A Comparison of Current AGMA, ISO and API Gear Rating Methods

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Introduction

There are many different gear rating methods in use today, and they can give substantially different results for any given gearset. This paper will make it easy to understand the choices and the impact the choices have on gearbox design. Eight standards are included — AGMA 2001; AGMA 6011; AGMA 6013; ISO 6336; API 613; API 617; API 672; and API 677. A brief introduction and history of each standard is presented, and the basic differences between them are highlighted. Two sets of examples are used to illustrate the differences. These examples are presented in both tabular and graphical format, and are fully discussed. The first set contains a wide range of gears, and each gearset is rated by each standard. The second set compares gears designed for a specific set of requirements according to each of these standards. The perils of increasing service factor are mentioned, particularly in regard to high pitch line velocity gears. Finally, there is a discussion of how to make a gearbox more reliable without changing the rating method or service factor. The choice of rating method can have a huge impact on the size of the gearbox, and this paper should help avoid specifying the wrong standard and having an oversized gearbox. It should also be useful as an aid to customers who are unsure of the differences between the standards.

Description of the Standards

API 613 — 5th edition (2003): ***Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.*** Most of the main gearboxes in refineries must conform to this specification. This is the most conservative standard, and if you specify this, you will probably pay substantially more for the gearbox than if another standard was specified. This standard is for parallel shaft helical gear units that are in continuous service without installed spare equipment. The gears may be single or double helical, one or two stage, and may be designed as reducers or speed increasers, but it does not apply to integrally geared units such as integrally geared compressors (which are covered by API 617 and 672). Most of its requirements do not apply to general purpose gears since they fall under API 677; however, gear ratings calculated according to API 613 and API 677 are the same. API 613 covers not only gear rating, but also the related lubricating systems, controls, and instrumentation. It was first published in 1968 based on AGMA formulas, but in 1977, the second edition was published with a very simplified approach. It was designed so preliminary sizing of gearing could easily be done with just a slide rule. It does require the Geometry Factor "J" from AGMA 908, but before the age of computers, this was often estimated from graphs. This simple method is still the one used in API 613, even though slide rules are hard to find and engineers who know how to use them are becoming quite rare. The very conservative ratings stem mainly from basing the material allowable stresses on the lowest grade materials (grade 1) from the AGMA standard in effect in 1977, even though use of the better "grade 2" materials is required. Although AGMA allowable stresses have increased over the years to reflect increasingly stricter metallurgical requirements, improved metallurgy, and extensive field experience, the API ratings have remained unchanged. The sixth edition is currently in development and may be published this year (2018). It appears that the rating equations will change to mirror those in AGMA 2001, but there will be a derating factor introduced so the resulting ratings may be similar to those of the prior editions. However, it does incorporate language to allow the use of alternate rating methods if the API method would result in excessive pitch line velocity or excessive face width.

API 617 — 8th *Edition* (2014): ***Axial and Centrifugal Compressors and Expander-Compressors; Part 3 — Integrally Geared Centrifugal Compressors***. This was first published in 1958 and covered only barrel-type centrifugal compressors, since integrally geared centrifugal compressors did not exist at that time. The 2002 seventh edition expanded the scope to cover Integrally Geared Centrifugal Compressors and Expander-compressors. It is now essentially three standards packaged as one. Each section has its own set of annexes, and for integrally geared centrifugal compressors, an annex in part 3 specifies a rating method based directly on ANSI/AGMA 2001. This method specifies how each factor is to be calculated, and then imposes an additional 20% derating factor. So, it is quite conservative, but not nearly as conservative as API 613. The eighth edition of API 617 was published in 2014 and did not change this rating method.

ANSI/AGMA 2001-D04 (2004): ***Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth***. AGMA 2001 and 2101 (the metric version) are the basic AGMA gear rating standards that most other AGMA rating standards are based on, and they have evolved from standards originally published in 1946. The ratings calculated by these standards have slowly risen over the years as a result of higher allowable stress numbers that have been introduced along with stricter metallurgical requirements. The user is given some flexibility in selecting the values of the factors to be used in the rating, so even given complete information on a gearset, two engineers may use different values for some factors and come up with different ratings using this standard. Therefore, specific application standards such as API 617 part 3, AGMA 6011, or AGMA 6013, provide guidance on selecting the factors to be used in the rating. The AGMA Helical Gear Rating Committee has been working for many years to revise this standard, but it may be a while before a new revision is released.

ANSI/AGMA 6013-B16 (2016): ***Standard for Industrial Enclosed Gear Drives.*** This standard is for low- to moderate-speed gears. This, and its metric version AGMA 6113-B16, is a combination of prior standards ANSI/AGMA 6009-A00 and ANSI/AGMA 6010-F97 — which in turn were based on AGMA 480, AGMA 460, and AGMA 420. It presents general guidelines for design, rating, and lubrication of parallel, concentric, and right-angle shaft drives. However, this paper will only consider the rating of parallel shaft gearboxes. For these gearboxes, this standard only applies when the pitch line velocity does not exceed 7,000 ft/min (35.56 m/s). It specifies that ANSI/AGMA 2001-D04 is to be used for the rating, and provides the specific factors to be used. The rating is for 10,000 operating hours, using the least conservative life factors.

ANSI/AGMA 6011-J14 (2014): ***Specification for High-Speed Helical Gear Units.*** The first high-speed gear unit standard was adopted in 1943 and has evolved over time. It is now based on ANSI/AGMA 2001-D04 and applies when the pitch line velocity exceeds 6,890 ft/min (35 m/s). The factors to be used for rating are either specified or a specific calculation procedure is given. The rating is for a minimum of 40,000 operating hours, using the most conservative stress cycle (life) factor. However, if the number of stress cycles exceeds the stress cycle factor graph endpoint, then the designer has the option of using the graph endpoint or extrapolating the curve to lower values.

ISO 6336-2006 (with the exception of part 5, released in 2003): ***Calculation of Load Capacity of Spur and Helical Gears.*** This standard, which is composed of five separate parts, is largely based on prior DIN standards and is generally accepted everywhere outside of the United States. It contains multiple methods to establish ratings, including method "A" (testing the gears under simulated or actual operating conditions) and various calculation methods. In general, method "B" should be used. There are a number of fundamental differences between the AGMA and ISO rating methods. The ISO standard finds the calculation points for bending strength by fitting an equilateral triangle into the base of the tooth, whereas the AGMA method is to use the Lewis parabola. The ISO dynamic factor is based on shaft vibration and proximity to a critical speed based on a very simplistic model of the shaft, while the AGMA dynamic factor is based mainly on allowable single tooth pitch variation. Yet despite these and other differences, the gear ratings are often fairly similar. The working group ISO/TC60/SC2/WG6 is currently revising Parts 1–3, and a new edition might be published in 2018 or 2019.

API 672 — 4th edition (2004): ***Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical, and Gas Industry Services.*** This was originally published in 1979, with the fourth edition published in 2004. This standard directs the user to rate the gears according to ANSI/AGMA 6011.

API 677 — 3rd edition (2006): ***General-Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.*** This was first published in 1989 and used a modified K factor rating method. The 1997 second edition changed the rating method to that given in API 613. The current third edition was published in 2006.

Some Standards Use Service Factors, Others Use Safety Factors

Service factors have long been used as a simple method to provide an appropriate margin when designing gears. API 617, API 672, AGMA 6011, and AGMA 6013 use a service factor that includes the combined effects of safety factor, overload, and reliability (for pitting, these factors are *SH*, K_O, Y_z, and for bending SF, K_O, Y_z). API 613 and API 677 use the service factor as the sole factor, so their service factors also include the dynamic, size, load distribution, stress cycle (life), and temperature factors — plus either surface condition factor (for pitting) or rim thickness factor (for bending strength).

AGMA 2001 allows the use of either service factor or safety factor — but they are NOT interchangeable. ISO 6336 uses safety factors, and in addition to a lot of other factors also uses an application factor. It should be noted that, with the exception of the load distribution factor, the factors used in ISO are calculated quite differently from those used in AGMA.

Differences between Ratings Standards for Specific Gearsets

In this section the maximum power ratings according to six different gear rating methods will be compared for fourteen sets of gears covering a range of sizes and speeds. There are only six unique methods in the eight gear rating standards mentioned here. API 672 states that the gears shall be rated to ANSI/AGMA 6011. Similarly, the section on gear rating in API 677 has the same equations, factors, and limits as API 613, except for a minor difference in allowable L/d ratio (pinion face width to reference diameter) for nitrided gears.

The gearsets used in this comparison are presented in Table 1. All are grade 2 (MQ for ISO) alloy steel, and carburized (58 Rc), nitrided (R 15N 90), or through hardened (321 BN) as noted. No profile shift was used and all sets were run on standard center distance. Speeds range from 700 to 45,000 RPM. The resulting ratings range from 200 to over 20,000 HP. An even wider range of gears could have been analyzed, and additional examples could show more variability, but that probably would not change the general conclusions of this study. The values and factors chosen are sufficient for the purposes of this study, but they were selected for simplicity; they do not represent actual gears in production and should not be used as a recommendation or guide for gear design.

Ratings are for 20 years of continuous operation, except ANSI/AGMA 6011-J14, which specifies that ratings are for a minimum of 40,000 hours. Therefore, for comparison, ANSI/AGMA 6011 ratings are presented both for 40,000 hours and 175,200 hours (20 years). The ANSI/AGMA 6013 ratings are for 10,000 hours, as stipulated. The rating results are presented even if the pinion speed or the pitch line velocity was too high or low for the standard to apply.

Table 1 Geometry and speeds of example gear sets

Set Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Type	increase	increase	increase	increase	increase	increase	reduce	increase	reduce	increase	increase	reduce	reduce	reduce
Bull gear teeth	151	167	151	151	167	167	167	97	173	367	151	173	59	97
Pinion teeth	29	35	29	29	35	35	35	29	35	30	29	35	35	29
Module, mm	5	3	5	5	3	3	3	6	2	2	5	2	3	6
Pressure Angle	20°	25°	20°	20°	25°	20°	25°	25°	25°	25°	20°	25°	25°	25°
Helix Angle	18°	16°	18°	18° Double	16°	16°	16°	25° Double	16°	20°	18°	16°	15°	25° Double
Center distance	18.63	12.41	18.63	18.63	12.41	12.41	12.41	16.42	8.52	16.63	18.63	8.52	5.75	16.42
Face width, inch	6.25	5.50	6.25	8.25	5.50	5.50	5.50	8.00	3.00	2.75	6.25	3.00	4.50	8.00
Reference diameter, inch	6.00	4.30	6.00	6.00	4.30	4.30	4.30	7.56	2.87	2.51	6.00	2.87	4.28	7.56
Input Speed, RPM	3600	3600	3600	3600	3600	3600	3600	4500	3600	3600	3600	3600	3600	4500
Output Speed, RPM	18,745	17,177	18,745	18,745	17,177	17,177	754	1,345	728	44,040	18,745	728	2,136	1,345
Pitch line velocity, ft/min	28,796	19,339	29,456	29,456	19,339	19,339	4,053	8,905	2,702	28,983	28,796	2,702	4,034	8,905
Heat Treatment	Nitrided	Nitrided	Carb.	Carb.	Carb.	Carb.	Carb.	Carb.	Carb.	Carb.	Thru Hard	Thru Hard	Thru Hard	Thru Hard
Notes	RPM above 6013 limit	RPM above 6013 limit	RPM above 6013 limit	RPM above 6013 limit	RPM above 6013 limit	RPM above 6013 limit	RPM below 613, 677, 672, 6011 limits	RPM above 6013 limit	RPM below 613, 677, 672, 6011 limits	RPM above 6013 limit	RPM above 6013 limit	RPM below 613, 677, 672, 6011 limits	RPM below 613, 677, 672, 6011 limits	RPM above 6013 limit

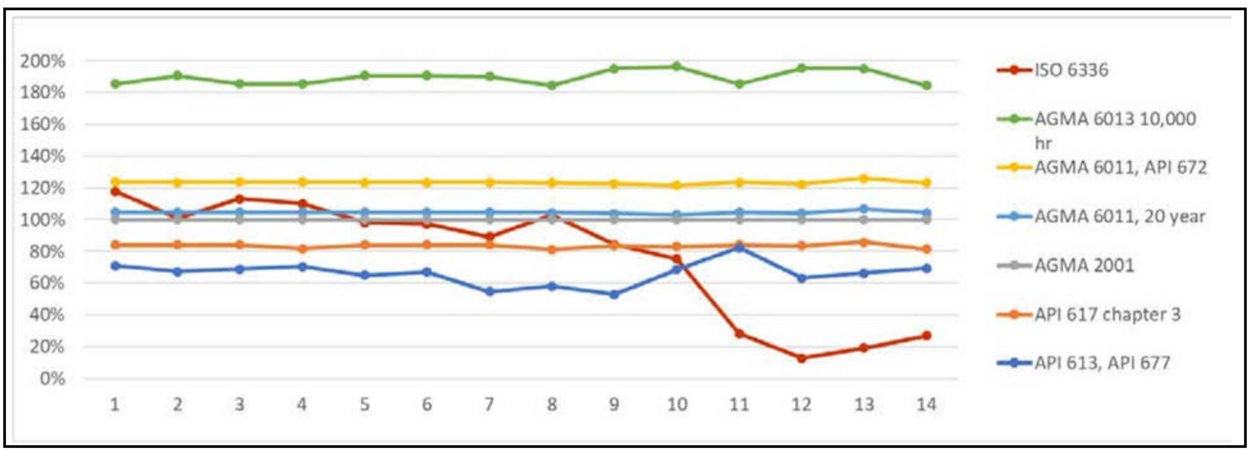

Figure 1 Pitting ratings as a ratio to AGMA 2001 pitting rating.

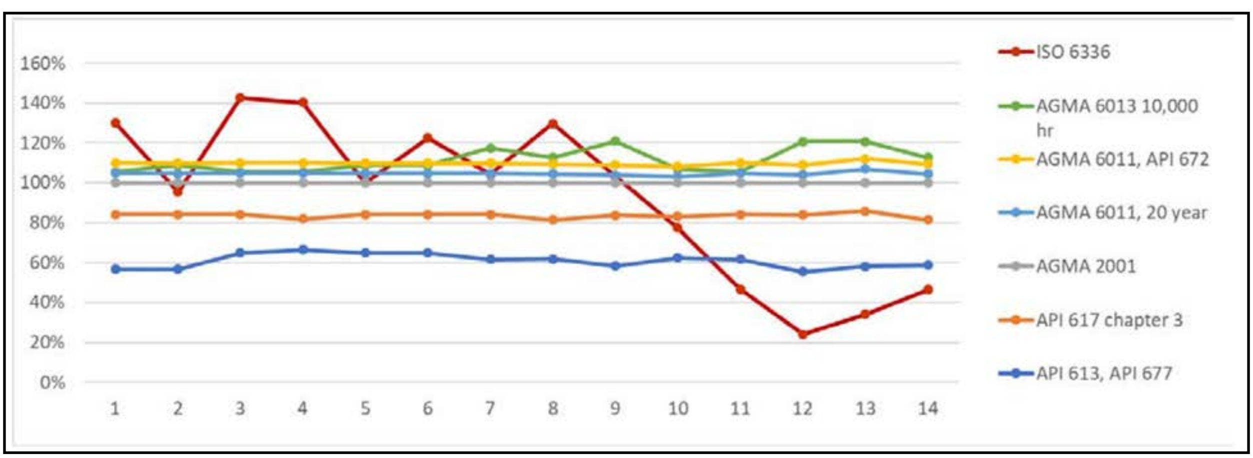

Figure 2 Bending ratings as a ratio to AGMA 2001 bending rating.

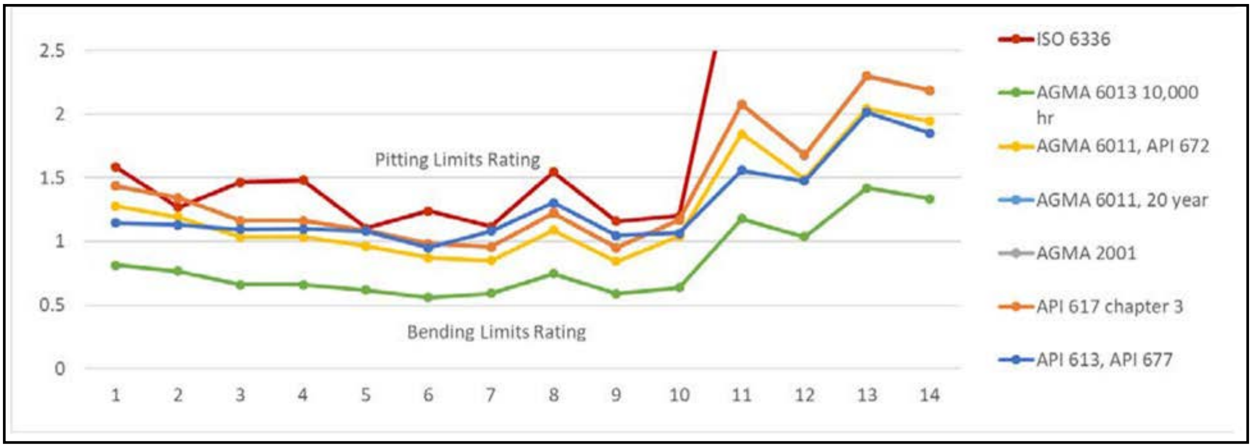

Figure 3 Ratio of bending rating to pitting rating.

Because of the wide range of power these sets are capable of transmitting, the results in Figures 1 and 2 are presented as the ratio of the rating to the ANSI/AGMA 2001-D04 rating. Each line represents one rating standard. A line chart is used for clarity; it is not meant to imply any relationship between different gearsets other than they are being rated with the same method. The order of the sets is arbitrary, except that the nitrided sets are presented first, followed by the carburized sets, and then the through hardened ones. For the pitting ratings shown (Fig. 1), all the ratings that use AGMA methods as their basis are quite consistent for the cases studied. API 613 ratios show a lot more variability, due to factors in the AGMA standards that API 613 does not use. The major change comes with a change to through hardened material (sets 11–14), and ISO rates through hardened steels far lower than AGMA does. This may be due to historical differences — particularly cleanliness — between the through hardening steels used in Europe and those used in the United States.

For most of the example gearsets, the AGMA 6011 ratings are about double the API 613 ratings. This is a staggering difference! The API 613 ratings for case and surface hardened gears are consistently the lowest, both for bending and pitting. The highest ratings come from ISO 6336 and ANSI/AGMA 6013, though the inclusion of 6013 may be a bit unfair since it uses stress cycle factors for only 10,000 hours of operation. All the other AGMA ratings are fairly consistent.

Figure 2 compares the bending ratings to ANSI/AGMA 2001-D04. Again, all the ratings that use AGMA methods as their basis are quite consistent for the cases studied. It is not surprising that the ISO 6336 methods do not track the AGMA method very well at all, since the rating methods are quite different. Also, the low ISO ratings for sets 11–14 correspond to the through hardened gearsets.

The ratio of bending rating to pitting rating is shown for each example and each rating method in Figure 3. When the ratio is above 1.0, i.e. — when the bending rating is above the pitting rating — bending ratings are ignored and the surface durability ratings determine the gearset ratings. It can be seen that whether it is pitting or bending that determines the overall rating, both depend on the gearset in question and the rating standard used. For any standard, examples can always be found where pitting limits the set rating, and other examples will show that bending limits the rating.

Many designers strive for gearsets that have close to "balanced" ratings, but often with the pitting rating slightly lower than the bending rating. This means that the gears are more likely to pit than break. It is far better for the gears to become noisy due to pitting and therefore get inspected and repaired or replaced, rather than breaking and potentially ruining the whole gearbox. But a balanced gearset according to one method may not be balanced according to another method.

It should be noted that when using AGMA or API standards, usually the same service factor is used for both the pitting rating and bending rating. However, when using ISO 6336, often a much higher safety factor is used for bending than is used for pitting.

It is interesting to note that the graphs show that the ratings remain consistent even outside the scope specified in the standards. However, a standard should not be specified if the application is not within the scope.

Most gear experts recognize that the ratings from the standards are just a rough approximation of the power that can be safely transmitted through the gears. The truth of this becomes obvious as the results of this study are examined. There is only one power level that will cause failure after a specific number of hours of operation, yet different standards give vastly different approximations of what that load is. Since gear failures are not common, clearly even the least conservative standards are sufficient for most applications. Yet when a standard has been specified, the gear vendor must ensure that the gear rating according to the specified standard meets the specified power.

The Positive and Negative Consequences of Imposing a More "Conservative" Design

Purchasers sometimes try to assure themselves that gears will be very reliable by the selection of a "conservative" rating standard or by increasing the required safety or service factors. The advantage of doing this is the supposedly lower chance of failure. However, if an adequately sized gearset will not fail, it is already sufficiently reliable. A larger gearset will not be more reliable. For low-speed sets, the only negative consequences of being "conservative" may be size, price, and slightly higher operating costs due to higher losses. For high-speed sets, being "conservative" can lead to high face widths or high pitch line velocities that can have significant negative consequences. Increased face width not only makes the gearset more sensitive to alignment, it is detrimental due to the heating of the oil, which is transported across the face width as the contact line sweeps across. The further the oil travels across the face, the higher its temperature gets. Increased pitch line velocity leads to increased sliding velocities, which also lead to a higher temperature in the contact zone and higher risk of varnishing or scuffing. In some cases, high tooth temperatures have resulted in a metallurgical transformation that distorted the helix, thereby adversely affecting the load distribution across the tooth flanks. As John Amendola (CEO, Artec Machine Systems; AGMA standards committees) has said: "So bigger is not necessarily more conservative. In reality, the most important factors are good load distribution, low sliding velocities, and proper lubrication."

How to Reduce the Risk of Failure

The load that will cause failure depends on many things, so an accurate rating can only be determined by testing. However, in many cases, testing to determine a safe load over the full life of a gearbox is not practical — *which is why rating standards exist*. The rating standards provide minimum requirements that must be met for the rating to be valid. The gear cost can be minimized by just meeting these minimum requirements, but by going beyond them, an extra margin of safety can be achieved. Rather than simply increasing the required service or safety factors or specifying the use of a very conservative rating standard, every aspect of the gearbox should be carefully examined. The first step is to determine the maximum load and the load spectrum based on a full analysis of the application. Additionally, there are many things that should always be considered —especially for critical applications. There are many standards — such as those from AGMA and ISO, as well as many books — that provide a great deal more information on these topics. The following very brief list just touches on some of the things that should be considered to reduce the risk of a failure:

- ***Lubricant used.*** The viscosity, the FZG load stage, the base stock, and the additives used all have a significant role in the life of a gearset. The lubricant can make the difference between successful operation and failure not only for pitting, but also for scuffing and micropitting. It is essential to keep the oil free of water and to change it at appropriate intervals. Proper filtration of the lubricant is critical, since entrained particles can result in wear. In some cases, use of an electrostatic filter to remove submicron particles may even be justified. See ANSI/AGMA 9005-F16 for more information on lubricants.
- ***Application of lubricant to the gear teeth.*** While in some cases, occasionally painting tar on the teeth of very large and slow-moving gears may be sufficient, and dip or splash lubrication is adequate for moderate speed gearing (up to about 15 m/s or 3000 ft/min pitch line velocity), high speed gears require spray lubrication. This spray may be directed into the in-mesh of the gears, or on higher speed gears into the out-mesh where the partial vacuum created by the separating teeth helps suck the oil mist onto the tooth flanks, or the system may use multiple nozzles on both the in-mesh and out-mesh to provide optimal lubrication and cooling. When spraying both the in-mesh and out-mesh, usually about one third of the flow goes to the incoming side for lubrication and the rest goes to the outgoing side for cooling.
- ***Temperature of the gear teeth.*** The gear teeth normally are cooled by the flow of lubricant, both on the teeth themselves and on their sides. While sufficient lubrication is essential, with high speed gears, excessive lubricant flow can be detrimental and lead to excessive heat generation and power losses. In high speed gears, oil that gets between the teeth is often ejected axially, sometimes at supersonic speeds when the gears have high pitch line velocity and low helix angles. Excessive oil mist surrounding the gears can lead to high windage losses, raising the bulk temperature of the gears. Excessive temperatures in the contact zone can lead to varnishing, scuffing, or other problems. With pressure-fed systems, the oil temperature is typically controlled with oil coolers. When the gearbox is in a cold environment, it is good practice to preheat and circulate the oil prior to startup so it has an acceptable viscosity during startup.
- ***Micro-geometry of the gear teeth.*** Proper profile modifications will decrease the chance of problems. Highly loaded gears often require tip relief to avoid the tip of the driven gear from gouging into the flank of the driving gear. Helix (lead) modification can, and in many cases should, be used to compensate for tooth deformations that will occur during operation, both from the load and the temperature profile of the tooth flanks. The use of ISO1328-1 class 4 or better tolerances for the tooth flanks may be appropriate for some gears to assure that the specified modifications are achieved, although the use of such tight tolerances may not be appropriate for general purpose or low speed gears where class 6 or 7 is considered good.
- ***Alignment.*** The best gears in the world can fail if not properly aligned. In addition to the parallelism of the bores machined into the gearbox, bearing play, differential thermal growth, and internal or external load-induced distortions of either the gearbox or gears themselves should be accounted for.
- ***Material used.*** The gear material is obviously critical to the life of the gears. It is important to consider the specific material chemistry, the material cleanliness, its processing (hot or cold worked, total reduction ratio, forged or rolled), and heat treatment. The following brief comments barely scratch the surface of gear metallurgy. For more information, see AGMA 923-B05 or consult with a gear metallurgist.
  - ¤ The appropriate alloy should be selected for the application. Some steels are easier to harden than others, but note that there can be significant differences between different batches of the same alloy. The material chemistry of the specific batch can affect the hardenability. Jominy end-quench tests can be used to assess hardenability, and published ranges can be used to aid in the selection of which alloy to use. They may also be incorporated into the specification of the properties the alloy must have.
  - ¤ Material cleanliness is critical, since inclusions can be stress risers and be the initiation points for failures. Cleaner steels can safely carry higher loads.
  - ¤ The processing of steel from billet to final part can have an effect on the

life of the part. Sufficient reduction ratios are beneficial, and appropriate forging, such as pancake forging for bull gear disks, can result in favorable grain size and structure.

¤ Heat treatment is used to obtain the proper hardness distribution in the gear. Specification of a better hardenability material can be negated by improper heat treatment. The spacing of the gears in the furnace and during quenching, the quenchant used, and the flow rate and amount of agitation of the quenchant will all affect the heat treatment results. Larger sections are more difficult to properly heat treat than small ones, and so may require materials with better hardenability.
  - ◊ Hardness and strength are generally proportional, so the harder the gear, the higher the rating will be. For a given required power, it is not unusual for a higher hardness specification to result in a less expensive gear since the harder gear can be smaller. For case or surface hardened gears, just as critical as the hardness is the hardness profile. If the hardness falls too rapidly with depth, then at some depth from the surface, the sub-surface stress can exceed the strength, leading to a subsurface failure that can grow to the surface. Jominy data along with knowledge of the part size, heat treatment, and quench severity is useful to predict the hardness profile.
  - ◊ Use of through hardened gears is common, even though their hardness is considerably lower than that of surface or case hardened gears. Since they are heat treated before machining, they can be machined to final size without worrying about the changes that can occur during heat treatment. Machining becomes more difficult or impossible as hardness increases, but the hardness cutoff point for through hardened gears varies by manufacturer, and it has increased over the years due to advances in manufacturing technology.
  - ◊ Flame or induction hardening can produce a hardened surface layer, and dual frequency induction hardening can produce a particularly good surface layer. However, API 613 and 677 do not recognize flame or induction hardening. Also, these hardening processes require numerous test pieces to certify the process, so they may not be suitable for very low volume or one-off production.
  - ◊ Nitriding produces a very thin but very hard surface layer, so it is very good at reducing the chance of pitting.
  - ◊ Some people consider case carburized gears to be the best, and in some cases, they may also be the least expensive since they can be smaller than other gears rated for the same power. Case depth needs to be controlled to be sure that it is sufficient to avoid a subsurface failure, but not excessive since gear tooth tips may become brittle and break.
  - ◊ It is not unusual to use different hardness for the pinion and bull gear specifications. When there is a difference, the pinions are usually harder due to higher stress in the pinions, resulting from their tooth shape and their having more stress cycles.

- ***Surface finish***: Improved surface finish generally leads to improved gear performance. In addition to minimizing surface roughness, the lay of any machining or grinding marks can be important. There used to be a theory that some roughness was required to hold an oil film, but testing on isotropic superfinished surfaces has disproved that. Careful grinding can produce a $16r_a$ (micro-inch) finish, while isotropic superfinishing can bring it down to $2r_a$. Claimed benefits include reduced noise, reduced gear wear, increased power output, increased part life, and lower operating costs. Of course, as with all manufacturing processes, a cost benefit analysis should be performed to determine the optimal level of surface finish for the application.
- ***Dynamic loads including vibration***: It is critical to know the maximum load that the gearset will ever see, and preferably the lifetime load spectrum will be known. The entire wind energy business was almost brought to a complete halt due to miscommunication of maximum loads. Vibration, either lateral or torsional (which may be difficult to detect), can ruin gears. Proper analysis during the design stage can generally be used to guide any necessary changes so damaging vibrations will not occur during operation.

A good gearbox designer or vendor will look at all of these, and thus be able provide a very reliable design no matter which standard is specified. However, the size and therefore the price of the gearbox will be affected by the rating standard chosen.

Effect of Rating Standards on the Size of a Gearset Designed for a Specific Application

As an example of the effect the rating standard can have on the size of a gearbox, Table 2 presents designs of gearsets that are rated at 4,800 HP for 20-year life, according to five standards. In all cases, the rating is pitting limited. The only changes made to meet the rating were to adjust the module and face width, keeping the L/D ratio for the pinion at approximately 1.0. While it would be very unusual to actually make gears with such odd modules, this example serves to illustrate the average effect rating standards have on one particular set of design conditions. Actual designs would use standard modules, so changes in numbers of teeth would be made to get close to the rating. If only number of teeth were changed, then for designs such as this, which are close to being balanced between pitting and bending, increasing the number of teeth could cause the set to become bending limited.

Since the cost of a gearbox is roughly proportional to the volume of the gears, the API 613 gearbox will cost about 60% more, even if all other design criteria are kept the same. But even if the extra cost of the gearbox is not a concern, the increased pitch line velocity and increased face width should be. It can be seen that for this case, use of API 613 results in almost 20% higher face width and pitch line velocity than that which would result from designing to AGMA 6011. While this may not be a serious issue when the pitch line velocity is not very high, it can become a major problem when the power and speed requirements require a pitch line velocity approaching or exceeding 30,000 ft/min (150 m/s). So being "conservative" in the specifications can sometimes result in a compromised design.

Conclusions

When a gearbox is properly specified and built so it will not fail, then there is no way to make it more reliable. There is an old engineering saying: good enough is best. Specifying a different standard or increasing service or safety factors can make the gear box more expensive, but if the gearbox would be adequate without the additional expense, then nothing is gained by adding requirements. In fact, being too conservative in the specification of a gearbox may have negative consequences.

It is important to fully understand all the loads and environmental conditions the gearbox will be subjected to so that the gearbox requirements can be properly specified. It is very important to properly specify all loads, the expected operating life, and any special circumstances so the proper factors can be specified for the rating. The standard specified for gearbox rating and the service or safety factors should be appropriate for the application and should not be excessively conservative.

Table 2 Gearbox size as a function of rating standard

	units	ISO 6336	ANSI/AGMA 6011 20 year	ANSI/AGMA 2001	API 617 chapter 3	API 613, API 677
Number of teeth, bull gear		173	173	173	173	173
Number of teeth, pinion		35	35	35	35	35
Module	mm	2.84	2.97	3	3.18	3.54
Pressure Angle	deg	25	25	25	25	25
Helix Angle	deg	16	16	16	16	16
Material		carburized	carburized	carburized	carburized	carburized
Face Width	inch	4.03	4.2	4.3	4.65	5.06
Pinion Pitch Diameter	inch	4.071	4.257	4.300	4.558	5.075
L/D		0.990	0.987	1.000	1.020	0.997
Gear Pitch Diameter	inch	20.123	21.044	21.257	22.532	25.083
Pinion volume	inch^3	52.5	59.8	62.5	75.9	102.3
Gear volume	inch^3	1281.7	1460.8	1526.0	1854.1	2500.3
Total volume	inch^3	1334.1	1520.6	1588.4	1930.0	2602.6
Input Speed	rpm	3600	3600	3600	3600	3600
Output Speed	rpm	17794	17794	17794	17794	17794
Pitch line velocity	ft/min	18965	19833	20034	21236	23640
Pitch line velocity as % of ANSI/AGMA		94.7%	2001 99.0%	100.0%	106.0%	118.0%
Volume ratio to 2001		84.0%	95.7%	100.0%	121.5%	163.8%

Note: The ANSI/AGMA 6013 standard was not included in this comparison since it specifies 10,000-hour life, as opposed to the 175,200-hour (20-year) life used in these examples.

References

1. API 613 Sixth Edition: Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.
2. API 617 Eighth Edition: Axial and Centrifugal Compressors and Expander-compressors; Part 3 — Integrally Geared Centrifugal Compressors.
3. API 672 Fourth Edition: Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical, and Gas Industry Services.
4. API 677 Third Edition: General-Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.
5. AGMA 2001-D04: Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

John Rinaldo is retired from Atlas Copco Comptec LLC where for 25 years he designed gears for high-speed, integrally geared centrifugal compressors. He is currently a member of the API 613 taskforce, and serves as the vice chair of the AGMA Gear Accuracy committee and the Nomenclature committee. He is the convener of ISO TC60/SC1/WG4 "Terminology and notation of gears" and is the U.S. delegate to ISO TC60/WG2 "Accuracy of gears" working group. His varied career started with the aerodynamic design of compressor impellers, shifted to the design of compressor control systems and then moved to general research and development of centrifugal compressors. He has been licensed as a Professional Engineer in both Wisconsin and New York, has been granted 4 patents, and is a recipient of the AGMA Distinguished Service award.

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Appendix – Example 1 runs

ISO 6336 2006 Rating, version 2.0031 | Data Set: 1 | Page 1
FTM Paper Gear Set 1 | 2017/07/27 | 13:54:24
151-29 5 mn a 20 18 helix | American Gear Manufacturers Association
Nitrided | Gear Rating Suite - GUI Version 3.0.170

**** Gear Geometry Error Messages ****

42) Note: Zero backlash x factors are not being used for rating. The sum of X1 + X2, -0.1648 does not correspond to the value 0.0000 calculated from the center distance and the pressure angle.

**** Velocity Error Messages ****

4) WARNING: X-Factors are outside limits for mesh stiffness calculation.

**** Load Distribution Error Messages ****

5) Note: Mesh misalignment is approximated from gear quality.

**** Durability Factors Error Messages ****

3) Note: Pinion cycles above 1.E10, graph of flank (pitting) life factor extrapolated to 1.9705E11
4) Note: Gear cycles above 1.E10, graph of flank (pitting) life factor extrapolated to 3.7843E10

**** Strength Factors Error Messages ****

4) Note: Pinion cycles above 1.E10, graph of root (bending) life factor extrapolated to 1.9705E11
13) Note: Gear cycles above 1.E10, graph of root (bending life factor extrapolated to 3.7843E10

	**** Gear Geometry** (External Gears) **	Pinion		Gear (Wheel)	
	Gear Set Type		Single Helical		
z	Number of Teeth	29		151	
u	Gear Ratio (Hunting Tooth Set)		5.2069		
m_n	Normal Module		5.0000		mm
a	Center Distance		18.6283		inch
a_s	Standard Center Distance		18.6283		inch
b	Face Width	6.2500		6.2500	inch
b_{eff}	Effective Face Width		6.2500		inch
n	Speed	18,744.8		3,600.0	rpm
v_t	Pitch Line Velocity		29,456.3		ft/min
α_n	Normal Reference Pressure Angle		20.0000		degrees
α_{wt}	Transverse Operating Pressure Angle		20.9419		degrees
β	Helix Angle		18.0000		degrees
β_w	Operating Helix Angle		18.0000		degrees
h_t	Whole depth	0.4887		0.4887	inch
c	Tip to Root Clearance	0.0950		0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root			

	** Diameters **	Pinion	Gear (Wheel)	
d_a	Tip Diameter	6.3961	31.648	inch
h_a	Addendum	1.0000	1.0000	normalized
d	Reference Pitch Diameter	6.0024	31.254	inch
d_w	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
d_{SOI}	Start of Involute Diameter	5.6625	30.798	inch
d_b	Base Diameter	5.6059	29.1896	inch
d_f	Root Diameter	5.4188	30.670	inch

	** Ratios **	Pinion		Gear (Wheel)
ε_α	Transverse (Profile) Contact Ratio		1.6405	
ε_β	Axial (Face) Contact Ratio		3.1230	
ε_γ	Total Contact Ratio		4.7635	
b_{eff} $/d_w$	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
b_{eff} $/a$	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root   Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

	** Tool Data - Same for Pinion & Gear **	Hob or Rack Type Cutter	
h_{aP}	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
s_0	Measured Tool Tooth Thickness	1.5708	normalized
pr	Protuberance of Tool	0.0000	inch
q	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

	** Surface Finish **	Pinion	Gear (Wheel)	
R_a	Flank Roughness, Arithmetic Average	32.000	32.000	micro-inch
R_a	Root Roughness, Arithmetic Average	64.000	64.000	micro-inch

	**** Tooth Thickness ****	Pinion	Gear (Wheel)	
s_{an}	Normal Tip Tooth Thickness	0.1347	0.1499	inch
	Normal Tip Tooth Thickness	0.6843	0.7613	normalized
a	Center Distance for Calculation of Zero Backlash (Mean)	18.6283		inch
$\Delta x/2$	Thinning for Backlash (on ref. diameter)	0.0600	0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000	0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness		
j_t	Transverse Circular Backlash	0.0248		inch

	**** Configuration Data ****	Pinion	Gear (Wheel)	
	Gear Blank Construction	Solid	Solid	
l	Pinion Shaft Bearing Span	8.0000		inch
s	Pinion Offset	0.0000		inch
d_{sh}	Pinion Shaft External Diameter	3.0000		inch
d_{shi}	Pinion Shaft Internal Diameter	0.0000		inch
	Tooth Alignment Correction	None		
ρ_F	Set Arrangement	ISO 6336-1 figure 13 A		
	Contact Pattern	Favorable		
υ_{40}	Kinematic Viscosity of Lubricant at 40 C	32.000		cSt
C_a	Design Tip Modification	0.0000		0.0001 in

	**** ISO Materials ****	Pinion	Gear (Wheel)
	Material	NT: Gas Nitrided Steel	NT: Gas Nitrided Steel
	Material Sub-class		
	Material Quality	MQ	MQ

	**** Material Hardness ****	Pinion	Gear (Wheel)
	Surface Hardness	90 Rockwell 15N	90 Rockwell 15N
	Note: Hardness conversions are approximate		

	**** Application Data** (Wheel Driving) **	Pinion	Gear (Wheel)	
n	Speed	18,744.8	3,600.0	rpm
L	Design Life	20.0000		years
N_L	Design Life	1.9705E11	3.7843E10	cycles
	Contacts per Revolution	1	1	
	Idler?	No	No	

	**** Life Factor Data ****	Pinion	Gear (Wheel)
N_L	Number of Cycles	1.9705E11	3.7843E10
Z_N	Pitting Durability Stress Cycle Factor (input)	0.0000	0.0000
Y_N	Bending Strength Stress Cycle Factor (input)	0.0000	0.0000
Z_{N10}	Pitting Durability Cycle Factor at 10^10	0.8500	0.8500
Y_{N10}	Bending Strength Cycle Factor at 10^10	0.8500	0.8500

	**** Tolerances ****	Pinion	Gear (Wheel)
	ISO 1328-1 Accuracy Grade	6.0000	6.0000

** ISO 6336 2006 Rating Output **

Power Rating, Calculate from Safety Factor

**** ISO Factors ****

K_A	Application Factor	1.4000	
S_{Hmin}	Minimum Safety Factor, Durability	1.2000	
S_{Fmin}	Minimum Safety Factor, Strength	1.4000	
	Face Load Factor, Strength	Calculated	

**** Dynamic Factor ****

K_v	Dynamic Factor (Method B)	1.1665	
m_{red}	Reduced Mass of Pair	0.0741	lb/in
c'	Max.Single Pair Stiffness	12.6634	lb/(in μin)
$c_{\gamma\alpha}$	Mean Value Mesh Stiffness per Unit Face - for K_v	18.7465	lb/(in μin)
N_S	Resonance Ratio	3.5800	

**** Load Distribution Factor ****

	Tooth Alignment Correction	None	
	Set arrangement	ISO 6336-1 figure 13 A	
	Contact Pattern	Favorable	
$K_{H\beta}$	Face Load Factor, flank (Method B)	1.0962	
$K_{F\beta}$	Face Load Factor, root (Method B)	1.0884	
$K_{H\alpha}$	Trans.Load Factor, flank (Method B)	1.1332	
$K_{F\alpha}$	Trans Load Factor, root (Method B)	1.1332	
f_{sh0}	Unit Load Shaft Deflection	0.0249	0.0001 in
$F_{\beta x}$	Initial Equivalent Misalignment	7.2653	0.0001 in
$F_{\beta y}$	Effective Equiv Misalignment	6.1755	0.0001 in
$c_{\gamma\beta}$	Mesh stiffness per Unit Face - for $K_{H\beta}$	15.9345	lb/(in μin)

Type of Rating: **Power Rating, Calculate from Safety Factor**

	**** Surface Durability Rating Factors ****	Pinion		Gear (Wheel)	
Z_H	Zone Factor		2.3944		
Z_E	Elastic Factor		189.812		$(lb/in^2)^{1/2}$
Z_ε	Contact Ratio Factor		0.7808		
Z_β	Helix Angle Factor		1.0254		
Z_B , Z_D	Single Pair Tooth Contact Factor	1.0000		1.0000	
Z_{NT}	Life Factor, static	1.0000		1.0000	
	Life Factor, reference	0.8500		0.8500	
Z_L	Lubrication Factor, static		1.0000		
	Lubrication Factor, reference		0.9224		
Z_R	Roughness Factor, static		1.0000		
	Roughness Factor, reference		0.9833		
Z_V	Velocity Factor, static		1.0000		
	Velocity Factor, reference		1.0690		
Z_W	Work Hardening Factor, static	1.0000		1.0000	
	Work Hardening Factor, reference	1.0000		1.0000	
Z_X	Size Factor		1.0000		

	**** Bending Strength Rating Factors ****	Pinion		Gear (Wheel)
Y_F	Tooth Form Factor	1.5013		1.2643
Y_S	Stress Correction Factor	1.7976		2.1428
	Contact Ratio		0.6686	
Y_{DT}	Deep Tooth Factor		1.0000	
	Rim Thickness Factor	1.0000		1.0000
Y_β	Helix Angle Factor		0.8500	
Y_{NT}	Life Factor, static	1.0000		1.0000
	Life Factor, reference	0.8500		0.8500
$Y_{\delta relT}$	Relative Notch Sensitivity Factor, static	0.9595		1.0286
	Relative Notch Sensitivity Factor, reference	0.9616		0.9989
Y_{RrelT}	Relative Surface Factor, static	1.0000		1.0000
	Relative Surface Factor, reference	0.9948		0.9948
Y_X	Size Factor, static	1.0000		1.0000
	Size Factor, reference	1.0000		1.0000

** MAIN RATING VALUES **

	** Surface Durability Ratings **	Pinion	Gear (Wheel)	
σ_{Hlim}	Allowable Stress Number, contact	1,250.00	1,250.00	
σ_{HG}	Pitting Stress Limit, static	1,212.04	1,212.04	
	Pitting Stress Limit, reference	1,030.24	1,030.24	
σ_{HP}	Permissible Contact Stress, static	1,010.03	1,010.03	
	Permissible Contact Stress, reference	858.53	858.53	
σ_{HP}	Permissible contact Stress	811.06	837.00	
σ_{H0}	Nominal Contact Stress	569.43		
σ_{H}	Contact Stress	811.06	811.06	
S_H	Durability Safety Factor	1.2000	1.2384	
	** Bending Strength Ratings **	Pinion	Gear (Wheel)	
σ_{Flim}	Allowable Bending Stress	420.00	420.00	
σ_{FG}	Tooth Root Stress Limit, static	803.52	834.73	
	Tooth Root Stress Limit, reference	682.99	709.52	
σ_{FP}	Permissible Tooth Root Stress, static	573.94	596.23	
	Permissible Tooth Root Stress, reference	487.85	506.80	
σ_{FP}	Permissible Tooth Root Stress	459.57	493.46	
σ_{F0}	Nominal Tooth Root Stress	143.712	144.271	
σ_{F}	Tooth Root Stress	289.480	290.605	
S_F	Strength Safety Factor	2.2226	2.3773	
	** POWER SUMMARY **	Pinion	Gear (Wheel)	
F_t	Tangential Force	11,179.1		lbf
	Torque	33,551.	174,697.	in-lb
	Power at Specified Safety factor	9,978.7		hp

**** AGMA 6011 Error Messages ****

Note: All 6011 warnings also apply to API 613
7) Note, see AGMA 6011 I03 Table 2 for recommended accuracy grades as a function of pitch line velocity

**** API 613 Error Messages ****

5) Warning, standard violated: Pinion Tooth accuracy must be ISO 1328-1 grade 4 or better
6) Warning, standard violated: Gear Tooth accuracy must be ISO 1328-1 grade 4 or better

	**** Gear Geometry** (External Gears) **	Pinion		Gear (Wheel)	
	Gear Set Type		Single Helical		
N_P N_G	Number of Teeth	29		151	
m_G	Gear Ratio (Hunting Tooth Set)		5.2069		
m_n	Normal Module		5.0000		mm
C	Center Distance		18.6283		inch
	Standard Center Distance		18.6283		inch
F	Face Width	6.2500		6.2500	inch
F	Effective Face Width		6.2500		inch
n	Speed	18,744.8		3,600.0	rpm
ν_t	Pitch Line Velocity		29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle		20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle		20.9419		degrees
ψ_s	Helix Angle		18.0000		degrees
	Operating Helix Angle		18.0000		degrees
h_t	Whole depth	0.4887		0.4887	inch
c	Tip to Root Clearance	0.0950		0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root			

	**** Diameters ****	Pinion	Gear (Wheel)	
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

**** Tool Data - Same for Pinion & Gear **** Hob or Rack Type Cutter

h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

**** Tooth Thickness ****

		Pinion	Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347	0.1499	inch
	Normal Tip Tooth Thickness	0.6843	0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)	18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600	0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000	0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness		
B_t	Transverse Circular Backlash	0.0248		inch

**** API Materials ****

	Pinion	Gear (Wheel)
	Material is Steel	
Heat Treatment	Nitrided	Nitrided
Surface Hardness	90.0 Rockwell 15N	90.0 Rockwell 15N

Note: Hardness conversions are approximate

**** Application Data** (Wheel Driving) **

		Pinion	Gear (Wheel)	
n_p	Speed	18,744.8	3,600.0	rpm
q	Contacts per Revolution	1	1	
	Idler?	No	No	

	** API 613 Data **	Pinion	Gear (Wheel)	
I_m	Material Index Number (pitting allowable)	300.23	300.23	psi
S_a	Bending Stress Number (allowable)	27,557.2	27,557.2	psi
	Type of Rating:	**Power Rating, Calculate from Service Factor**		
SF	API 613 Service Factor (input)	1.4000		

	** AGMA 908 DATA (normalized) **	Pinion	Gear (Wheel)
K_f	Stress Correction Factor	1.4277	1.5500
I	I-Factor	0.2363	
J	J-Factor	0.5467	0.6264

** API 613 RATING OUTPUT **

**** PITTING ****

K_a	Tooth Pitting Index, allowable	214.449	psi
	Allowable Power at input Service Factor	6,024.2	hp

** BENDING **	Pinion	Gear (Wheel)	
Allowable Power at input Service Factor	6,903.3	7,909.8	hp

**** POWER SUMMARY ****

Allowable Power at Input Service Factor	6,024.2	hp

**** Strength and Stress Cycle Factor Error Messages ****

172) WARNING: Number of cycles exceeds the range defined in the standard, stress cycle factors extrapolated beyond 1E10 cycles

**** Effective Case Error Messages ****

213) WARNING: Contact stress is not known, case depth as a function of contact stresses is undefined
214) WARNING: Contact stress is not known, core hardness coefficent is undefined

**** AGMA 6011 Error Messages ****

7) Note, see AGMA 6011 I03 Table 2 for recommended accuracy grades as a function of pitch line velocity

	**** Gear Geometry** (External Gears) **	Pinion	Gear (Wheel)	
	Gear Set Type	Single Helical		
N_P N_G	Number of Teeth	29	151	
m_G	Gear Ratio (Hunting Tooth Set)	5.2069		
m_n	Normal Module	5.0000		mm
C	Center Distance	18.6283		inch
	Standard Center Distance	18.6283		inch
F	Face Width	6.2500	6.2500	inch
F	Effective Face Width	6.2500		inch
n	Speed	18,744.8	3,600.0	rpm
ν_t	Pitch Line Velocity	29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle	20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle	20.9419		degrees
ψ_s	Helix Angle	18.0000		degrees
	Operating Helix Angle	18.0000		degrees
h_t	Whole depth	0.4887	0.4887	inch
c	Tip to Root Clearance	0.0950	0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root		

	**** Diameters ****	Pinion	Gear (Wheel)	
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

	**** Tool Data - Same for Pinion & Gear ****	Hob or Rack Type Cutter	
h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

	**** Surface Finish ****	Pinion	Gear (Wheel)	
f_p	Flank Roughness, Arithmetic Average	32.000	32.000	micro-inch

	**** Tooth Thickness ****	Pinion		Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347		0.1499	inch
	Normal Tip Tooth Thickness	0.6843		0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)		18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600		0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000		0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness			
B_t	Transverse Circular Backlash		0.0248		inch

	**** Configuration Data ****	Pinion	Gear (Wheel)	
	Gear Blank Construction	Solid	Solid	
S	Pinion Shaft Bearing Span	8.0000		inch
S_1	Pinion Offset	Not used for 6011		

Symbol	**** AGMA Materials ****	Pinion	Gear (Wheel)	Unit
	Material	Steel	Steel	
	Material Sub Class	Nitralloy 135M	Nitralloy 135M	
	Heat Treatment	Nitrided	Nitrided	
	Material Grade	2	2	
μ_P μ_G	Poisson's Ratio	0.3000	0.3000	
E_P E_G	Modulus of Elasticity	29,500,000.	29,500,000.	psi

	**** Material Hardness ****	Pinion	Gear (Wheel)
	Surface Hardness	90 Rockwell 15N	90 Rockwell 15N
	Core Hardness	321 Brinell	321 Brinell
	Note: Hardness conversions are approximate		

Symbol	**** Application Data** (Wheel Driving) **	Pinion	Gear (Wheel)	Unit
n_p	Speed	18,744.8	3,600.0	rpm
L	Design Life	40,000.		hours
N	Design Life	4.4988E10	8.6400E09	cycles
q	Contacts per Revolution	1	1	
	Idler?	No	No	

	**** Tolerances ****	Pinion	Gear (Wheel)
	AGMA 2000 Quality Number	Q12	Q12

** AGMA 6011-I03 Rating Output **

Power Rating, Calculate from Service Factor

Symbol	**** Effective Case Data ****	Pinion	Gear (Wheel)	Unit
U_c	Core Hardness Coefficient	0.0000	0.0000	
	Total Case Depth	0.0000	0.0000	inch
	Figure 15 Heavy Minimum Total Case Depth	0.0237	0.0237	inch
	Figure 15 Normal Minimum Total Case Depth	0.0171	0.0171	inch

Symbol	**** Dynamic Factor ****	Value
K_v	Dynamic Factor (input)	1.1300
A_v	Required Transmission Accuracy	A 4

Symbol	**** Load Distribution Factor ****	Value
	Intended Service (per std)	Precision Enclosed Gearing
	Leads Properly Modified? (per std)	Yes
	Lapped or Adjusted at Assembly? (per std)	Yes
C_{mc}	Lead Correction Factor (per std)	0.8000
C_{pf}	Pinion Proportion Factor	0.1447
C_{pm}	Pinion Proportion Modifier (per std)	1.0000
C_{ma}	Mesh Alignment Factor	0.1439
C_e	Mesh Align Correction Factor (per std)	0.8000
K_m	Load Distribution Factor	1.2079

	**** AGMA 908 DATA (normalized) ****	Pinion		Gear (Wheel)	
	Minimum Contact Length		10.5500		inch
K_f	Stress Correction Factor	1.4277		1.5500	
I	I-Factor		0.2363		
J	J-Factor	0.5467		0.6264	

	**** Yield Strength Factors ****	Pinion		Gear (Wheel)	
	Application Requirements (for yield strength factor):		Industrial Practice		
K_y	Yield Strength Factor	0.7500		0.7500	
K_{my}	Load Distribution Factor - Overload		1.1600		
W_{max}	Maximum Tangential Load		11,739.8		lbf
s_{ay}	Allowable Yield Strength	121,922.		121,922.	psi
	Yield Strength Safety Factor	5.0849		6.3252	

	**** General Factors ****		
K_s	Size Factor	1.0000	
K_T	Temperature Factor	1.0000	
W_t	Tangential Load	11,739.8	lbf

	**** Pitting Durability Stress Factors Summary ****	Pinion		Gear (Wheel)	
C_f	Surface Condition Factor		1.0000		
C_G	Gear Ratio Factor		0.8389		
C_H	Hardness Ratio Factor		1.0000		
C_p	Elastic Coefficient		2,271.44		(lb/in^2)^.5
Z_N	Pitting Durability Stress Cycle Factor	0.6243		0.6848	

	**** Bending Strength Stress Factors Summary ****	Pinion		Gear (Wheel)
C_H	Hardness Ratio Factor		1.0000	
K_B	Rim Thickness Factor	1.0000		1.0000
Y_N	Bending Strength Stress Cycle Factor	0.7621		0.8038

** MAIN RATING VALUES **

	**** PITTING ****	Pinion	Gear (Wheel)	
K	Contact Load Factor	373.03		psi
s_{ac}	Allowable Contact Stress Number	183,000.	183,000.	psi
P_{acu}	Allowable Transmitted Power at Unity Service Factor	14,670.8	17,648.7	hp
C_{SF}	Service Factor (minimum, input)	1.4000		
P_{ac}	Allowable Power at Input Service Factor	10,479.1	12,606.2	hp

	**** BENDING ****	Pinion	Gear (Wheel)	
U_L	Unit Load	9,542.1		psi
P_{atu}	Allowable Transmitted Power at Unity Service Factor	18,743.6	22,652.5	hp
s_{at}	Allowable Bending Stress Number	53,180.	53,180.	psi
K_{SF}	Service Factor (minimum, input)	1.4000		
P_{at}	Allowable Power at Input Service Factor	13,388.3	16,180.4	hp

	**** POWER SUMMARY ****	Pinion	Gear (Wheel)	
W_t	Tangential Force	11,739.8		lbf
T_P T_G	Member Torque	35,234.	183,459.	in-lb
P_a	Allowable Power at Input Service Factor	10,479.1		hp

**** Strength and Stress Cycle Factor Error Messages ****

172) WARNING: Number of cycles exceeds the range defined in the standard, stress cycle factors extrapolated beyond 1E10 cycles

**** Effective Case Error Messages ****

213) WARNING: Contact stress is not known, case depth as a function of contact stresses is undefined
214) WARNING: Contact stress is not known, core hardness coefficent is undefined

**** AGMA 6011 Error Messages ****

7) Note, see AGMA 6011 I03 Table 2 for recommended accuracy grades as a function of pitch line velocity
18) Note: standard recommends rating at 40,000 hours

	**** Gear Geometry** (External Gears) **	Pinion	Gear (Wheel)	
	Gear Set Type	Single Helical		
N_P N_G	Number of Teeth	29	151	
m_G	Gear Ratio (Hunting Tooth Set)	5.2069		
m_n	Normal Module	5.0000		mm
C	Center Distance	18.6283		inch
	Standard Center Distance	18.6283		inch
F	Face Width	6.2500	6.2500	inch
F	Effective Face Width	6.2500		inch
n	Speed	18,744.8	3,600.0	rpm
ν_t	Pitch Line Velocity	29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle	20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle	20.9419		degrees
ψ_s	Helix Angle	18.0000		degrees
	Operating Helix Angle	18.0000		degrees
h_t	Whole depth	0.4887	0.4887	inch
c	Tip to Root Clearance	0.0950	0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root		

	**** Diameters ****	Pinion	Gear (Wheel)	
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

	**** Tool Data - Same for Pinion & Gear ****	Hob or Rack Type Cutter	
h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

	**** Surface Finish ****	Pinion	Gear (Wheel)	
f_p	Flank Roughness, Arithmetic Average	32.000	32.000	micro-inch

	**** Tooth Thickness ****	Pinion		Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347		0.1499	inch
	Normal Tip Tooth Thickness	0.6843		0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)		18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600		0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000		0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness			
B_t	Transverse Circular Backlash		0.0248		inch

	**** Configuration Data ****	Pinion	Gear (Wheel)	
	Gear Blank Construction	Solid	Solid	
S	Pinion Shaft Bearing Span	8.0000		inch
S_1	Pinion Offset	Not used for 6011		

	**** AGMA Materials ****	Pinion	Gear (Wheel)	
	Material	Steel	Steel	
	Material Sub Class	Nitralloy 135M	Nitralloy 135M	
	Heat Treatment	Nitrided	Nitrided	
	Material Grade	2	2	
μ_P μ_G	Poisson's Ratio	0.3000	0.3000	
E_P E_G	Modulus of Elasticity	29,500,000.	29,500,000.	psi

	**** Material Hardness ****	Pinion	Gear (Wheel)
	Surface Hardness	90 Rockwell 15N	90 Rockwell 15N
	Core Hardness	321 Brinell	321 Brinell

Note: Hardness conversions are approximate

	**** Application Data** (Wheel Driving) **	Pinion	Gear (Wheel)	
n_p	Speed	18,744.8	3,600.0	rpm
L	Design Life	175,200.		hours
N	Design Life	1.9705E11	3.7843E10	cycles
q	Contacts per Revolution	1	1	
	Idler?	No	No	

**** Tolerances ****	Pinion	Gear (Wheel)
AGMA 2000 Quality Number	Q12	Q12

** AGMA 6011-I03 Rating Output **

Power Rating, Calculate from Service Factor

	**** Effective Case Data ****	Pinion	Gear (Wheel)	
U_c	Core Hardness Coefficient	0.0000	0.0000	
	Total Case Depth	0.0000	0.0000	inch
	Figure 15 Heavy Minimum Total Case Depth	0.0237	0.0237	inch
	Figure 15 Normal Minimum Total Case Depth	0.0171	0.0171	inch

	**** Dynamic Factor ****	
K_v	Dynamic Factor (input)	1.1300
A_v	Required Transmission Accuracy	A 4

	**** Load Distribution Factor ****	
	Intended Service (per std)	Precision Enclosed Gearing
	Leads Properly Modified? (per std)	Yes
	Lapped or Adjusted at Assembly? (per std)	Yes
C_{mc}	Lead Correction Factor (per std)	0.8000
C_{pf}	Pinion Proportion Factor	0.1447
C_{pm}	Pinion Proportion Modifier (per std)	1.0000
C_{ma}	Mesh Alignment Factor	0.1439
C_e	Mesh Align Correction Factor (per std)	0.8000
K_m	Load Distribution Factor	1.2079

	**** AGMA 908 DATA (normalized) ****	Pinion		Gear (Wheel)	
	Minimum Contact Length		10.5500		inch
K_f	Stress Correction Factor	1.4277		1.5500	
I	I-Factor		0.2363		
J	J-Factor	0.5467		0.6264	

	**** Yield Strength Factors ****	Pinion		Gear (Wheel)	
	Application Requirements (for yield strength factor):		Industrial Practice		
K_y	Yield Strength Factor	0.7500		0.7500	
K_{my}	Load Distribution Factor - Overload		1.1600		
W_{max}	Maximum Tangential Load		9,949.8		lbf
s_{ay}	Allowable Yield Strength	121,922.		121,922.	psi
	Yield Strength Safety Factor	5.9998		7.4632	

	**** General Factors ****		
K_s	Size Factor	1.0000	
K_T	Temperature Factor	1.0000	
W_t	Tangential Load	9,949.8	lbf

	**** Pitting Durability Stress Factors Summary ****	Pinion		Gear (Wheel)	
C_f	Surface Condition Factor		1.0000		
C_G	Gear Ratio Factor		0.8389		
C_H	Hardness Ratio Factor		1.0000		
C_p	Elastic Coefficient		2,271.44		(lb/in^2)^.5
Z_N	Pitting Durability Stress Cycle Factor	0.5748		0.6304	

	**** Bending Strength Stress Factors Summary ****	Pinion		Gear (Wheel)
C_H	Hardness Ratio Factor		1.0000	
K_B	Rim Thickness Factor	1.0000		1.0000
Y_N	Bending Strength Stress Cycle Factor	0.7265		0.7663

** MAIN RATING VALUES **

	** PITTING **	Pinion	Gear (Wheel)	
K	Contact Load Factor	316.16		psi
s_{ac}	Allowable Contact Stress Number	183,000.	183,000.	psi
P_{acu}	Allowable Transmitted Power at Unity Service Factor	12,433.8	14,957.7	hp
C_{SF}	Service Factor (minimum, input)	1.4000		
P_{ac}	Allowable Power at Input Service Factor	8,881.3	10,684.1	hp

	** BENDING **	Pinion	Gear (Wheel)	
U_L	Unit Load	8,087.2		psi
P_{atu}	Allowable Transmitted Power at Unity Service Factor	17,870.0	21,596.9	hp
s_{at}	Allowable Bending Stress Number	53,180.	53,180.	psi
K_{SF}	Service Factor (minimum, input)	1.4000		
P_{at}	Allowable Power at Input Service Factor	12,764.3	15,426.3	hp

	** POWER SUMMARY **	Pinion	Gear (Wheel)	
W_t	Tangential Force	9,949.8		lbf
T_P T_G	Member Torque	29,861.5	155,486.	in-lb
P_a	Allowable Power at Input Service Factor	8,881.3		hp

AGMA 2001-D04 Rating, rating engine version 1.0031 Data Set: 1 Page 1
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151-29 5 mn a 20 18 helix American Gear Manufacturers Association
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**** Dynamic Factor Error Messages ****

58) Note: Dynamic Factor (1.1100) set per maximum (most consertive) value for 'very accurate gearing' in figure 1.

**** Strength and Stress Cycle Factor Error Messages ****

172) WARNING: Number of cycles exceeds the range defined in the standard, stress cycle factors extrapolated beyond 1E10 cycles

**** Effective Case Error Messages ****

213) WARNING: Contact stress is not known, case depth as a function of contact stresses is undefined
214) WARNING: Contact stress is not known, core hardness coefficent is undefined

	**** Gear Geometry** (External Gears) **	Pinion	Gear (Wheel)	
	Gear Set Type	Single Helical		
N_P N_G	Number of Teeth	29	151	
m_G	Gear Ratio (Hunting Tooth Set)	5.2069		
m_n	Normal Module	5.0000		mm
C	Center Distance	18.6283		inch
	Standard Center Distance	18.6283		inch
F	Face Width	6.2500	6.2500	inch
F	Effective Face Width	6.2500		inch
n	Speed	18,744.8	3,600.0	rpm
ν_t	Pitch Line Velocity	29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle	20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle	20.9419		degrees
ψ_s	Helix Angle	18.0000		degrees
	Operating Helix Angle	18.0000		degrees
h_t	Whole depth	0.4887	0.4887	inch
c	Tip to Root Clearance	0.0950	0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root		

	**** Diameters ****	Pinion	Gear (Wheel)	
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root   Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

	**** Tool Data - Same for Pinion & Gear ****	Hob or Rack Type Cutter	
h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

	**** Surface Finish ****	Pinion	Gear (Wheel)	
f_p	Flank Roughness, Arithmetic Average	32.000	32.000	micro-inch

	**** Tooth Thickness ****	Pinion		Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347		0.1499	inch
	Normal Tip Tooth Thickness	0.6843		0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)		18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600		0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000		0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness			
B_t	Transverse Circular Backlash		0.0248		inch

	**** Configuration Data ****	Pinion	Gear (Wheel)	
	Gear Blank Construction	Solid	Solid	
S	Pinion Shaft Bearing Span	8.0000		inch
S_1	Pinion Offset	0.0000		inch

	**** AGMA Materials ****	Pinion	Gear (Wheel)	
	Material	Steel	Steel	
	Material Sub Class	Nitralloy 135M	Nitralloy 135M	
	Heat Treatment	Nitrided	Nitrided	
	Material Grade	2	2	
μ_P μ_G	Poisson's Ratio	0.3000	0.3000	
E_P E_G	Modulus of Elasticity	29,500,000.	29,500,000.	psi

	**** Material Hardness ****	Pinion	Gear (Wheel)
	Surface Hardness	90 Rockwell 15N	90 Rockwell 15N
	Core Hardness	321 Brinell	321 Brinell
	Note: Hardness conversions are approximate		

	**** Application Data** (Wheel Driving) **	Pinion		Gear (Wheel)	
n_p	Speed	18,744.8		3,600.0	rpm
L	Design Life		20.0000		years
N	Design Life	1.9705E11		3.7843E10	cycles
q	Contacts per Revolution	1		1	
	Idler?	No		No	

	**** Life Factor Data ****	Pinion	Gear (Wheel)
	Number of Cycles	1.9705E11	3.7843E10
Z_N	Pitting Durability Stress Cycle Factor (input)	0.0000	0.0000
Y_N	Bending Strength Stress Cycle Factor (input)	0.0000	0.0000
	Pitting Durability Cycle Factor at 10^10	0.6792	0.6792
	Bending Strength Cycle Factor at 10^10	0.8000	0.8000

	**** Tolerances ****	Pinion	Gear (Wheel)
	AGMA 2000 Quality Number	Q12	Q12

** AGMA 2001-D04 Rating Output **

Power Rating, Calculate from Service Factor

	**** Effective Case Data ****	Pinion	Gear (Wheel)	
U_c	Core Hardness Coefficient	0.0000	0.0000	
	Total Case Depth	0.0000	0.0000	inch
	Figure 15 Heavy Minimum Total Case Depth	0.0237	0.0237	inch
	Figure 15 Normal Minimum Total Case Depth	0.0171	0.0171	inch

	**** Dynamic Factor ****				
V_p	Pitch Variation (input)	2.0866		2.7559	0.0001 in
A_v	Transmission Accuracy Number		5.0000		
K_v	Dynamic Factor		1.1100		

	**** Load Distribution Factor ****			
	Intended Service (input)	Precision Enclosed Gearing		
	Leads Properly Modified? (input)	No		
	Lapped or Adjusted at Assembly? (input)	No		
C_{mc}	Lead Correction Factor (input)	1.0000		
C_{pf}	Pinion Proportion Factor	0.1447		
C_{pm}	Pinion Proportion Modifier (input)	1.0000		
C_{ma}	Mesh Alignment Factor	0.1439		
C_e	Mesh Align Correction Factor (input)	1.0000		
K_m	Load Distribution Factor	1.2886		

	**** AGMA 908 DATA (normalized) ****	Pinion	Gear (Wheel)	
	Minimum Contact Length	10.5500		inch
K_f	Stress Correction Factor	1.4277	1.5500	
I	I-Factor	0.2363		
J	J-Factor	0.5467	0.6264	

	**** Yield Strength Factors ****	Pinion	Gear (Wheel)	
	Application Requirements (for yield strength factor):	Industrial Practice		
K_y	Yield Strength Factor	0.7500	0.7500	
K_{my}	Load Distribution Factor - Overload	1.1600		
W_{max}	Maximum Tangential Load	9,494.6		lbf
	Stress due to Wmax	14,543.7	11,691.8	psi
s_{ay}	Allowable Yield Strength	121,922.	121,922.	psi
	Yield Strength Safety Factor	6.2874	7.8210	

	**** General Factors ****		
K_s	Size Factor	1.0000	
K_T	Temperature Factor	1.0000	
W_t	Tangential Load	9,494.6	lbf

	**** Pitting Durability Stress Factors Summary ****	Pinion	Gear (Wheel)	
C_f	Surface Condition Factor	1.0000		
C_G	Gear Ratio Factor	0.8389		
C_H	Hardness Ratio Factor	1.0000		
C_p	Elastic Coefficient	2,271.44		(lb/in^2)^.5
Z_N	Pitting Durability Stress Cycle Factor	0.5748	0.6304	

	**** Bending Strength Stress Factors Summary ****	Pinion	Gear (Wheel)
C_H	Hardness Ratio Factor	1.0000	
K_B	Rim Thickness Factor	1.0000	1.0000
Y_N	Bending Strength Stress Cycle Factor	0.7265	0.7663

** MAIN RATING VALUES **

	**** PITTING ****	Pinion	Gear (Wheel)	
K	Contact Load Factor	301.69		psi
s_{ac}	Allowable Contact Stress Number	183,000.	183,000.	psi
P_{acu}	Allowable Transmitted Power at Unity Service Factor	11,865.1	14,273.4	hp
C_{SF}	Service Factor (minimum, input)	1.4000		
P_{ac}	Allowable Power at Input Service Factor	8,475.1	10,195.3	hp

	**** BENDING ****	Pinion	Gear (Wheel)	
U_L	Unit Load	7,717.2		psi
P_{atu}	Allowable Transmitted Power at Unity Service Factor	17,052.1	20,608.3	hp
s_{at}	Allowable Bending Stress Number	53,180.	53,180.	psi
K_{SF}	Service Factor (minimum, input)	1.4000		
P_{at}	Allowable Power at Input Service Factor	12,180.1	14,720.2	hp

	**** POWER SUMMARY ****	Pinion	Gear (Wheel)	
W_t	Tangential Force	9,494.6		lbf
T_P T_G	Member Torque	28,495.5	148,373.	in-lb
P_a	Allowable Power at Input Service Factor	8,475.1		hp

**** Effective Case Error Messages ****

213) WARNING: Contact stress is not known, case depth as a function of contact stresses is undefined
214) WARNING: Contact stress is not known, core hardness coefficent is undefined

Symbol	**** Gear Geometry** (External Gears) **	Pinion	Gear (Wheel)	Units
	Gear Set Type	Single Helical		
N_P N_G	Number of Teeth	29	151	
m_G	Gear Ratio (Hunting Tooth Set)	5.2069		
m_n	Normal Module	5.0000		mm
C	Center Distance	18.6283		inch
	Standard Center Distance	18.6283		inch
F	Face Width	6.2500	6.2500	inch
F	Effective Face Width	6.2500		inch
n	Speed	18,744.8	3,600.0	rpm
ν_t	Pitch Line Velocity	29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle	20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle	20.9419		degrees
ψ_s	Helix Angle	18.0000		degrees
	Operating Helix Angle	18.0000		degrees
h_t	Whole depth	0.4887	0.4887	inch
c	Tip to Root Clearance	0.0950	0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root		

Symbol	**** Diameters ****	Pinion	Gear (Wheel)	Units
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

	**** Tool Data - Same for Pinion & Gear ****	Hob or Rack Type Cutter	
h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

	**** Surface Finish ****	Pinion	Gear (Wheel)	
f_p	Flank Roughness, Arithmetic Average	32.000	32.000	micro-inch

	**** Tooth Thickness ****	Pinion		Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347		0.1499	inch
	Normal Tip Tooth Thickness	0.6843		0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)		18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600		0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000		0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness			
B_t	Transverse Circular Backlash		0.0248		inch

	**** Configuration Data ****	Pinion	Gear (Wheel)	
	Gear Blank Construction	Solid	Solid	
S	Pinion Shaft Bearing Span	8.0000		inch
S_1	Pinion Offset	0.0000		inch

	**** AGMA Materials ****	Pinion	Gear (Wheel)	
	Material	Steel	Steel	
	Material Sub Class	Nitralloy 135M	Nitralloy 135M	
	Heat Treatment	Nitrided	Nitrided	
	Material Grade	2	2	
μ_P μ_G	Poisson's Ratio	0.3000	0.3000	
E_P E_G	Modulus of Elasticity	29,500,000.	29,500,000.	psi

	**** Material Hardness ****	Pinion	Gear (Wheel)
	Surface Hardness	90 Rockwell 15N	90 Rockwell 15N
	Core Hardness	321 Brinell	321 Brinell

Note: Hardness conversions are approximate

	**** Application Data** (Wheel Driving) **	Pinion		Gear (Wheel)	
n_p	Speed	18,744.8		3,600.0	rpm
L	Design Life		175,200.		hours
N	Design Life	1.9705E11		3.7843E10	cycles
q	Contacts per Revolution	1		1	
	Idler?	No		No	

	**** Tolerances ****	Pinion	Gear (Wheel)
	AGMA 2000 Quality Number	Q12	Q12

** API 617 Seventh edit Output **

Power Rating, Calculate from Service Factor

	**** Effective Case Data ****	Pinion	Gear (Wheel)	
U_c	Core Hardness Coefficient	0.0000	0.0000	
	Total Case Depth	0.0000	0.0000	inch
	Figure 15 Heavy Minimum Total Case Depth	0.0237	0.0237	inch
	Figure 15 Normal Minimum Total Case Depth	0.0171	0.0171	inch

	**** Dynamic Factor ****				
V_p	Pitch Variation (input)	2.0866		2.7559	0.0001 in
A_v	Transmission Accuracy Number		0.0000		
K_v	Dynamic Factor		1.1200		

	**** Load Distribution Factor ****	
K_m	Load Distribution Factor (input)	1.2162

	**** AGMA 908 DATA (normalized) ****	Pinion		Gear (Wheel)	
	Minimum Contact Length		10.5500		inch
K_f	Stress Correction Factor	1.4277		1.5500	
I	I-Factor		0.2363		
J	J-Factor	0.5467		0.6264	

Symbol	** Yield Strength Factors **	Pinion		Gear (Wheel)	Units
	Application Requirements (for yield strength factor):		Industrial Practice		
K_y	Yield Strength Factor	0.7500		0.7500	
K_{my}	Load Distribution Factor - Overload		1.1600		
W_{max}	Maximum Tangential Load		7,975.6		lbf
s_{ay}	Allowable Yield Strength	121,922.		121,922.	psi
	Yield Strength Safety Factor	7.4849		9.3105	

Symbol	** General Factors **		
K_s	Size Factor	1.0000	
K_T	Temperature Factor	1.2500	
W_t	Tangential Load	7,975.6	lbf

Symbol	** Pitting Durability Stress Factors Summary **	Pinion		Gear (Wheel)	Units
C_f	Surface Condition Factor		1.0000		
C_G	Gear Ratio Factor		0.8389		
C_H	Hardness Ratio Factor		1.0000		
C_p	Elastic Coefficient		2,271.44		(lb/in^2)^.5
Z_N	Pitting Durability Stress Cycle Factor	0.5747		0.6304	

Symbol	** Bending Strength Stress Factors Summary **	Pinion		Gear (Wheel)
C_H	Hardness Ratio Factor		1.0000	
K_B	Rim Thickness Factor	1.0000		1.0000
Y_N	Bending Strength Stress Cycle Factor	0.7266		0.7664

** MAIN RATING VALUES **

	** PITTING **	Pinion	Gear (Wheel)	
K	Contact Load Factor	253.426		psi
s_{ac}	Allowable Contact Stress Number	183,000.	183,000.	psi
P_{acu}	Allowable Transmitted Power at Unity Service Factor	9,966.8	11,989.8	hp
C_{SF}	Service Factor (minimum, input)	1.4000		
P_{ac}	Allowable Power at Input Service Factor	7,119.1	8,564.2	hp

	** BENDING **	Pinion	Gear (Wheel)	
U_L	Unit Load	6,482.6		psi
P_{atu}	Allowable Transmitted Power at Unity Service Factor	14,326.2	17,313.6	hp
s_{at}	Allowable Bending Stress Number	53,180.	53,180.	psi
K_{SF}	Service Factor (minimum, input)	1.4000		
P_{at}	Allowable Power at Input Service Factor	10,233.0	12,366.8	hp

	** POWER SUMMARY **	Pinion	Gear (Wheel)	
W_t	Tangential Force	7,975.6		lbf
T_P T_G	Member Torque	23,936.6	124,635.	in-lb
P_a	Allowable Power at Input Service Factor	7,119.1		hp

**** AGMA 6011 Error Messages ****

Note: All 6011 warnings also apply to API 613

7) Note, see AGMA 6011 I03 Table 2 for recommended accuracy grades as a function of pitch line velocity

**** API 613 Error Messages ****

5) Warning, standard violated: Pinion Tooth accuracy must be ISO 1328-1 grade 4 or better

6) Warning, standard violated: Gear Tooth accuracy must be ISO 1328-1 grade 4 or better

	**** Gear Geometry** (External Gears) **	Pinion		Gear (Wheel)	
	Gear Set Type		Single Helical		
N_P N_G	Number of Teeth	29		151	
m_G	Gear Ratio (Hunting Tooth Set)		5.2069		
m_n	Normal Module		5.0000		mm
C	Center Distance		18.6283		inch
	Standard Center Distance		18.6283		inch
F	Face Width	6.2500		6.2500	inch
F	Effective Face Width		6.2500		inch
n	Speed	18,744.8		3,600.0	rpm
ν_t	Pitch Line Velocity		29,456.3		ft/min
ϕ_n	Normal Reference Pressure Angle		20.0000		degrees
ϕ_t	Transverse Operating Pressure Angle		20.9419		degrees
ψ_s	Helix Angle		18.0000		degrees
	Operating Helix Angle		18.0000		degrees
h_t	Whole depth	0.4887		0.4887	inch
c	Tip to Root Clearance	0.0950		0.0950	inch
		Pinion Tip to Gear Root / Gear Tip to Pinion Root			

	**** Diameters ****	Pinion	Gear (Wheel)	
d_o D_o	Tip Diameter	6.3961	31.648	inch
a_{oP} a_{oG}	Addendum	1.0000	1.0000	normalized
D	Reference Pitch Diameter	6.0024	31.254	inch
d	Operating (working) Pitch Diameter	6.0024	31.254	inch
d_{SAP}	Start of Active Profile (Minimum)	5.7104	30.932	inch
	Start of Involute Diameter	5.6625	30.798	inch
D_b	Base Diameter	5.6059	29.1896	inch
D_R	Root Diameter	5.4188	30.670	inch

	**** Ratios ****	Pinion		Gear (Wheel)
m_p	Transverse (Profile) Contact Ratio		1.6405	
m_F	Axial (Face) Contact Ratio		3.1230	
m_t	Total Contact Ratio		4.7635	
	Facewidth to Operating Pitch Diameter Ratio	1.0412		0.2000
	Facewidth to Center Distance Ratio	0.3355		0.3355

**** Line of Action Data ****

Gear Driving, First Contact Near Gear Root Sliding velocity is for pinion, change sign for gear sliding velocity
Point C1 determined by gear tip diameter

Points on line of action	Distance on line of action	Pinion Roll Angle	Pinion Diameter inch	Gear Roll Angle	Gear Diameter inch	Sliding Velocity in/sec	Specific Sliding Pinion	Specific Sliding Gear
C1 Gear End of Active Profile	0.5435	11.1106	5.7104	24.0045	31.648	-1,238.19	-1.1605	0.5371
C2 Gear Highest Point STC	0.9325	19.0616	5.9080	22.4775	31.355	-328.02	-0.1792	0.1520
C3 Working Pitch Point	1.0727	21.9271	6.0024	21.9271	31.254	0.0000	0.0000	0.0000
C4 Gear Lowest Point STC	1.1508	23.5244	6.0601	21.6204	31.199	182.845	0.0809	-0.0881
C5 Gear Start of Active Profile	1.5398	31.4754	6.3961	20.0934	30.932	1,093.01	0.3616	-0.5665
C6 Total Line of Action Length	6.6581 inch							

Point C5 determined by Pinion Tip diameter
Percent Approach Action: 46.89%
Percent Recess Action: 53.11%

**** Tool Data - Same for Pinion & Gear **** Hob or Rack Type Cutter

h_a	ISO (1/2 pitch) Tool Addendum (from ref. line)	1.4000	normalized
t_m	Measured Tool Tooth Thickness	1.5708	normalized
δ_{a0}	Protuberance of Tool	0.0000	inch
	Finishing Stock Allowance - Normal	0.0000	inch
r_T	Tool Tip Radius	0.3936	normalized
h_{a0}	Hypothetical Tool Addendum	1.4000	normalized

**** Tooth Thickness ****

		Pinion	Gear (Wheel)	
t_o	Normal Tip Tooth Thickness	0.1347	0.1499	inch
	Normal Tip Tooth Thickness	0.6843	0.7613	normalized
C	Center Distance for Calculation of Zero Backlash (Mean)	18.6283		inch
Δ_n	Thinning for Backlash (on ref. diameter)	0.0600	0.0600	normalized
x	Profile Shift Coefficient (Zero Backlash x Factor)	0.0000	0.0000	normalized
		Rating Based on Nominal (with thinning) Thickness		
B_t	Transverse Circular Backlash	0.0248		inch

**** API Materials ****

	Pinion	Gear (Wheel)
	Material is Steel	
Heat Treatment	Nitrided	Nitrided
Surface Hardness	90.0 Rockwell 15N	90.0 Rockwell 15N

Note: Hardness conversions are approximate

**** Application Data** (Wheel Driving) **

		Pinion	Gear (Wheel)	
n_p	Speed	18,744.8	3,600.0	rpm
q	Contacts per Revolution	1	1	
	Idler?	No	No	

	** API 613 Data **	Pinion		Gear (Wheel)	
I_m	Material Index Number (pitting allowable)	300.23		300.23	psi
S_a	Bending Stress Number (allowable)	27,557.2		27,557.2	psi
	Type of Rating:	**Power Rating, Calculate from Service Factor**			
SF	API 613 Service Factor (input)		1.4000		

	** AGMA 908 DATA (normalized) **	Pinion		Gear (Wheel)	
K_f	Stress Correction Factor	1.4277		1.5500	
I	I-Factor		0.2363		
J	J-Factor	0.5467		0.6264	

** API 613 RATING OUTPUT **

	** PITTING **				
K_a	Tooth Pitting Index, allowable		214.449		psi
	Allowable Power at input Service Factor		6,024.2		hp

	** BENDING **	Pinion		Gear (Wheel)	
	Allowable Power at input Service Factor	6,903.3		7,909.8	hp

	** POWER SUMMARY **				
	Allowable Power at Input Service Factor		6,024.2		hp