REVIEW OF API VERSUS AGMA GEAR STANDARDS—RATING, DATA SHEET COMPLETION, AND GEAR SELECTION GUIDELINES

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ABSTRACT

There are many gear tooth and gearbox rating standards existing in the world. For a given gearbox, the rating system that is used can give very different answers in the amount of power that can be transmitted. If a user is not specific or does not have a basic understanding of the different rating systems, the price and the reliability of the gearbox can be dramatically affected.

The intent of this tutorial is to simplify, then compare the current API, AGMA, ISO, and DIN gear standards to help the inexperienced or casual user make intelligent decisions. The probable changes to API and ISO that should occur in the near future are also discussed.

INTRODUCTION

The basis for the gear rating standards in the United States has been developed by the participants in the American Gear Manufacturers Association (AGMA). AGMA, founded in 1916, has developed rating standards by consensus using volunteers from the gear manufacturing companies and other interested parties who wish to participate. Currently, the basic gear tooth rating formulas are in AGMA 2001 (1995). The two product specific AGMA standards that are discussed in this paper are 6010 (1997) and 6011 (1998), the "low speed" and "high speed" standards, respectively.

In 1977, the American Petroleum Institute (API) released the second edition of the special purpose API 613 gear standard, which applied to high-speed gearing. The rating formulas were simplified from the AGMA standards and more conservative stresses were required. The general-purpose gear standard, API 677, was first released in 1989 using a slightly modified API 613 formula. In 1997, the rating formulas were changed to be identical with the API 613 standard. The rating methods used in the API 613 (1995) and API 677 (1997) standards are highly valued by many because they are consistent between manufacturers and easily checked by purchasers and users.

In Europe, both the German originated specification DIN 3990 and the AGMA standards are used. The International Organization for Standardization (ISO) modified DIN 3990 and released ISO 6336 in 1996.

A draft of the new international standard equivalent to API 613 (1995), ISO 13691 (2000), was submitted for ballot on February 8, 2000. The rating is based on ISO 6336 (1996), and results in slightly different gear ratings than API 613 (1995).

There are committees currently working on revisions of API 672 (1996), integrally geared air compressors, and API 617 (1995), centrifugal compressors. At this time, it appears that gear sets that have a ratio higher than seven to one will be rated on a simplified version of AGMA 2001 (1995), but using a derating factor to gain conservatism.

Understanding the manner in which the various rating standards evolved, it is logical to expect them to give different answers. In addition to confusing the purchaser and user, gear manufacturers are also often confused when transferring back and forth between rating standards. This confusion is increased when comparing international and domestic standards. Even if the supply requirement is simply "AGMA," or to a small degree, "API," the overlap of the standards can supply a gearbox that is a surprise! The end result can be disappointing performance in the field. The intent of this paper is to educate the purchaser and user to know what to specify for better understanding of gear ratings and to hopefully reduce gear problems.

There are also changes occurring in the gear tooth quality standards. Problems have been recognized with AGMA 2000 (1993), so it will be revised or withdrawn. ISO 1328-1 (1995) and ISO 1328-2 (1997) have just been released as an ANSI/AGMA document and probably will become the replacement for AGMA 2000 (1993).

OVERVIEW OF API 613

The API 613 (1995) standard is a gear tooth rating standard and includes detailed quality assurance requirements and more detailed testing requirements as compared with the AGMA standards. It is primarily intended for gears that are in continuous service without installed spare equipment. Following is a brief history of the standard:

- First Edition—August 1968
  - Units rated per AGMA 421.06
  - Application
    - Pinion speed: >3600 rpm
    - Pitch line velocity (PLV): >4000 fpm (20 mps)
- Second Edition—February 1977
  - Conservative K factor rating method
  - Application
    - Pinion speed: >2900 rpm
    - PLV: >5000 fpm (25 mps)
- Third Edition—April 1988
  - Continued use of K factor rating method
  - Speed and PLV guidelines removed
  - Quality assurance (QA) procedures and documentation enhanced
- Fourth Edition—June 1995
  - Maintained basic scope of Third Edition
  - Established minimum instrumentation requirements

As the Standard evolved, basic requirements and features were changed or added. A summary of these changes is as follows:

- First Edition features (1968)
  - AGMA 421.06 rating
  - Four hour mechanical test
  - Bearing thermometers
  - Split journal bearings
- Second Edition additions (1977)
  - Conservative K factor rating
  - Provision for torsiograph
  - Tilt-pad thrust bearing
  - Provisions for vibration probes
  - SST internal piping
  - Studded flange oil connections
  - QA procedures and documentation
  - Lateral critical speed analysis
  - Axially-split shaft seals
  - Material certification
  - Four and 1/4 hour mechanical test
- Third Edition additions (1988)
  - 20 year design life
  - Hobbing as a finishing operation
  - "Observed" versus "witnessed" inspection
  - Drawing and data requirements
  - New allowable unbalance procedure
- Fourth Edition additions (1995)
  - Gear tooth charts > 30,000 fpm (150 mps)
  - Minimum instrumentation requirements
  - Four radial vibration probes
  - Two axial vibration probes
  - Two accelerometers
  - 12 temperature sensors
  - Residual magnetism and runout checks
  - 20 year QA records availability
  - Additional vibration data during test

OVERVIEW OF API 677

The API 677 (1997) standard is for general-purpose gears that are usually spared or are in noncritical applications. It is limited to gearboxes with gear tooth pitch line velocities below 12,000 fpm (60 mps) for parallel shafts or 8000 fpm (40 mps) for bevel shafts. It is generally limited to 2000 hp. Following is a brief history of the Standard:

- First Edition—March 1988
  - Uses modified K factor rating method
  - Application
    - Rated power: < 2000 hp
    - PLV: < 12,000 fpm (60 mps)
- Second Edition—July 1997
  - Rating changed to be identical to API 613 (1995)
  - "Generally limited" to < 2000 hp

As compared with the API 613 (1995) standard, the quality assurance requirements are slightly less stringent and the testing requirements are much less stringent. However, lubrication systems and auxiliary equipment are included. Following are the basic requirements of the current standard, API 677, Second Edition (1997):

- 90 dBA sound pressure level test
- Stainless steel breather cap
- Shaper cut or hobbed gearing
- Antifriction or hydrodynamic bearings
- Axial stability check
- Three tooth contact checks: one at checking stand, two in casing (pre/post test)
- One hour full speed, no load mechanical run test
- Housing vibration check during mechanical test
- Aluminum labyrinth oil seals >800 fpm (4 mps)
- Dynamic balancing of gear elements
- Vertical jackscrews and dowel pin starter holes
- QA documentation on file at vendor's plant for 20 years
- Mass elastic drawing

OVERVIEW OF AGMA 6011

The AGMA Standard 6011 (1998) is a specification for high-speed enclosed helical gear units. It does not apply to bevel or internal gearing. This standard is applicable in a single reduction gearbox if the pinion is over 4000 rpm or the gear tooth pitch line velocity is over 6500 fpm (35 mps). In a multireduction gearbox, the gear tooth pitch line velocity must be over 6500 fpm (33 mps) in the fastest gear set and at least 1500 fpm (8 mps) in other gear sets. The gear tooth rating of this standard seems to reasonably repeat when the same gear set is compared among different gearbox manufacturers.

History of AGMA 6011

The first high speed AGMA gear standard was adopted in 1943 as 421.01. The original standard contained formulas for computing the durability horsepower rating of gearing. In later years, the strength rating was added. The standard evolved through 421.06 (1968) before the numbering system was changed. The new numbering system would include the standard number, a hyphen, the revision letter, and the year of the release. In 1992, 6011 replaced the old numbering system and the standard became 6011-G92. At this revision, the formulas for the durability and strength horsepower rating were removed from the standard and were replaced by referring to the basic rating standard AGMA 2001 (1995) (2001 was the 218 standard before the new numbering system). The 6011-G92 was revised to 6011-H98 in 1998. The rating methods are now per AGMA 2101, which is the metric version of AGMA 2001 (1995).

OVERVIEW OF AGMA 6010

The AGMA Standard 6010 (1997) is a specification for lower speed gear units that can apply to helical, spur, and bevel gears. The limitations are speeds up to 4500 rpm and pitch line velocities not over 7000 fpm (35 mps). (Both the speed and pitch line velocity overlap with the AGMA 6011 (1998) standard, so care must be taken to specify which standard prevails in the overlap situation.) Its gear tooth rating system refers to the formulas in the basic rating standard AGMA 2001 (1995). Several of the variables in the gear tooth rating system in AGMA 6010 (1997) are allowed a range, resulting in a wide variation of ratings between manufacturers for the same gearbox. The magnitude of this variation is about ± 20 percent. It is recognized as having a good thermal rating method.

History of AGMA 6010

The low speed enclosed gearbox standard was originally known as AGMA 420. The standard AGMA 420.04 (released in 1975) used a series of formulas and graphs included in the body of the standard to calculate the strength and durability rating of the gear set. As is typical of the AGMA rating systems, the term “service factor” was used to describe the ratio between the maximum and mean torque for a specific application. In 1988, a revision was released with the new AGMA numbering system as AGMA 6010-E88. It substituted the term “application factor” for “service factor” and referred to the basic rating formulas used in AGMA 218.01 instead of having them in the body of the standard. The current standard is AGMA 6010-F97 (1997). It refers to the formulas in AGMA 2001-C95 (1995) for the durability and strength rating. The application factor reverted to service factor and the thermal rating section was substantially improved.

API 613 AND 677 RATING METHOD

API, working with the gear manufacturers, developed a simplified rating formula that first appeared in API 613, Second Edition, 1977. The API 677, Second Edition (1997), standard utilized the same method. The method, simplified from AGMA 2001 (1995) formulas, has the two typical criteria of any gear tooth rating system, the durability of the gear tooth and the strength of the gear tooth. The durability of the gear tooth is calculated using “K factor,” the universal term used for determining and comparing gear sizes. The strength of the gear tooth is calculated using a “bending stress number” so that the limit is below preset values based on hardness.

API 613 AND 677 FORMULAS AND EXAMPLE

The K factor is usually calculated at the “gear rated power” stamped on the gearbox nameplate. “Gear rated power” is defined in API 613 (1995), paragraph 1.4.5. The K factor is defined as follows:

$$K = [W_t / dF_w][(R+1)/R] \tag{1}$$

In SI units:

$$W_t = \left[(1.91 \times 10^7) P_g\right] / N_p d \tag{2}$$

In US customary units, W_t can be expressed as follows:

$$W_t = \left(126{,}000 P_g\right) / N_p d \tag{3}$$

where:
K = Tooth pitting index in MPa (lb/in^2)
W_t = Transmitted tangential load at operating pitch diameter, in N (lb)
F_w = Net face width, in mm (in)
d = Pinion pitch diameter, in mm (in)
R = Number of teeth in gear divided by number of teeth in pinion
P_g = Gear rated power, in kW (hp)
N_p = Pinion speed, in rpm

The allowable K factor at the gear 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:

$$K_a = I_m / (SF) \tag{4}$$

where:
K_a = Allowable K factor
I_m = Material index number (from Table 3 and Figure 3 in API 613 (1995), Fourth Edition)
SF = Minimum gear service factor (from Table 2 in API 613 (1995), Fourth Edition)

The strength of the gear tooth is calculated using the bending stress number. The allowable bending stress number depends on materials selected for the gear teeth, the tooth hardening processes used, and the service factor. It is calculated at the gear rated power. The bending stress number is calculated as follows:

In SI units:

$$S = [W_t / \left(m_n F_w\right)](SF)[(1.8 cos\ \gamma)/J] \tag{5}$$

In US customary units:

$$S = [\left(W_t P_{nd}\right)/F_w](SF)[(1.8 cos\ \gamma)/J] \tag{6}$$

where:
S = Bending stress number
P_{nd} = Normal diametral pitch
γ = Helix angle
J = Geometry factor (from AGMA 908 (1999))
m_n = Module number, in mm

As an example, let us go through an actual petrochemical plant gearbox calculation. The conditions are a synchronous motor driving a centrifugal compressor through a gearbox. The motor is nameplated at 9000 hp, 1.0 service factor, operating at 1800 rpm. The information on the API 613 (1995) data sheet is as follows:

Page 1, line 42: Net face width, “Fw” *10.5* in; Pinion L/d *1.52*
Page 1, line 37: Pitch dia, in Pinion *8.721*; Gear *33.279*
Page 1, line 35: Number of teeth Pinion *38*; Gear *145*
Page 1, line 43: Normal diametral pitch *5*; Backlash *0.016-0.026* in
Page 1, line 40: Helix angle *29.3749* degrees
Page 1, line 39: Pinion *0.55*; Gear *0.58*
Page 1, line 22: Material index number (Fig 2, Table 3) *440*
Page 1, line 21: Gear service factor (2.2.2.1) *1.4* (min)
Page 2, line 36: Pinion(s) *AISI 9310H VD*; Hardness *58 RC minimum*
Page 2, line 37: Gear rim(s) *AISI 9310H VD*; Hardness *58 RC minimum*

Solving for the transmitted tangential load in pounds, substitute the above into Equation (3):

$$\begin{aligned} W_t &= (126{,}000 P_g)/N_p d \\ W_t &= (126{,}000 \times 9000)/(1800 \times 145/38)8.721 \\ W_t &= 18{,}932\ \textit{pounds} \end{aligned} \tag{7}$$

Solving for the K factor at rated conditions, substitute the above into Equation (1):

$$K = [W_t/dF_w][(R+1)/R]$$
$$K = ((18{,}932/8.721\times10.5))((145/38)+1)/(145/38)) \quad (8)$$
$$K = 261$$

The allowable K factor is calculated from Equation (4). The material index number is based on the hardness and the heat-treating process. The value can be found either in Table 3 or Figure 3 on page 7 of API 613 (1995), Fourth Edition, as well as the data sheets. The minimum gear service factors are in Table 2 on page 6. They will also be on the data sheets. Substituting into Equation (4):

$$K_a = 440/1.4$$
$$K_a = 314 \quad (9)$$

The actual K factor of 261 is less than the allowable K factor of 314, therefore the durability portion of the rating meets API 613 (1995) standard. Gearbox users sometimes question the manufacturer when the actual K factor is the same or only slightly below the allowable K factor. The response usually is that being substantially below the allowable only increases the service factor, which is already very high, therefore, it is not necessary.

The strength rating is checked by using the formula for the bending stress number. The bending stress number formula is given in Equation (6).

The geometry factor (designated by "J") is to account for stress concentration in the root of the tooth. It will vary slightly among manufacturers depending on the shape of the roughing and finishing tool as well as heat-treating distortion. It will be different from the pinion to the gear. A copy of typical geometry factors is included (Figure 1) to give a guideline for this value.

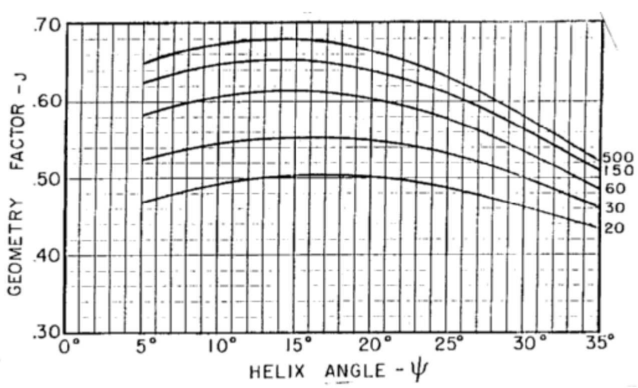

Figure 1. Typical J Factor.

Notice that the pinion will have a different bending stress number than the gear, so both must be checked. Substituting into Equation (6) to solve for the pinion bending stress number:

$$S = ((18{,}932\times8.721)/10.5)(1.4)(1.8\times cos29.3749)/.55)$$
$$S = 36{,}025 \quad (10)$$

Substituting into Equation (6) to solve for the gear bending stress number:

$$S = ((18{,}932\times8.721)/10.5)(1.4)((1.8\times cos29.3749)/.58)$$
$$S = 34{,}113 \quad (11)$$

The bending stress number for the gear and pinion can be compared to the allowable bending stress number in Figure 4 on page 8 in API 613 (1995). The hardness in this example is the same on both parts at 58 RC so the same maximum allowable bending stress applies, but often the pinion is harder than the gear. The maximum allowable bending stress per Figure 4 is 38,500, which is more than either the pinion actual bending stress (36,025) or the gear actual bending stress (34,113). We conclude that the gear set strength rating meets the API 613 (1995) standard.

API LENGTH-TO-DIAMETER RATIOS

All gear tooth rating standards recognize that it is difficult to maintain equal loading across the width of the gear tooth. The generally accepted method for controlling this problem is to limit the shape of the pinion. If a pinion diameter is large compared with its length, then the dynamic bending and twisting of the pinion are less than if it were smaller in diameter and longer. API 613 (1995) and 677 (1997) control the shape by giving limits on the length-to-diameter ratio, usually referred to as the L/d ratio. The guidelines are listed in paragraph 2.2.3.5 and Figure 3 of API 613 (1995), Fourth Edition. For single helical gear sets, the calculation is typically based on the shorter of the pinion or the gear tooth length, if they are different. For double helical gear sets, the calculation should include the gap. The gap is usually required because of the manufacturing process, therefore the gap can change slightly depending on the type of process used. Shown in Table 1 is typical information on gap widths as a function of the normal diametral pitch (Pnd):

Table 1. Typical Gap Widths for Double Helical Gears.

Pnd	Gap Width (Inches)
2	4
4	2.75
6	2.375
8	2.125
10	2
12	2

For the petrochemical plant gearbox example, the total face width of the pinion and the gear is the sum of the net face width plus the gap.

$$L/d = (10.5+2.75)/8.721$$
$$L/d = 1.52 \quad (12)$$

Per Table 3 of API 613 (1995), Fourth Edition, the maximum allowable L/d ratio for this example is 1.6, therefore, the L/d meets the API 613 (1995) standard without the justification per paragraph 2.2.3.6. If the L/d had calculated higher than 1.6, then the manufacturer could have submitted a detailed analysis of the gear tooth deflection and loading per paragraph 2.2.3.6 and discussed this with the purchaser. In some examples, such as very high pitch line velocities, this may be recommended and should be carefully considered.

COMPARISONS OF API 613 AND 677 WITH AGMA 6010 AND AGMA 6011

There are many ways to compare the API gear tooth ratings to the AGMA ratings. For our purposes, first, the comparison will be AGMA 6011 (1998) to API 613 (1995) for a turbine driving a generator. The second will compare AGMA 6011 (1998) to API 613 (1995) for a synchronous motor driving a compressor. The third will compare AGMA 6010 (1997) to API 677 (1997) for an induction motor driving a fan. The fourth and final comparison will present the data in a different manner, using an example that fits into the overlap region where both AGMA 6011 (1998) and 6010 (1997) and API 613 (1995) and 677 (1997) can apply.

Comparison of API 613 to AGMA 6011—
For a Turbine Driven Generator

The method used will be to size seven gear sets at different powers of gas turbines (1000 hp, 5000 hp, 10,000 hp, etc.) per API 613 (1995), Fourth Edition, and then to rate the same gear sets per AGMA 6011-H98 (1998). For this application, both API 613 (1995) and AGMA 6011 (1998) require a 1.1 service factor. The gas turbine speed has been selected at 5400 rpm and the generator speed at 3600 rpm. Figure 2 shows the service factor calculated by AGMA 6011 (1998) and API 613 (1995) for identical gear sets.

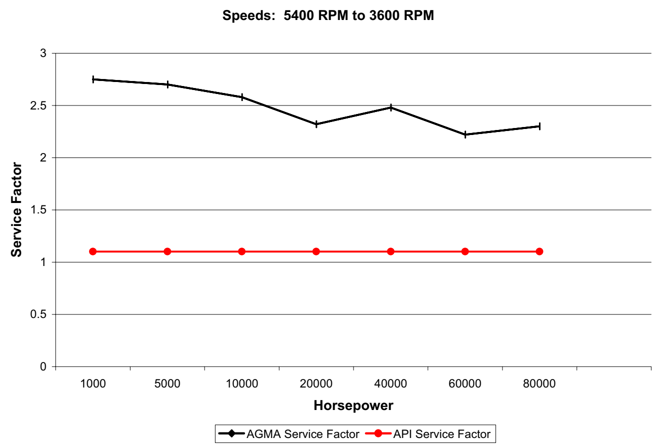

Figure 2. Turbine/Generator Comparison.

A different way to present the difference between the size of the gear set rated by both standards is to pictorially represent them. Referring to Figure 2, the 20,000 hp data point is pictorially represented (Figure 3) by an API 613 (1995) rated gear set on the left and an AGMA 6011 (1998) rated gear set on the right. The service factor in both gear sets is 1.1. The AGMA 6011 (1998) gear set scale is 79 percent of the scale of the API 613 (1995) gear set.

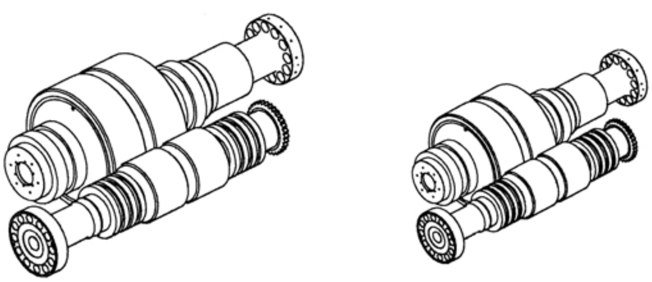

Figure 3. Size Comparison of API/AGMA.

Comparison of API 613 to AGMA 6011—
For a Synchronous Motor Driven Compressor

The most common application of the API 613 (1995) standard is a motor driving a compressor. This comparison is based on an 1800 rpm motor driving through a speed increasing gearbox to a centrifugal compressor at 6000 rpm. The service factor for both API 613 (1995) and AGMA 6011 (1998) is 1.4 for this application. Figure 4 is plotted for seven different gear sets at different power ratings.

The more robust API 613 (1995) gearbox is slightly less efficient than the smaller AGMA 6011 (1998) version. To understand how much the API 613 (1995) standard affects the efficiency, an example of a 3000 hp electric motor at 1785 rpm driving through a gearbox to a 5600 rpm centrifugal compressor is offered in Table 2. The decrease in efficiency of 1.3 hp or 0.04 percent of the transmitted power is very small.

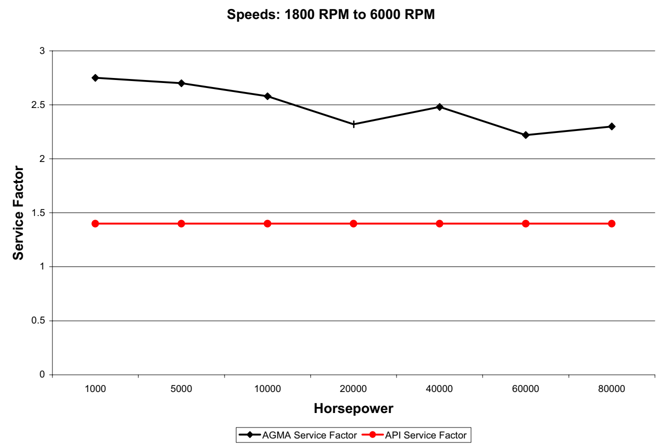

Figure 4. Motor/Compressor Comparison.

Table 2. Example of How API 613 Affects Efficiency.

Unit Data	AGMA 6011	API 613
Center distance (in)	14.00	16.00
Net face width (in)	7.75	9.75
Mechanical rating	4,652	7,428
HP loss	43.3	44.6
Efficiency	98.6%	98.5%
Oil flow (gpm)	17	23
Weight (lb)	3200	4400

Comparison of API 613 or API 677 to AGMA 6010

A typical application that could specify API 677 (1997) and also fit in the speed and velocity limitations of AGMA 6010 (1997) is a motor driving through a speed reducing gearbox to a fan. For this example, the motor speed is 1780 rpm and the fan is 400 rpm. Figure 5 is plotted for seven different gear sets at different power ratings.

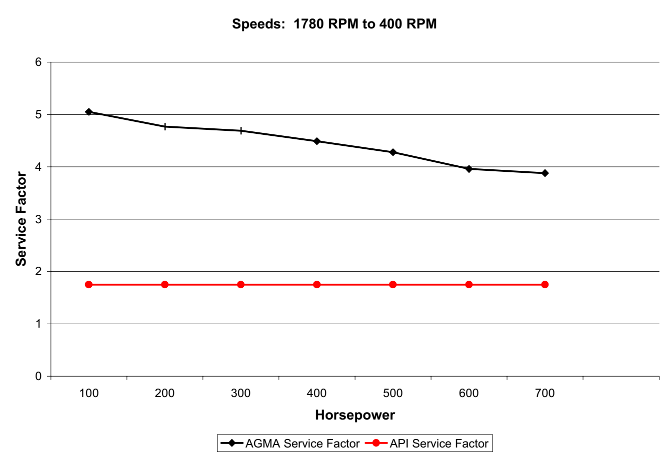

Figure 5. Motor/Fax Comparison.

Comparison of API 613, API 677, AGMA 6010, and AGMA 6011

As discussed earlier, there is an overlap area between AGMA 6010 (1997) and AGMA 6011 (1998). Fortunately, the overlap is small. The overlap exists when both the rpm are between 4000 and 4500 and the gear tooth pitch line velocity is between 6500 fpm (33 mps) and 7000 fpm (35 mps). An overlap can also occur between

API 613 (1995) and API 677 (1997). An example of a gearbox fitting into this overlap is an 1800 hp, 1800 rpm electric motor driving through a speed increasing gearbox to a 4000 rpm centrifugal compressor. Table 3 gives the resulting face width and center distance to meet the minimum service factor for that standard. Table 3 also gives the AGMA 6011 (1998) service factor for each gear set so that the robustness and cost can be compared.

Table 3. Service Factor Comparison.

	AGMA 6010	AGMA 6011	API 677	API 613
Center distance	10	10	14	14
Net face width	6	7	7	7
AGMA 6011	1.20	1.38	2.70	2.70
Cost	100%	110%	180%	290%

OVERVIEW OF DIN 3990

The DIN 3990, Part 21 (1989), standard is intended to be applied to gearboxes with a pinion that is rotating at 3000 rpm or greater. The latest release was in 1989. This standard is based on analyzing a gear set with 100 mm centers, then modifying the results by a series of factors such as size, speed, material, surface condition, and lubricant factors. Because some of the factors are load dependent, it is not possible to calculate the capacity of the gear set unless the load is known, unlike the AGMA 6011 (1998) standard.

It is generally recognized that the DIN 3990, Part 21 (1989), standard is not suitable for through-hardened gear sets. During testing, the test was stopped when pitting first appeared, which is not regarded as a failure point for through-hardened gearing. This resulted in the allowable stresses being set lower than necessary. As consensus on this point was gained, a new material grade, MX, was created and temporarily put into ISO 6336-5 (1996), “Strength and Quality of Materials,” for use with DIN 3990, Part 21 (1989).

Comparison of DIN 3990, Part 21, to AGMA 6011

An actual application that gives a comparison between DIN 3990, Part 21 (1989), and AGMA 6011 (1998) is an 1800 rpm synchronous motor, 4627 hp, driving a centrifugal compressor at 14,233 rpm. The gear set supplied is through-hardened with a 22 inch center distance. It is single helical with an effective face of 6.5 inches. The calculated AGMA 6011 (1998) service factor is 1.58 whereas DIN 3990, Part 21 (1989), would allow 17 percent more horsepower for the same service factor.

AGMA 6011-H98 (1998) has examples in “Annex E” that are often used as a basis for comparisons between gear standards. They are appropriate for comparing DIN 3990, Part 21 (1989), and AGMA 6011 (1998). Example #1 in “Annex E” has a 5000 rpm pinion driving a 1480 gear. The gear set is through-hardened and has a 15.748 center distance with an effective face width of 10 inches. AGMA would allow the driver to have 4400 hp with a 1.4 service factor. DIN 3990, Part 21 (1989), would allow 16 percent more horsepower for the same service factor.

Example #2 in “Annex E” could be a gas turbine at 8215 rpm driving a 3600 rpm generator. The gear set is carburized and has a 16.5 inch center distance with an effective face width of 10.23 inches. AGMA would allow the gas turbine to have 26,229 hp when using the correct 1.3 service factor. DIN 3990, Part 21 (1989), would allow 92 percent more power; however, this would probably be derated due to scoring calculations or require that special lubricants be used.

The conclusion is that DIN 3990, Part 21 (1989), calculates a minor increase over AGMA 6011 (1998) for through-hardened gearing and a major increase for carburized gearing.

OVERVIEW OF ISO 6336

ISO 6336 (1996), “Calculation of Load Capacity of Spur and Helical Gears,” is the gear rating standard that has been adopted by the European Community, the Eastern Bloc, and Japan. It was released in 1997. It evolved from the 1987 DIN 3990, Part 21 (1989), standard, so there is a strong similarity between the two. The appropriate AGMA standard for comparison is AGMA 2001 (1995). The ISO 6336 (1996) is recognized as being more complex and detailed that AGMA 2001 (1995), requiring about 20 more pieces of information to calculate a gear set rating.

The Standard is broken into four categories. They are as follows:

- *6336-1*—Definitions and Influence Factors, Such as the Dynamic and Load Distribution Factor
- *6336-2*—Calculation of Gear Tooth Surface Compressive Stress and Permissible Compressive Stress
- *6336-3*—Calculation of the Tensile Stress in the Root of the Gear Tooth and the Permissible Bending Stress
- *6336-5*—Strength and Quality of Materials

The Standard gives three methods to calculate the ratings, method A, B, or C, in decreasing order of accuracy. Method A often includes full size testing as would be appropriate in the aerospace industry. Method B uses detailed calculations to correlate field data to similar designs and is the method typically used in the industrial gear market. Method C is a simplified method used for narrow applications.

The theory in ISO 6336-2 (1996) is based on the fundamental Hertzian equations for surface stress, very similar to AGMA 2001 (1995). The ISO 6336-3 (1996) section is based on simplified cantilever beam theory somewhat similar to AGMA 2001 (1995), resulting in the J factor being reasonably close at a 20 degree pressure angle, but diverging at higher pressure angles. One large difference is the greater design detail required in ISO 6336 (1996). Another difference from AGMA 2001 (1995) is that since the ISO dynamic factor and the load distribution factor are dependent on load, the capacity of a gear set cannot be calculated until the load is known. It is necessary to iterate until the required safety factor is achieved. ISO 6336 (1996) does not directly calculate an allowable power for a gear set, nor does AGMA 2001 (1995) calculate a service factor.

When identical gear set ratings are compared calculated by ISO 6336 (1996) and AGMA 6011 (1998), substantial differences are found. The gear tooth strength ratings seem to be always higher using ISO 6336 (1996). The durability rating is about the same for through-hardened gear sets, but ISO 6336 (1996) has higher durability ratings for carburized gear sets.

FUTURE GEAR RATING STANDARDS

There are five areas of gear rating standards that probably will change in the near future.

Overview of ISO 13691

The draft copy of ISO 13691 (2000), “Gears—High Speed Special-Purpose Units for the Petroleum, Chemical, and Gas Industries,” has been released for voting. The voting terminates on July 3, 2000.

The Standard is obviously derived from API 613 (1995), to the extent that the layout, figures, testing, and data sheets are easily recognized. The methods used to rate the gear set are similar to API 613 (1995), but have been derived from ISO 9084 (Draft). To quote Annex G, paragraph G.6.3, “The allowable contact stress numbers (for surface durability) and allowable bending stress number (for bending strength) are established by applying the principal that the successful and satisfactory gear design experience when using Standard API 613 (1995) be fully maintained” (ISO 13691, 2000). Comparisons with API 613 (1995) have resulted in virtually the same durability ratings, but different strength ratings. The difference in strength ratings is probably because of the different methods in calculating the J factor.

Proposed Rating Change of High Ratio API 613 Gear Sets

A different rating method has been proposed for high ratio compressors that would have been rated by API 613 (1995). The API 617 (1995) committee is reviewing a proposal to change the method of rating the gear sets with a ratio greater than 7.0 to 1. These high ratio gear sets are typically used in integral compressor drives and often use a carburized pinion with a through-hardened gear. This hardness combination is not addressed in API 613 (1995). Gear manufacturers recognize that a more detailed analysis is important on high ratio gearing. This is to evaluate the uneven loading across the length of the tooth.

The proposed rating method is more complicated than API 613 (1995) because of the increased detail. The added areas are:

- A load distribution factor modifies the gear set rating for the dynamic deformation of the gear teeth.
- A life-cycle factor modifies the rating for the number of cycles that a gear tooth will see. For instance, some designs have many pinions in mesh with one gear. The gear has many cycles for every turn it makes.
- A dynamic factor has been added to compensate for the quality of the gear teeth. At the typical speeds of these gear sets, the accuracy has to be very good, so this factor has a relatively small effect.

The allowable rating factors are from AGMA 2101 (metric version of the 2001 (1995) standard) so they will be consistent with API 613 (1995). All the factors will be specified, so everyone should get the same answer.

In comparisons presented at the API 617 (1995) committee meeting in 1999, some gear set ratings were slightly higher and some were slightly lower than API 613 (1995). In a comparison with AGMA 6011 (1998), the AGMA ratings were from 20 to 93 percent higher based on durability and 18 to 40 percent higher based on strength than the proposed method.

Comparison of Gear Tooth Accuracy
Standards, AGMA 2000 and ISO 1328

It is recognized that the gear tooth accuracy standard, AGMA 2000 (1993), "Gear Classification and Inspection Handbook," has very lenient allowable errors in the lead of the gear tooth. As a result, the AGMA members voted to withdraw the standard. The decision was appealed and is currently going through the AGMA appeal process. Two new standards, ISO 1328-1 (1995) and ISO 1328-2 (1997) have been accepted by AGMA and released as ANSI/AGMA ISO 1328-1 (1995) and ANSI/AGMA ISO 1328-2 (1997), respectively. The information in ANSI/AGMA ISO 1328-1 (1995) covers virtually everything that is needed for the usual gear inspections.

ANSI/AGMA ISO 1328 (1995, 1997) is much different from AGMA 2000 (1993). The major difference is in the numbering system. The AGMA numbering system for different classes of accuracy is from Q3 to Q15, in order of increasing precision. The ANSI/AGMA ISO system is just the opposite, consisting of 13 classes with zero being the most precision and 12 the least precision. While is it impossible to define a direct comparison, the "Rule of 17" is typically used. Subtract the AGMA quality number from 17 and the answer is reasonably close to the ANSI/AGMA ISO class.

To compare the two standards in more detail, the allowable lead errors for wide face widths in ANSI/AGMA ISO 1328 (1995, 1997) are tighter than in AGMA 2000 (1993). This has long been a complaint of AGMA 2000 (1993). However, ANSI/AGMA ISO 1328 (1995, 1997) uses tables instead of formulas, so a minute change in one of the parameters can cause a large change in the allowable accuracy value. The gear tooth profile section of ANSI/AGMA ISO 1328 (1995, 1997) has improved K chart definitions as compared to AGMA 2000 (1993). The gear tooth spacing section of ANSI/AGMA ISO 1328 is regarded as an improvement over AGMA 2000 (1993).

In conclusion, most gear manufacturers agree that ISO 1328 (1995, 1997) is a better standard than AGMA 2000 (1993).

ISO 9084

An international standard specific for high-speed products, ISO 9084, is being drafted. This would be comparable to the AGMA 6011 (1998) standard.

ISO 9085

An international standard specific for low speed products, ISO 9085, is being drafted. This could be appropriately compared to the AGMA 6010 (1997) standard.

CONCLUSION

The conclusion is that the gear set rating standards in common use are very different and therefore very confusing. For some applications, the gear set ratings are close, but usually there are significant differences in the ratings when comparing different standards. However, there are some general statements that can be useful for the typical gear unit user:

- The API standards always result in a substantially more robust gear set. The results are very repeatable among manufacturers.
- AGMA 6011 (1998) has good repeatability among manufacturers, but is not as complex an analysis as the ISO standards.
- AGMA 6010 (1997) has a wide variation in ratings among manufacturers.
- Both ISO and DIN standards generally have higher ratings than AGMA standards for carburized gear sets and have a very complex analysis.
- ANSI/AGMA 1328 (1995, 1997) is an improvement over AGMA 2000 (1993).

To reduce some of the confusion that a user may have, it may be helpful to compare offers from different manufacturers by calculating and then comparing the API service factors. Be aware that the API service factors will probably be less than unity, but the comparison should indicate the most robust gear set. A manufacturer's perspective could be to use the ratings to gain a competitive edge and supply a less robust gear unit.

APPENDIX A—
GEAR SELECTION GUIDELINES

In this section, the guidelines for gear data sheet preparation and major criteria for the selection of gears will be briefly discussed.

GEAR DATA SHEET PREPARATION

Gear data sheets provide the basis for gear design and define the scope of supply by the supplier. Together with the API and AGMA gear standards, they form the basis for supplier proposal.

- Input data
- Rating and installation data
- Basic input data for the preparation of gear data sheet is as follows:
  - Type of gear (parallel shaft, and right angle, etc.)
  - Driven equipment and driver horsepower rating (normal and maximum)*
  - Torque at maximum continuous speed*
  - Input and output speeds*
  - Rotation direction*
  - Minimum gear service factor
  - Pinion hardness**
  - Electrical area classification

- Noise limitation*
- Lube oil **
- Mounting plates
- Coupling type
- Instruments
- Testing
- Piping

*By driven equipment vendor
**Usually by gear manufacturer

INQUIRY REQUISITION

The inquiry requisition comprises the completed gear data sheet along with the applicable specifications and any special project requirements. The inquiry requisition is dispatched to gear vendors from the approved bidder's list.

TECHNICAL BID EVALUATION

Upon receipt of the vendor quotations, technical and commercial bid evaluations are performed. This will determine if the bidders are in compliance with applicable specifications and data sheets. The following are the major technical areas that require detailed review prior to the selection of the specific gear design and its suppliers.

Basic Gear Data

- Gear service factor (actual)
- Mechanical rating
- Mechanical efficiency
- Pitch line velocity
- Tooth pitting index (actual/allowable)
- Tangential load
- Actual and allowable bending stress number
- Material index number
- Noise level
- Surface finish

Construction Features

- Gear type (single versus double helical, etc.)
- Length to diameter ratio (L/d), defined for hardness range
- Pinion/shaft attachment method
- Gear/shaft attachment method
- Gear casing fabrication method (cast versus fabricated)
- Tooth hardness/load distribution (through- versus case-hardened)
- Materials of construction (iron versus steel)

Bearings and Lubricant Requirements

- Radial bearing type/loading, journal velocity, expected temperature, etc.
- Thrust bearing type/area/loading/rating
- Oil flow, viscosity, and specialty oil

Experience

- Review of experience list for similar applications

Testing Requirements

- Full speed/no load (minimum test per API 613/617 (1995/1995))
- Full speed/part load (checks tooth contact and noise)
- Full speed/full load (checks temperature rise and scoring tendency)
- Full torque/slow roll (load applied at reduced speed/full torque reduced horsepower. Test proves gear can carry torque but no indication on effect of bearing design, balance, stability, and temperature life. (Not recommended.))

Note: Oil viscosity during shop test should be the same as a contract oil at 100°C/210°F.

GEAR SELECTION

Following is additional information that can be useful in the evaluation of gear selection.

Gear Types

Some of the most common gear types are:

- Single or double helical
- Spur
- Bevel
- Worm

The bevel and worm gears have limited application due to the size limitation on bevels and sliding velocity limits on the worm gear. Spur gearing is also limited since it must be very large to transmit a load equivalent to the same size helical gear. The majority of heavy industrial gears are of single or double helical gear design. Some of the major differences of each gear type are shown in Table A-1.

Table A-1. Gear Type Differences.

Single Helical	**Double Helical**
Have teeth of only one hand on each gear. (One direction of rotation.)	Have both right and left hand and operate on parallel axes, symmetrical loading.
Helix angle is usually in the range of 5° to 20°.	Helix angle is usually in the range of 20° to 45°.
Requires thrust bearings and thrust faces on each rotor.	Limited thrust bearing loads and smaller thrust bearings. (Equalized axial thrust from the gear teeth)
Less efficient due to thrust bearing load.	Higher efficiency due to low thrust bearing losses and less lube oil requirements.
Do not have apex runout.	Has some apex runout that produces additional loading.
Not as sensitive to coupling lockup or other external thrust load.	Sensitive to coupling lock-up (external thrust) load.
Due to longer teeth, the temperature rise of the oil is greater than double helical at the same operating speed. (Higher thermal distortion.)	Less thermal distortion due to shorter tooth. Symmetrical thermal distortion for the gear casing.
Less expensive to fabricate gear elements.	More expensive to fabricate gear elements.
Less commonly applied.	More commonly used.

Tooth Hardness Criteria

Gears are available in the hardness range of 220 to 360 BHN (through-hardened teeth) and 50 to 60 Rockwell C (case-hardened teeth) with considerations shown in Table A-2.

Table A-2. Case-Hardened Versus Through-Hardened Teeth.

Case- (Surface) Hardened Teeth (carburized or nitriding)	**Through-Hardened Teeth**
Hardness (50 to 60 Rockwell C).	Hardness (220 to 360 BHN)
Smaller size and less weight.	Large size compared to case-hardened teeth.
Higher efficiency due to smaller losses and lower lube oil requirements.	Lower efficiency due to larger losses.
(Higher scoring index susceptible to scoring and dynamic distortion due to higher loading and sliding velocities). Less tolerant to overload – sudden failure without warning.	More forgiving of operational errors and will wear progressively before failing.
Complex heat treatment procedure.	Simple heat treatment.
Lower pitch line velocity due to higher allowable unit loading.	Allows higher pitch line velocity because material has good stress relief property after heat-treatment.
Frequently used for higher power applications (for a given size, can transmit twice the power as 300 BHN).	Frequently used for medium power applications
Difficult to repair (hand grinding repair).	Easy to repair. (Can be repaired quickly by recutting gear and increasing pinion size.)

Verification of Gear Design and Construction Features

During bid evaluation, for gears outside the manufacturer's normal range, it may be useful to contact users of similar designs to obtain their experience and suggestions.

Gear Service Factor

Gear service factor is applied to tooth pitting index and bending stress. Requisition engineer must verify compliance with applicable standards (AGMA, API, etc.). The characteristic of driver and driven equipment will account for potential overload, shock load, or continuous oscillatory torque characteristics.

Pitch Line Velocity

Ensure that gear to shaft attachment is in accordance with API 613 (1995) criteria shown in Table A-3. Be aware that the gearbox will be oversped during testing. The design must be capable of overspeed.

Table A-3. Criteria for Gear to Shaft Attachment for API 613.

Maximum Pitch Line Velocity		Gear/Shaft Attachment
Feet/Min	Meters/Sec	
12,000	60	Shrunk-on forged rims
25,000	127	Welded with forged rims
30,000	152	Shrunk-on forged gears

Note: Gear/shaft attachment method is not addressed in API 677 (1997).

For designs based on API 613 (1995), pinions will be integrally forged with their shafts. For designs based on API 677 (1997), the guideline for pinion attachment is shown in Table A-4.

Table A-4. Criteria for Gear to Shaft Attachment for API 677.

Maximum Pitch Line Velocity		Pinion/Shaft Attachment
Feet/Min	Meters/Sec	
< 3000	< 15	Separate pinion and shaft fabrication
> 3000	> 15	Welded with forged rims

Tooth Pitting Index

The gears are sized on the basis of tooth pitting index called K factor. This includes the factors to account for:

- Radii of curvature of contacting tooth surfaces
- Gear extended life
- Reliability
- Material strength in terms of pitting index
- Dynamic load effects

Allowable K Factor

The allowable K factor varies with the selected material for gear teeth hardening process and service factor. The allowable K factor is defined as follows:

- K = Material index number/service factor

Material Index Number

Material index number represents the value defined based on hardness and heat-treatment process. The acceptable material index value should be verified based on API 613 (1995) material index and L/d ratio table.

Bending Stress

The bending stress value for both gear and pinion is a measure of gear tooth strength and is calculated per Equation (4) of API 613 (1995), paragraph 2.2.4.1.

Length to Diameter Ratio

The length to diameter ratio (L/d) is a guideline to provide acceptable loading across the gear tooth. This guideline is listed in paragraph 2.2.3.5 of API 613 (1995).

Casing Fabrication

Cast casings are more commonly used due to their lower cost and shorter delivery time, and to comply with low noise requirements.

The API 613 (1995), Fourth Edition, gear data sheet is shown in Figures A-1, A-2, A-3, and A-4. A gear testing comparison is shown in Table A-5.

SPECIAL PURPOSE GEAR UNITS
API 613 FOURTH EDITION
DATA SHEET
U.S. CUSTOMARY UNITS

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

Applicable To: ○ Proposal ○ Purchase ○ As Built

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**

○ RATING REQUIREMENTS

Driven Equip Power Normal ______ Max ______
Driver Power Rated ______ Max ______
Gear Rated Power (2.2.1) ______
Torque @ Max Cont. Speed ______ Lb Ft
Max Torque (2.2.1) ______ Lb Ft @ ______ RPM
Rated Speed, RPM (1.4):
Input ______ ○ Specified ○ Nominal
Output ______ ○ Specified ○ Nominal
Allow Var In Gear Ratio (1.4) (+) (-) ______ %
Max Continuous Speed (1.4) ______ RPM
Trip Speed (1.4) ______ RPM
Gear Service Factor (2.2.2.1) ______ (Min)
Pinion Hardness (2.2.2.2) ______
Shaft Assembly Designation (2.1.17.2) ______
HS Shaft Rot Fac'g Cpl'g (2.1.17.2) ○ CW ○ CCW
LS Shaft Rot Fac'g Cpl'g (2.1.17.2) ○ CW ○ CCW
HS Shaft End: ○ Cylindrical ○ Taper ○ 1-Key ○ 2-Keys
○ Hydraulic Taper ○ Integral Flange
LS Shaft End: ○ Cylindrical ○ Taper ○ 1-Key ○ 2-Keys
○ Hydraulic Taper ○ Integral Flange
External Loads (2.1.13) ______
Other Operating Conditions (2.2.3.4) (2.6.1.4) ______

○ INSTALLATION DATA

○ Indoor ○ Heated ○ Under Roof
○ Outdoor ○ Unheated ○ Partial Sides
○ Grade ○ Mezzanine ○ ______
○ Winterization Req'd ○ Tropicalization Req'd
Electrical Area (2.1.7) Class ____ Grp ____ Div ____
Max Allow SPL (2.1.4) ______ 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 ○ ______

NOTES: ______

☐ BASIC GEAR DATA

Mechanical Rating (1.4) ______ HP@ ______ RPM
Full Load Power Loss ______ HP
Mechanical Efficiency ______ %
Pitch Line Velocity ______ FPM
Tooth Pitting Index, "K" (2.2.3):
Actual ______ Allowable ______
Tangential Load, "W_t" (2.2.3.2) ______ Lb
Bending Stress Number, "S_t" (2.2.4.2):

	Pinion	Gear
Actual	______	______
Allowable	______	______

Material Index Number (Fig 2, Table 3) ______
Anticipated SPL (2.1.4) ______ dBA @ ______ Ft
Journal Static Weight Loads (2.6.2.1):
Pinion ______ Lb Gear ______ Lb
WR^2 Referred To LS Shaft ______ Lb Ft^2
Breakaway Torque ______ Lb Ft @ LS Shaft

☐ CONSTRUCTION FEATURES

TYPE OF GEAR ☐ Reducer ☐ Increaser
☐ Single Stage ☐ Double Stage
☐ Single Helical ☐ Double Helical
☐ Epicyclic ☐ ______

TEETH
Number of Teeth Pinion ______ Gear ______
Gear Ratio ______ Center Dist ______ In
Pitch Dia, In Pinion ______ Gear ______
Finish ____ RA AGMA Geometry Factor "J":
Pinion ______ Gear ______
Helix Angle ______ Degrees
Normal Pressure Angle ______ Degrees
Net Face Width, "Fw" ______ In Pinion L/D ______
Normal Diametral Pitch ______ Backlash ______ In
Tooth Plating (2.5.1.4) ☐ Recom'd ☐ Not Recom'd

MANUFACTURING METHODS
Teeth Generated By The ______ Process
Teeth Finished By The ______ Process
Teeth Hardening Method ______
Gear To Shaft (2.5.3.2) ☐ Integral ☐ Shrunk-on
Rim Attachment (2.5.3.2) ______

Figure A-1. API Gear Data Sheet One. (Courtesy American Petroleum Institute)

SPECIAL PURPOSE GEAR UNITS
API 613 FOURTH EDITION
DATA SHEET
U.S. CUSTOMARY UNITS

Job No. ____ Item No.
Date ____ By ____
Revision No. ____ By ____

○ ADDITIONAL REQUIREMENTS

MOUNTING PLATES (3.3.1)
○ Gear Furnished With (3.3.3.1):
○ Baseplate ○ Soleplate(s) ○ Subplate(s)
○ Mounting Plate(s) Furnished By(3.3.1.1) ____
○ Equipment On Baseplate (3.3.2.1) ____

○ Baseplate With Leveling Pads (3.3.2.3)
○ Baseplate Suitable For Column Mounting (3.3.2.4)
○ Grout Type (3.3.1.2.5) ○ Epoxy ○ ____
● **PAINTING** (4.4.3.1) ○ ____
MISCELLANEOUS
● Undamped Critical Analysis Report (2.6.1.6):
○ With Damped Rotor Response Analysis Report (2.6.1.6)
○ Torsional Analysis By (2.6.1.8): ○ 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 And Drain Connections ____

● VIBRATION DETECTORS (3.4.2)

RADIAL (3.4.2.1)
○ Manufacturer ____
○ No. At Each Shaft Bearing ____ Total No. ____
○ Oscillator-Demodulators Supplied By ____
○ Manufacturer ____
○ Monitor Supplied By (3.4.2.5) ____
○ Location ____ Enclosure ____
○ Manufacturer ____
○ Alarm ____ ○ Shutdown ____
○ Shutdown Time Delay ____ Seconds
AXIAL (3.4.2.1)
○ Manufacturer ____ No. Required ____
○ Location ____
○ Oscillator-Demodulators Supplied By ____
○ Manufacturer ____
○ Monitor Supplied By (3.4.2.5) ____
○ Location ____ Enclosure ____
○ Manufacturer ____
○ Alarm ____ ○ Shutdown ____
○ Shutdown: Time Delay ____ Seconds ____
ACCELEROMETER (3.4.3.1)
Manufacturer ____ No. Required ____
Location ____
Monitor Supplied By (3.4.2.5) ____

☐ RADIAL BEARINGS

	Pinion	Gear
Type		
Diameter, In		
Length, In		
Journal Velocity, FPS		
Loading, PSI		
Clearance (min-max), In		
Span, In		

☐ THRUST BEARING(S)

Location		
Manufacturer		
Type		
Size		
Area, In^2		
Loading, PSI		
Rating, PSI		
Int. Thrust Load, Lb (+)(-)		
Ext. Thrust Load, Lb (+)(-)		

☐ COUPLING(S)

Manufacturer		
Model		
Cplg. Rating, HP/100 RPM		
Cplg. Gear Pitch Dia., In		
Cplg. Press. Angle, Deg.		
Cylindrical / 1-Key	☐	☐
Cylindrical / 2-Keys	☐	☐
Tapered / 1-Key	☐	☐
Tapered / 2-Keys	☐	☐
Tapered / Keyless	☐	☐

☐ MATERIALS

Gear Casing ____ Oil Seals ____
Radial Bearings ____
Thrust Bearing(s) ____
HS Shaft ____ LS Shaft ____
Pinion(s) ____ Hardness ____
Gear Rim(s) ____ Hardness ____
Low Temp. Operation (2.9.4) ____

☐ PIPING CONNECTIONS

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

NOTES: ____

Figure A-2. API Gear Data Sheet Two. (Courtesy American Petroleum Institute)

SPECIAL PURPOSE GEAR UNITS
API 613 FOURTH EDITION
DATA SHEET
U.S. CUSTOMARY UNITS

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

INSTRUMENTS

- Mercury Thermometers (3.4.1.1) ____________
- Bearing Metal Temp. Sensors (2.7.1.3) ____________

CONTRACT DATA

- Vendor's Rep At Site (2.1.10)
- Test Data Prior To Shipment ____________
- Progress Reports (5.3.4) ____________

SHIPMENT (4.4.1)

	Contract Unit	Spares
Export Boxing (4.4.3.9)	O	O
Domestic Boxing	O	O
Outdoor Storage Over 6 mos.	O	O

COUPLINGS AND GUARDS.

	High Speed	Low Speed
Coupling Furnished By		
Coupling Type		
Coupling Lubrication		
Mount Coupling Halves (3.2.1)		
Taper		
Limited End Float		
Cplg. Guard Furnished By		

LUBRICATION REQUIREMENTS

- Oil System Furnished By (2.8.3) ____________
- Oil Visc.: ____SSU@100 F____SSU@210 F (2.8.6)

INSPECTIONS AND TESTS (4.1)

	Req'd	Wit-ness	Ob-served	Test Log
Shop Inspection (4.1.1.1)	O	O	O	
Cleanliness Inspection (4.2.3.2)	O	O	O	
Hardness Verification Inspection (4.2.3.3)	O	O	O	O
Dismantle-Reassembly Inspection (4.3.2.3.1)	O	O	O	
Contact Check (2.5.2.2)	O	O	O	O
Contact Check Tape Lift (2.5.2.2)	O	O	O	O
Journal Runout Check (2.5.2.1)	O	O	O	O
Axial Stability Check (2.5.2.3)	O	O	O	O
Rotor Balancing Machine Sensitivity Check (2.6.2.3)	O	O	O	O
Residual Unbalance Check (2.6.2.3.3)	O	O	O	O
Mechanical Run Test (4.3.2)	O	O	O	O
Mechanical Run Test (Spare Rotors) (4.3.2.4)	O	O	O	O

LUBRICATION REQUIREMENTS

Min. Startup Oil Temperature	____________	F
Unit Oil Flow (Total)	____________	GPM(US)
Unit Oil Pressure	____________	PSI
Oil Flow, Mesh	____________	GPM(US)
Oil Flow, HS Bearings	____________	GPM(US)
Oil Flow, LS Bearings	____________	GPM(US)
Oil Flow, Thrust Bearing(s)	____________	GPM(US)

ADDITIONAL REQUIREMENTS

Filter Breather Location (2.3.1.10) ____________

GEAR DATA

Power Loss Each HS Bearing ____________
Power Loss Each LS Bearing ____________
Power Loss Each Thrust Bearing ____________

	Pinion	Gear
Outside Diameter, In		
Root Diameter, In		
Center Groove Diameter, In		
Durability Power		
Strength Power		
Face Overlap Ratio		
Transverse Contact Ratio		
Length Line of Action, In		

NOTES

Figure A-3. API Gear Data Sheet Three. (Courtesy American Petroleum Institute)

**API 613 FOURTH EDITION
DATA SHEET
U.S. CUSTOMARY UNITS**

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

O INSPECTIONS AND TESTS (Con't.)	Req'd.	Witness	Observe	Test Log
Add'l. Mechanical Tests(4.3.2.2.15)	O	O	O	O
Part or Full Load And Full Speed Test (4.3.3.1)	O	O	O	O
Full Torque, Slow Roll Test (4.3.3.2)	O	O	O	O
Full Torque Static Test (4.3.3.3)	O	O	O	O
Back-To-Back Locked Torque Test (4.3.3.4)	O	O	O	O
Sound Level Test (4.3.3.5)	O	O	O	O
Additional Gear Tooth Test (4.2.2.8)	O	O	O	O
Use Shop Lube System	O	O	O	
Use Job Lube System	O	O	O	
Use Shop Vibration Probes, Etc.	O	O	O	
Use Job Vibration Probes, Etc.	O	O	O	
Other (4.2.1.2)	O	O	O	
Final Assembly, Maintenance & Running Clearance (4.2.1.1.e)	O	O	O	O
Oil System Cleanliness (4.3.2.1.3)	O	O	O	
Oil System-Casing Joint Tightness (4.3.2.1.4)	O	O	O	
Warning And Protection Devices (4.3.2.1.5)	O	O	O	

NOTES

NOTES

Figure A-4. API Gear Data Sheet Four. (Courtesy American Petroleum Institute)

Table A-5. Gear Testing Comparison.

	Full Speed No Load Test	Full Speed Part Load Test	Full Torque Slow Roll Test	Back-to-Back Locked Torque Test	Full Torque Static Test	Full Speed Full Load
Speed	Rated	Rated	Slow roll	Rated	Static	Rated
Load	No load	Part load	Part load	Full	No load	Full
Torque	----	Partial	Full	Full	Full	Full
Contract vibration equipment by instrumentation	✓	✓	----	✓	----	✓
Contract ½ coupling and spaced simulated weight	✓	✓	----	----	----	✓
Overspeed test (110% of MCS for 15 minutes)	✓	✓	Not applicable	Not stated	Not applicable	Yes
Verify lateral critical speed	✓	✓	No	Yes	No	✓
Verify vibration and bearing oil temp duration	4 hours (API 613) 1 hour (API 677)	Yes	Yes	Yes	----	Yes
Disassembly inspection (bearings, tooth contact)	Yes	Yes	----	Yes	----	----
Remarks	Mandatory test per API 613/677 Detects bearing temperature and critical speed problems	Optional test Tooth contact and noise check	Optional test Demonstrate tooth contact and load carrying capability	Optional test Demonstrates tooth contact and torque carrying capability at full torque load and speed (with one gearbox operating in reverse tooth loading)	Not recommended Checks ability of shaft, keys, and gear teeth to carry load	Optional test Verify bearing design, tooth contact, etc. Difficult to determine balance and dynamic performance due to bearing reaction in opposite direction
Budget cost (for 10,000 to 20,000 hp unit)	\$3000	\$5000	\$8500	\$32,000	\$3400	\$5000 (up to 2000 hp load)

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