element shall be recorded on the residual unbalance work sheet (see Appendix G).

2.6.6.2 A residual unbalance check shall be performed in the balance machine and recorded in accordance with the residual unbalance work sheet (see Appendix G).

2.6.6.3 The vendor shall supply to the purchaser the journal static loading, in kg (lb.).

2.6.6.4 The maximum allowable residual unbalance shall be calculated from the following equation:

In SI units:

$$U_{max} = 6350W/N \tag{9}$$

In U.S. customary units:

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

where

U_{max} = amount of residual rotor unbalance, in gram-millimeters (oz-in.),

W = journal static loading, in kilograms (pounds),

N = maximum continuous speed, in rpm.

2.6.6.5 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 mil), whichever is less:

In SI units:

$$A = 25.4\sqrt{\frac{12000}{N}} \tag{10}$$

In Customary units:

$$A = \sqrt{\frac{12000}{N}}$$

where

A = amplitude of unfiltered vibration, in μm (mil) true peak to peak,

N = maximum continuous speed, in rpm.

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.

Note: These limits are not to be confused with the limits specified in 2.6.3 for shop verification of unbalanced response.

2.6.6.6 Electrical and mechanical runout shall be determined by rotating the rotor through the full 360 degrees supported in V blocks at the journal centers while measuring runout with a non-contacting vibration probe and a dial indicator at the centerline of each probe location and one probe-tip diameter to either side.

Note: It should be pointed out that the rotor runout determined in V blocks generally will not be reproduced 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 based on the slow roll speed and rotor weight.

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

2.6.6.8 If the vendor can demonstrate that electrical or mechanical runout is present, the actual combined runout measured on the test up to a maximum of 25% of the test level calculated from Equation 10 or 6.5 μm (0.25 mil), whichever is less, may be vectorially subtracted from the vibration signal measured during the factory test.

2.6.6.9 Casing Vibration

The overall vibration levels during testing shall not exceed the values shown in Table 9.

Table 9—Casing Vibration Levels

	Velocity (rms)	Acceleration (true peak)
Frequency Range	10 Hz – 2.5 kHz	2.5 kHz – 10 kHz
Overall	2.9 mm/sec (0.11 in./sec)	4 g's; g = 9.81 m/s^2
Discrete Frequencies	1.8 mm/sec (0.07 in./sec)	(g = 32.16 ft/s^2)

Note: Discrete frequency shall be determined from FFT spectrum.

2.7 BEARINGS AND BEARING HOUSINGS

2.7.1 General

2.7.1.1 Radial and thrust bearings shall be of the hydrodynamic fluid film type.

2.7.1.2 Bearings shall be designed to prevent incorrect positioning.

Note: This is typically accomplished by the non-symmetric positioning of the anti-rotation pins.

2.7.1.3 Unless otherwise specified, thrust bearings and radial bearings shall be fitted with bearing-metal temperature sensors and installed as specified in API Std 670. There shall be a minimum of two elements per radial bearing and two per face on thrust bearings.

2.7.2 Radial Bearings

2.7.2.1 Hydrodynamic radial bearings shall be split for ease of assembly, precision bored, and of the sleeve or pad type, with steel-backed, babbitted replaceable liners, pads, or shells. These bearings shall be equipped with anti-rotation pins and shall be positively secured in the axial direction. When approved by the purchaser, copper-chrome may be used instead of steel for the backing.

2.7.2.2 The bearing design shall suppress hydrodynamic instabilities and provide sufficient damping over the entire range of allowable bearing clearances to limit rotor vibration to the maximum specified amplitudes (see 2.6.6) while the equipment is operating loaded or unloaded at specified operating speeds, including operation at any critical frequency.

2.7.2.3 The liners, pads, or shells shall be in axially split bearing housings and shall be replaceable without having to remove the coupling hub.

2.7.2.4 The bearing loading shall not exceed 34 bar (500 lb./in.^{2}) on a projected-area basis; however, for any specified operating condition, the load capacity of the bearing shall also be limited by a minimum film thickness of 25 μm (0.001 in.).

2.7.3 Thrust Bearings

2.7.3.1 Unless otherwise specified by the purchaser, thrust bearings shall be provided on the low-speed shaft for all double-helical gears and shall be provided on each shaft for all single-helical gears. When gears are supplied without thrust bearings, limited-end float or diaphragm-type couplings shall be used to maintain positive axial positioning of the connected rotors. (See Figure C-1, panels A-B, Appendix C, for typical system arrangements in which thrust bearings may be eliminated from double-helical gears.) All gear units without thrust bearings shall be supplied with locating collars on the low-speed shaft to prevent contact of the rotating elements with the gear casing. On gear units without a thrust bearing, axial float shall not be less than 13 mm (0.5 in.).

2.7.3.2 Thrust bearings shall be of the steel-backed, babbitted multiple-segment type, designed for equal thrust capacity in both directions and arranged for continuous pressurized lubrication to each side. Both sides shall be of the tilting-pad type, incorporating a self-leveling feature that ensures that each pad carries an equal share of the thrust load with minor variation in pad thickness. With the approval of the purchaser, tapered land thrust bearings may be used on the low speed shafts operating below 2000 rpm.

2.7.3.3 Each thrust pad shall be designed and manufactured with the dimensional precision (thickness variation) that will allow the interchange or replacement of individual pads.

2.7.3.4 Integral thrust collars are preferred for hydrodynamic thrust bearings. When integral collars are furnished, they shall be provided with at least 3 mm ($^1/_8$ in.) of additional stock to enable refinishing if the collar is damaged. When replaceable collars are furnished (for assembly and maintenance purposes), they shall be positively locked to the shaft to prevent fretting.

2.7.3.5 Both faces of thrust collars for hydrodynamic thrust bearings shall have a surface finish of 0.4 microns (16 micro-inches) Ra, or better and after mounting the axial total indicated runout of either face shall not exceed 13 µm (0.0005 in.)

2.7.3.6 Thrust bearings shall be sized for continuous operation under all specified conditions, including all external forces transmitted by the couplings. The external axial force transmitted by the coupling shall be considered as being numerically additive to any internal thrust forces.

2.7.3.7 Thrust bearings shall be selected such that under any operating condition the load does not exceed 50% of the bearing manufacturer's ultimate load rating. The ultimate load rating is the load that will produce the minimum acceptable oil-film thickness without inducing failure during continuous service or the load that will not exceed the creep-initiation or yield strength of the babbitt at the location of maximum temperature on the pad, whichever load is less. (See Appendix D for bearing selections based on loading, speed, and thrust shoe area.) In sizing thrust bearings, consideration shall be given 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 turbulence of the oil film.

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

2.7.3.8 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.

2.7.3.8.1 For couplings located so that the external force is produced by a single-ended drive motor with sleeve bearings, the external force shall be considered equal to the maximum magnetic centering force of the motor. If the maximum magnetic centering force is not specified, a force of 1.5 newtons per kilowatt (250 pounds per 1000 horsepower) motor rated power shall be used.

2.7.3.8.2 For gear-type couplings (located other than as described in 2.7.3.8.1, the external force shall be calculated from the following formula:

In SI units:

$$F = \frac{(0.25)(9{,}550)P_R}{(N_R D_P)}$$

In U.S. customary units:

$$F = \frac{(0.25)(63{,}000)P_R}{(N_R D_P)}$$

where

F = external force, in kilonewtons (pounds),
0.25 = the applied coefficient of gear tooth friction,
P_R = rated power, in kilowatts (horsepower),
N_R = rated speed, in revolutions per minute,
D_P = shaft diameter at the coupling, in millimeters (in.).

Note: Shaft diameter is an approximation of the coupling pitch radius.

2.7.3.8.3 Thrust forces from metallic flexible element couplings shall be calculated on the basis of the maximum allowable deflection permitted by the coupling manufacturer.

2.7.4 Bearing Housings

2.7.4.1 In this standard, the term "bearing housing" refers to all bearing enclosures, including the gear casing.

2.7.4.2 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.

2.7.4.3 Gaskets shall not be used on housing end covers where the gasket thickness would affect the end play or clearance of the thrust bearing.

2.7.4.4 Bearing housings shall be equipped with replaceable labyrinth-type end seals and deflectors where the shaft passes through the housing; lip-type seals shall not be used. The seals and deflectors shall be made of non-sparking materials. The design of the seals and deflectors shall effectively retain oil in the housing and prevent entry of foreign material into the housing.

2.7.4.5 Provision shall be made for mounting the following:

1 one event per revolution probe at input and output
2 axial probes at each thrust bearing
1 axial probe on any shaft without a thrust bearing
2 radial probes per radial bearing
2 accelerometers—1 input and 1 output on the coupling ends

2.7.4.6 Unless otherwise specified, the following shall be provided:

1 one event per revolution probe at input and output (Total = 2)
2 axial probes at each thrust bearing (Total = 2)
2 radial probes per radial bearing, coupling ends only (Total = 4)
2 accelerometers – 1 per coupling end (Total = 2)
The probe installation shall be as specified in API Std 670.

Note: The number and position of axial probes should consider the type of gear (double or single helical) and thrust bearing location.

2.8 LUBRICATION

2.8.1 The gear unit shall be pressure lubricated. Spray nozzles for the teeth shall be provided.

2.8.2 The gear unit and lubrication system shall be designed to limit the temperature of the casing drain oil to 77ºC (170ºF), with an inlet oil temperature of 49ºC (120ºF) to ensure high mechanical efficiency and dimensional stability and to limit the maximum temperature of the tooth metal. Where oil inlet temperatures exceed 49ºC (120ºF), special consideration shall be given to bearing design, oil flows, and allowable temperature rise. Oil outlets from thrust bearings shall be tangential in the control ring or in the thrust bearing cartridge if oil control rings are not used.

● **2.8.3** The oil system shall be furnished by either the purchaser or the vendor, as specified.

2.8.4 Unless otherwise specified, bearings, bearing housings and gears shall be arranged for oil lubrication using a mineral oil in accordance with ISO 3448. If synthetic oil is specified, all components of the gearbox shall be compatible with the type of synthetic oil specified.

Note: Synthetic lubricants may have advantages over mineral oils, particularly in certain classes of machinery operating at high temperatures and/or high pressures. A bearing designed for synthetics will not easily run on mineral oil lubricants due to cooling and space considerations. An owner would need a relatively sophisticated inventory control system to prevent inadvertent mixing of mineral oils and synthetics, which are chemically incompatible. Synthetic lubricants may also be incompatible with certain paints and coatings and they may be difficult to dispose of.

2.8.5 Unless otherwise specified, pressurized oil systems shall conform to the requirements of Special-Purpose Oil Systems as defined in Chapters 1 and 2 of API Std 614.

2.8.6 Where oil is supplied from a common system to two or more components of a machinery train (such as a compressor, a gear, and a motor), the vendor having unit responsibility shall ensure compatibility of type, grade, pressure and temperature of oil for all equipment served by the common system.

Note: The usual lubricant employed in a common oil system is a mineral oil that corresponds to ISO 3448 Grade 32. Compatibility of lube oil requirements needs to be agreed among the purchaser and all vendors supplying equipment served by the common system. In some cases, there can be significant differences in individual component needs. For example, a refrigeration compressor may need low pour point oil, a gear may need high viscosity and a turbine may need a conventional mineral oil. In such cases, it may be necessary to change the design of a component or to provide separate oil systems.

2.9 MATERIALS

2.9.1 General

2.9.1.1 Except as required or prohibited by this standard or by the purchaser, materials of construction shall be selected by the manufacturer for the operating and site environmental conditions specified.

2.9.1.2 The materials of construction of all major components shall be clearly stated in the vendor's proposal. Materials shall be identified by reference to applicable standards, including the material grade (see Appendix E). When no such designation is available, the vendor's material specification, giving physical properties, chemical composition, and test requirements shall be included in the proposal.

2.9.1.3 Low-carbon steels can be notch sensitive and susceptible to brittle fracture at ambient or lower temperatures. Therefore, only fully killed, normalized steels made to fine-grain practice are acceptable. The use of steel made to coarse austenitic grain size practice (such as ASTM A515) is prohibited.

2.9.1.4 The vendor shall specify the optional tests and inspection procedures that may be necessary to ensure that materials are satisfactory for the service. Such tests and inspections shall be listed in the proposal.

Note: The purchaser may specify additional optional tests and inspections, especially for materials used for critical components or in critical services.

2.9.1.5 Materials used in gear wheel and pinion teeth shall be forged or hot rolled alloy steel of high quality selected to meet the criteria for tooth pitting index and strength outlined in 2.2. In selecting the material, the vendor shall consider whether the gear wheel and pinion are to be through hardened, case hardened, or nitrided. The material quality of gear teeth will conform to ISO 6336-5 material quality Grade ME for case hardened or nitrided steels and quality Grade MX for through hardened steels. The material and manufacturing method will be approved by the purchaser.

2.9.1.6 Casings shall be sound and free from porosity, hot tears, shrink holes, blow holes, cracks, scale, blisters, and similar defects. Surfaces of castings shall be cleaned by sandblasting, shotblasting, chemical cleaning, or other standard methods. Mold-parting fins and the remains of gates and risers shall be chipped, filed or ground flush.

2.9.1.7 Fully enclosed cored voids, which become fully enclosed by methods such as plugging, welding, or assembly, are prohibited.

2.9.2 Welding of Rotating Elements

2.9.2.1 Welding of piping, pressure-containing parts, rotating parts and other highly stressed parts, weld repairs and any dissimilar-metal welds shall be performed and inspected by operators and procedures qualified in accordance with internationally recognized standards such as Section VIII, Division 1, and Section IX of the ASME Code.

2.9.2.2 All welds shall be continuous full-penetration welds. All welds shall be double welded, except when only one side is accessible; in such instances a backup ring, a consumable insert, or an inert gas shield with an internal gas purge backup shall be used.

2.9.2.3 The vendor shall be responsible for the review of all repairs and repair welds to ensure that they are properly heat treated and nondestructively examined for soundness and compliance with the applicable qualified procedures. Repair welds shall be nondestructively tested by the same method used to detect the original flaw, however, the minimum level of inspection after the repair shall be by the magnetic particle method in accordance with 4.2.2.4 for magnetic material and by the liquid penetrant method in accordance with 4.2.2.5 for nonmagnetic material. Unless otherwise specified, procedures for major repairs shall be subject to review by the purchaser before any repair is made.

2.9.2.4 Repair welding in the area of the gear teeth is prohibited.

2.9.3 Heat Treatment

2.9.3.1 After through hardened gear materials have been rough machined to the approximate final contour of the blank and heat treated, the tooth area shall be checked for proper hardness. After surface hardened gear materials have been completely heat treated, the surface hardness and case depth shall be checked on a representative coupon of suitable size that has accompanied the part during all heat treat processes.

2.9.3.2 Casings, whether of cast or fabricated construction, shall be stress relieved before final machining and after any welding, including repairs.

2.10 NAMEPLATES AND ROTATION ARROWS

Nameplates and rotation arrows shall be of Series 300 stainless steel or of nickel-copper alloy (Monel or its equivalent) attached by pins of similar material and located for easy visibility. As a minimum, the following data shall be clearly stamped on the nameplate:

a. The vendor's name.
b. The size and model of the gear unit.
c. The gear ratio.
d. The serial number.
e. The gear rated power.
f. The rated input speed in revolutions per minute.
g. The rated output speed in revolutions per minute.
h. The owner's item number.
i. The number of gear wheel teeth.
j. First lateral critical speed of pinion.
k. First lateral critical speed of gear wheel.

3 Accessories

● 3.1 GENERAL

The purchaser shall specify the accessories to be supplied by the vendor.

3.2 COUPLINGS AND GUARDS

3.2.1 Unless otherwise specified, flexible couplings and guards between drivers, gear units and driven equipment shall be supplied by the manufacturer of the driven equipment.

Note: Appendix C provides a guide to the selection of coupling types and the location of thrust bearings in equipment trains that employ gears.

3.2.2 When an integral-flanged shaft end is not furnished, the gear unit coupling hubs shall be mounted by the gear manufacturer unless otherwise specified (see 2.5.4.2).

3.2.3 Couplings and guards shall conform to API Std 671. The make, type, and mounting arrangement of the coupling shall be agreed upon by the purchaser and the vendors of the drivers, gear units, and driven equipment.

3.2.4 Information on shafts, keyway dimensions (if any), and shaft end movements due to end play and thermal effects shall be furnished by the vendor to the purchaser.

3.2.5 The coupling-to-shaft juncture shall be designed and manufactured to be capable of transmitting power at least equal to the power rating of the coupling.

3.2.6 The purchaser of the coupling shall supply a moment simulator, as required for the mechanical running tests (see 4.3.2.1.6).

3.3 MOUNTING PLATES

3.3.1 General

● **3.3.1.1** If specified, the gear unit shall be furnished with soleplates or a baseplate.

3.3.1.2 In 3.3.1.2.1 through 3.3.1.2.10, the term “mounting plate” refers to both baseplates and soleplates.

3.3.1.2.1 The upper and lower surfaces of mounting plates and any separate pedestals mounted thereon shall be machined parallel.

3.3.1.2.2 The mounting plate or plates shall be furnished with horizontal (axial and lateral) jackscrews, the same size or larger than the vertical jackscrews. The lugs holding these jackscrews shall be attached to the mounting plates in such a manner that they do not interfere with the installation of the equipment, jackscrews or shims. Precautions shall be taken to prevent vertical jackscrews in the equipment feet from marring the shimming surfaces. Alternative methods of lifting equipment for the removal or insertion of shims or for moving equipment horizontally, such as provision for the use of hydraulic jacks, may be proposed. Such arrangements should be proposed for equipment that is too heavy to be lifted or moved horizontally using jackscrews. Jack screws shall be plated for rust resistance.

3.3.1.2.3 Machinery supports shall be designed to limit the relative displacement of the shaft end caused by the worst combination of pressure, torque and allowable piping stress, to 50 µm (0.002 in.).

3.3.1.2.4 Unless otherwise specified, epoxy grout shall be used for machines mounted on concrete foundations. The vendor shall blast-clean in accordance with ISO 8501 Grade $S_a 2$ (SSPC SP6), all grout contact surfaces of the mounting plates and coat those surfaces with inorganic zinc silicate in preparation for epoxy grout.

Notes:
1. Epoxy primers have a limited life after application. The grout manufacturer should be consulted to ensure proper field preparation of the mounting plate for satisfactory bonding of the grout.
2. Inorganic zinc silicate is compatible with epoxy grout, does not exhibit limited life after application as do most epoxy primers, and is environmentally acceptable.

3.3.1.2.5 Anchor bolts shall not be used to fasten machinery to the mounting plates. There shall be a 1/4-in. diametrical clearance between the anchor bolts and the anchor bolt holes in the mounting plate.

3.3.1.2.6 Mounting plates shall conform to the following:

a. Mounting plates shall not be drilled for equipment to be mounted by others.
b. Mounting plates shall be supplied with leveling screws.
c. Outside corners of mounting plates which are in contact with the grout shall have 50 mm (2 in.) minimum radiused outside corners (in the plan view).
d. All machinery mounting surfaces shall be treated with a rust preventive immediately after machining.
e. Mounting plates shall extend at least 25 mm (1 in.) beyond the outer three sides of equipment feet.
f. Mounting plates shall be machined to a finish of 6 µm (0.00025 in.) arithmetic average roughness (Ra) or better.

Note: Item c: Radiused corners are recommended to prevent the potential of cracking the grout.

Note: Item e: This requirement allows handling of shims and mounting level or laser type instruments to check alignment.

3.3.1.2.7 The vendor of the mounting plates shall furnish a solid stainless steel shim ground on both sides with a thickness of at least 5 mm (1/4 in.) between the equipment feet and the mounting plates. Each shim shall straddle the hold-down bolts and be at least 5 mm (1/4 in.) larger on all sides than the equipment feet.

3.3.1.2.8 Unless otherwise specified, anchor bolts will be furnished by the purchaser.

3.3.1.2.9 When the mounting plate is supplied by the vendor, the vendor shall also supply the hold-down bolts for attaching the gear unit to the mounting plate.

3.3.1.2.10 Equipment shall be designed for installation in accordance with API RP 686.

3.3.2 Baseplate

3.3.2.1 When a baseplate is specified, the purchaser shall indicate the major equipment to be mounted on it. A baseplate shall be a single fabricated steel unit, unless the purchaser and the vendor agree that it may be fabricated in multiple sections. Multiple-section baseplates shall have machined and doweled mating surfaces which shall be bolted together to ensure accurate field reassembly.

Note: A baseplate with a nominal length of more than 12 m (40 ft) or a nominal width of more than 4 m (12 ft) may have to be fabricated in multiple sections because of shipping restrictions.

3.3.2.2 Unless otherwise specified, the baseplate shall extend under the drive-train components so that any leakage from these components is contained within the baseplate.

Note: This is a housekeeping consideration for personnel safety and to prevent damage resulting from shipping.

● **3.3.2.3** If specified, the baseplate shall be designed to facilitate the use of optical, laser-based or other instruments for accurate leveling in the field. The details of such facilities shall be agreed by the purchaser and vendor. Where the requirement is satisfied by the provisions of leveling pads and/or targets, they shall be accessible with the baseplate on the foundation and the equipment mounted. Removable protective covers shall be provided. For column-mounted baseplates (see 3.3.2.4), leveling pads or targets shall be located close to the support points. For noncolumn-mounted baseplates, a pad or target should be located at each corner. When required for long units, additional pads shall be located at intermediate points.

● **3.3.2.4** If specified, the baseplate shall be suitable for column mounting (that is, of sufficient rigidity to be supported at specified points) without continuous grouting under structural members. The baseplate design shall be agreed upon by the purchaser and the vendor.

3.3.2.5 The baseplate shall be provided with lifting lugs for a four-point lift. Lifting the baseplate complete with all equipment mounted shall not permanently distort or otherwise damage the baseplate or the equipment mounted on it.

3.3.2.6 The bottom of the baseplate between structural members shall be open. When the baseplate is installed on a concrete foundation, it shall be provided with at least one grout hole having a clear area of at least 0.01 m^2 (19 $in.^2$) and no dimension less than 75 mm (3 in.) in each bulkhead section. These holes shall be located to permit grouting under all load-carrying structural members. Where practical, the holes shall be accessible for grouting with the equipment installed. The holes shall have 13 mm ($^1/_2$ in.) raised-lip edges, and if located in an area where liquids could impinge on the exposed grout metallic covers with a minimum thickness of 16 gauge shall be provided. Vent holes at least 13 mm ($^1/_2$ in.) in size shall be provided at the highest point in each bulkhead section of the baseplate.

3.3.2.7 The underside mounting surfaces of the baseplate shall be in one plane to permit use of a single-level foundation. When multi-section baseplates are provided, the mounting pads shall be in one plane after the baseplate sections are doweled and bolted together.

3.3.2.8 Unless otherwise specified, nonskid decking covering all walk and work areas shall be provided on the top of the baseplate.

3.3.2.9 All baseplate mounting surfaces:

a. Shall be machined after the baseplate is fabricated.
b. In the same horizontal plane shall be within 25 µm (0.001 in.) to prevent a soft foot.
c. Shall have each mounting surface machined within a flatness of 42 µm per linear meter (0.0005 in./linear ft) of mounting surface.
d. With different mounting planes shall be parallel to each other within 50 µm (0.002 in.).

The above tolerances shall be recorded and verified by inspection in unrestrained condition on a flat machined surface at the place of manufacture.

Note: The surfaces being discussed are those on which the equipment is mounted and on the bottom of the baseplate.

● **3.3.2.10** If specified, sub-sole plates shall be provided by the vendor.

3.3.3 Soleplates and Subsoleplates

3.3.3.1 When soleplates are specified, they shall meet the requirements of 3.3.3.1.1 and 3.3.3.1.2 in addition to those of 3.3.1.

Note: Refer to Appendix L, Typical Mounting Plates.

3.3.3.1.1 Adequate working clearance shall be provided at the bolting locations to allow the use of standard socket or box wrenches and to allow the equipment to be moved using the horizontal and vertical jackscrews.

3.3.3.1.2 Soleplates shall be steel plates that are thick enough to transmit the expected loads from the equipment feet to the foundation, but in no case shall the plates be less than 40 mm ($1^1/_2$ in.) thick.

3.3.3.2 When subsoleplates are specified, they shall be steel plates at least 25 mm (1 in.) thick. The finish of the subsoleplates' mating surfaces shall match that of the soleplates.

3.4 CONTROLS AND INSTRUMENTATION

3.4.1 General

3.4.1.1 Instrumentation and installation shall conform to any detailed specifications in the purchasers inquiry or order or both. When no detailed specifications are furnished, instrumentation and installation shall conform to the requirements of API Stds 614 and 670.

3.4.1.2 Unless otherwise specified, controls and instrumentation shall be designed for outdoor installation and shall meet the requirements of IP65 as detailed in IEC 60034-5 (NEMA 4 as detailed in Publication 250).

3.4.1.3 Instrumentation and controls shall be designed and manufactured for use in the area classification (class, group, and division or zone) specified.

3.4.1.4 All conduit, armored cable and supports shall be designed and installed so that it can be easily removed without damage and shall be located so that it does not hamper removal of bearings, seals, or equipment internals.

3.4.2 Vibration, Position, and Temperature Detectors

3.4.2.1 Radial shaft vibration probes, accelerometers, and axial position probes shall be installed and calibrated in accordance with API Std 670.

Note: For the required number of probes, see 2.7.4.5 and 2.7.4.6.

3.4.2.2 Accelerometers shall be located horizontally, on the input and output bearing housings below the split line unless otherwise specified.

3.4.2.3 Radial shaft vibration probes shall be located in the *x-y* positions on the input and output shaft adjacent to the bearing on the coupling side. Provisions for *x-y* probes, including shaft runout of the target areas per paragraph 2.5.4.4, shall be provided on all radial bearings unless otherwise specified.

3.4.2.4 Axial position probes shall be located at the blind end of the input and output shafts unless otherwise specified.

Note: Gear shafts without thrust bearings may have axial floats that require axial position probes with extended linear range.

● **3.4.2.5** If specified, radial shaft vibration and axial position monitors shall be supplied and calibrated in accordance with API Std 670.

● **3.4.2.6** If specified, casing vibration monitors shall be supplied, installed, and calibrated in accordance with API Std 670.

● **3.4.2.7** The type of temperature detectors shall be specified. The temperature detectors shall be installed and calibrated in accordance with API Std 670.

3.5 PIPING AND APPURTENANCES

Unless otherwise specified, lube-oil piping and appurtenances shall conform to the requirements of API Std 614.

3.6 SPECIAL TOOLS

3.6.1 When special tools and fixtures are required to disassemble, assemble, or maintain the unit, they shall be included in the quotation and furnished as part of the initial supply of the machine. For multiple-unit installations, the requirements for quantities of special tools and fixtures shall be mutually agreed upon by the purchaser and the vendor. These or similar special tools shall be used during shop assembly and post-test disassembly of the equipment.

● **3.6.2** If specified, hanging fixtures shall be provided for vertical storage of the rotors. See 2.5.4.8 for shaft provisions for these fixtures.

3.6.3 When special tools are provided, they shall be packaged in a separate, rugged metal box and shall be marked "special tools for (provide tag/item number)." Each tool shall be stamped or tagged to indicate its intended use.

4 Inspection, Testing, and Preparation for Shipment

4.1 GENERAL

● **4.1.1** The purchaser shall specify the extent of participation in the inspection and testing.

4.1.1.1 When shop inspection and testing have been specified, the purchaser and the vendor shall coordinate manufacturing hold points and inspectors' visits.

4.1.1.2 Witnessed mechanical running or optional tests require confirmation of the successful completion of a preliminary test.

● **4.1.1.3** If specified, the purchaser's representative, the vendor's representative, or both shall indicate compliance in accordance with the inspector's checklist (Appendix I) by initialing, dating, and submitting the completed checklist to the purchaser prior to shipment.

4.1.1.4 After advance notification to the vendor, the purchaser's representative shall have entry to all vendor and subvendor plants where manufacturing, testing, or inspection of the equipment is in progress.

4.1.2 The vendor shall notify subvendors of the purchaser's inspection and testing requirements.

● **4.1.3** The purchaser shall specify the amount of advanced notification required for a witness or observed inspection or test.

4.1.4 Equipment, materials, and utilities for the specified inspection and tests shall be provided by the vendor.

4.1.5 The purchaser's representative shall have access to the vendor's quality program for review.

4.2 INSPECTION

4.2.1 General

4.2.1.1 The vendor shall keep the following data available for at least 20 years from the date of shipment of the equipment. For parts that are repaired, the data shall be kept for 20 years from the date of shipment of the repaired part. This data shall be available for examination or reproduction by the purchaser or his representative upon request:

a. Necessary certification of materials, such as mill test reports.

b. Purchase specifications for all items on bills of materials.

c. Test data to verify that the requirements of the specification are being met.

d. Results of documented tests and inspections, including fully identified records of all heat treatment and radiography.

● e. If specified, final-assembly maintenance and running clearances.

● **4.2.1.2** The purchaser may specify the following:

a. Parts to be subjected to surface and subsurface examination.

b. The type of inspection required, such as magnetic particle, dye penetrant, radiographic, and ultrasonic examination.

4.2.2 Material Inspection

● 4.2.2.1 General

If radiographic, ultrasonic, magnetic particle, or liquid penetrant inspection of welds or materials is required or specified, the criteria in 4.2.2.2 through 4.2.2.5 shall apply, unless other criteria are specified by the purchaser. Cast iron may be inspected in accordance with 4.2.2.4 and 4.2.2.5. Welds, cast steel, and wrought material may be inspected in accordance with 4.2.2.2 through 4.2.2.5.

4.2.2.2 Radiography

4.2.2.2.1 Radiography shall be in accordance with ASTM E94.

4.2.2.2.2 The acceptance standard used for welding fabrications shall be Section VIII, Division 1, UW 51 (for 100% radiography) UW-52 (for spot radiography), of the ASME Code. The acceptance standard used for castings shall be VIII, Division 1, Appendix 7, of the ASME Code.

4.2.2.3 Ultrasonic Inspection

4.2.2.3.1 Ultrasonic inspection shall be in accordance with Section V, Articles 5 and 23, of the ASME Code.

4.2.2.3.2 The acceptance standard used for welded fabrications shall be section VIII, Division 1, Appendix 12, of the ASME Code. The acceptance standard used for castings shall be Section VIII, Division 1, Appendix 7, of the ASME Code.

4.2.2.4 Magnetic Particle Inspection

4.2.2.4.1 Both wet and dry methods of magnetic particle inspection shall be in accordance with ASTM E709.

4.2.2.4.2 The acceptance standard used for welded fabrications shall be Section VIII, Division 1, Appendices 6 and Section V, Article 25, of the ASME Code. The acceptability of defects in castings shall be based on a comparison with the photographs in ASTM E125. For each type of defect, the degree of severity shall not exceed the limits specified in Table 10.

Table 10—Maximum Severity of Defects in Castings

Type	Defect	Severity Level
I	Linear Discontinuities	1
II	Shrinkage	2
III	Inclusions	2
IV	Chills and Chaplets	1
V	Porosity	1
VI	Welds	1

4.2.2.5 Liquid Penetrant Inspection

4.2.2.5.1 Liquid penetrant inspection shall be in accordance with Section V, Article 6, of the ASME Code.

4.2.2.5.2 The acceptance standard used for welded fabrications shall be Section VIII, Division 1, Appendix 8 and Section V Article 24, of the ASME Code. The acceptance standard used for castings shall be Section VIII, Division 1, Appendix 7, of the ASME Code.

4.2.2.6 Rotating Elements

● **4.2.2.6.1** Certified information on the major rotating elements shall be kept available for 20 years from the date of shipment and shall be obtainable by the purchaser upon request. All records of work, whether performed in the normal course of manufacture or as part of a repair procedure, shall be fully identified. This information shall include the following:

a. Chemical and physical data from specimens made and heat treated with the parts.

b. Records of all heat treatment and resulting hardnesses.

c. Radiographs and records of ultrasonic inspections, if radiography and ultrasonic inspection are performed.

d. For surface-hardened gears, hardnesses and case depths determined on coupons treated with the work piece.

e. For through-hardened gears, hardnesses determined in at least three dispersed locations in the tooth blank area.

4.2.2.6.2 All welds in rotating elements, including those attaching gears to shafts, shall receive 100% inspection. Accessible surfaces of welds shall be inspected after back chipping or gouging and again after stress relieving. Magnetic particle or ultrasonic inspection is preferred. Other methods, such as dye penetrant and radiography, are acceptable only as agreed upon by the purchaser and the vendor.

4.2.2.6.3 All gear wheel and pinion teeth shall be subjected to 100% magnetic particle inspection in accordance with ASTM A275. Cracks are not acceptable. Linear indications that result from nonmetallic inclusions in the tooth flanks or roots that are larger than 1.5 mm (0.06 in.) shall be reported to the purchaser for disposition. Linear indications are defined as those having a length equal to or greater than three times the width. Acceptance or rejection shall be decided on a case-by-case basis and shall be agreed upon by the purchaser and the vendor.

4.2.2.7 Forgings and Hot-rolled Steel Shafts

4.2.2.7.1 Forgings and hot-rolled steel shafts shall be free from cracks, seams, laps, shrinkage, and other similar defects.

4.2.2.7.2 After rough machining, forgings and bar stock for major rotating elements shall be subjected to 100 % ultrasonic inspection in accordance with ASTM A388. Acceptance criteria shall be agreed upon by the purchaser and the vendor.

● **4.2.2.8 Additional Inspection**

Additional gear tooth inspections may be required when agreed upon by the purchaser and the vendor. These may include (but are not limited to) inspections of tooth spacing, pitch line runout, profile, and lead.

4.2.3 Mechanical Inspection

4.2.3.1 During assembly of the system and prior to testing, each component (including cast-in passages of these components) and all piping and appurtenances shall be cleaned by pickling or by another appropriate method. They shall be inspected to ensure that they have been cleaned and are free of foreign materials, corrosion products, and mill scale.

● **4.2.3.2** All oil systems shall meet the cleanliness requirements of API Std 614. When specified, the purchaser may inspect for cleanliness the equipment and all piping and appurtenances furnished by or through the vendor before piping is finally assembled.

● **4.2.3.3** If specified, the hardness of parts, welds, and heat-affected zones shall be verified as being within the allowable values by testing of the parts, welds, or zones. The method, extent, documentation, and witnessing of the testing shall be agreed upon by the purchaser and the vendor.

4.3 TESTING

4.3.1 General

4.3.1.1 Gear units shall be mechanically tested in accordance with 4.3.2. Other tests that may be specified by the purchaser are described in 4.3.3.

4.3.1.2 At least 6 weeks before the first scheduled running test, the vendor shall submit to the purchaser, for his review and comment, detailed procedures for all running tests, including acceptance criteria for all monitored parameters.

4.3.1.3 The vendor shall notify the purchaser not less than 5 working days prior to the date the equipment will be ready for testing. If the testing is rescheduled, the vendor shall notify the purchaser not less than 5 working days prior to the new test date.

4.3.2 Mechanical Running Tests

4.3.2.1 Before the mechanical test is performed, the requirements of 4.3.2.1.1 through 4.3.2.1.9 shall be met.

4.3.2.1.1 The contract bearings shall be used in the machine for the mechanical running tests.

4.3.2.1.2 All oil pressures, viscosities, and temperatures shall be at the operating values recommended in the manufacturer's operating instructions for the specific unit being tested. The overall oil flow rate to the gear unit shall be recorded.

4.3.2.1.3 Test-stand oil filtration shall be 10 microns or better. Oil system components downstream of the filters shall meet the cleanliness requirements of API Std 614 before any test is started.

4.3.2.1.4 The joints and connections in the casing and the oil system shall be checked for tightness, and any leaks shall be corrected.

4.3.2.1.5 All warning, protective, and control devices used during the test shall be checked, and adjustment shall be made as required.

4.3.2.1.6 The mechanical running tests shall be conducted with the contract half couplings mounted and the moment simulators attached, resulting in a moment equal to one-half of the contract coupling spacer. If the coupling halves are not available, or are not compatible with the vendor's shop equipment, the moment simulator must simulate the moment equal to the contract half coupling plus one-half of the coupling spacer. When all testing is completed, if requested by the purchaser, the moment simulators shall be returned to the purchaser (see 3.2.6).

4.3.2.1.7 All purchased vibration probes, cables, oscillator demodulators, accelerometers and temperature sensors shall be in use during the test. When vibration transducers are not furnished by the vendor or if the purchased units are not compatible with shop readout facilities, then shop transducers and readouts that meet the accuracy requirements of API Std 670 may be used.

4.3.2.1.8 Shop test facilities shall include instrumentation with the capability of continuously monitoring and plotting revolutions per minute, peak-to-peak displacement, and phase angle (*x-y-y'*). Presentation of vibration displacement and phase marker shall also be by oscilloscope.

4.3.2.1.9 The vendor shall clean and coat a minimum of 5 teeth across the full face of each helix at four locations equally spaced around the gear wheel and two locations equally spaced around the pinion with hard layout laquer.

4.3.2.2 During the mechanical running tests, the requirements of 4.3.2.2.1 through 4.3.2.2.12 shall be met.

4.3.2.2.1 The vendor shall keep a detailed log of the final tests, making entries every 15 minutes for the first hour and every 30 minutes for the duration of the tests. Each entry shall include the following information:

a. Inlet oil temperature and pressure.
b. Oil flow.
c. Outlet oil (drain) temperature.
d. Shaft vibration, frequency, and amplitude, both filtered (synchronous) and unfiltered (raw).
e. Bearing temperatures.
f. Casing vibration (refer to Table 9).

4.3.2.2.2 The vibration characteristics determined by the use of the instrumentation specified in 4.3.2.1.7 and 4.3.2.1.8 shall serve as the basis for acceptance or rejection of the machine (see 2.6.6).

● **4.3.2.2.3** If seismic test values are specified, vibration data (minimum and maximum values) shall be recorded and located (clock angle) in a radial plane transverse to each bearing centerline (if possible) using shop instrumentation during the tests.

4.3.2.2.4 The gear unit shall be operated at maximum continuous speed until the bearing temperatures and the lube-oil temperatures have stabilized.

4.3.2.2.5 The gear unit shall be operated at 110 % of maximum continuous speed for a minimum of 15 minutes.

4.3.2.2.6 To verify the lateral critical speeds, the speed shall be momentarily increased to 120% of maximum continuous speed, and phase amplitude plots shall be made as the unit coasts down to 10% of maximum continuous speed (see 2.6.2).

Note: The bearing temperatures may rise measurably during this test.

4.3.2.2.7 The gear unit shall be operated at maximum continuous speed for 4 hours.

Note: The sequence of steps listed in 4.3.2.2.5 through 4.3.2.2.7 is optional.

4.3.2.2.8 The mechanical operation of all equipment being tested and the operation of the test instrumentation shall be satisfactory. The measured unfiltered vibration shall not exceed the limits of 2.6.6.5 and shall be recorded throughout the operating speed range.

4.3.2.2.9 For shaft vibration, measure with non-contacting transducers while the gear unit is operating at maximum continuous speed, a sweep shall be made for vibration amplitudes at frequencies other than synchronous. As a minimum, the sweep shall cover a frequency range from 0.25 times the synchronous speed of the gear wheel to 4 times the synchronous speed of the pinion, but not less than 1500 hz. If the amplitude of any discrete, nonsynchronous vibration exceeds 20% of the pinion shaft (or 40% for shafts running below 2000 rpm) of the allowable vibration as defined in 2.6.6.5, the purchaser and the vendor shall mutually agree on requirements for any additional testing and on the gear unit's suitability for shipment.

4.3.2.2.10 During loaded tests unfiltered peak-to-peak shaft axial vibration for double helical gears shall not exceed 50 µm (2.0 mils) (see 2.5.2.3). During no-load or light load test, double-helical pinions may exhibit random axial movement greater than 50 µm (2.0 mils).

4.3.2.2.11 For casing vibration, measured with casing mounted acceleration transducers, while the gear unit is operating at maximum continuous speed, a sweep shall be made at a frequency range from 10 Hz up to, and including 10 kHz. The values of velocity and acceleration shall not exceed the allowable levels shown on Table 9.

● **4.3.2.2.12** Plots showing synchronous vibration amplitude and phase angle versus speed for deceleration shall be made before and after the 4-hour run. Plots shall be made of both the filtered (one per revolution) and the unfiltered vibration levels, if specified, these data shall also be furnished in polar form. The speed range covered by these plots shall be 400 rpm to the specified driver trip speed.

● **4.3.2.2.13** If specified, lube-oil inlet pressures and temperatures shall be varied through the range permitted in the operating manual. This shall be done during the 4-hour test.

● **4.3.2.2.14** If specified, tape recordings shall be made of all real-time vibration data.

● **4.3.2.2.15** If specified, the tape recordings of real-time vibration data shall be provided to the purchaser.

● **4.3.2.2.16** Testing at any additional speeds, the duration of testing at each speed, and the data to be recorded shall be specified by the purchaser at the time of purchase.

4.3.2.3 After the mechanical running test is completed, the requirements of 4.3.2.3.1 through 4.3.2.3.3 shall be met.

4.3.2.3.1 The top half of the casing shall be removed and the tooth mesh shall be inspected for proper contact and for surface damage resulting from the test.

4.3.2.3.2 Hydrodynamic bearings shall be removed, inspected, and reassembled after the mechanical running tests are completed.

4.3.2.3.3 If replacement or modification of bearings or seals, or dismantling of the case to replace or modify other parts is required to correct mechanical or performance deficiencies, the initial tests shall not be acceptable, and the final shop tests shall be run after these replacements or corrections are made.

4.3.2.4 When spare gear elements are ordered to permit concurrent manufacture, each spare element shall also be given mechanical running tests in the gear unit in accordance with the requirements of this standard.

● 4.3.3 Optional Tests

If specified, the shop tests described in 4.3.3.1 through 4.3.3.5 shall be performed.

4.3.3.1 Full-speed/Full- or Part-load Test

The gear unit shall be tested to transmit its partial or full rated power, as agreed upon by the purchaser and the vendor, at its rated input speed. The load shall be applied by a mechanical or a hydraulic method (dynamometers, prony brakes, and so forth) until the bearing and lube-oil temperatures have stabilized. Details of the test, including vibration limits, shall be negotiated before the order. Where shop dynamometers have a sufficient torque rating but an insufficient speed rating, shop stepdown gear units shall be used in lieu of testing at a reduced speed.

4.3.3.1.1 When partial- or full-load tests are specified, the vendor's test stand lubrication system shall include an automatic low lube-oil shut-down system.

4.3.3.2 Full-torque/Reduced Speed Test

When a full speed full load test cannot be performed, a full torque reduced speed test may be specified to demonstrate a loaded tooth contact pattern. For gears where there is a significant amount of thermal and/or centrifugal distortion at operating speed, a full torque reduced speed test may not produce a test contact pattern representative of field contact operating pattern. Prior to the test, the vendor shall provide the purchaser with a drawing showing the acceptable test contact pattern.

Vibration and temperature limitations, as outlined elsewhere in this standard, should not be applied.

The unit's full torque shall be calculated at the gear rated power and the rated input or output speed. The full torque shall then be applied to its respective shaft by mechanical or hydraulic means at a speed convenient for the vendor's test-stand equipment.

The duration of the test shall be negotiated before the time of order.

The vendor shall clean and coat a minimum of five teeth across the full face of each helix at four locations equally spaced around the gear wheel and two locations equally spaced around the pinion with hard layout laquer.

When this test is a requirement, the vendor must assure the purchaser that the reduced speed limits are within the oil's capability to maintain an adequate film, thereby avoiding risk of damaging the tooth surface and/or bearings.

4.3.3.3 Full-torque/Static Test

One shaft of the gear unit shall be locked. The full torque, as calculated in 4.3.3.2, shall then be applied to the other shaft by mechanical or hydraulic means. This procedure shall be repeated at several mesh points of gear set. The number of load applications shall be negotiated prior to the time of order.

Note: For gears where a significant portion of the lead modification is due to thermal distortion, a full torque static test may not produce a test contact pattern representative of field contact operating pattern.

4.3.3.4 Back-to-back Locked-torque Test

Test of a minimum of two contract gearboxes are required for the execution of this test, one gear unit to serve as a slave unit to load the test gear unit. The gear units shall be coupled together, input shaft to input shaft and output shaft to output shaft. The full torque, as calculated in 4.3.3.2, shall then be introduced by removing the spacer and turning one shaft against its mate while the other shafts remain coupled. This torque shall be maintained in the system by mechanical or hydraulic means (for example, coupling halves keyed on shafts, torque introduced, and then coupling halves bolted together). The units shall then be run at their rated input and output speeds. The duration of the run shall be negotiated before the time of order.

Note: For purposes of this test the slave unit shall require a lead and profile modification suitable for loading in the testing mode. For modified leads, this shall require final modification of the gears suitable for the contract application. The vendor and purchaser shall agree on the extent of this work. At the conclusion of the back-to-back tests, the slave unit will require a test of its own since the back-to-back configuration cannot be duplicated for that purpose. The vendor and purchaser shall agree on the test to be performed.

The slave unit is loaded on the flanks that are normally not loaded, the bearing loads are in the opposite direction, and the stub shafts used to complete the torque path may have to be removed. Therefore, the slave unit and often also the tested unit will have to be modified after the test (see Figure 8).

4.3.3.5 Sound-level Test

The sound-level test shall be performed in accordance with ANSI/AGMA 6025-D98 or other agreed standard.

4.4 PREPARATION FOR SHIPMENT

4.4.1 Equipment shall be suitably prepared for the type of shipment specified, including blocking of rotor when necessary (such as when the gear unit does not contain thrust bearings). The preparation shall make the equipment suitable for 6 months of outdoor storage from the time of shipment, with no disassembly required before operation. If storage for a

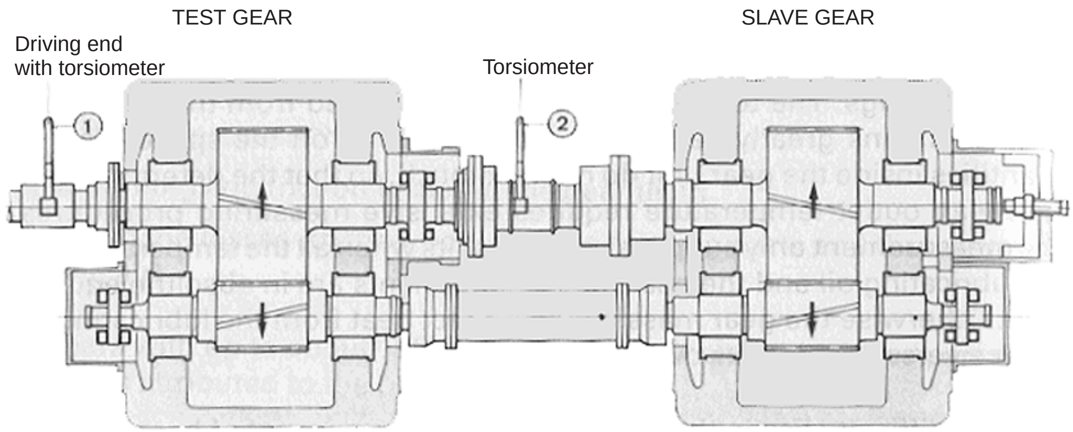


Figure 8—Back-to-back Locked-torque Test (Typical)

longer period is contemplated, the purchaser should consult with the vendor regarding the recommended procedures to be followed.

4.4.2 The vendor shall provide the purchaser with the instructions necessary to preserve the integrity of the storage preparation after the equipment arrives at the job site and before start-up, as described in Chapter 3 of API RP 686 *Recommended Practices for Machinery Installation and Installation Design*.

4.4.3 The equipment shall be prepared for shipment after all testing and inspection has been completed and the equipment has been approved by the purchaser. The preparation shall include that specified in 4.4.3.1 through 4.4.3.11.

4.4.3.1 Except for machined surfaces, all exterior surfaces that may corrode during shipment, storage or in service, shall be given at least one coat of the manufacturer's standard paint. The paint shall not contain lead or chromates.

Note: Austenitic stainless steels are typically not painted.

4.4.3.2 Exterior machined surfaces except for corrosion-resistant material shall be coated with a rust preventative.

4.4.3.3 The interior of the gear unit shall be clean; free from scale, welding spatter, and foreign objects; and sprayed or flushed with a rust preventive that can be removed with solvent. The rust preventive shall be applied through all openings while the gear unit is slow-rolled.

4.4.3.4 Internal steel areas of bearing housings and carbon steel oil systems' auxiliary equipment (such as reservoirs, vessels, and piping) shall be coated with a oil-soluble rust preventive.

4.4.3.5 Flanged openings shall be provided with metal closures at least 5 mm ($^3/_{16}$ in.) thick with elastomeric gaskets and at least four full-diameter bolts. For studded openings, all nuts needed for the intended service shall be used to secure closures. Each opening shall be secured with a tamperpoof seal (car seal) so that the protective cover cannot be removed without the seal being broken.

4.4.3.6 Threaded openings shall be provided with steel caps or round-head steel plugs in accordance with ANSI B16.11. The caps or plugs shall be of material equal to or better than that of the pressure casing. In no case shall non-metallic (such as plastic) caps or plugs be used.

4.4.3.7 Lifting points and the center of gravity shall be clearly identified on the equipment package. The vendor shall provide the recommended lifting arrangement.

4.4.3.8 The equipment shall be identified with item and serial numbers. Material shipped separately shall be identified with securely affixed, corrosion resistant metal tags indicating the item and serial number of the equipment for which the material is intended. In addition, crated equipment shall be shipped with duplicate packing lists, one inside and one on the outside of the shipping container.

On the outside of the container shall be stated the date of preservation, type of preservation, and date for reapplication of preservative. A warning shall also be placed on the outside that opening of the box to inspect the contents may require reapplication of preservative.

● **4.4.3.9** If spare gear elements are purchased, the rotors shall be suitably prepared for unheated indoor storage for a period of at least 3 years. The rotors shall be treated with a rust preventative and shall be housed in a vapor barrier envelope with a slow-release vapor-phase inhibitor. The rotors shall be crated for domestic or export shipment, as specified. A purchaser-approved resilient material 3 mm ($^1/_8$-in. thick) (not tetrafluorotethylene [TFE] or polytetrafluorotethylene [PTFE]) shall be used between the rotors and the cradle at the support areas. The rotors shall not be supported at journals (unless mounted in gear unit), probe areas, seal areas, coupling fit areas, or gear teeth. The probe target areas shall be marked “Probe Area, Do Not Cut” (see 2.5.4.8 and 3.6.2 for vertical storage provisions).

Note: TFE and PTFE are not recommended as cradle support liners since they cold flow and impregnate into the surface.

4.4.3.10 Exposed shafts and shaft couplings shall be wrapped with waterproof, moldable waxed cloth or vapor-phase-inhibitor paper. The seams shall be sealed with oil-proof adhesive tape.

4.4.3.11 The vendor shall clean and coat a minimum of five teeth across the full face of each helix at four locations equally spaced around the gear wheel and two locations equally spaced around the pinion with hard layout laquer.

Note: This will allow later evaluation of field tooth contact pattern.

● **4.4.4** If specified, metal “Top Hat” covers shall be bolted to the gear unit, totally enclosing and sealing the shaft ends to allow for purging of the gear unit with minimal leakage.

● **4.4.5** If specified, a steel rotor container(s) shall be supplied. Steel rotor shipping containers shall have provisions for storage of a rotor in both a horizontal position and a vertical position. The placement of the runners should allow for a fork truck to be able to move the container and the rotor while placed in the horizontal position. Additionally, lugs should be provided to allow a crane to lift the container while the rotor is in the container. The container, valves, and connections shall be designed for a minimum of 0.35 bar (5 psig) pressure. The purge gas should be specified. Containers shall be cylindrical, horizontally split and the top is to be sealed, bolted and doweled to prevent movement and leakage. The pressure in the container is to be maintained during storage at a minimum of 0.07 bar (1 psig).

4.4.6 One copy of the manufacturer’s installation instructions shall be packed and shipped with the equipment.

5 Vendor’s Data

5.1 GENERAL

5.1.1 The information to be furnished by the vendor is specified in 5.2 and 5.3.

5.1.2 The data shall be identified on transmittal (cover) letters, title pages and in title blocks or other prominent position on drawings, with the following information:

a. The purchaser’s/owner’s corporate name.
b. The job/project number.
c. The equipment item number and service name.
d. The inquiry or purchase order number.
e. Any other identification specified in the inquiry or purchase order.
f. The vendor’s identifying proposal number, shop order number, serial number, or other reference required to completely identify return correspondence.

5.1.3 A coordination meeting shall be held, preferably at the vendor’s plant, within 4 – 6 weeks after order commitment. Unless otherwise specified, the vendor shall prepare and distribute an agenda prior to this meeting, which, as a minimum, shall include a review of the following items:

a. The purchase order, scope of supply, unit responsibility, sub-vendor items and lines of communications.
b. The data sheets.
c. Applicable specifications and previously agreed exceptions.
d. Schedules for the transmittal of data, production and testing.
e. The quality assurance program and procedures.
f. Inspection, expediting, and testing.
g. Schematics and bills of materials for auxiliary systems.
h. The physical orientation of the equipment, shaft rotation, piping and auxiliary systems, including access for operation and maintenance.
i. Coupling selection and rating.
j. Thrust and radial bearing sizing, estimated loadings and specific configurations.
k. Rotor dynamics analysis (lateral, torsional and transient torsional, as required).
l. Audit of sub-suppliers.
m. Equipment performance, alternate operating conditions, startup, shutdown and any operating limitations.
n. Instrumentation and controls.
o. Inspection, related acceptance criteria, and testing.
p. Expediting.
q. Other technical items.

5.2 PROPOSALS

5.2.1 General

The vendor shall forward the original proposal, with the specified number of copies, to the addressee specified in the inquiry documents. The proposal shall include, as a minimum, the data specified in 5.2.2 through 5.2.4 and a specific statement that the gear unit and all its components and auxiliaries are in strict accordance with this standard. If the gear unit or any of its components or auxiliaries are not in strict accordance, the vendor shall include a list that details and explains each deviation. The vendor shall provide sufficient detail to enable the purchaser to evaluate any proposed alternative designs. All correspondence shall be clearly identified in accordance with 5.1.2.

5.2.2 Drawings

5.2.2.1 The drawings indicated on the Vendor Drawing and Data Requirements (VDDR) form (see Appendix F) shall be included in the proposal. As a minimum, the following shall be included:

a. A general arrangement or outline drawing for each gear unit showing overall dimensions, maintenance clearance dimensions, overall weights, erection weights, and the maintenance weight for the heaviest single component. The direction of rotation, and the size and location of major purchaser connections shall also be indicated.
b. Cross-section drawings showing the details of the proposed equipment.
c. Sketches that show methods of lifting the assembled gear unit. (This information may be included on the drawings specified in item a, above.)

5.2.2.2 If typical drawings, schematics, and bills of materials are used, they shall be marked up to show the correct weight and dimension data and to reflect the actual equipment and scope proposed.

5.2.3 Technical Data

The following data shall be included in the proposal:

a. The purchaser's data sheets, with completed vendor's information entered thereon and with literature to fully describe details of the offering.
b. The predicted sound data (see 2.1.6).
c. The VDDR form (see Appendix F), indicating the schedule according to which the vendor agrees to transmit all the data specified as part of the contract.
d. A schedule for shipment of the equipment, in weeks after receipt of the order.
e. A list of major wearing components, showing any interchangeability with the owner's other gear units.
f. A list of recommended start-up and normal maintenance spares, including any items that the vendor's experience indicates are likely to be required.
g. A list of the special tools furnished for maintenance. The vendor shall identify any metric items included in the offering.
h. A description of any special weather protection and winterization required for start-up, operation, and periods of idleness, under the site conditions specified on the data sheets. This description shall clearly indicated the protection to be furnished by the purchaser, as well as that included in the vendor's scope of supply.
i. A complete tabulation of the utility requirements, including the type, quantity and supply pressure of lube oil required, the heat load to be removed by the oil, and the nameplate power rating. Approximate data shall be clearly indicated as such.
j. A description of any optional or additional tests and inspection procedures for materials, as required by 2.9.1.4.
k. A description of any special requirements whether specified in the purchaser's inquiry or as outlined in 2.9.1.2.
l. A list of similar machines installed and operating under conditions analogous to those specified in the proposal.
m. Any start-up, shutdown, or operating restrictions required to protect the integrity of the equipment.
n. Vendor and purchaser shall discuss vendor's experience with proposed machine components.

5.2.4 Optional Tests

The vendor shall furnish an outline of the procedures to be used for each of the special or optional tests that have been specified by the purchaser or proposed by the vendor.

5.3 CONTRACT DATA

5.3.1 General

5.3.1.1 Contract data shall be furnished by the vendor in accordance with the agreed VDDR form (see Appendix F).

5.3.1.2 Each drawing shall have a title block in the lower right-hand corner with the date of certification, identification data specified in 5.1.2, revision number and date and title. Similar information shall be provided on all other documents including sub-vendor items.

5.3.1.3 The purchaser will promptly review the vendor's data upon receipt; however, this review shall not constitute permission to deviate from any requirements in the order unless specifically agreed upon in writing. After the data have been reviewed and accepted, the vendor shall furnish certified copies in the quantities specified.

5.3.1.4 A complete list of vendor data shall be included with the first issue of major drawings. This list shall contain titles, drawing numbers, and a schedule for transmittal of each item listed. This list shall cross-reference data with respect to the VDDR form in Appendix F.

5.3.2 Drawings and Technical Data

The drawings and data furnished by the vendor shall contain sufficient information so that together with the manuals specified in 5.3.5, the purchaser can properly install, operate, and maintain the equipment covered by the purchase order. All contract drawings and data shall be clearly legible (8-point minimum font size even if reduced from a larger size drawing), shall cover the scope of the agreed VDDR form, and shall satisfy the applicable detailed descriptions in Appendix F.

● 5.3.3 Progress Reports

The vendor shall submit progress reports to the purchaser at intervals specified.

Note: Refer to the description of item 28 in Appendix F for content of these reports.

5.3.4 Parts Lists and Recommended Spares

5.3.4.1 Parts Lists and Recommended Spares

The vendor shall submit complete parts lists for all equipment and accessories supplied. These lists shall include part names, manufacturers' unique part numbers, materials of construction (identified by applicable international standards). Each part shall be completely identified and shown on appropriate cross-sectional, assembly-type cutaway or exploded-view isometric drawings. Interchangeable parts shall be identified as such. Parts that have been modified from standard dimensions or finish to satisfy specific performance requirements shall be uniquely identified by part number. Standard purchased items shall be identified by the original manufacture's name and part number.

5.3.4.2 The vendor shall indicate on each of these complete parts lists all those parts that are recommended as start-up or maintenance spares, and the recommended stocking quantities of each. These should include spare parts recommendations of sub-suppliers that were not available for inclusion in the vendor's original proposal.

5.3.5 Installation, Operation, Maintenance and Data Manuals

5.3.5.1 The vendor shall provide sufficient written instructions and all necessary drawings to enable the purchaser to install, operate, and maintain all of the equipment covered by the purchase order. This information shall be compiled in a manual or manuals with a cover sheet showing the information listed in 5.1.2, an index sheet, and a complete list of the enclosed drawings by title and drawing number. The manual or manuals shall be prepared specifically for the equipment covered by the purchase order. "Typical" manuals are unacceptable.

5.3.5.2 Installation Manual

All information required for the proper installation of the gear unit and its auxiliaries shall be compiled in a manual that must be issued no later than the time of issue of final certified drawings. For this reason, it may be separate from the operating and maintenance instructions. This manual shall contain information on alignment and grouting procedures, normal and maximum utility requirements, centers of mass, rigging provisions and procedures, and all other installation data. All drawings and data specified in 5.2.2 and 5.2.3 that are pertinent to proper installation shall be included as a part of this manual (see also description of line item 25 in Appendix F).

5.3.5.3 Operating and Maintenance Manual

A manual containing all required operating and maintenance instructions shall be supplied not later than 2 weeks after all specified tests have been successfully completed. In addition to covering operation at all specified process conditions, this manual shall also contain separate sections covering operation under any specified extreme environmental conditions (see also description of line item 25 in Appendix F).

● 5.3.5.4 Technical Data Manual

When specified, the vendor shall provide the purchaser with a technical data manual within 30 days of completion of shop testing (see description of line item 26 in Appendix F for minimum requirements of this manual). Vendor may combine the technical manual with the operating and maintenance manual.

APPENDIX A—SPECIAL PURPOSE GEAR UNITS DATA SHEETS

Page 1 of 4

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
SI UNITS

Job No. ________ Item No. ________
P.O. No. ________ Date ________
Requisition No. ________
Inquiry No. ________
Revision ________ Date ________ By ________

Applicable To: ○ Proposal ○ Purchase ○ As Built ○ US Standards (1.7) ○ ISO Standards (1.7)

For	________	Manufacturer	________
Site	________	Model No.	________
Unit	________	Serial No.	________
Service	________	Driver Type	________
No. Required	________	Driven Equipment	________

NOTE: Numbers Within ( ) Refer To Applicable API Standard 613 Paragraphs

○ Information To Be Completed By Purchaser ☐ Information To Be Completed By Manufacturer

○ UNITS OF MEASUREMENT (1.8)

○ US Customary Units
● SI Units

○ RATING REQUIREMENTS

Driven Equip. kw (2.1.4): Norm. ________ Max ________
Driver kw: Rated ________ Max ________
Normal Transmitted Power (2.2.2) ________ kw
Gear Unit Rated Power (2.2.1) ________ kw
Torque @ Max Cont Speed ________ kg m
Max Torque (2.2.1) ________ kg m @ ________ RPM
○ Reducer ○ Increaser
Rated Speed, RPM (2.1.5) :
Input ________ ○ Specified ○ Nominal
Output ________ ○ Specified ○ Nominal
Allow Var In Gear Ratio (2.1.5) (+)(-) ________ %
Max Continuous Speed (1.5) ________ RPM
Trip Speed (1.5)(2.1.7) ________ RPM
Gear Service Factor (2.2.3.1) ________ (Min)
Hardness (2.2.3.2) Pinion ________ Gear ________
Shaft Assembly Designation (2.1.19) ________
HS Shaft Rot Fac'g Cpl'g (2.1.20.2) ○ CW ○ CCW
LS Shaft Rot Fac'g Cpl'g (2.1.20.2) ○ CW ○ CCW
External Loads (2.1.16) ________
Other Operating Conditions (2.2.2) (2.6.1.3) ________

○ INSTALLATION DATA (2.1.14)

○ Indoor ○ Heated ○ Under Roof
○ Outdoor ○ Unheated ○ Partial Sides
○ Grade ○ Mezzanine ○ ________
○ Winterization Req'd ○ Tropicalization Required
Elec. Area (2.1.10) Class ____ Grp ____ Div ____
Elec'l Area (2.1.10) Class ____ Zone ____ Temp ____
Max Allow SPL (2.1.6) ________ dBA @ ________ m
Elevation ________ Ft Barometer ________ kPa abs
Range Of Ambient Temperatures :

	Dry Bulb	Wet Bulb
Normal	________ °C	________ °C
Maximum	________ °C	________ °C
Minimum	________ °C	________ °C

Unusual Conditions ○ Dust ○ Fumes
○ ________

☐ BASIC GEAR DATA

☐ SIngle Stage ☐ Single Helical
☐ Double Stage ☐ Double Helical
☐ Epicyclic ☐ ________

Mechanical Rating (1.5.16) kw ________ @ ________ RPM
Gear Service Factor (2.2.3.1) ________ (Actual)
Full Load Gear Unit Power Loss ________ kw
Gear Unit Mechanical Efficiency ________ %
Rating Speed, RPM Pinion ________ Gear ________
Hardness used for
Rating, (HB or Rc) Pinion ________ Gear ________
Tooth Pitting Index, "K" mpa (2.2.4.1)(2.2.4.2) :
Allowable ________ Actual ________
Material Index Number, mpa (Fig 3, Table 4) ________
Bending stress number, 'St" mpa (2.2.5.1)(2.2.5.2)
Pinion: Allowable ________ Actual ________
Gear: Allowable ________ Actual ________
Pitch Line Velocity ________ m/sec
Anticipated SPL (2.1.6) ________ dBA @ ________ m
WR^2 Referred To LS Shaft ________ kg mm^2
Breakaway Torque ________ kg m
Pinion teeth Hardness Range ________
Pinion teeth hardening method: ________
☐ HS Shaft Separate Hardness Range ________
Gear Teeth (rim) Hardness Range ________
Gear teeth hardening method: ________
Gear Hub:
☐ Forged Cylinder ☐ Forged & Coped ☐ Fabricated
Gear To Shaft fit method (2.5.3.2)
☐ Integral ☐ Keyed inteference ☐ Keyless Inteference
Rim Attachment (2.5.3.2) ________
LS Shaft Hardness Range ________
Journal static Weight loads (2.6.6.3)
Pinion ________ kg Gear ________ kg
Total Gear Unit Assembled Weight ________ kg
NOTES: ________

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
SI UNITS

Job No. ________ Item No. ________
Date ________ By ________
Revision No. ________ By ________

☐ GEAR DATA

	Pinion	Gear
Number Of Teeth	____	____
Gear Ratio	____	
Tangential Load, "Wt", N (2.2.4.2)	____	
AGMA Geometry Factor "J":	____	____
Pitch Diameter, mm	____	____
Outside Diameter, mm	____	____
Root Diameter, mm	____	____
Center Groove Diameter, mm	____	____
Normal Pressure Angle, Degrees	____	
Normal Diametral Pitch	____	
Helix Angle, Degrees	____	
Center Distance	____	
Backlash, mm	____	
Net Face Width, "Fw", mm	____	
Pinion L/D	____	
Face Overlap Ratio	____	
Transverse Contact Ratio	____	
AGMA 6011 Service Factor	____	
Ratings based on ANSI/AGMA 6011 with SF=1.0		
Durability Power	____	
Strength Power	____	
Tooth Surface Finish µm RA	____	____
Tooth Generation Process	____	____
Tooth Finishing Process	____	____
Lead Modification (2.2.4.6)	☐ Not Req'd	☐ Req'd
Calculated Total Lead Mismatch µm	____	

☐ SCUFFING DATA (2.2.6)

Scuffing Data per ANSI/AGMA 925-A03

Scuffing Risk (2.2.6) ________ Calculation Method ________

Composite surface roughnexss σ_x	____	µm
Specific film thickness, EHL λ_{min}	____	µm
Tooth temperature, θ_M	____	°C
Maximum contact temperature, $\theta_{B\,max}$	____	°C

☐ MATERIALS

Gear Casing ________ Oil Seals ________
Pinion (s) ________
Gear Rim(s) ________
HS Shaft ________ LS Shaft ________
Radial Bearings ________ Backing ________
Thrust Bearing(s) ________ Backing ________

○ SHAFT END DETAIL

○ Gear Shaft End for Coupling Integral Flange (2.5.4.2)
○ Pinion Shaft End for Coupling Integral Flange (2.5.4.2)
○ Other: ________

○ COUPLINGS AND GUARDS

	Pinion Shaft	Gear Shaft
Coupling Furnished By (3.2.1)		
Mount Coupling Halves (3.2.2)		
Cplg. Guard Adapter By (3.2.3)		
Cplg. Guard Furnished By (3.2.3)		
Coupling vendor (3.2.3)		
Vendor's model number (3.2.3)		
Coupling weight on shaft, kg		
CG Inboard/outboard of shaft end		
Hub Drill Template Provided	○	○

☐ SHAFT END DETAIL (2.5.4.2)

Shaft end detail specified by: ○ Purchaser ☐ Gear Vendor
(Integral Unless Otherwise Specified)

Shaft end detail if 'Otherwise' specified:	Pinion	Gear
Tapered/Keyless	○☐	○☐
Tapered / 1-Key	○☐	○☐
Tapered / 2-Keys	○☐	○☐
Cylindrical / 1-Key	○☐	○☐
Cylindrical / 2-Keys	○☐	○☐
Other ________	○☐	○☐
Shaft Diameter, mm	____	____

(If integral flange use diameter imediately adjacent to flange)

☐ RADIAL BEARINGS

	Pinion	Gear
Type	____	____
Diameter, mm	____	____
Length, mm	____	____
Journal Velocity, m/sec	____	____
Loading, kPa	____	____
Clearance (min-max), mm	____	____
Span, mm	____	____
Power loss each brg., kw	____	____
Oil flow each brg., m^3/hr	____	____

☐ THRUST BEARING(S)

Location	____	____
Manufacturer	____	____
Type	____	____
Size	____	____
Area, mm^2	____	____
Loading, kPa	____	____
Rating, Kpa	____	____
Int.Thrust Load, N (+)(-)	____	____
Ext. Thrust Load, N (+)(-)	____	____
Power loss each, kw	____	____
Oil flow each, m^3/hr	____	____

NOTES: ________

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
SI UNITS

Job No. ______ Item No. ______
Date ______ By ______
Revision No. ______ By ______

○ LUBRICATION REQUIREMENTS

○ Oil System Furnished By (2.8.3):
○ Other ______ ○ Gear Vendor
○ Oil Visc.: ______ cP @ 40C ______ cP @ 100C (2.8.6)
○ ISO Grade ______ Load Stage (2.2.6) ______

☐ MESH

Mesh and windage power loss ______ kw
Oil Flow, Mesh ______ m^3/hr

☐ LUBRICATION REQUIREMENTS

Min. Startup Oil Temperature ______ °C
Normal Oil Inlet temperature ______ °C
Maximum Oil Inlet Temperature ______ °C
Unit Oil Flow (Total) ______ m3/h
Unit Oil Pressure ______ kPa
☐ Oil Visc.: ______ cP @ 40C ______ cP @ 100C (2.8.6)
☐ ISO Grade ______ Load Stage (2.2.6) ______

☐ PIPING CONNECTIONS

Service	No.	Size	Type
Lube Oil Inlet			
Lube Oil Outlet			
Casing Drain*			
Vent			
Casing Purge			

*Casing drain is 'dead bottom' housing drain, not lube oil outlet

○ MOUNTING PLATES

○ Gear Furnished w/ (3.3.1.1)
○ Baseplate ○ Soleplate ○ Subplate(s) (3.3.2.10)
○ Mounting Plate(s) Furnished By (3.3.1.1) ______
○ Baseplate Leveling (3.3.2.3) ______
○ Baseplate With Leveling Pads (3.3.2.3)
○ Baseplate Suitable For Column Mounting (3.3.2.4)
○ Vendor Review of Purchaser's Foundation Dwgs. (2.1.13)
○ Grout Type (3.3.1.2.4): ______

○ CONTRACT DATA

○ Test Data Prior To Shipment ______
○ Progress Reports (5.3.3) ______
○ Vendor Signoff of Inspector Checklist (4.1.1.3)
● Information Retained for 20 years (4.2.1.1) (4.2.2.6.1)
○ Technical Manual (5.3.5.4)
○ **PAINTING** (4.4.3.1) ○ ______
○ Painting Housing Interior not Allowed (2.3.1.14)

○ SHIPMENT

○ Steel Rotor Storage Container (4.4.5) ○ Shaft Covers (4.4.4)

	Contract Unit	Spares
Export Boxing	○	○
Domestic Boxing	○	○
Outdoor Storage Over 6 Mos.	○	3 yr Indoor ○ (4.4.3.9)

○ ADDITIONAL REQUIREMENTS

○ VIBRATION DETECTORS

RADIAL (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ______ ○ Total No. ______
○ X-Y Probes Pinion bearings ○ Coupling End ○ Blind End
○ X-Y Probes Gear bearings ○ Coupling End ○ Blind End
○ Other ______

AXIAL (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ______ ○ Total No. ______
○ Dual Probes at each thrust bearing
○ Single Probe any shaft without thrust bearing
○ Other ______

ONE EVENT PER REVOLUTION PROBE (2.7.4.5) (2.7.4.6)
○ Manufacturer ______ Total No. ______
○ One on Input Shaft ○ One on Output Shaft
○ Other ______

ACCELEROMETER (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ______ ○ No. Required ______
○ Pinion Coupling End ○ Gear Coupling End
○ Other ______

○ TEMPERATURE DETECTORS

○ Dial Type Thermometers (3.4.2.7) ______
○ Type Brg. Temp. Sensors (3.4.2.7) (2.7.1.3) ______
○ RTD ○ Thermocouple / ○ Simplex ○ Duplex
Calibration ______
○ HS/LS Bearings no. sensors each ______ / ______
○ Thrust - number of sensing elements each face ______

○ OTHER VIBRATION AND TEMPERATURE

(3.4.2.5)(3.4.2.6)	Other	Gear Vendor
Oscillator Demodulators Supplied By	○	○
Vibration Monitor Supplied By	○	○
○ Vibration Shutdown Delay Time ______ Seconds		
Temperature Monitor Supplied By	○	○
Oscillator Demodulator J-box By	○	○
Temp. Sensor Termination J-box By	○	○

J-box Type: ______ Mount: ______

○ MISCELLANEOUS

● Undamped Critical Analysis Report (2.6.2.1) **:**
○ w/ Dampd Rotr Respns Analys Rprt (2.6.2.4.3) (2.6.2.6)
Torsional Analysis By (2.6.5.2) ○ Gear Vendor ○ Other
○ Spare Set Of Gear Rotors (4.3.2.4)
○ Gear Case Furnished With Inlet Purge Connection (2.4.3)
○ Orientation Of Oil Inlet & Drain Conns. (2.4.6) ______
○ Filter Breather Location (2.3.1.12) ______
○ Torsional Device Provisions (2.5.4.5) ______
○ Rotor Vertical Storage Provisions (2.5.4.8) ______
○ Rotor Vertical Storage Fixture(s) (3.6.2) ______
○ Vendor Service Rep. on Site (2.1.13)

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
SI UNITS

Job No. ________ Item No. ________
Date ________ By ________
Revision No. ________ By ________

No.	○ INSPECTIONS AND TESTS (4.1)				
2	○ Advance Notice of Witness Testing Required (4.1.3)				
3	Number Calendar Days ________				
4			Wit-	Ob-	Test
5		Req'd	ness	serve	Log
6	Shop Inspection (4.1.1)	○	○	○	
7	Cleanliness Inspection (4.2.3.1)	○	○	○	
8	Hardness Verification (4.2.3.3)	○	○	○	○
9	Dismantle-Reassembly (4.3.2.3.1)	●	○	○	○
10	Contact Check (2.5.2.2)	●	○	○	○
11	Contact Check Tape Lift (2.5.2.2)	○	○	○	○
12	Gear Accuracy Check (2.5.2.1)	●	○	○	○
13	Double Helical Axial Stability (2.5.2.3)	●	○	○	○
14	Special testing integral forged gears (2.5.3.3)				
15	(testing per mutual agreement)	○	○	○	○
16	Residual Unbalance Chk. (2.6.6.2)	●	○	○	●
17	Mechanical Run Test (Main) (4.3.2)	●	○	○	●
18	Mech. Run Test (Spare) (4.3.2.4)	○	○	○	○
19	Add'l. Mechanical Tests (4.3.2.2.16)	○	○	○	○
20	Part Or Full Load And Full Speed				
21	Test (4.3.3.1)	○	○	○	○
22	Full Torque, Reduced Speed (4.3.3.2)	○	○	○	○
23	Full Torque Static Test (4.3.3.3)	○	○	○	○
24	Back-To-Back Locked Torque				
25	Test (4.3.3.4)	○	○	○	○
26	Sound Level Test (4.3.3.5)	●	○	○	○
27	Additional Gear Tooth Test (4.2.2.8)	○	○	○	○
28	Use Shop Lube System	○	○	○	
29	Use Job Lube System	○	○	○	
30	Use Shop Vibration Probes, Etc.	○	○	○	
31	Use Job Vibration Probes, Etc.	○	○	○	
32	Final Assembly, Maintenance &				
33	Running Clearance (4.2.1.1.e)	○	○	○	○
34	Oil System Cleanliness (4.2.3.2)	○	○	○	
35	Oil System-Casing Joint				
36	Tightness (4.3.2.1.4)	●	○	○	
37	Warning And Protection				
38	Devices (4.3.2.1.5)	●	○	○	
39	Seismic Vibration Data (4.3.2.2.3)	○	○	○	○
40	Vibration, Phase Plots (4.3.2.2.12)	●	○	○	○
41	Oil Inlet Range Test (4.3.2.2.13)	○	○	○	○
42	Tape Recorded Vibration Data	○	○	○	○
43	(4.3.2.2.14)(4.3.2.2.15)	○	Purchaser copy		

No.	**NON-DESTRUCTIVE TESTING (4.2.1.2) (4.2.2.1)**				
45		Surface	Sub-Surface		Log
46	○ Casing	○	○	________	○
47	○ Rot. Elemts.	○	○	________	○
48	○ Bearings	○	○	________	○
49	○ Other :	○	○	________	○
50	(Specify) ________				

NOTES:

Page 1 of 4

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
US CUSTOMARY UNITS

Job No. ______ Item No. ______
P.O. No. ______ Date ______
Requisition No. ______
Inquiry No. ______
Revision ______ Date ______ By ______

Applicable To: ○ Proposal ○ Purchase ○ As Built ○ US Standards (1.7) ○ ISO Standards (1.7)

For	______	Manufacturer	______
Site	______	Model No.	______
Unit	______	Serial No.	______
Service	______	Driver Type	______
No. Required	______	Driven Equipment	______

NOTE: Numbers Within ( ) Refer To Applicable API Standard 613 Paragraphs

○ Information To Be Completed By Purchaser ☐ Information To Be Completed By Manufacturer

○ UNITS OF MEASUREMENT (1.8)

● US Customary Units
○ SI Units

○ RATING REQUIREMENTS

Driven Equip. HP (2.1.4): Norm. ______ Max ______
Driver HP: Rated ______ Max ______
Normal Transmitted Power (2.2.2) ______ HP
Gear Unit Rated Power (2.2.1) ______ HP
Torque @ Max Cont Speed ______ Lb Ft
Max Torque (2.2.1) ______ Lb Ft @ ______ RPM
○ Reducer ○ Increaser
Rated Speed, RPM (2.1.5) :
Input ______ ○ Specified ○ Nominal
Output ______ ○ Specified ○ Nominal
Allow Var In Gear Ratio (2.1.5) (+)(-) ______ %
Max Continuous Speed (1.5) ______ RPM
Trip Speed (1.5)(2.1.7) ______ RPM
Gear Service Factor (2.2.3.1) ______ (Min)
Hardness (2.2.3.2) Pinion ______ Gear ______
Shaft Assembly Designation (2.1.19) ______
HS Shaft Rot Fac'g Cpl'g (2.1.20.2) ○ CW ○ CCW
LS Shaft Rot Fac'g Cpl'g (2.1.20.2) ○ CW ○ CCW
External Loads (2.1.16) ______
Other Operating Conditions (2.2.2) (2.6.1.3) ______

○ INSTALLATION DATA (2.1.14)

○ Indoor ○ Heated ○ Under Roof
○ Outdoor ○ Unheated ○ Partial Sides
○ Grade ○ Mezzanine ○ ______
○ Winterization Req'd ○ Tropicalization Required
Elec. Area (2.1.10) Class ______ Grp ______ Div ______
Elec'l Area (2.1.10) Class ______ Zone ______ Temp ______
Max Allow SPL (2.1.6) ______ dBA @ ______ Ft
Elevation ______ Ft Barometer ______ PSIA
Range Of Ambient Temperatures :

	Dry Bulb	Wet Bulb
Normal	______ °F	______ °F
Maximum	______ °F	______ °F
Minimum	______ °F	______ °F

Unusual Conditions ○ Dust ○ Fumes
○ ______

☐ BASIC GEAR DATA

☐ SIngle Stage ☐ Single Helical
☐ Double Stage ☐ Double Helical
☐ Epicyclic ☐ ______

Mechanical Rating (1.5.16) HP ______ @ ______ RPM
Gear Service Factor (2.2.3.1) ______ (Actual)
Full Load Gear Unit Power Loss ______ HP
Gear Unit Mechanical Efficiency ______ %
Rating Speed, RPM Pinion ______ Gear ______
Hardness used for
Rating, (HB or Rc) Pinion ______ Gear ______
Tooth Pitting Index, "K" lb/in^2 (2.2.4.1)(2.2.4.2) :
Allowable ______ Actual ______
Material Index Number, lb/in^2 (Fig 3, Table 4) ______
Bending stress number, 'St" lb/in^2 (2.2.5.1)(2.2.5.2)
Pinion: Allowable ______ Actual ______
Gear: Allowable ______ Actual ______
Pitch Line Velocity ______ FPM
Anticipated SPL (2.1.6) ______ dBA @ ______ Ft
WR^2 Referred To LS Shaft ______ Lb Ft^2
Breakaway Torque ______ Lb Ft @ LS Shaft
Pinion teeth Hardness Range ______
Pinion teeth hardening method: ______
☐ HS Shaft Separate Hardness Range ______
Gear Teeth (rim) Hardness Range ______
Gear teeth hardening method: ______
Gear Hub:
☐ Forged Cylinder ☐ Forged & Coped ☐ Fabricated
Gear To Shaft fit method (2.5.3.2)
☐ Integral ☐ Keyed inteference ☐ Keyless Inteference
Rim Attachment (2.5.3.2) ______
LS Shaft Hardness Range ______
Journal static Weight loads (2.6.6.3)
Pinion ______ Lb Gear ______ Lb
Total Gear Unit Assembled Weight ______ Lb
NOTES: ______

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
US CUSTOMARY UNITS

Job No. ________ Item No. ________
Date ________ By ________
Revision No. ________ By ________

☐ GEAR DATA

	Pinion	Gear
Number Of Teeth		
Gear Ratio		
Tangential Load, "Wt", lb (2.2.4.2)		
AGMA Geometry Factor "J":		
Pitch Diameter, In		
Outside Diameter, In		
Root Diameter, In		
Center Groove Diameter, In		
Normal Pressure Angle, Degrees		
Normal Diametral Pitch		
Helix Angle, Degrees		
Center Distance		
Backlash, In		
Net Face Width, "Fw", In		
Pinion L/D		
Face Overlap Ratio		
Transverse Contact Ratio		
AGMA 6011 Service Factor		
Ratings based on ANSI/AGMA 6011 with SF=1.0		
Durability Power		
Strength Power		
Tooth surface finish, υin RA		
Tooth Generation Process		
Tooth Finishing Process		

Lead Modification (2.2.4.6) ☐ Not Req'd ☐ Req'd
Calculated Total Lead Mismatch, In (2.2.4.5) ________

☐ SCUFFING DATA (2.2.6)

Scuffing Data per ANSI/AGMA 925-A03
Scuffing Risk (2.2.6) ________ Calculation Method ________

Composite surface roughnexss σ_x		μin
Specific film thickness, EHL λ_{min}		μin
Tooth temperature, θ_M		°F
Maximum contact temperature, $\theta_{B\,max}$		°F

☐ MATERIALS

Gear Casing ________ Oil Seals ________
Pinion (s) ________
Gear Rim(s) ________
HS Shaft ________ LS Shaft ________
Radial Bearings ________ Backing ________
Thrust Bearing(s) ________ Backing ________

○ SHAFT END DETAIL

○ Gear Shaft End for Coupling Integral Flange (2.5.4.2)
○ Pinion Shaft End for Coupling Integral Flange (2.5.4.2)
○ Other ________

○ COUPLINGS AND GUARDS

	Pinion Shaft	Gear Shaft
Coupling Furnished By (3.2.1)		
Mount Coupling Halves (3.2.2)		
Cplg. Guard Adapter By (3.2.3)		
Cplg. Guard Furnished By (3.2.3)		
Coupling vendor (3.2.3)		
Vendor's model number (3.2.3)		
Coupling weight on shaft, lb		
CG Inboard/outboard of shaft end		
Hub Drill Template Provided	○	○

☐ SHAFT END DETAIL (2.5.4.2)

Shaft end detail specified by: ○ Purchaser ☐ Gear Vendor
(Integral Unless Otherwise Specified)

Shaft end detail if 'Otherwise' specified:	Pinion	Gear
Tapered/Keyless	○☐	○☐
Tapered / 1-Key	○☐	○☐
Tapered / 2-Keys	○☐	○☐
Cylindrical / 1-Key	○☐	○☐
Cylindrical / 2-Keys	○☐	○☐
Other ________	○☐	○☐
Shaft Diameter, In		

(If integral flange use diameter imediately adjacent to flange)

☐ RADIAL BEARINGS

	Pinion	Gear
Type		
Diameter, In		
Length, In		
Journal Velocity, FPS		
Loading, PSI		
Clearance (min-max), In		
Span, In		
Power loss each brg., HP		
Oil flow each brg.,GPM(US)		

☐ THRUST BEARING(S)

Location		
Manufacturer		
Type		
Size		
Area, In^2		
Loading, PSI		
Rating, PSI		
Int.Thrust Load, Lb (+)(-)		
Ext. Thrust Load, Lb (+)(-)		
Power loss each, HP		
Oil flow each, GPM (US)		

NOTES: ________

Page 3 of 4

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
US CUSTOMARY UNITS

Job No. ________ Item No. ________
Date ________ By ________
Revision No. ________ By ________

○ LUBRICATION REQUIREMENTS

○ Oil System Furnished By (2.8.3):
○ Other ________ ○ Gear Vendor
○ Oil Visc.: ____ cP @ 100F ____ cP @ 120F (2.8.6)
○ ISO Grade ____ Load Stage (2.2.6) ____

☐ MESH

Mesh and windage power loss ____ HP
Oil Flow, Mesh ____ GPM (US)

☐ LUBRICATION REQUIREMENTS

Min. Startup Oil Temperature ____ °F
Normal Oil Inlet temperature ____ °F
Maximum Oil Inlet Temperature ____ °F
Unit Oil Flow (Total) ____ GPM (US)
Unit Oil Pressure ____ PSI
☐ Oil Visc.: ____ cP @ 100F ____ cP @ 210F (2.8.6)
☐ ISO Grade ____ Load Stage (2.2.6) ____

☐ PIPING CONNECTIONS

Service	No.	Size	Type
Lube Oil Inlet			
Lube Oil Outlet			
Casing Drain*			
Vent			
Casing Purge			

*Casing drain is 'dead bottom' housing drain, not lube oil outlet

○ MOUNTING PLATES

○ Gear Furnished w/ (3.3.1.1)
○ Baseplate ○ Soleplate ○ Subplate(s) (3.3.2.10)
○ Mounting Plate(s) Furnished By (3.3.1.1) ____
○ Baseplate Leveling (3.3.2.3) ____
○ Baseplate With Leveling Pads (3.3.2.3)
○ Baseplate Suitable For Column Mounting (3.3.2.4)
○ Vendor Review of Purchaser's Foundation Dwgs. (2.1.13)
○ Grout Type (3.3.1.2.4): ____

○ CONTRACT DATA

○ Test Data Prior To Shipment ____
○ Progress Reports (5.3.3) ____
○ Vendor Signoff of Inspector Checklist (4.1.1.3)
● Information Retained for 20 years (4.2.1.1) (4.2.2.6.1)
○ Technical Manual (5.3.5.4)
○ **PAINTING** (4.4.3.1) ○ ____
○ Painting Housing Interior not Allowed (2.3.1.14)

○ SHIPMENT

○ Steel Rotor Storage Container (4.4.5) ○ Shaft Covers (4.4.4)

	Contract Unit		Spares	
Export Boxing	○		○	
Domestic Boxing	○		○	
Outdoor Storage Over 6 Mos.	○	3 yr Indoor	○	(4.4.3.9)

○ ADDITIONAL REQUIREMENTS

○ VIBRATION DETECTORS

RADIAL (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ____ ○ Total No. ____
○ X-Y Probes Pinion bearings ○ Coupling End ○ Blind End
○ X-Y Probes Gear bearings ○ Coupling End ○ Blind End
○ Other ____

AXIAL (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ____ ○ Total No. ____
○ Dual Probes at each thrust bearing
○ Single Probe any shaft without thrust bearing
○ Other ____

ONE EVENT PER REVOLUTION PROBE (2.7.4.5) (2.7.4.6)
○ Manufacturer ____ Total No. ____
○ One on Input Shaft ○ One on Output Shaft
○ Other ____

ACCELEROMETER (3.4.2.1) (2.7.4.5) (2.7.4.6)
○ Manufacturer ____ ○ No. Required ____
○ Pinion Coupling End ○ Gear Coupling End
○ Other ____

○ TEMPERATURE DETECTORS

○ Dial Type Thermometers (3.4.2.7) ____
○ Type Brg. Temp. Sensors (3.4.2.7) (2.7.1.3) ____
○ RTD ○ Thermocouple / ○ Simplex ○ Duplex
Calibration ____
○ HS/LS Bearings no. sensors each ____ / ____
○ Thrust - number of sensing elements each face ____

○ OTHER VIBRATION AND TEMPERATURE

(3.4.2.5 & 3.4.2.6)	Other	Gear Vendor
Oscillator Demodulators Supplied By	○	○
Vibration Monitor Supplied By	○	○
○ Vibration Shutdown Delay Time ____		Seconds
Temperature Monitor Supplied By	○	○
Oscillator Demodulator J-box By	○	○
Temp. Sensor Termination J-box By	○	○

J-box Type: ____ Mount: ____

○ MISCELLANEOUS

● Undamped Critical Analysis Report (2.6.2.1) **:**
○ w/ Dampd Rotr Respns Analys Rprt (2.6.2.4.3) (2.6.2.6)
Torsional Analysis By (2.6.5.2) ○ Gear Vendor ○ Other
○ Spare Set Of Gear Rotors (4.3.2.4)
○ Gear Case Furnished With Inlet Purge Connection (2.4.3)
○ Orientation Of Oil Inlet & Drain Conns. (2.4.6) ____
○ Filter Breather Location (2.3.1.12) ____
○ Torsional Device Provisions (2.5.4.5) ____
○ Rotor Vertical Storage Provisions (2.5.4.8) ____
○ Rotor Vertical Storage Fixture(s) (3.6.2) ____
○ Vendor Service Rep. on Site (2.1.13)

Page 4 of 4

SPECIAL PURPOSE GEAR UNITS
API 613 FIFTH EDITION
DATA SHEET
US CUSTOMARY UNITS

Job No. ____________ Item No. ____________
Date ____________ By ____________
Revision No. ____________ By ____________

○ INSPECTIONS AND TESTS (4.1)

○ Advance Notice of Witness Testing Required (4.1.3)
Number Calendar Days ________

	Req'd	Wit-ness	Ob-serve	Test Log
Shop Inspection (4.1.1)	○	○	○	
Cleanliness Inspection (4.2.3.1)	○	○	○	
Hardness Verification (4.2.3.3)	○	○	○	○
Dismantle-Reassembly (4.3.2.3.1)	●	○	○	○
Contact Check (2.5.2.2)	●	○	○	○
Contact Check Tape Lift (2.5.2.2)	○	○	○	○
Gear Accuracy Check (2.5.2.1)	●	○	○	○
Double Helical Axial Stability (2.5.2.3)	●	○	○	○
Special testing integral forged gears (2.5.3.3)				
(testing per mutual agreement)	○	○	○	○
Residual Unbalance Chk. (2.6.6.2)	●	○	○	●
Mechanical Run Test (Main) (4.3.2)	●	○	○	●
Mech. Run Test (Spare) (4.3.2.4)	○	○	○	○
Add'l. Mechanical Tests (4.3.2.2.16)	○	○	○	○
Part Or Full Load And Full Speed				
Test (4.3.3.1)	○	○	○	○
Full Torque, Reduced Speed (4.3.3.2)	○	○	○	○
Full Torque Static Test (4.3.3.3)	○	○	○	○
Back-To-Back Locked Torque				
Test (4.3.3.4)	○	○	○	○
Sound Level Test (4.3.3.5)	●	○	○	○
Additional Gear Tooth Test (4.2.2.8)	○	○	○	○
Use Shop Lube System	○	○	○	
Use Job Lube System	○	○	○	
Use Shop Vibration Probes, Etc.	○	○	○	
Use Job Vibration Probes, Etc.	○	○	○	
Final Assembly, Maintenance &				
Running Clearance (4.2.1.1.e)	○	○	○	○
Oil System Cleanliness (4.2.3.2)	○	○	○	
Oil System-Casing Joint				
Tightness (4.3.2.1.4)	●	○	○	
Warning And Protection				
Devices (4.3.2.1.5)	●	○	○	
Seismic Vibration Data (4.3.2.2.3)	○	○	○	○
Vibration, Phase Plots (4.3.2.2.12)	●	○	○	○
Oil Inlet Range Test (4.3.2.2.13)	○	○	○	○
Tape Recorded Vibration Data	○	○	○	○
(4.3.2.2.14)(4.3.2.2.15)	○	Purchaser copy		

NON-DESTRUCTIVE TESTING (4.2.1.2) (4.2.2.1)

	Surface	Sub-Surface		Log
○ Casing	○	○	____________	○
○ Rot. Elemts.	○	○	____________	○
○ Bearings	○	○	____________	○
○ Other :	○	○	____________	○

(Specify) ____________

NOTES:

APPENDIX B—REFERENCES

API

Std 614	*Lubrication, Shaft-Sealing, and Control-Oil Systems and Auxiliaries for Petroleum, Chemical and Gas Industry Services*
Std 617	*Centrifugal Compressors for Petroleum, Chemical and Gas Industry Services*
Std 670	*Vibration, Axial-Position, and Bearing-Temperature Monitoring Systems*
Std 671	*Special Purpose Couplings for Petroleum, Chemical and Gas Industry Services*
Std 672	*Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical, and Gas Industry Services*
Std 677	*General-Purpose Gear Units for Petroleum, Chemical and Gas Industry Services*
Publ 684	*Tutorial on the API Standard Paragraphs Covering Rotor Dynamics and Balance (An Introduction to Lateral Critical and Train Torsional Analysis and Rotor Balancing)*
RP 686	*Machinery Installation and Design*

AGMA[1]

908	*Information Sheet—Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth*
925	*Effect of Lubrication on Gear Surface Distress*

ANSI/AGMA[2]

2001	*Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth (U.S. Units)*
2101	*Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth (Metric)*
6001	*Design and Selection of Components for Enclosed Gear Drives*
6010	*Standard for Spur, Helical, Herringbone, and Bevel Enclosed Drives*
6011	*Specification for High Speed Helical Gear Units*
6025	*Sound for Enclosed Helical, Herringbone and Spiral Bevel Gears*

ANSI/AGMA/ISO[3]

1328-1	*Cylindrical Gears – ISO system of accuracy—Part 1: Definitions and Allowable Values of Deviations Relevant to Corresponding Flanks of Gear Teeth*

ANSI/ASME[4]

B1.1	*Unified Screw Threads (UN and UNR Thread Form)*
B16.1	*Cast Iron Pipe Flanges and Flange Fittings, Classes 25, 125 and 250*
B16.5	*Pipe Flanges and Flanged Fittings NPS 1/2 Through NPS 250*
B16.11	*Forged Fittings, Socket—Welding and Threaded*
B16.42	*Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300*
B17.1	*Keyways and Key Sets*
B1.20.1	*Pipe Threads, General Purpose (Inch)*

ASME

Boiler and Pressure Vessel Code

Section V “Nondestructive Examination”

Section VIII “Rules for Construction of Pressure Vessels”

Section IX “Qualification Standard for Welding and Brazing Qualifications”

ASTM[5]

A192	*Standard Reference Radiographs for Investment Steel Castings of Aerospace Applications*
A269	*Standard Specification for Seamless an Welded Austenitic Stainless Steel Tubing for General Service*
A275	*Standard Method for Magnetic Particle Examination of Steel Forgings*
A312	*Standard Practice for Description and Selection of Conditions for Photographing Specimens*
A388	*Standard Practice for Ultrasonic Examination of Heavy Steel Forgings*
A515	*Standard Specification for Pressure Vessel Plates, Carbon Steel, for Intermediate-and Higher-Temperature Service*

[1]American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.
[2]American National Standards Institute, 11 West 42nd Street, New York, New York 10036.
[3]International Standards Organization, 11 West 42nd Street, New York, New York 10036.
[4]ASME International, 3 Park Avenue, New York, New York 10016-5990.
[5]American Society for Testing and Materials, 100 Barr Harbor Drive, West Conshohocken, Pennsylvania 19428-2959.

DS 56B	*Metals and Alloys in the Unified Numbering System*
E94	*Standard Guide for Radiographic Examination*
E125	*Standard Reference Photographs for Magnetic Particle Indications on Ferrous Castings*
E709	*Standard Practice for Magnetic Particle Examination*

CEN[6]

10083	*Quenched and Tempered Steels*
10084	*Case Hardening Steels*
1561	*(BS) Founding—Grey Cast Irons*
1563	*(BS) Founding—Spheriodal Graphite Cast Irons*

IEC[7]

IEC 60079-0	*Electrical Apparatus for Explosive Gas Atmospheres*
60034-5	*Degrees of Protection Provided by the Integral Design of Rotating Electrical Machines (IP Code)*

ISO

7-1	*Pipe threads where pressure-tight joints are made on the threads*
261	*ISO general-purpose metric screw threads—General plan*
262	*ISO general-purpose metric screw threads—Selected sizes for screws, bolts and nuts*
724	*ISO general-purpose metric screw threads—Basic dimensions*
965	*ISO general purpose metric screw threads—Tolerances*
3448	*Industrial liquid lubricants—ISO viscosity classification*
6336-5	*Calculation of load capacity of spur and helical gears—Part 5: Strength and quality of materials*
6708	*Pipework Components—Definition and Selection of DN (Nominal Size)*
7005-1	*Metallic Flanges—Part 1: Steel Flanges*
7005-2	*Metallic Flanges—Part 2: Cast Iron Flanges*
1507-1	*(or 15071 Check use in spec.) Hexagonal Bolts with Flange—Small Series—Product Grade A*
8501	*Preparation of Steel Substrates Before Application of Paints and Related Products*
TR-10064-1	*Cylindrical gears—Code of inspection practice Part 1: Inspection of corresponding flanks of gear teeth*
14365-1	*Gears—FZG test procedures Part 1: FZG test method A/8, 3/90 for relative scuffing load-carrying capacity of oils*

NEMA[8]

SM23	*Steam Turbines for Mechanical Drive*
Publ 250	*Enclosures for Electrical Equipment (1000 Volts Maximum)*

NFPA[9]

NFPA 70	*National Electrical Code*, Articles 500, 501, 502 & 504

SAE[10]

J518	*Hydraulic Flanged Tube Pipe, and Hose Connections, Four Bolt Split Flange Type*

SSPC[11]

SP-6	*Commercial Blast Cleaning*

[6]European Committee for Standardization, 36 rue de Stassart, B-1050 Brussels, Belgium.
[7]International Electrochemical Commission, 1 rue de Varembe, Geneva, Switzerland.
[8]National Electrical Manufacturers Association, 1300 North 17th Street, Suite 1847, Rosslyn, Virginia 22209.
[9]National Fire Protection Association, 1 Batterymarch Park, Quincy, Massachusetts 02269.
[10]SAE International, Society of Automotive Engineers, 400 Commonwealth Drive, Warrendale, Pennsylvania 15096-0001.
[11]Steel Structures Painting Council, 40 24th Street, Suite 600, Pittsburgh, Pennsylvania 15222.

APPENDIX C—COUPLINGS FOR HIGH SPEED GEAR UNITS

C.1 General

C.1.1 Appendix C is intended to provide a guide to the selection of coupling types and the location of thrust bearings in equipment trains that employ gear units. This appendix is not intended to supersede this standard or the information contained in the data sheets, or API Std 671.

C.1.2 Gear units must be connected to driving and driven machines by means of couplings that will not impose harmful forces on the rotating elements of the gear unit. This is necessary to maintain uniform distribution of the tooth loading across the face of the gears throughout varying thermal and load conditions. Excessive moment forces exerted across a coupling will cause the pinion or gear wheel to cock in its bearings, resulting in a shift in the tooth loading toward one end of the gear teeth. Excessive axial force transmitted across a coupling will cause one helix of a double-helical gear to be more heavily loaded than the other if the arrangement of machinery is such that the axial force is transmitted across the gear-tooth mesh to reach an opposing thrust bearing.

C.2 Coupling Types

C.2.1 The coupling types listed in C.2.2 through C.2.4 are not intended to be all inclusive, but are intended to include the most popular types used in conjunction with gear units.

C.2.2 Generic types (popular designations) of couplings include:

a. Gear-tooth or gear-type.
b. Grid-type.
c. Flexible-element.
d. Rubber bushing.
e. Flexible shaft (quill).
f. Rigid flange (solid) types.

C.2.3 Any of the couplings listed in C.2.2 may be arranged to have a limited end float, that is, to limit the axial freedom of one connected shaft with respect to the other.

C.2.4 GEAR TYPE COUPLINGS

In gear-tooth couplings, the axial force is indeterminate, since in these the slip force is directly related to the coefficient of friction that the coupling is exhibiting at the moment. Consideration should be given to the possibility of overloading the teeth of a double-helical gear in the event of coupling hang-up caused by sludging or excessive wear.

Gear type couplings may be designed as limited end float couplings.

C.3 Flexible Element Couplings

Flexible element couplings have a distinct advantage with regard to axial force problems, since the force required to displace one half of a coupling with respect to the other half is quite predictable. Once the machine is properly installed and correct axial settings are obtained, the maximum axial force that may be transmitted across a gear mesh or carried by opposing thrust bearings is known. Thrust bearings can be selected for optimum conditions. The axial forces to be transmitted across the gear mesh can readily be included with other factors in the sizing of the gear unit.

Flexible element couplings are inherently limited end float couplings.

C.4 Limited-end-float Couplings

C.4.1 The use of limited-end-float couplings makes it possible to eliminate or reduce the potential problem of excessive thrust. In a machinery train in which only one unit (such as a compressor) requires a thrust bearing to maintain the internal axial clearances between the stator and the rotor, having one limited-end-float coupling between the compressor and the gear unit and having a second limited-end-float coupling between the gear unit and the motor eliminate the need for thrust bearings on either the gear unit or the motor. This arrangement minimizes the load on the compressor thrust bearing, since the most it will feel from the connected machinery will be equal to the motor centering force (see Figure C-1, Panel A).

C.4.2 In a machinery train that involves a steam- or gas-turbine prime mover, a double-helical gear unit, and a compressor, both the turbine and the compressor require thrust bearings. In this case, the use of a single limited-end-float coupling, can eliminate the thrust bearing from the gear unit. Selecting the coupling that has the higher calculated thrust-transmitting potential (usually the low-speed coupling) as a limited-end-float coupling, plus eliminating the thrust bearing from the gear unit can drastically reduce the thrust on the machine connected to that gear unit shaft. Selecting smaller thrust bearings with improved machine efficiency is possible (see Figure C-1, Panel B).

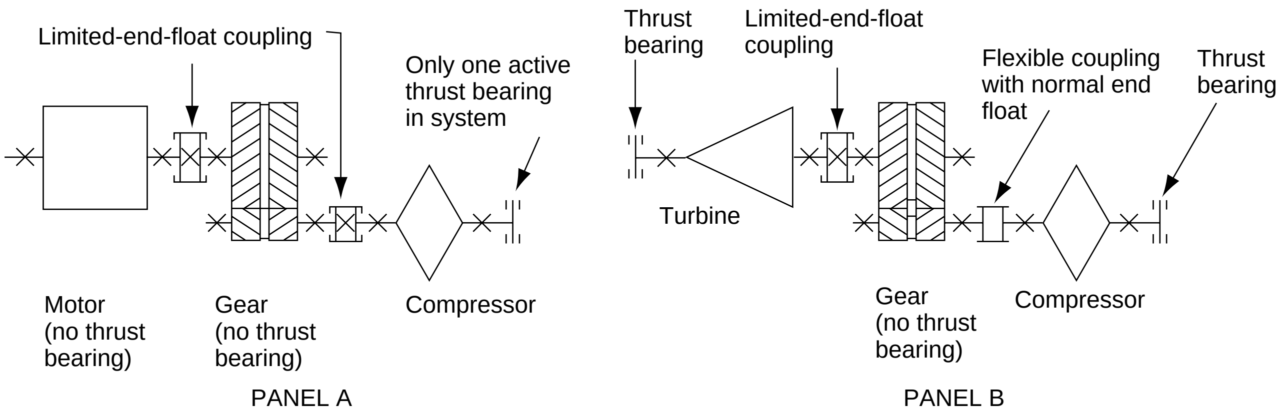
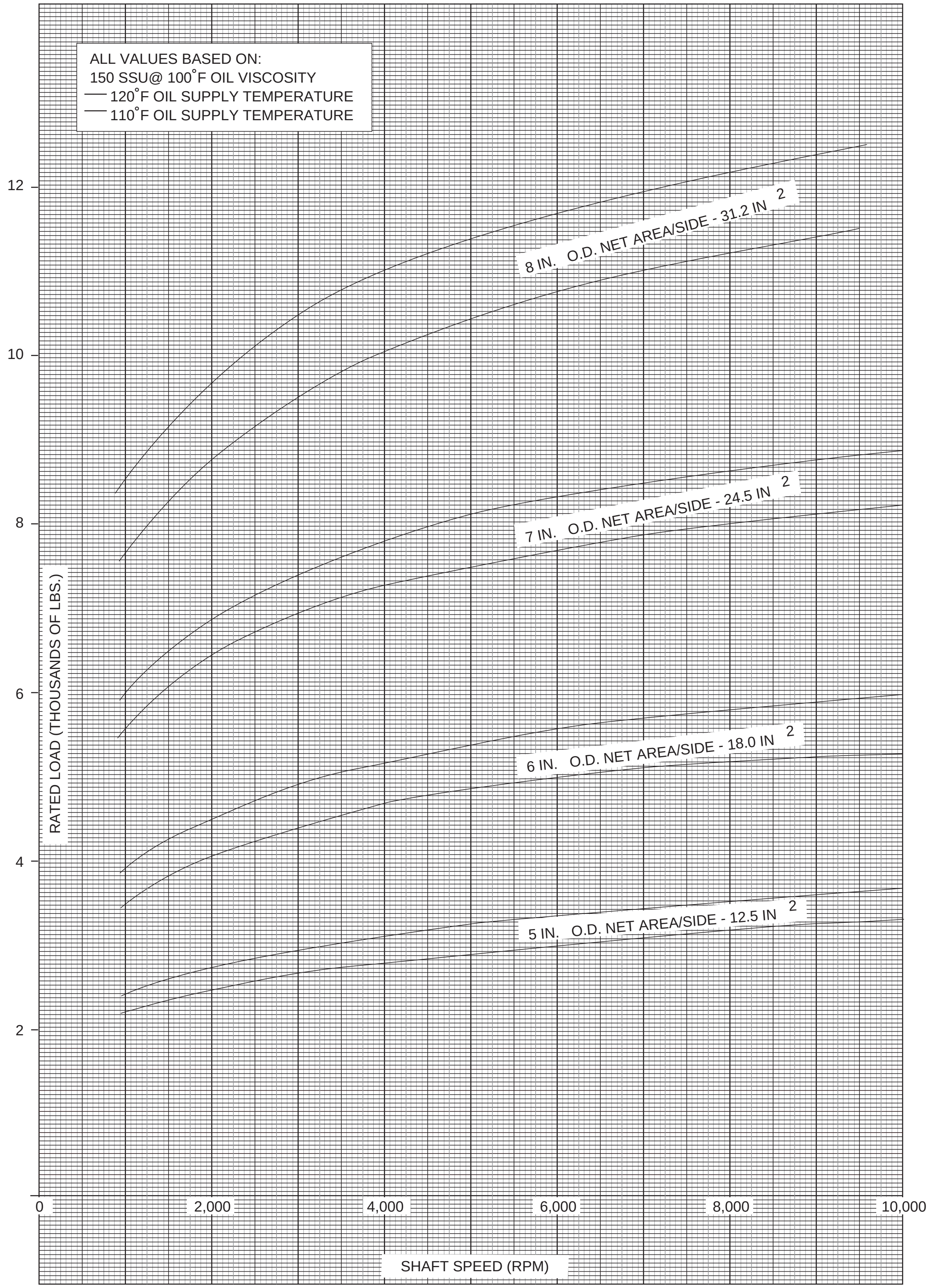


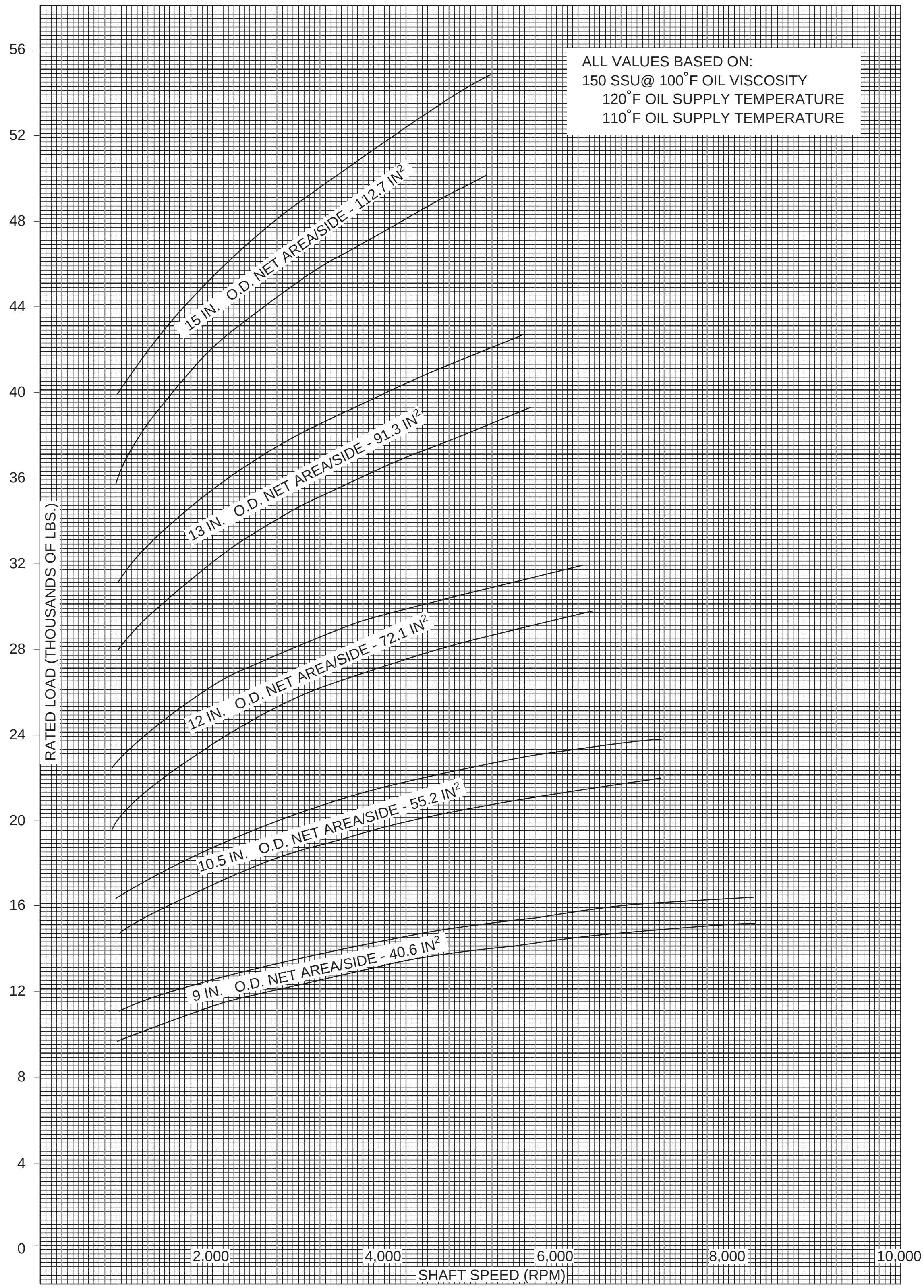
Figure C-1—Couplings for Gear Units

APPENDIX D—RATED LOAD CURVES FOR THRUST BEARINGS WITH STANDARD 6 × 6 SHOES
(See following pages for figures.)



Note: OD = outside diameter. All values are based on an oil viscosity of 32 square millimeter per second at 38˚C (150 Saybolt universal seconds at 100˚F)

Figure D-1—Rated Load Curves for Thurst Bearings with Standard 6 × 6 Shoes: Outside Diameters from 5 in. – 8 in.



Note: OD = outside diameter. All values are based on an oil viscosity of 32 square millimeter per second at 38˚C (150 Saybolt universal seconds at 100˚F)

Figure D-2—Rated Load Curves for Thrust Bearings with Standard 6 × 6 Shoes: Outside Diameters from 9 in. – 15 in.

APPENDIX E—MATERIAL SPECIFICATIONS FOR SPECIAL PURPOSE GEAR UNITS PAGE

(See following page for tables.)

Table E-1—Gear Unit Housings

Cast	Material[a, b]
Cast	ASTM A27 Grade 65-35
Cast	ASTM A48 Class 30 Minimum
Cast	EN JL1040
Cast	EN JL1050
Cast	EN JL1030
Cast	EN JL1015
Fabricated	**Material[a ,b]**
Hot-rolled bars	ASTM A575, A 576
Plate	ASTM A131
Plate	ASTM A283
Plate	ASTM A284, Grade B
Plate	ASTM A285
Plate	ASTM A516
Plate or Shape	ASTM A6
Plate or Shape	ASTM A36
Plate or Shape	AISI 1010
Plate or Shape	AISI 1020
	EN C45E
	EN C15E
	EN S235JRG

Table E-2—Shafts, Pinions[c], and Gear Wheels[c]

Heat Treatment	Material[a b]
Through hardened	AISI 4140
	AISI 4145
	AISI 4340
	EN 42CrMo4
	EN 34CrNiMo6
	EN 30CrNiMo8
	EN 36CrNiMo16
Nitrided	SAE/AMS 6470
	SAE/AMS 6475
	AISI 4140
	AISI 4340
	EN 31CrMoV9
	EN 32CrMoV12-9
Carburized	AISI 3310
	AISI 4320
	AISI 9310
	EN 12NiCrMo7
	EN 18CrNiMo7-6[d]

Table E-3—Fabricated Gears

Web	Material[a, b]
Plate	ASTM A283, Grade B
Plate	ASTM A285, Grade B&C
Plate or Shapes	ASTM A36
	EN C45E
	EN C15E
	EN S235JRG
Hub	**Material[a ,b]**
	AISI 1020
	AISI 4140
	AISI 4340
	EN C45E
	EN C15E
	EN S235JRG
Rim[c]	**Material[a ,b]**
Through hardened	AISI 4130
	AISI 4135
	AISI 4140
	AISI 4145
	AISI 4340
	EN 42CrMo4
	EN 34CrNiMo6
	EN 30CrNiMo8
	EN 36CrNiMo16
Nitrided	SAE/AMS 6470
	SAE/AMS 6475
	AISI 4140
	AISI 4340
	EN 31CrMoV9
	EN 32CrMoV12-9
Carburized	AISI 3310
	AISI 4320
	AISI 9310
	EN 12NiCrMo7
	EN 18CrNiMo7-6

[a]Descriptions of AISI designations can be found in ASTM DS 56B. For alloy steel the material chemistry shown can be both standard and H. Also sometime the designation E is shown in front of the AISI number to indicate electric furnace melted and is also acceptable. For instance, when 4340 is the listed material, 4340, 4340H, E4340 and E4340H are all acceptable materials.

[b]Descriptions of EN or Euronorm designations can be found in EN10083 & EN10084 & EN1561 & EN1563.

[c]Forgings and hot rolled bar stock used on gear wheels, gear wheel rims, and pinions shall as a minimum be electric furnace melted and vacuum degassed and meet the requirements of ANSI/AGMA ISO 6336-5 quality grade ME for case hardened or nitrided and quality grade MX for through hardened.

[d]New designation for 17CrNiMo6.

APPENDIX F—VENDOR DRAWING AND DATA REQUIREMENTS

SPECIAL PURPOSE GEAR UNITS VENDOR DRAWING AND DATA REQUIREMENTS

JOB NO. ____________ ITEM NO. ____________
PURCHASE ORDER NO. ____________ DATE ____________
REQUISITION NO. ____________ DATE ____________
INQUIRY NO. ____________ DATE ____________
PAGE 1 OF 2 BY ____________

FOR ____________ REVISION ____________
SITE ____________ UNIT ____________
SERVICE ____________ NO. REQUIRED ____________

Proposal[a] Bidder shall furnish ______ copies of data for all items indicated by an X.

Review[b] Vendor shall furnish ______ copies and ______ transparencies of drawings and data indicated.

Final[c] Vendor shall furnish ______ copies and ______ transparencies of drawings and data indicated.
Vendor shall furnish ______ operating and maintenance manuals.

DISTRIBUTION RECORD

Final—Received from vendor
Final—Due from vendor[c]
Review—Returned to vendor
Review—Received from vendor
Review—Due from vendor[c]

Proposal	Review	Final	DESCRIPTION	Review—Due from vendor	Review—Received from vendor	Review—Returned to vendor	Final—Due from vendor	Final—Received from vendor
			1. Certified dimensional outline drawing and list of connections.					
			2. Cross-sectional drawings and part numbers.[d]					
			3. Rotor assembly drawings and part numbers.[d]					
			4. Thrust-bearing assembly drawing and part numbers.[d]					
			5. Journal-bearing assembly drawings and bills of material.					
			6. Coupling assembly drawing and bill of material.[d,e]					
			7. Lube-oil schematic and bill of material.					
			8. Lube-oil component drawings and data.[d]					
			9. Electrical and instrumentation schematics and bills of material.					
			10. Electrical and instrumentation arrangement drawing and list of connections.					
			11. Anticipated tooth contact drawing and specifications.[d]					
			12. Record of deviations from manufacturing process control system.[f]					
			13. Mass elastic data.					
			14. Lateral critical speed analysis report.[f]					
			15. Torsional analysis report.[f]					
			16. Input and output shaft position diagram.					
			17. Weld procedures.[f]					
			18. Hydrostatic test logs (oil system).					
			19. Mechanical running test logs.[f]					
			20. Rotor balancing logs.					
			21. Rotor mechanical and electrical runout.[f]					
			22. As-built data sheets.					
			23. As-built dimensions or data.[d]					
			24. Installation manual.					
			25. Operating and maintenance manual.					
			26. Technical manual.					
			27. Spare parts recommendation and price list.					
			28. Progress reports and delivery schedule.					
			29. Preservation, packaging, and shipping procedures.					

[a]Proposal drawings and data do not have to be certified or as-built.
[b]Purchaser will indicate in this column the desired time frame for submission of materials, using the nomenclature given at the end of the form.
[c]Bidder shall complete this column to reflect his actual distribution schedule and shall include this form with the proposal.
[d]These items are normally provided only in instruction manuals.
[e]If furnished by the vendor.
[f]If specified.

SPECIAL PURPOSE GEAR UNITS VENDOR DRAWING AND DATA REQUIREMENTS

JOB NO. ____________ ITEM NO. ____________

PAGE 2 OF 2 BY ____________

DATE ____________ REVISION ____________

Proposal[a] Bidder shall furnish ______ copies of data for all items indicated by an X.

Review[b] Vendor shall furnish ______ copies and ______ transparencies of drawings and data indicated.

Final[c] Vendor shall furnish ______ copies and ______ transparencies of drawings and data indicated.
Vendor shall furnish ______ operating and maintenance manuals.

DISTRIBUTION RECORD

Final—Received from vendor
Final—Due from vendor[c]
Review—Returned to vendor
Review—Received from vendor
Review—Due from vendor[c]

Proposal	Review	Final	DESCRIPTION	Review—Due from vendor	Review—Received from vendor	Review—Returned to vendor	Final—Due from vendor	Final—Received from vendor
			30. List of special tools furnished for maintenance.					
			31. Nondestructive test procedures and acceptance criteria.					
			32. Book with all Quality Assurance documents.					

[a]Proposal drawings and data do not have to be certified or as-built.
[b]Purchaser will indicate in this column the desired time frame for submission of materials, using the nomenclature given at the end of the form.
[c]Bidder shall complete this column to reflect his actual distribution schedule and shall include this form with the proposal.
[d]These items are normally provided only in instruction manuals.
[e]If furnished by the vendor.
[f]If specified.

Notes:
1. Where necessary to meet the scheduled shipping date, the vendor shall proceed with manufacture upon receipt of the order and without awaiting the purchaser's approval of drawings.
2. The vendor shall send all drawings and data to the following:

____

3. All drawings and data shall show project, purchase order, and item numbers as well as plant location and unit. One set of the drawings and instructions necessary for field installation, in addition to the copies specified above, shall be forwarded with shipment.
4. See the descriptions of required items that follow.
5. All of the information indicated on the distribution schedule shall be received before final payment is made.

Nomenclature:
S—number of weeks before shipment.
F—number of weeks after firm order.
D—number of weeks after receipt of approved drawings.

Vendor ____

Date ________________ Vendor Reference ________________________

Signature ____
(Signature acknowledges receipt of all instructions)

1. Certified dimensional outline drawing and list of connections including the following:

    a. The size, rating, and location of all customer connections.
    b. Approximate overall and handling weights.
    c. Overall dimensions, and maintenance and dismantling clearances.
    d. Shaft centerline height.
    e. Dimensions of baseplates (if furnished) complete with diameters, number, and locations of bolt holes and the thicknesses of sections through which the bolts must pass.
    f. Grouting details.
    g. Center of gravity and lifting points.
    h. Shaft end separation and alignment data.
    i. Direction of rotation.
    j. Winterization, tropicalization, and/or noise attenuation details, when required.

2. Cross-sectional drawings and part numbers.
3. Rotor assembly drawings and part numbers.
4. Thrust bearing assembly drawing and part numbers.
5. Journal bearing assembly drawings and bills of material.
6. Coupling assembly drawing and bill of material.
7. Lube oil schematic and bill of material including the following:

    a. Oil type, oil flows, temperatures, and pressures at each use point.
    b. Control, alarm, and trip settings (pressure and recommended temperatures).
    c. Total heat loads.
    d. Utility requirements, as applicable.
    e. Pipe, valve, and orifice sizes.
    f. Instrumentation, safety devices, control schemes, and wiring diagrams.
    g. Oil inlet, outlet, and casing purge and drain connections.

8. Lube oil component drawings and data.
9. Electrical and instrumentation schematics and bills of material including the following:

    a. Vibration alarm and shutdown limits.
    b. Bearing temperature alarm and shutdown limits.

10. Electrical and instrumentation arrangement drawing and list of connections.
11. Anticipated tooth contact drawing and specifications.
12. Record of deviations from manufacturing process control system.
13. Mass elastic data.
14. Lateral critical speed analysis report.
15. Torsional analysis report.
16. Input and output shaft position diagram.
17. Weld procedures.
18. Hydrostatic test logs (oil system).
19. Mechanical running test logs.
20. Rotor balancing logs.
21. Rotor mechanical and electrical runout.
22. As-built data sheets.
23. As-built dimensions or data.
24. Installation manual.
    SECTION 1—Installation:

    a. Storage.
    b. Foundation.
    c. Grouting.
    d. Setting equipment, rigging procedures, component weights and lifting diagram.
    e. Alignment.

f. Piping recommendations.
g. Composite outline drawing, including anchor bolt locations.
h. Dismantling clearances.

25. Operating and maintenance manual.
SECTION 1—Operation:

a. Start-up procedures, including tests and checks before start-up.
b. Routine operational procedures.
c. Lube oil recommendations (pressure, flow, viscosity, other).

SECTION 2—Disassembly and assembly:

a. Rotor in gear casing.
b. Journal bearings.
c. Thrust bearings (including clearance and preload on antifriction bearings).
d. Seals.
e. Thrust collars, if applicable.
f. Allowable wear of running clearances.
g. Fits and clearances for rebuilding.
h. Routine maintenance procedures and intervals.

SECTION 3—Drawing and data requirements:

a. Certified dimensional outline drawing and list of connections.
b. Cross-sectional drawing and bill of materials.
c. Instrumentation arrangement and list of connections.

26. Technical Manual.
SECTION 1—Vibration data:

a. Vibration analysis data.
b. Lateral critical speed analysis report.
c. Torsional critical speed analysis report, if specified.
d. Mass elastic data.

SECTION 2—As-built data:

a. As-built data sheets.
b. As-built clearances.
c. Rotor balance logs and 6-point residual unbalance check.
d. Noise data sheets.
e. Input and output shaft position diagram.
f. Weld procedures.
g. Gear checker charts (lead, profile and spacing).
h. Tooth contact blue transfer tapes (checking stand).
i. Tooth contact blue transfer tapes (contract housing).
j. Mechanical running test logs and vibration data.
k. Rotor mechanical and electrical runout.
l. As-built dimensions or data

27. Spare parts recommendation and price list.
28. Progress reports on major component procurement, engineering milestones, assembly, testing and delivery schedule.
29. Preservation, packaging and shipping procedures.
30. List of special tools furnished for maintenance.
31. Non-destructive test procedures and acceptance criteria.
32. Book with all Quality Assurance documents.

APPENDIX G—RESIDUAL UNBALANCE WORK SHEETS (SEE FOLLOWING PAGES FOR WORKSHEETS.)

G.1 General

This appendix describes the procedure to be used to determine residual unbalance in machine rotors. Although some balancing machines may be set up to read out the exact amount of unbalance, the calibration can be in error. The only sure method of determining is to test the rotor with a known amount of unbalance.

G.2 Residual Unbalance

Residual unbalance is the amount of unbalance remaining in a rotor after balancing. Unless otherwise specified, residual unbalance shall be expressed in g-mm (g-in).

G.3 Maximum Allowable Residual Unbalance

G.3.1 The maximum allowable residual unbalance, per plane, shall be calculated according to the paragraph from the standard to which this appendix is attached.

G.3.2 The static weight on each journal shall be determined by physical measurement. (Calculation methods may introduce errors). It should NOT simply be assumed that that rotor weight is equally divided between the two journals. There can be great discrepancies in the journal weight to the point of being very low (even negative on over-hung rotors). In the example problem, the left plane has a journal weight of 530.7 kg (1170 lb). The right plane has a journal weight of 571.5 kg (1260 lb).

G.4 Residual Unbalance Check

G.4.1 GENERAL

G.4.1.1 When the balancing machine readings indicate that the rotor has been balanced within the specified tolerance, a residual unbalance check shall be performed before the rotor is removed from the balancing machine.

G.4.1.2 To check the residual unbalance, a known trial weight is attached to the rotor sequentially in 6 equally spaced radial positions (60 degrees apart), each at the same radius. (i.e., same moment [g-in.]). The check is run at each balance machine readout plane, and the readings in each plane are tabulated and plotted on the polar graph using the procedure specified in G.4.2.

G.4.2 PROCEDURE

G.4.2.1 Select a trial weight and radius that will be equivalent to between one and two times the maximum allowable residual unbalance (e.g., if U_{max} is 488.4 g-mm [19.2 g-in.], the trial weight should cause 488.4 to 976.8 g-mm [19.2 to 38.4 g-in.] of unbalance). This trial weight and radius must be sufficient so that the resulting plot in G.4.2.5 encompasses the origin of the polar plot.

G.4.2.2 Starting at a convenient reference plane (i.e., ~last heavy spot), mark off the specified 6 radial positions (60 degree increments) around the rotor. Add the trial weight near the last known heavy spot for that plane. Verify that the balance machine is responding and is within the range and graph selected for taking the residual unbalance check.

G.4.2.3 Verify that the balancing machine is responding reasonably (i.e., no faulty sensors or displays). For example if the trial weight is added to the last known heavy spot, the first meter reading should be at least twice as much as the last reading taken before the trial weight was added. Little or no meter reading generally indicates that the rotor was not balanced to the correct tolerance, the balancing machine was not sensitive enough, or that a balancing machine fault exists (i.e., faulty pickup). Proceed, if this check is OK.

G.4.2.4 Remove the trial weight and rotate the trial weight to the next trial position (that is, 60, 120, 180, 240, 300 and 360 degrees from the initial trial weight position). Repeat the initial position as a check for repeatability on the Residual Unbalance Worksheet. All verification shall be performed using only one sensitivity range on the balance machine.

G.4.2.5 Plot the balancing machine amplitude readout versus angular location of trial weight (NOT balancing machine phase angle) on the Residual Unbalance Worksheet and calculate the amount of residual unbalance (refer to work sheets, Figures G-3 & GG-5).

Note: The maximum reading occurs when the trial weight is placed at the rotor's remaining heavy spot; the minimum reading occurs when the trial weight is placed opposite the rotor's heavy spot (light spot). The plotted readings should form an approximate circle around the origin of the polar chart. The balance machine angular location readout should approximate the location of the trial weight. The maximum deviation (highest reading) is the heavy spot (represents the plane of the residual unbalance). Blank work sheets are Figures G-1 & G-2.

G.4.2.6 Repeat the steps described in G.4.2.1 through G.4.2.5 for each balance machine readout plane. If the specified maximum allowable residual unbalance has been exceeded in any balance machine readout plane, the rotor shall be balanced more precisely and checked again. If a balance correction is made in any balance machine readout plane, then the residual unbalance check shall be repeated in all balance machine readout planes.

G.4.2.7 For stacked component balanced rotors, a residual unbalance check shall be performed after the addition and balancing of the rotor after the addition of the first rotor component, and at the completion of balancing of the entire rotor, as a minimum.

Notes:
1. This ensures that time is not wasted and rotor components are not subjected to unnecessary material removal in attempting to balance a multiple component rotor with a faulty balancing machine.
2. For large multi-stage rotors, the journal reactions may be considerably different from the case of a partially stacked to a completely stacked rotor.

Customer: ____________________
Job/Project Number: ____________________
OEM Equipment S/N: ____________________
Rotor Identification Number: ____________________
Repair Purchase Order Number: ____________________
Vendor Job Number: ____________________
Correction Plane (Left or Right)—use sketch ____________________ (plane)

Balancing Speed [ ] (rpm)
Maximum Rotor Operating Speed (N) [ ] (rpm)
Static Journal Weight Closest to This Correction Plane (W) [ ] (kg) [ ] (lb.)
Trial Weight Radius (R)—the radius at which the trial weight will be placed [ ] (mm) [ ] (in.)

Calculate Maximum Allowable Residual Unbalance (U_{max}):
SI Units:

$$U_{max} = \frac{(6350) \times (W)}{(N)} = \frac{(6350) \times [\ \ ]}{[\ \ ]} = [\ \ ] \text{ (g-mm)}$$

Customary Units:

$$U_{max} = \frac{(113.4) \times (W)}{(N)} = \frac{(113.4) \times [\ \ ]}{[\ \ ]} = [\ \ ] \text{ (g-in.)}$$

Calculate the trial unbalance(TU):
Trial Unbalance (TU) is between ($1 \times U_{max}$) and ($2 \times U_{max}$) (1×) to (2×) (Selected Multiplier is) [ ]
SI Units: [ ] to [ ] = [ ] (g-mm)
Customary units: [ ] to [ ] = [ ] (g-in.)

Calculate the trial weight (TW):

$$\text{Trial Weight } (TW) = \frac{U_{max}}{R} = \frac{[\ \ ] \text{ g-mm}}{[\ \ ] \text{ mm}} \text{ or } \frac{[\ \ ] \text{ g-in.}}{[\ \ ] \text{ in.}} = [\ \ ] \text{ (g)}$$

Conversion Information:
1 kg = 2.2046 lb. 1 oz. = 28.345 g

Obtain the test data and complete the table:

Test Data

Position	Trial Weight Angular Location on Rotor (degrees)	Balancing Mach Readout Amplitude (grams)	Phase Angle (degrees)
1	0		
2	60		
3	120		
4	180		
5	240		
6	300		
Repeat 1	0		

Sketch the rotor configuration:

Rotor Sketch

PROCEDURE:
Step 1: Plot the balancing machine amplitude versus trial weight angular location on the polar chart (Figure G-2) such that the largest and smallest values will fit.
Step 2: The points located on the polar chart should closely approximate a circle. If it does not, then it is probably that the recorded data it is in error and the test should be repeated.
Step 3: Determine the maximum and minimum balancing machine amplitude readings.
Step 4: Using the worksheet, (Figure G-2), determine the Y and Z values required for the residual unbalance calculation.
Step 5: Using the worksheet, (Figure G-2), calculate the residual unbalance remaining in the rotor.
Step 6: Verify that the determined residual unbalance is equal to or less than the maximum allowable residual unbalance (U_{max}).

Half Keys Used for Rotor Balancing
(add sketch for clarification if necessary)

Location	Weight

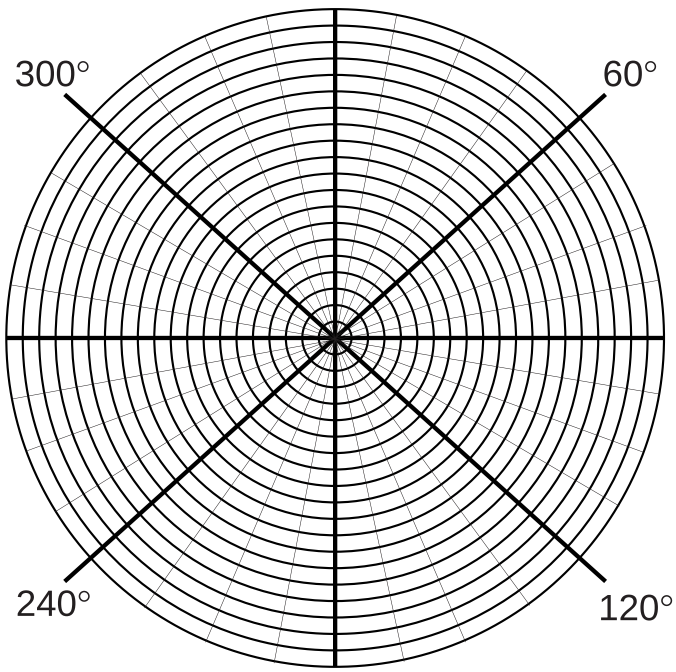
NOTES:
1) The trial weight angular location should be referenced to a keyway or some other permanent marking on the rotor. The preferred location is the location of the once-per-revolution mark (for the phase reference transducer).
2) The balancing machine amplitude readout for the Repeat of 1 should be the same as Position 1, indicating repeatability.
3) A primary source for error is not maintaining the same radius for each trial weight location.

Balanced By: __________ Date: ________
Approved By: __________ Date: ________

Figure G-1—(Blank) Residual Unbalance Work Sheet

Customer: ____________
Job/Project Number: ____________
OEM Equipment S/N: ____________
Rotor Identification Number: ____________
Repair Purchase Order Number: ____________
Vendor Job Number: ____________
Correction Plane (Left or Right)—use sketch ____________

RESIDUAL UNBALANCE POLAR PLOT



Rotor Rotation: ☐ CCW ☐ CW   Phase is layed out: ☐ CCW ☐ CW

Calculate *Y* and *Z* values:

Maximum amplitude value is:	[ ] g		Minimum amplitude value is:		[ ]
Y = (Maximum – Minimum)/2 (	[ ]	–	[ ]	) / 2 =	[ ]
Z = (Maximum + Minimum)/2 (	[ ]	+	[ ]	) / 2 =	[ ]

Residual Unbalance Left in Rotor =	(*TU*)	×	(*Y*)	/	(*Z*)		
SI Units:	[ ]	×	[ ]	/	[ ]	=	[ ]
Customary Units:	[ ]	×	[ ]	/	[ ]	=	[ ]

Allowable Unbalance Tolerance = U_{max} = [ ] gm-mm [ ] gm-in.

RESULT: Residual unbalance left in the rotor is equal to or less than the allowable unbalance tolerance?

☐ PASS ☐ FAIL

☐ As Received ☐ Final ☐ Other: ____________

Balanced By: ____________ Date: ____________
Approved By: ____________ Date: ____________

Figure G-2—(Blank) Residual Unbalance Polar Plot Work Sheet

Customer:	ABC Refining Co.	
Job/Project Number:	00 – 1234	
OEM Equipment S/N:	C – 1234	
Rotor Identification Number:	1234 – C – 4320	
Repair Purchase Order Number:	PO 12345678	
Vendor Job Number:	Shop – 00 – 1234	
Correction Plane (Left or Right)—use sketch	Left	(plane)

Balancing Speed	800	(rpm)		
Maximum Rotor Operating Speed (N)	6900	(rpm)		
Static Journal Weight Closest to This Correction Plane (W)	530.7	(kg)	1170	(lb.)
Trial Weight Radius (R)—the radius at which the trial weight will be placed	381	(mm)	15	(in.)

Calculate Maximum Allowable Residual Unbalance (U_{max}):

SI Units:

$$U_{max} = \frac{(6350) \times (W)}{(N)} = \frac{(6350)}{6900} \times 530.7 = \boxed{488.4} \text{ (g-mm)}$$

Customary Units:

$$U_{max} = \frac{(113.4) \times (W)}{(N)} = \frac{(113.4)}{6900} \times 1170 = \boxed{19.2} \text{ (g-in.)}$$

Calculate the trial unbalance(TU):

Trial Unbalance (TU) is between ($1 \times U_{max}$) and ($2 \times U_{max}$)

	(1×)	to	(2×)	(Selected Multiplier is)	1.6
SI Units	488.4	to	976.8	is	781.4 (g-mm)
Customary units:	19.2	to	38.5	is	30.8 (g-in.)

Calculate the trial weight (TW):

$$\text{Trial Weight } (TW) = \frac{U_{max}}{R} = \frac{781 \text{ g-mm}}{381 \text{ mm}} \text{ or } \frac{31 \text{ g-in.}}{15 \text{ in.}} = \boxed{2.1} \text{ (g)}$$

Conversion Information:

1 kg = 2.2046 lb. 1 oz. = 28.345 g

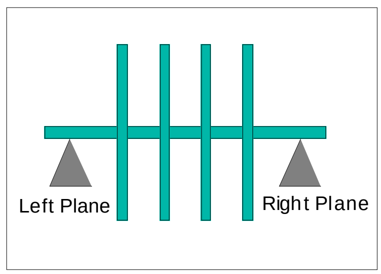
Obtain the test data and complete the table:

Test Data

Position	Trial Weight Angular Location on Rotor (degrees)	Balancing Mach Readout Amplitude (grams)	Phase Angle (degrees)
1	0	1.60	358
2	60	1.11	59
3	120	1.58	123
4	180	2.21	182
5	240	3.00	241
6	300	2.30	301
Repeat 1	0	1.58	359

Sketch the rotor configuration:

Rotor Sketch



PROCEDURE:

Step 1: Plot the balancing machine amplitude versus trial weight angular location on the polar chart (Figure G-2) such that the largest and smallest values will fit.

Step 2: The points located on the polar chart should closely approximate a circle. If it does not, then it is probably that the recorded data it is in error and the test should be repeated.

Step 3: Determine the maximum and minimum balancing machine amplitude readings.

Step 4: Using the worksheet, (Figure G-2), determine the Y and Z values required for the residual unbalance calculation.

Step 5: Using the worksheet, (Figure G-2), calculate the residual unbalance remaining in the rotor.

Step 6: Verify that the determined residual unbalance is equal to or less than the maximum allowable residual unbalance (U_{max}).

Half Keys Used for Rotor Balancing
(add sketch for clarification if necessary)

Location	Weight

NOTES:

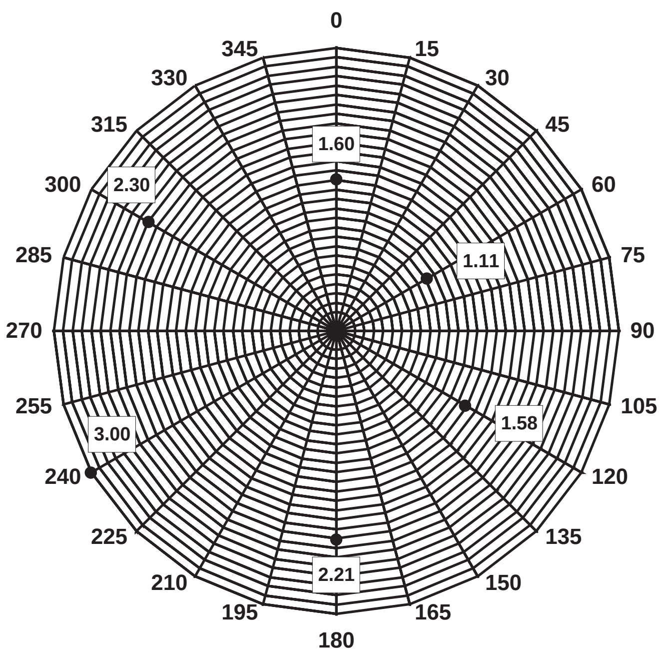
1) The trial weight angular location should be referenced to a keyway or some other permanent marking on the rotor. The preferred location is the location of the once-per-revolution mark (for the phase reference transducer).
2) The balancing machine amplitude readout for the Repeat of 1 should be the same as Position 1, indicating repeatability.
3) A primary source for error is not maintaining the same radius for each trial weight location.

Balanced By:	CJ, TR, & RC	Date:	05/24/00
Approved By:	CC	Date:	05/24/00

Figure G-3—Sample Residual Unbalance Work Sheet for Left Plane

Customer:	ABC Refining Co.
Job/Project Number:	00 – 1234
OEM Equipment S/N:	C – 1234
Rotor Identification Number:	1234 – C – 4320
Repair Purchase Order Number:	PO 12345678
Vendor Job Number:	Shop – 00 – 1234
Correction Plane (Left or Right)—use sketch	Left (plane)

RESIDUAL UNBALANCE POLAR PLOT



Rotor Rotation: ☒ CCW ☐ CW

Phase is layed out: ☐ CCW ☒ CW

Calculate *Y* and *Z* values:

Maximum amplitude value is:		3.00	g	Minimum amplitude value is:			1.11	g
Y = (Maximum – Minimum)/2	(	3.00	–	1.11	) /2	=	0.9	
Z = (Maximum + Minimum)/2	(	3.00	+	1.11	) /2	=	2.1	

Residual Unbalance Left in Rotor =	(*TU*)	×	(*Y*)	/	(*Z*)	=		
SI Units:	781	×	0.95	/	2.06	=	359.3	g-mm
Customary Units:	31	×	0.95	/	2.06	=	14.1	g-in.

Allowable Unbalance Tolerance = U_{max} = 488.4 g-mm 19.2 g-in.

RESULT: Residual unbalance left in the rotor is equal to or less than the allowable unbalance tolerance? **PASS**

☐ As Received ☐ Final ☒ Other: w/o trim hardware

Balanced By:	CJ, TR & RC	Date:	05/24/00
Approved By:	CC	Date:	05/24/00

Figure G-4—Sample Residual Unbalance Polar Plot Work Sheet for Left Plane

Customer:	ABC Refining Co.
Job/Project Number:	00 – 1234
OEM Equipment S/N:	C – 1234
Rotor Identification Number:	1234 – C – 4320
Repair Purchase Order Number:	PO 12345678
Vendor Job Number:	Shop – 00 – 1234
Correction Plane (Left or Right)—use sketch	Right (plane)

Balancing Speed	800	(rpm)		
Maximum Rotor Operating Speed (N)	6900	(rpm)		
Static Journal Weight Closest to This Correction Plane (W)	571.5	(kg)	1260	(lb.)
Trial Weight Radius (R)—the radius at which the trial weight will be placed	203	(mm)	8	(in.)

Calculate Maximum Allowable Residual Unbalance (U_{max}):

Si Units:

$$U_{max} = \frac{(6350) \times (W)}{(N)} = \frac{(6350) \times 571.5}{6900} = \boxed{525.9} \text{ (g-mm)}$$

Customary Units:

$$U_{max} = \frac{(113.4) \times (W)}{(N)} = \frac{(113.4) \times 1260}{6900} = \boxed{20.7} \text{ (g-in.)}$$

Calculate the trial unbalance (TU) :

Trial Unbalance (TU) is between $(1 \times U_{max})$ and $(2 \times U_{max})$

	(1×)		(2×)		(Selected Multiplier is) 1.6
SI Units:	525.9	to	1051.9	is	841.5 (g-mm)
Customary units:	20.7	to	41.4	is	33.1 (g-in.)

Calculate the trial weight (TW):

$$\text{Trial Weight } (TW) = \frac{U_{max}}{R} = \frac{842 \text{ g-mm}}{203 \text{ mm}} \text{ or } \frac{33 \text{ g-in.}}{8 \text{ in.}} = \boxed{4.1} \text{ (g)}$$

Conversion Information:

1 kg = 2.2046 lb. 1 oz. = 28.345 g

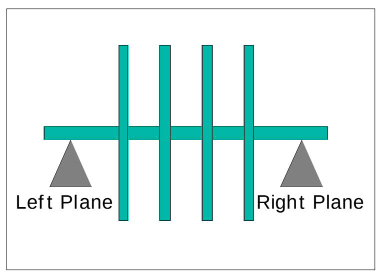
Obtain the test data and complete the table:

Test Data

Position	Trial Weight Angular Location on Rotor (degrees)	Balancing Mach Readout	
		Amplitude (grams)	Phase Angle (degrees)
1	0	4.60	3
2	60	4.20	58
3	120	4.70	121
4	180	5.20	180
5	240	5.80	235
6	300	5.10	301
Repeat 1	0	4.60	2

Sketch the rotor configuration:

Rotor Sketch



PROCEDURE:

Step 1: Plot the balancing machine amplitude versus trial weight angular location on the polar chart, (Figure G-2) such that the largest and smallest values will fit.

Step 2: The points located on the polar chart should closely approximate a circle. If it does not, then it is probably that the recorded data it is in error and the test should be repeated.

Step 3: Determine the maximum and minimum balancing machine amplitude readings.

Step 4: Using the worksheet, (Figure G-2), determine the Y and Z values required for the residual unbalance calculation.

Step 5: Using the worksheet, (Figure G-2), calculate the residual unbalance remaining in the rotor.

Step 6: Verify that the determined residual unbalance is equal to or less than the maximum allowable residual unbalance (U_{max}).

Half Keys Used for Rotor Balancing
(add sketch for clarification if necessary)

Location	Weight

NOTES:

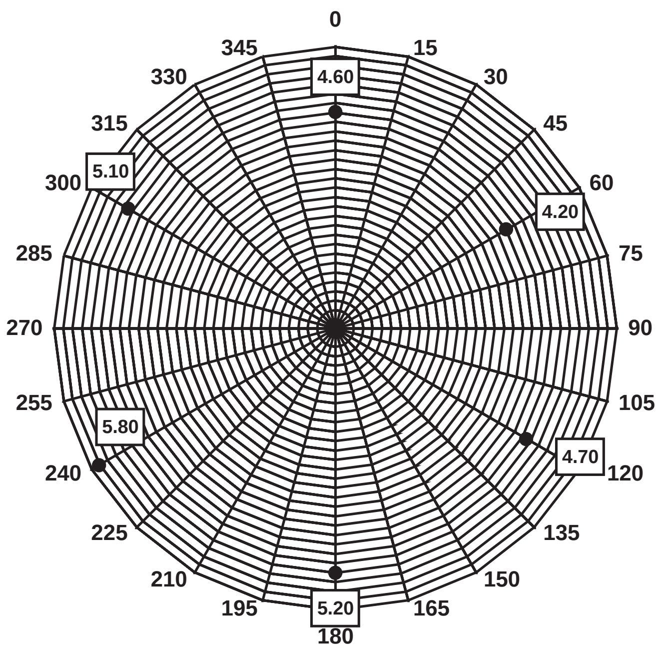
1) The trial weight angular location should be referenced to a keyway or some other permanent marking on the rotor. The preferred location is the location of the once-per-revolution mark (for the phase reference transducer).
2) The balancing machine amplitude readout for the Repeat of 1 should be the same as Position 1, indicating repeatability.
3) A primary source for error is not maintaining the same radius for each trial weight location.

Balanced By:	CJ, TR, & RC	Date:	05/24/00
Approved By:	CC	Date:	05/24/00

Figure G-5—Sample Residual Unbalance Work Sheet for Right Plane

Customer:	ABC Refining Co.
Job/Project Number:	00 – 1234
OEM Equipment S/N:	C – 1234
Rotor Identification Number:	1234 – C – 4320
Repair Purchase Order Number:	PO 12345678
Vendor Job Number:	Shop – 00 – 1234
Correction Plane (Left or Right)—use sketch	Right (plane)

RESIDUAL UNBALANCE POLAR PLOT



Rotor Rotation: ☒ CCW ☐ CW Phase is layed out: ☐ CCW ☒ CW

Calculate Y and Z values:

Maximum amplitude value is:	5.80		Minimum ampltude value is:			4.20	g
Y = (Maximum – Minimum)/2 (	5.80	–	4.20	) /2	=	0.8	
Z = (Maximum + Minimum)/2 (	5.80	+	4.20	) /2	=	5.0	

Residual Unbalance Left in Rotor =	(TU)	×	(Y)	/	(Z)			
SI Units:	842	×	0.8	/	5	=	134.6	g-mm
Customary Units:	33	×	0.8	/	5	=	5.3	g-in.

Allowable Unbalance Tolerance = U_{max} = 525.9 g-mm 20.7 g-in.

RESULT: Residual unbalance left in the rotor is equal to or less than the allowable unbalance tolerance? PASS

☐ As Received ☐ Final ☒ Other: w/o trim hardware

Balanced By:	CJ, TR & RC	Date:	05/24/00
Approved By:	CC	Date:	05/24/00

Figure G-6—Sample Residual Unbalance Polar Plot Work Sheet for Right Plane

Customer:
Job / Project Number:
OEM Equipment S / N:
Rotor Identification Number:
Repair Purchase Order Number:
Vendor Job Number:
Correction Plane (Left or Right) - use sketch (plane)

Balancing Speed (rpm)
Maximum Rotor Operating Speed (N) (rpm)
Static Journal Weight Closest to This Correction Plane (W) (lbs)
Trial Weight Radius (R) - the radius at which the trial weight will be placed (in)

Calculate Maximum Allowable Residual Unbalance (Umax):

Customary Units:

Umax = $\frac{(113.4) \times (W)}{(N)}$ = $\frac{(113.4) \quad X \qquad 0}{0}$ = ____ (g-in)

Calculate the trial unbalance (TU):
Trial Unbalance (TU) is between (1 X Umax) and (2 X Umax) (1 X) to (2 X) (Selected Multiplier is) ____
Customary units: to is 0.0 (g-in)

Calculate the trial weight (TW):

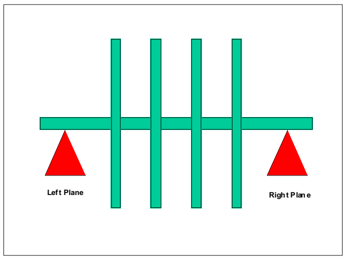
Trial Weight (TW) = $\frac{\text{Umax}}{\text{R}}$ = $\frac{0 \text{ g-in}}{0 \text{ in}}$ = ____ (g)

Conversion Information:
1kg = 2.2046 lbs **1 ounce = 28.345 grams**

Sketch the rotor configuration:

Obtain the test data and complete the table:

Rotor Sketch



Test Data

Position	Trial Weight Angular Location on Rotor (degrees)	Balancing Mach Readout: Amplitude (grams)	Balancing Mach Readout: Phase Angle (degrees)
1	0		
2	60		
3	120		
4	180		
5	240		
6	300		
Repeat 1	0		

HALF KEYS USED FOR ROTOR BALANCING
(add sketch for clarification if necessary)

Location	Weight

PROCEDURE:

Step 1: Plot the balancing machine amplitude versus trial weight angular location on the polar chart (Figure G-2) such that the largest and smallest values will fit.

Step 2: The points located on the Polar Chart should closely approximate a circle. If it does not, then it is probabloy that the recorded data it is in error and the test should be repeated.

Step 3: Determine the maximum and minimum balancing machine amplitude readings .

Step 4: Using the worksheet, (Figure G-2), determine the Y and Z values required for the residual unbalance calculation.

Step 5: Using the worksheet, (Figure G-2), calculate the residual unbalance remaining in the rotor.

Step 6: Verify that the determined residual unbalance is equal to or less than the maximum allowable residual unbalance (Umax).

NOTES:

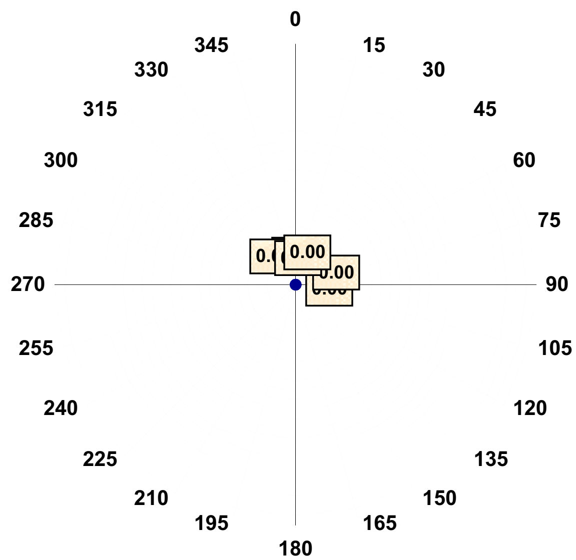
1) The trial weight angular location should be referenced to a keyway or some other permanent marking on the rotor. The preferred location is the location of the once-per-revolution mark (for the phase reference transducer).
2) The balancing machine amplitude readout for the Repeat of 1 should be the same as Position 1, indicating repeatability.
3) A primary source for error is not maintaining the same radius for each trial weight location.

Balanced By: ____ **Date:** ____
Approved By: ____ **Date:** ____

Figure G-7—Work Sheet—Customary

Customer:	**0**
Job / Project Number:	**0**
OEM Equipment S / N:	**0**
Rotor Identification Number:	**0**
Repair Purchase Order Number:	**0**
Vendor Job Number:	**0**
Correction Plane (Left or Right) - use sketch	**0** **(plane)**

RESIDUAL UNBALANCE POLAR PLOT



Rotor Rotation: ☐ **CCW** ☐ **CW** **Phase is layed out:** ☐ **CCW** ☐ **CW**

Calculate Y and Z values:

Maximum amplitude value is:	**0.00**	**grams**	**Minimum ampltude value is:**			**0.00**	**grams**
Y = (Maximum - Minimum) / 2 (	**0.00**	**-**	**0.00)**	**/ 2**	**=**	**0.0**	
Z = (Maximum + Minimum) / 2 (	**0.00**	**+**	**0.00)**	**/ 2**	**=**	**0.0**	

Residual Unbalance

Left in Rotor =	**(TU)**	**X**	**(Y)**	**/**	**(Z)**			
		X	**0**	**/**	**0**	**=**		**g-in**

Allowable Unbalance Tolerance = Umax = ______ **g-in**

RESULT: **Residual unbalance left in the rotor is equal to or less than the allowable unbalance tolerance?**

☐ **As Received** ☐ **Final** ☐ **Other: w/o trim hardware**

Balanced By:		**Date:**	
Approved By:		**Date:**	

Figure G-8—Polar Plot—Customary

Customer:
Job / Project Number:
OEM Equipment S / N:
Rotor Identification Number:
Repair Purchase Order Number:
Vendor Job Number:
Correction Plane (Left or Right) - use sketch (plane)

Balancing Speed (rpm)
Maximum Rotor Operating Speed (N) (rpm)
Static Journal Weight Closest to This Correction Plane (W) (kg)
Trial Weight Radius (R) - the radius at which the trial weight will be placed (mm)

Calculate Maximum Allowable Residual Unbalance (Umax):
Si Units:

$$\text{Umax} = \frac{(6350) \times (W)}{(N)} = \frac{(6350) \times 0}{0} = \quad \text{(g-mm)}$$

Calculate the trial unbalance (TU):
Trial Unbalance (TU) is between (1 X Umax) and (2 X Umax) (1 X) to (2 X) (Selected Multiplier is)
SI Units: 0.0 to 0.0 is 0.0 (g-mm)

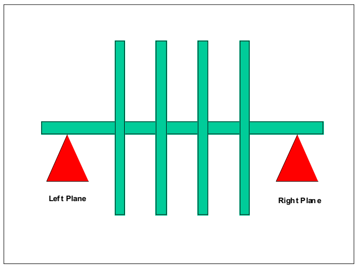
Calculate the trial weight (TW):

$$\text{Trial Weight (TW)} = \frac{\text{Umax}}{R} = \frac{0 \text{ g-mm}}{0 \text{ mm}} = \quad \text{(g)}$$

Conversion Information:
1kg = 2.2046 lbs   1 ounce = 28.345 grams

Sketch the rotor configuration:

Rotor Sketch



Obtain the test data and complete the table:

Test Data

Position	Trial Weight Angular Location on Rotor (degrees)	Balancing Mach Readout: Amplitude (grams)	Balancing Mach Readout: Phase Angle (degrees)
1	0		
2	60		
3	120		
4	180		
5	240		
6	300		
Repeat 1	0		

HALF KEYS USED FOR ROTOR BALANCING
(add sketch for clarification if necessary)

Location	Weight

PROCEDURE:

Step 1: Plot the balancing machine amplitude versus trial weight angular location on the polar chart (Figure G-2) such that the largest and smallest values will fit.
Step 2: The points located on the Polar Chart should closely approximate a circle. If it does not, then it is probabloy that the recorded data it is in error and the test should be repeated.
Step 3: Determine the maximum and minimum balancing machine amplitude readings .
Step 4: Using the worksheet, (Figure G-2), determine the Y and Z values required for the residual unbalance calculation.
Step 5: Using the worksheet, (Figure G-2), calculate the residual unbalance remaining in the rotor.
Step 6: Verify that the determined residual unbalance is equal to or less than the maximum allowable residual unbalance (Umax).

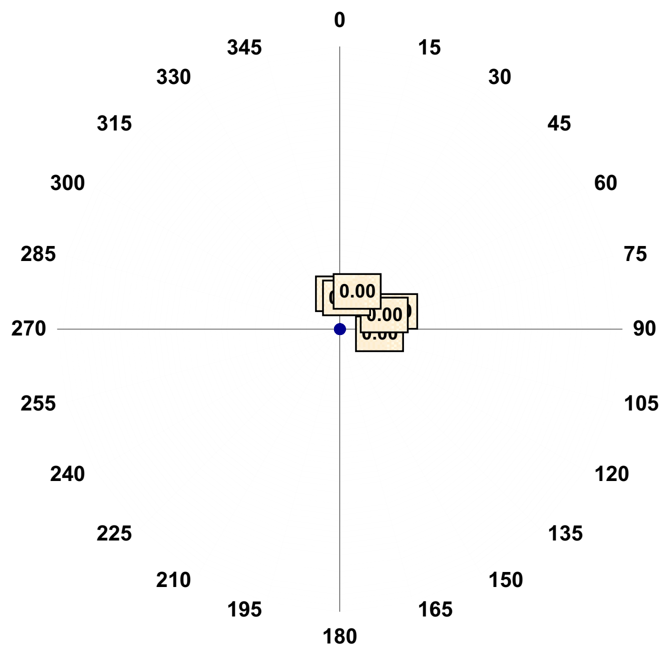
NOTES:
1) The trial weight angular location should be referenced to a keyway or some other permanent marking on the rotor. The preferred location is the location of the once-per-revolution mark (for the phase reference transducer).
2) The balancing machine amplitude readout for the Repeat of 1 should be the same as Position 1, indicating repeatability.
3) A primary source for error is not maintaining the same radius for each trial weight location.

Balanced By: **Date:**
Approved By: **Date:**

Figure G-9—Work Sheet—SI

Customer:	**0**
Job / Project Number:	**0**
OEM Equipment S / N:	**0**
Rotor Identification Number:	**0**
Repair Purchase Order Number:	**0**
Vendor Job Number:	**0**
Correction Plane (Left or Right) - use sketch	**0** **(plane)**

RESIDUAL UNBALANCE POLAR PLOT



Rotor Rotation: ☐ **CCW** ☐ **CW** **Phase is layed out:** ☐ **CCW** ☐ **CW**

Calculate Y and Z values:

Maximum amplitude value is:	0.00	**grams**	**Minimum ampltude value is:**			**0.00** **grams**
Y = (Maximum - Minimum) / 2 (	0.00	**-**	**0.00** **)**	**/ 2**	**=**	**0.0**
Z = (Maximum + Minimum) / 2 (	0.00	**+**	**0.00** **)**	**/ 2**	**=**	**0.0**

Residual Unbalance

Left in Rotor =	**(TU)**	**X**	**(Y)**	**/**	**(Z)**			
		X	**0.0**	**/**	**0.0**	**=**		**g-mm**

Allowable Unbalance Tolerance = Umax = ______ **g-mm**

RESULT: **Residual unbalance left in the rotor is equal to or less than the allowable unbalance tolerance?**

☐ **As Received** ☐ **Final** ☐ **Other:** **w/o trim hardware**

Balanced By: **Date:**
Approved By: **Date:**

Figure G-10—Polar Plot—SI

APPENDIX H—GEAR INSPECTION (INFORMATIVE)

H.1 General

H.1.1 ANSI/GMA ISO 1328-1 *Cylindrical Gears - ISO System of Accuracy - Part 1: Definitions and Allowable Values of Deviations Relevant to Corresponding Flanks of Gear Teeth* and ISO/TR 10064-1 *Recommendations relative to gear blanks, shaft center distance and parallelism of axes*, describes gear measuring methods and is a general summary of the different procedures used. Many of these procedures may not be applicable due to manufacturing methods and measuring equipment available.

H.1.2 Gears meeting API Std 613 generally have large diameters and wide face width and by necessity are manufactured in matched sets with the tolerances in terms of mismatch between the contacting tooth surfaces.

H.1.3 Limits on deviations given in ANSI/AGMA ISO 1328-1 apply to each element and do not cover gears pairs as such. Actual matched gear set accuracy depends on the mismatch as discussed in H.1.2. Individual elements may meet the accuracy requirements and still not be suitable for service. Conversely individual elements may not meet accuracy requirements but with favorable mis-match the gear set may be suitable. The vendor and the purchaser should be prepared to discuss the actual acceptance values for each element to achieve the desired gear set accuracy.

H.1.4 The measuring methods described in this appendix cannot be used to replace tooth contact checking procedures used to verify the gear tooth fit in the job casing in the vendor's shop as described in 2.5.2.2 and the contact checking after field installation and alignment.

H.2 Double Helical Gears

The measuring methods described in ANSI/AGMA ISO 1328 and ISO/TR 10064-1 are for single helical or spur gears.

H.2.1 Double helical gears are two single helical gears on the same rotor. These gears shall require data for both the right-hand and left-hand helixes.

H.2.2 When performing tooth contact inspections, it is necessary that the pinion be allowed to move axially to obtain true tooth contact patterns.

H.2.3 Apex runout measurements are required in accordance with 2.5.2.3 for double helical gears.

H.3 Modified Tooth Flanks

H.3.1 LEAD MODIFICATION

H.3.1.1 Every gear under loaded operating conditions is subject to deformation of the entire rotor, in three ways, bending deflection, torsional windup and thermal distortion of the rotor. All three influence the tooth form under operating conditions.

Increased pitch line velocities especially above 140 m/s or 28,000 fpm, result in considerable greater windage & mesh losses which increase the thermal tooth distortion along the flank of the gear teeth. The two variables which most influence the gear set rating are the center distance and the face width of the rotors. If the gear designer elects to reduce the center distance in an attempt to reduce the pitch line velocity and accordingly the windage and mesh losses, he is expected to recover the rating required by increasing the face width.

Increasing the face width increases the bearing span, which may result in a less rigid rotor than one where the support bearings are closer together. At the same time a torsional windup of the gear set rotors also takes place. Longer face widths also increase the thermal distortion of the rotors as the gear lubricant is sprayed onto the rotors and pumped along the rotor flanks to the tooth end(s). The accumulated distortion increases with increased bearing span.

In order to obtain an even load distribution across the entire face width when transmitting the rated power, the pinion teeth should be manufactured with a lead modification having the shape of the inversion of the combined deflection. The total lead modification is the superposition of the modification for the mechanical deflections, bending & torsional and the modification for the thermal distortion. The typical shape of such a modified lead is shown in Figure H-1.

Note: A method for reducing, but not eliminating, the effects of thermal distortion is through effective (non-uniform) oil spray distribution.

H.3.1.2 Working flanks provided with such lead modifications should not be used to align a gear, when checked under no load, if the blue tooth contact pattern is too short. When the predicted contact pattern is too short, the non-working flanks of the pinion and gear wheel should be ground without correction (parallel) and used as a basis for correct alignment of the gears. Purchaser and Vendor shall agree on the patterns obtained in the housing and test stand.

Note: In cases when the *L/d* ratio is less than shown in Table 3 and the total lead mismatch is less than the limitations prescribed in 2.2.4.5, lead modifications may be made to improve contact pattern under load.

H.3.2 PROFILE MODIFICATION

H.3.2.1 To prevent engagement shocks due to tooth bending deformations on the working flanks of the pinion and/or gear wheel, the rotors are manufactured with profile modifications to obtain an even trapezoidal load distribution along the path of contact in the transverse section as shown in Figure H-2.

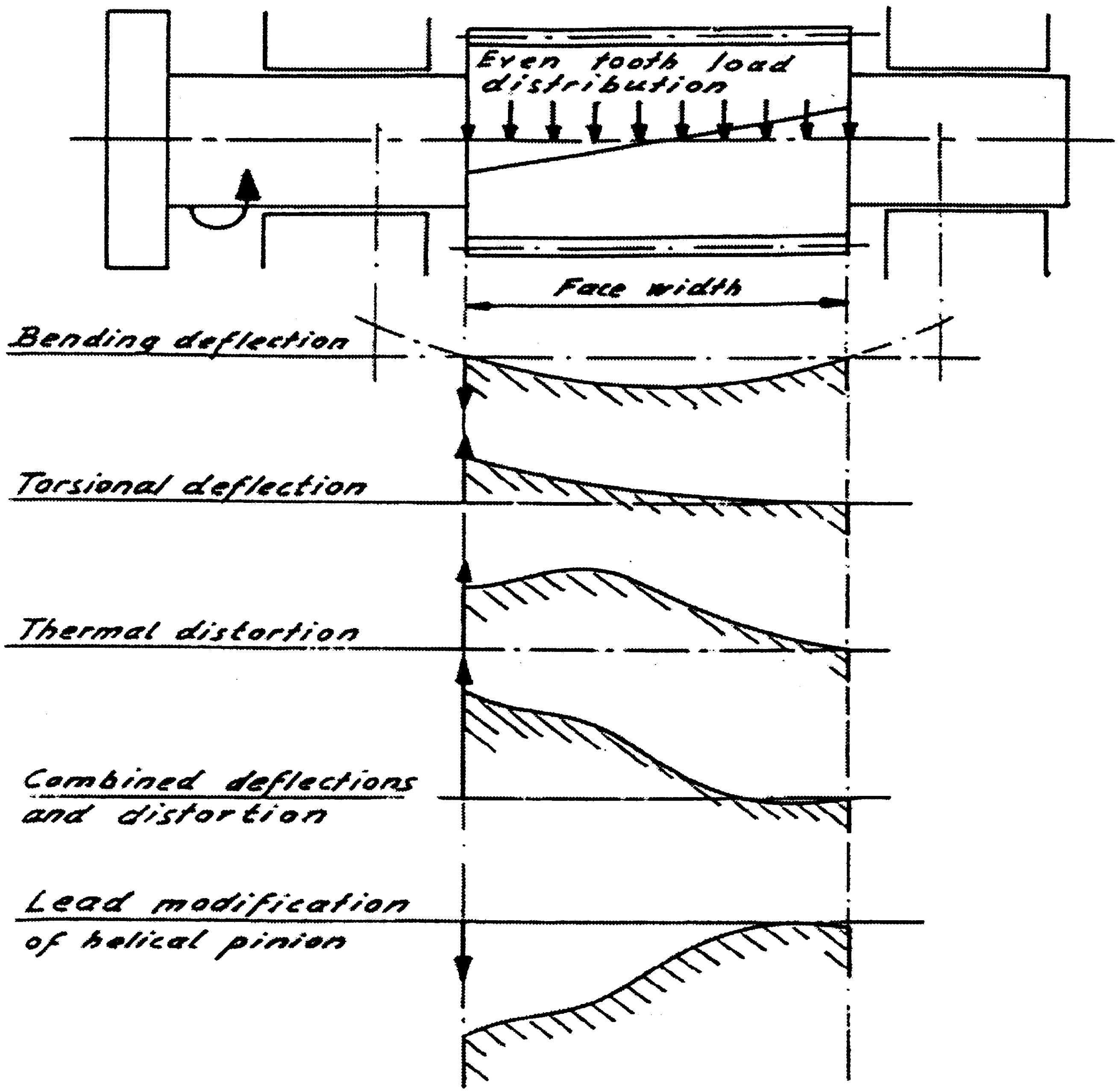


Figure H-1—Tooth Alignment (Lead) Modification of Helical Pinion

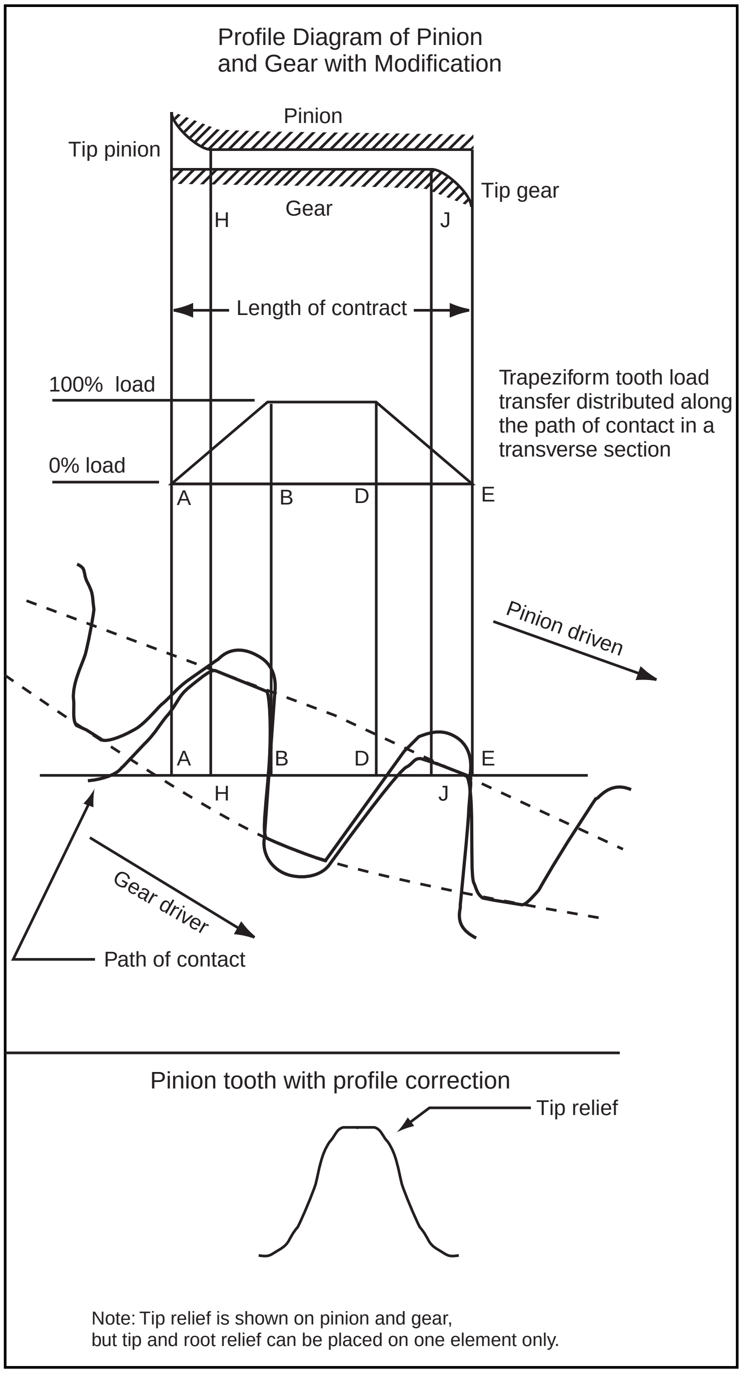


Figure H-2—Profile Modification

APPENDIX I—INSPECTOR'S CHECKLIST

Item	Inspection Agency	Inspector's Initials	Date	Extent W/O/*	API Std 613 Reference
1. Basic Design					
a. Special tools shipped with gear, such as bearing mandrel.	C			*	3.6.1 & 3.6.3
b. Hanging provisions and fixtures (if specified).	C			*	2.5.4.8 & 3.6.2
c. Shaft assembly in accordance with approved data sheets.	V			*	2.1.19
d. Shaft rotation arrows in accordance with approved general arrangement drawings and data sheets.	C			*	2.1.20.1 & 2.1.20.2
2. Casings					
a. Doweling provisions.	C			*	2.3.1.2
b. Internal piping.	C			*	2.3.1.7
c. Piping and tubing.	V			*	2.3.1.8
d. Support of internal piping.	C			*	2.3.1.9
e. Cleanliness of internal piping.	C			*	2.3.1.10
f. Lube oil drainage.	C & V			*	2.3.1.11
g. Filter breather.	C			*	2.3.1.12
h. Inspection covers.	C			*	2.3.1.13
i. Interior coatings.	V			*	2.3.1.14
j. Sealing of horizontal joint.	C			*	2.3.2
k. All doweling of parts.	C			*	2.3.4.2
3. Bolting					
a. Through bolting.	V			*	2.3.3.1
b. Studded connections.	V			*	2.3.3.3
c. Bolting clearance.	C			*	2.3.3.4
4. Assembly and Disassembly					
a. Oil piping Configuration.	V & C			*	2.3.4.1
b. Jackscrews, relieved surfaces, and lifting lugs (w/ warning labels as required).	C			*	2.3.4.2
5. Casing Connections					
a. Single lube oil supply.	V			*	2.4.1
b. Single lube oil drain and its size.	V			*	2.4.2
c. Inlet purge connection (when specified).	C			*	2.4.3
d. Unacceptable pipe sizes.	V			*	2.4.4
e. Accessibility of customer connections.	C			*	2.4.5
f. Drain and oil connection size and location per latest approved general assembly drawing.	C			*	2.4.6
g. Installation of threaded connections.	C			*	2.4.6.1
h. Flanges in accordance with ISO 7005-1 or ISO 7005-2 (ASME B 16.1, B 16.5, B 16.42 or SAE J518).	C			*	2.4.7
i. Studs installed.	V & C			*	2.4.8
j. Threaded connections.	C			*	2.4.9
k. Tapped openings not connected to piping.	C			*	2.4.10
6. Gear Elements					
a. Gear tooth surface finish.	V & C			W	2.5.1.2
7. Critical Speeds and Balancing					
a. Critical speeds.	V			*	2.6.2.4
b. Rotor balancing.	C			W	2.6.6.1
c. Rotor balance machine calibration.	C			*	2.6.6.2
d. Coupling balance.	V			W	API Std 671
e. Mechanical and electrical runout (in vee blocks).	C			W	2.6.6.6
f. Gauss levels.	C			W	API Std 670
8. Bearings					
a. Anti-rotation pins and axially secured.	V			*	2.7.1.2 & 2.7.2.1
b. Bearing fitted with RTDs.	V & C			*	2.7.1.3

c. Bearing white metal (journal and thrust).	C			W	2.7.2.1 & 2.7.3.2
d. Integral thrust collar stock.	V			*	2.7.3.4
e. Thrust collar finish and runout.	C			W	2.7.3.5
9. Vibration, Position and Acceleration Detectors					
a. Installed in accordance with API Std 670.	V & C			*	3.4.2.1 through 3.4.2.4
10. Materials					
a. Review vendor in-house quality control checks on materials and manufacturing methods concerning gears, pinions, bearings.	C			*	2.5.2
b. Review manufacturer's in-house quality control rejects with written explanation of rejects pieces disposition.	C			*	
c. No major repairs to be made without customer approval.	V & C			W	2.9.2.3
11. Nameplate versus Data Sheets					
a. Provide nameplate rubbing.	C			*	2.10
12. Mounting Plates					
a. Plated vertical jackscrews or alternate jacking facility.	C			*	3.3.1.2.2 & 3.3.1.2.3
b. Plated horizontal jackscrews.	C			*	3.3.1.2.2
c. Preparation of mounting plates for epoxy grout.	V & C			O	3.3.1.2.4
d. Mounting bolts.	C			*	3.3.1.2.9
e. Gear dowel holes.	C			*	2.3.1.2
f. Stainless steel shim packs per API 686.	C			*	3.3.1.2.7
g. Foundation bolt hole location versus item 1.e, Appendix F.	C			*	VDDR, Item 1.e
13. Controls and Instrumentation					
a. Instrumentation supplied against approved data sheets.	C			*	3.4, Appendix A
14. Piping and Appurtenances					
a. Site flow glasses.	C			*	3.5
15. Inspection					
a. Mill test reports[d] for all gear element components.	C & V			*	4.2.1.1.a
b. UT of all gear element components after rough machinery.	C & V			W	4.2.2.7.2
c. Record of all heat treatment and resulting hardnesses.	C & V			*	4.2.2.6.1.b
d. Records of all radiographs and UT inspections.	C & V			W	4.2.2.6.1.c
e. Hardness versus case depth of teeth.	C & V			W	2.2.3.2 4.2.2.6.1.d
f. Hardness of through hardened gears.	C & V			W	2.2.3.2 4.2.2.6.1.e
g. Stress relieve of casing.	V			*	2.9.3.2
h. Hardness of welds and heat effected zones on gear elements.	V & C			*	4.2.3.3
i. UT or MP inspection of all welds on rotating elements.	V & C			W	4.2.2.6.2
j. MP of gear and pinion teeth.	V & C			W	4.2.2.6.3
k. Purchase specification[e].	V			*	4.2.1.1.b
l. Maintenance and running clearance data (if specified).	V			*	4.2.1.1.e
m. Results of quality control checks.					2.5.2 & 4.2.1.1.d
1. Gear tooth accuracy check.	V			*	2.5.2.1
2. Check stand contact.	V & C			W	2.5.2.2
3. Contact check in job casing.	V & C			W	2.5.2.2
4. Axial stability.	V & C			O	2.5.2.3
n. Setting of thrust bearing.	V & C			W	
o. Cleanliness of equipment including lube oil system prior to running any tests; cleanliness per API Std 614.				O	4.3.2.1.3
p. Final assembly clearances (journal bearing, thrust bearing, etc.).				W	VDDR Item 26 4.2.1.1.e
16. Special Tools					
a. Separately packaged and labeled with item number.				*	3.6.3
b. Each tool tagged for intended use.				*	3.6.3
c. Hanger fixtures (if specified).				W	3.6.2
17. Testing					
a. Check alignment prior to running any tests.	C			W	

b. Mechanical running test.	C			W	4.3.2
c. 25, 50, 75 and 100 % full torque slow roll tests, optional tests.	C			W	4.3.3
d. 110% maximum continuous speed test.	C			W	4.3.2.2.6
e. Critical speed rundown check from 120%.	C			W	4.3.2.2.5
f. Check position of lube oil sprays.	C			W	2.8.1
18. Preparation for Shipment					
a. Painting.	C			*	4.4.3.1
b. Rust prevention.	C			*	4.4.3.2
c. Interior of gear preserved.	C			*	4.4.3.3 & 4.4.3.4
d. Metal closures.	C			*	4.4.3.5
e. Car sealing of metal closure prior to shipment.	C			*	4.4.3.5
f. Threaded opening capped or plugged.	C			*	4.4.3.6
g. Lifting points clearly marked.	C			*	4.4.3.7
h. Exposed shafts and couplings wrapped with waterproof moldable wound cloth or VPI inhibitor paper and seams sealed with oil proof adhesive tape.	C			*	4.4.3.10
i. Shipping container and rotor blocking.	C			*	4.4.1
j. Identification of items and preservation details.	C			*	4.4.3.8
k. Spare elements storage container and preservation.	C			*	4.4.3.9 & 4.4.5
l. Installation instructions.	C			*	4.4.6
m. Bill of lading for each packaged box against box contents.	C			*	5.3.5

Note: W = witnessed; O = observed; * = designated inspection agency is to confirm that the requirement is satisfied; V = vendor; C = contractor; VDDR = vendor drawing and data requirements; UT = ultrasonic inspection; MP = wet magnetic particle inspection.

APPENDIX J—RATING COMPARISON API 613 VS. AGMA 2101

J.1 General

J.1.1 This appendix describes how the rating formulas in ANSI/AGMA 2101-C95 are related to the rating methods in API Std 613 fifth edition. For definitions of terms refer to these two standards.

J.2 Tooth Pitting Index

J.2.1 Equation 27 from ANSI/AGMA 2101-C95 for calculating pitting resistance power rating is shown as Equation J1 below.

$$P_{azu} = \frac{\omega_1 b}{1.91 \times 10^7} \frac{Z_1}{K_V K_S K_H Z_R} \left(\frac{d_{w1} \sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \quad \text{(J1)}$$

J.2.2 There are terms in AGMA that do not precisely match API terms. They are:

AGMA		API	
u	=	R	Gear ratio
ω_1	=	N_p	Pinion speed
P_a	=	Pg	Gear rated power
P_{azu}	=	$SF \cdot P_g$	Mechanical rating
F_t	=	W_t	Tangential load
b	=	F_w	Face width
d_{w1}	=	d	Pinion pitch diameter

J.2.3 Equation J2 is a conservative and reasonably accurate approximation to the more complex method found ANSI/AGMA 2101-C95 for calculating the pitting geometry factor, Z_I. The API term for ratio, R, is used.

$$Z_1 = 0.225 \frac{R}{R+1} \quad \text{(J2)}$$

J.2.4 Using the simplified formula for Z_I, the API 613 terms, and rearranging Equation J1 algebraically results in Equation J3.

$$SFP_g = \frac{N_p d^2 F_w}{1.91 \times 10^7} \frac{0.225}{K_V K_S K_H Z_R} \left(\frac{R}{R+1} \right) \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \quad \text{(J3)}$$

J.2.5 The tangential load is related to power and speed as shown by Equation J4.

$$W_t = \frac{1.91 \times 10^7 P_g}{N_p d} \quad \text{(J4)}$$

J.2.6 Substituting tangential load, W_t, into Equation J3 and further rearranging gives.

$$\frac{W_t}{dF_w} \left(\frac{R+1}{R} \right) = \frac{0.225}{SFK_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \quad \text{(J5)}$$

Note: The terms to the left of the equal sign in Equation J5 is the K factor as defined in API Std 613 and the terms to the right is the material index number, I_m, divided by the service factor.

$$K = \frac{W_t}{dF_w} \left(\frac{R+1}{R} \right) \quad \text{(J6)}$$

$$\frac{I_m}{SF} = \frac{0.225}{SFK_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \quad \text{(J7)}$$

J.2.7 The service factor, SF, is applied to the material index number, I_m, to give an allowable factor, K_a.

$$K_a = \frac{I_m}{SF} \quad \text{(J8)}$$

J.2.8 ANSI/AGMA 2101-C95 calculates rating factors based on specific design considerations. However, API Std 613 has chosen to use conservative values for these factors and fix their numerical value. API uses the following values for the pitting rating factors:

$K_v = 1.1$	$K_s = 1.0$	$K_H = 1.3$	$Z_N = 0.68$
$Z_R = 1.0$	$Z_E = 190$	$Z_W = 1.0$	$Y_\theta = 1.0$

Note: In Equation J7 the product of the rating factors is a constant 2.0154×10^{-6} as shown in Equations J9, J10, and J11.

$$I_m = \frac{0.225}{K_V K_S K_H Z_R} \left(\frac{\sigma_{HP} Z_N Z_W}{Z_E Y_\theta} \right)^2 \quad \text{(J9)}$$

$$I_m = \frac{0.225}{1.1 \cdot 1.0 \cdot 1.3 \cdot 1.0} \left(\frac{\sigma_{HP} \cdot 0.68 \cdot 1.0}{190 \cdot 1.0} \right)^2 \quad \text{(J10)}$$

$$I_m = 2.0154 \times 10^{-6} (\sigma_{HP})^2 \quad \text{(J11)}$$

J.2.9 This constant shown in Equation J11 is multiplied by the allowable contact stress squared, σ_{HP}^2, and the result is the material index number, I_m, as shown in Table J-1.

Table J-1—Allowable Contact Stress Per API

Minimum Gear Hardness	API Material Index Number, I_m	API Contact Stress Number, σ_{HP}
302 HB†	1.38	827
321 HB†	1.53	872
341 HB†	1.70	918
363 HB†	1.89	969
90 HR15*	2.07	1013
58 Rc**	3.03	1227

†Through hardened *Nitrided **Carburized

J.3 Bending Stress Number

J.3.1 Equation 28 from ANSI/AGMA 2101-C95 for calculating bending strength power rating is shown as Equation J12.

$$P_{ayu} = \frac{w_1 d_{wl}}{1.91 \times 10^7 K_v} \frac{b m_t}{K_s} \frac{Y_J}{K_H K_B} \frac{\sigma_{FP} Y_N}{Y_\theta} \quad \text{(J12)}$$

J.3.2 Additional terms that are different in AGMA versus API standards are listed below. Note AGMA's use of the transverse module and API use of the normal module. (See J.2.1 also)

AGMA		API	
m_t	=	$m_n \div cos\,(\gamma)$	Module
P_{ayu}	=	$SF \cdot P_g$	Mechanical rating
β	=	γ	Helix angle
Y_J	=	J	Strength geometry factor

J.3.3 Using the API terms and rearranging Equation J12 algebraically results in Equation J13.

$$SFP_g = \frac{N_p d}{1.91 \cdot 10^7} \frac{F_w m_n J}{cos\,\gamma} \frac{\sigma_{FP} Y_N}{K_V K_S K_H K_B Y_\theta} \quad \text{(J13)}$$

J.3.4 Substituting tangential load, W_t, into Equation J13 and further rearranging gives:

$$\left[\frac{W_t}{m_n F_w}\right](SF)\left[\frac{cos\,\gamma}{J}\right]\frac{K_V K_S K_H K_B Y_\theta}{Y_N} = \sigma_{FP} \quad \text{(J14)}$$

J.3.5 ANSI/AGMA 2101-C95 calculates rating factors based on specific design considerations. However, API Std 613 has chosen to use conservative values for these factors and fix their numerical value. API uses the following values for the strength rating factors:

$K_V = 1.1$	$K_S = 1.0$	$K_H = 1.3$
$Y_\theta = 1.0$	$Y_N = 0.80$	$K_B = 1.0$

Note: In Equation J14, the product of the rating factors is very close to a value of 1.8.

$$1.8 \cong \frac{K_V K_S K_H K_B Y_\theta}{Y_N} \cong \frac{1.1 \cdot 1.0 \cdot 1.3 \cdot 1.0 \cdot 1.0}{0.8} \quad \text{(J15)}$$

J.3.6 Using the value shown in Equation J15 the terms to the left of the equal sign in Equation J14 is the bending stress number, *S*, as defined in API Std 613.

$$S = \left[\frac{W_t}{m_n F_w}\right](SF)\left[\frac{1.8\,cos\,\gamma}{J}\right] \quad \text{(J16)}$$

J.3.7 To arrive at the allowable bending stress number, S_a, API613 has applied an additional 25% reduction in bending stress number, σ_{FP}. This is done to increase the strength rating in relation to pitting resistance. While tooth surface damage can in many cases be detected before complete failure allowing time for scheduling repair, tooth bending failure usually leads to an immediate shutdown of the unit. The values used are shown in Equation J17 and Table J-2.

$$S_a = 0.75\,\sigma_{FP} \quad \text{(J17)}$$

Table J-2—Allowable Bending Stress Peer API

Minimum Gear Hardness	API Allowable Bending Stress Number, S_a	PI Bending Stress Number, σ_{FP}
302 HB†	180	240
321 HB†	191	255
341 HB†	203	270
363 HB†	215	287
90 HR15*	190	253
58 Rc**	266	354

†Through hardened *Nitrided **Carburized

J.4 AGMA Allowable Stress Numbers

J.4.1 ANSI/AGMA 2101-C95 gives allowable stresses, σ_{HP} and σ_{FP}, based on a material "grade" and a better grade of material allows higher stresses. API Std 613 specifies the use of higher grade materials, but to be conservative bases allowable stress on the lowest grade. Both API and AGMA specifications have been revised since the original values were chosen so there is some divergence in the values today. Therefore, the API values are close to but not exactly equal to ANSI/AGMA 2101-C95 "Grade 1" values. As a comparison, Table J-3 shows the AGMA Grade 1 allowable stress numbers. These may be compared to the values chosen by API and shown in Table J-1 and J-2.

Table J-3—AGMA Grade 1 Allowable Stress Numbers

Minimum Gear Hardness	AGMA Contact Stress Number, σ_{HP}	AGMA Bending Stress Number, σ_{FP}
302 HB†	870	249
321 HB†	906	259
341 HB†	950	270
363 HB†	1006	282
90 HR15*	1070	—
58 Rc**	1240	380

†Through hardened *Nitrided **Carburized

APPENDIX K—SHAFT END SIZING METHOD

K.1 General

K.1.1 This appendix describes the maximum torsional stress allowed in steel shaft ends where couplings are attached. Where shafts fail to meet the criteria of this appendix a more precise method as agreed to by the vendor and purchaser shall be used (such as ANSI/AGMA 6001-D97). The vendor shall provide calculations and supporting data necessary for the purchaser to evaluate the design.

K.1.2 The general equations in this appendix for calculated and allowable stress do not separately consider the effects of shaft size, surface finish, operating temperature, corrosion, residual stresses, reliability, notch sensitivity and stress concentration, maximum cyclic to mean stress ratio, number of stress cycles, and bending stress. Rather, they are combined into a single, simple calculated torsional stress equation and associated conservative allowable torsional stress values. If there is reason to believe a particular application is not suitable for this simplified method, then other methods as discussed in paragraph K.1.1 must be used.

K.1.3 The stress concentration factor for torsion, including notch sensitivity, at such locations as key joints, shoulders, grooves, splines and interference fits shall not be greater than 3.0. Where design constraints require the stress concentration factor be greater than 3.0, then other methods as discussed in paragraph K.1.1 must be used.

K.2 Torsional shaft stress criteria

K.2.1 Nominal shaft torsional stress is calculated using Equation K1 or K2 as applicable. Where the ratio (d_i/d_o) > 0.9, the shafts are thin wall and other methods as discussed in paragraph K.1.1 must be used. Torque, T, shall be based on the *gear rated power* and the rated speed of the shaft. (See section 1.4 in main body of standard.)

In SI units:

$$S_S = \frac{16000\,T d_o}{\pi(d_o^4 - d_i^4)} \qquad \text{(K1)}$$

In U.S. customary units:

$$S_S = \frac{16\,T d_o}{\pi(d_o^4 - d_i^4)} \qquad \text{(K2)}$$

where

s_S = calculated torsional shear stress, N/mm^2 (lb/in.^{2}),

T = shaft torque, Nm (lb/in.),

d_o = shaft outside diameter, mm (in.),

d_i = shaft inside diameter, mm (in.).

K.2.2 The allowable torsional shear stress for through hardened steel shafts is calculated using Equations K3 or K4 as applicable. The allowable stress for steel shafts that are hardened by processes such as case carburizing or nitriding should be based on the core hardness of the material.

In SI units:

$$S_{sa} = \frac{(-0.07HB^2 + 65HB - 5000)}{145} \qquad \text{(K3)}$$

In U.S. customary units:

$$S_{sa} = -0.07HB^2 + 65HB - 5000 \qquad \text{(K4)}$$

where

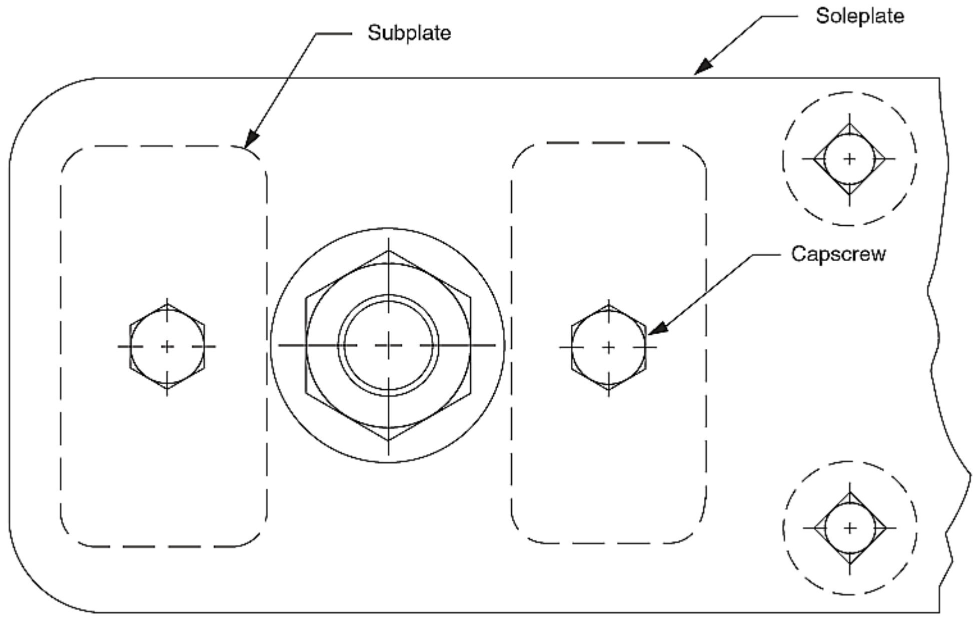
s_{sa} = allowable torsional shear stress, N/mm^2 (lb/in.^{2}),

HB = Brinell Hardness of the shaft.

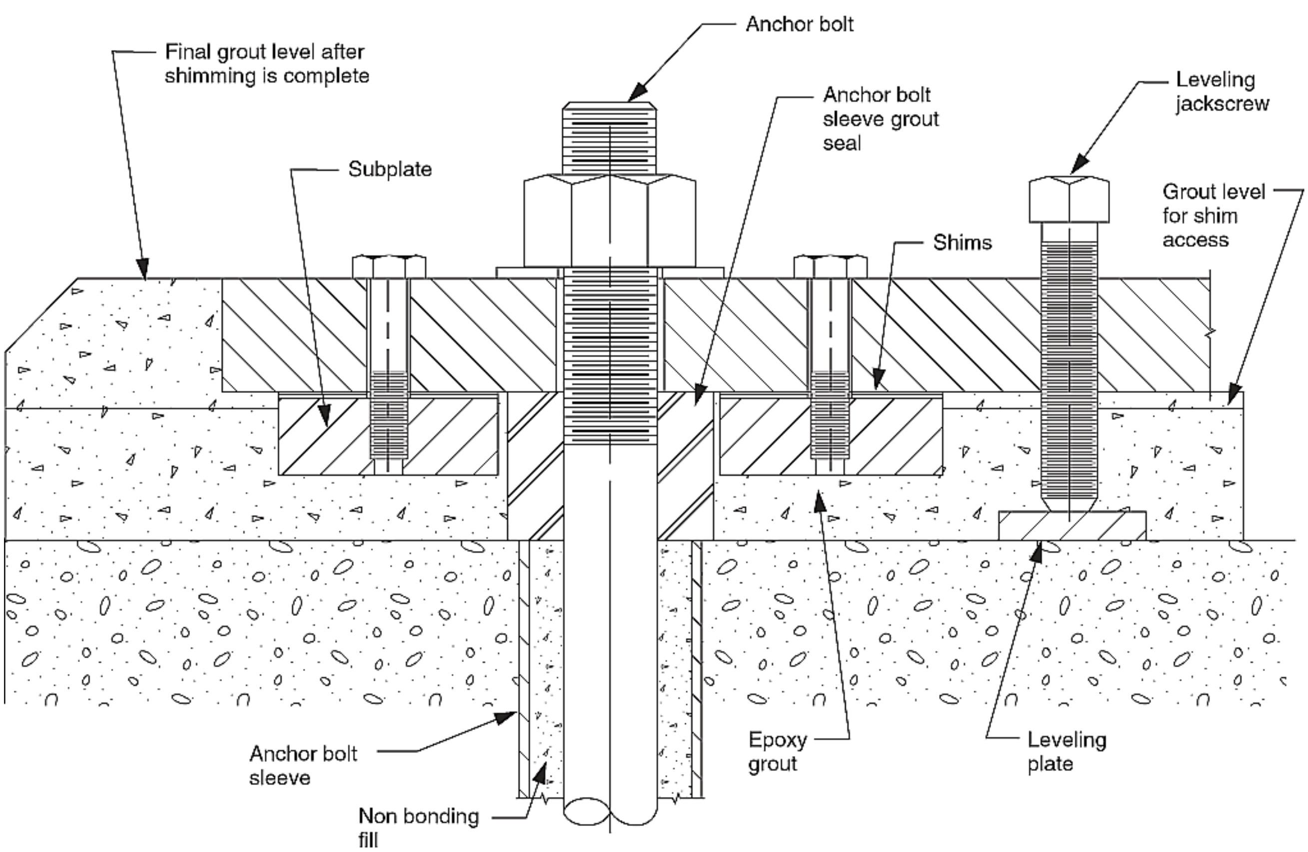
K.2.3 For steel shafts, the calculated stress due to torsion shall not exceed the allowable stress for torsion as shown in Equation K5.

$$S_s \le S_{sa} \qquad \text{(K5)}$$

APPENDIX L—TYPICAL MOUNTING PLATES

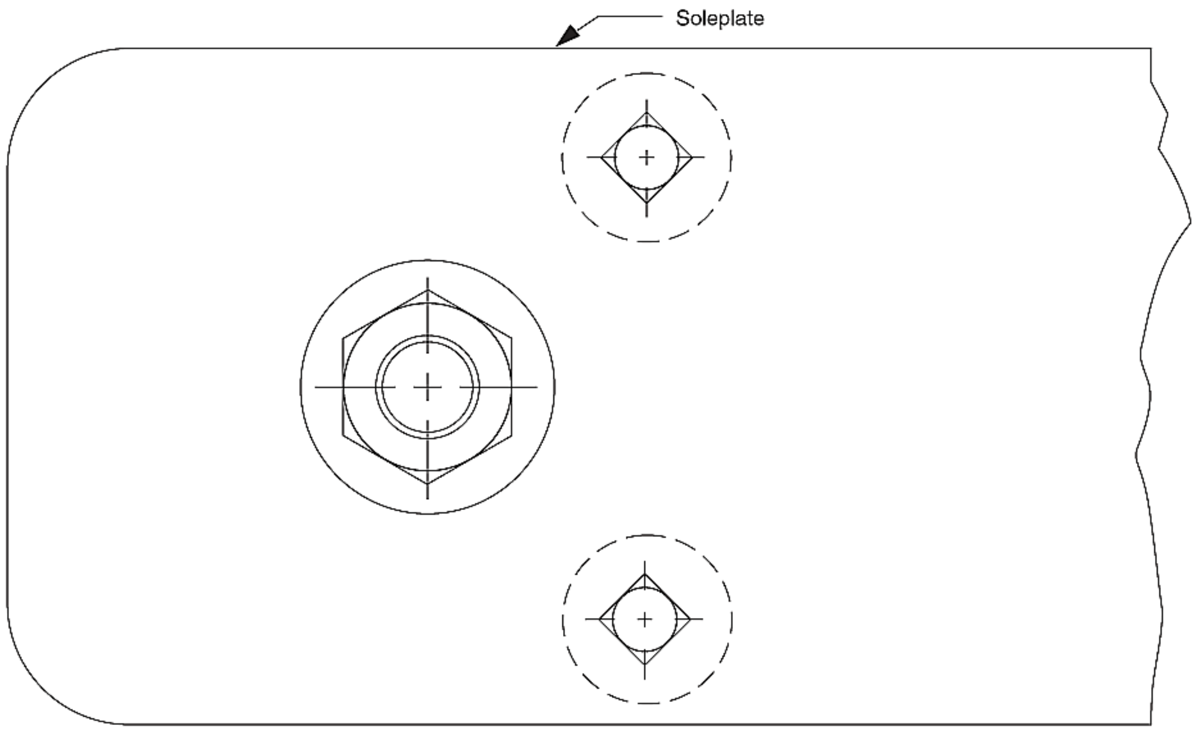


TOP VIEW OF FOUNDATION AT FOUNDATION BOLT

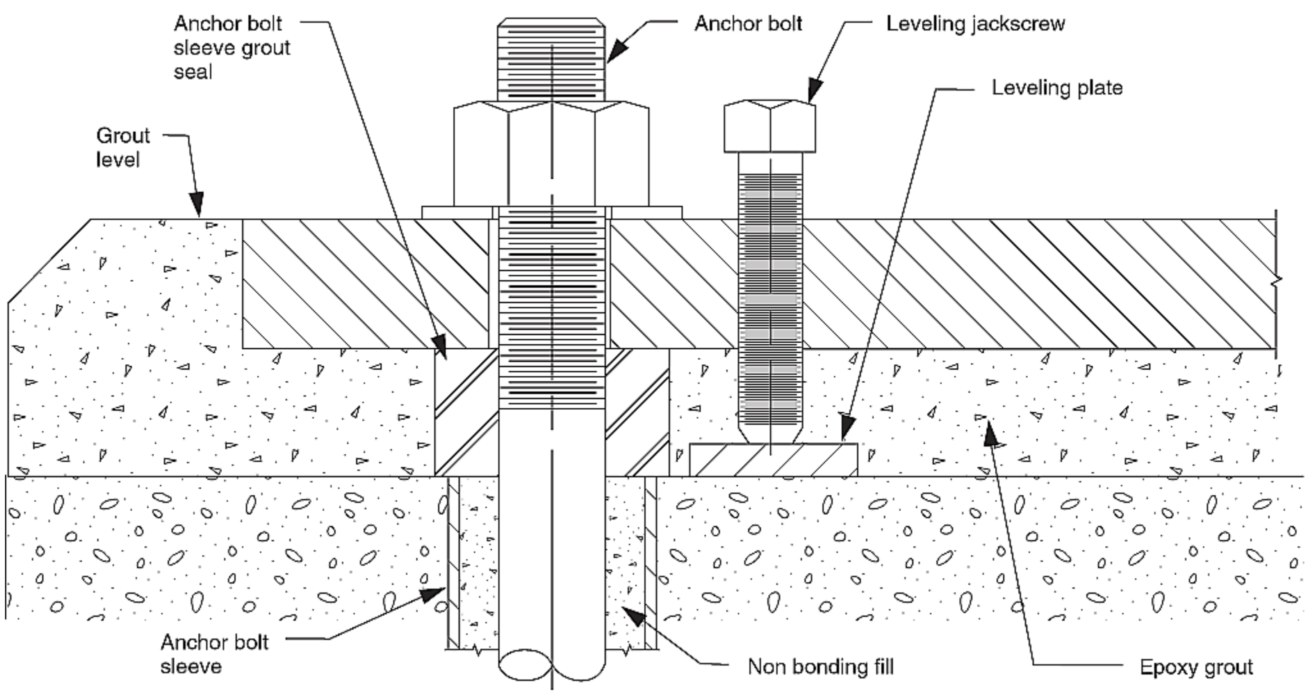


CROSS-SECTION OF FOUNDATION AT FOUNDATION BOLT

Figure L-1—Mounting Plate with Subplates



TOP VIEW OF FOUNDATION AT FOUNDATION BOLT



CROSS-SECTION OF FOUNDATION AT FOUNDATION BOLT

Figure L-2—Mounting Plate without Subplates

Additional copies are available through Global Engineering Documents at (800) 854-7179 or (303) 397-7956

Information about API Publications, Programs and Services is available on the World Wide Web at: http://www.api.org

American Petroleum Institute

1220 L Street, Northwest
Washington, D.C. 20005-4070
202-682-8000

Product No. C61305