How to Spec a Mill Gear

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This paper outlines the design considerations that go into construction of a drive system in order to explain the importance of specific data, why it is required, and where design freedom is necessary. Apart from loads and speeds, interface dimensions and site specific conditions are also needed. Deciding up front which gear rating practice to select can affect the torque capacity of the drive train by ~15%.

Introduction

An expert in whole-brain learning, Steven Snyder, once said, "There are only two problems in life: (1) You know what you want, and you don't know how to get it; and/or (2) You don't know what you want."

To solve these two problems, a clear understanding and good communication skills are necessary. In terms of getting a great gear set, it requires a coordinated effort between the end user, the gear manufacturer, the gear designer, the consultant, and the original equipment manufacturer. Each of these groups has a key piece of the puzzle necessary for the gear to fulfill its useful operational life. This paper will outline what information needs to be collected and passed on to the gear designer to develop a successful drive train for a specific area of use: gearing for cylindrical grinding mills. It will act as a checklist for information required, outline the impact of certain selections, and resolve ambiguities to address the two problems outlined above.

Background

A grinding mill circuit is an unusual installation for gearing when compared to traditional enclosed gear drive installations, but these applications have been utilized for over eighty six years. The grinding process, more accurately termed a tumbling process, uses horizontal rotating cylinders that contain the material to be broken, potentially augmented by grinding media (Fig. 1). The material moves up the wall of the drum until gravity overcomes centrifugal forces, and it drops to the bottom of the drum to collide with the remaining material. This breaks up the particles and reduces their size. Power required for this process ranges from 75 to 18,000 kW (100 to 24,000 HP), in either single- or dual-motor configurations.

In this type of application, the pinion is mounted on pillow blocks driven by a low-speed motor or a motor and enclosed gear drive. The gear is mounted on the mill using a flange bolted connection (see Figure 2 for one type of flange installation). Both the center distance and alignment are adjustable either by shimming the pillow blocks or moving the mill. Lubricant is typically either high-viscosity oil (1,260 cSt @ 100°C) sprayed on the gear in 15 minute intervals or a lower viscosity oil or grease product sprayed on the pinion every few minutes. Alternately, lubrication can be applied by continuous spray or dip immersion methods.)

Gear sizes can range up to 14 meters (46 feet) in diameter with face widths approaching 1.2 meters (50 inches). Typical tooth sizes range from 20 to 40 module (1.25 DP to 0.64 DP). Single-stage reduction gears range from 8:1 to as much as 20:1. Gear materials are typically through hardened cast steel, fabricated forged and rolled steel, or spheroidal graphitic iron. Pinions are carburized, induction hardened, or through hardened heat treated steels.

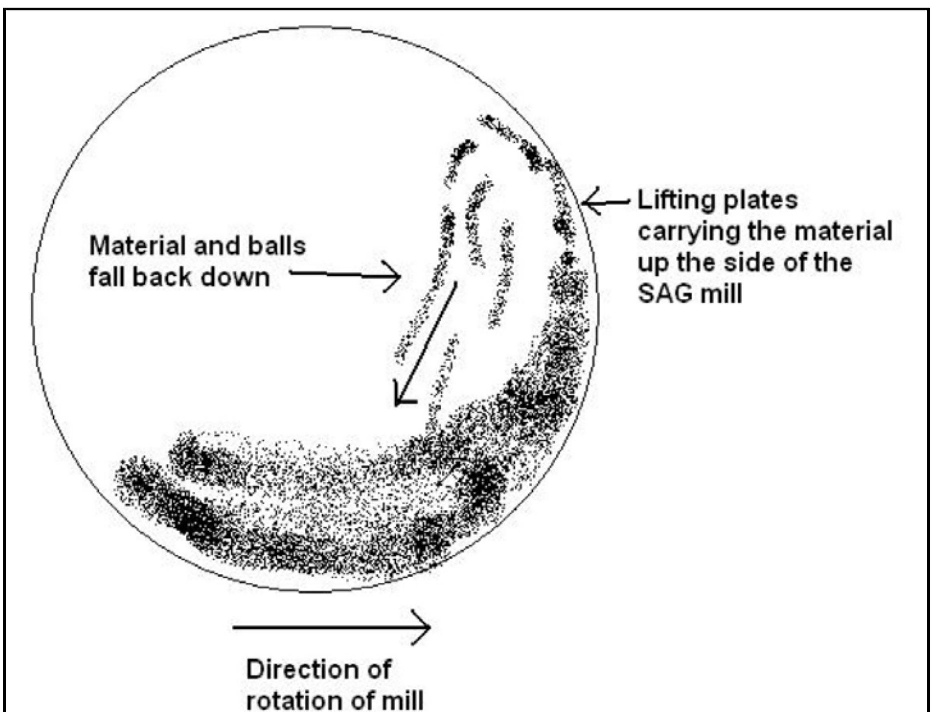

Figure 1 Grinding mill process.

Figure 2 Grinding mill installation.

For small installations, either a one- or two-piece design is used with the split joints located in the root of a tooth. Four- and six-piece designs are also utilized when the weight of the segments exceeds the crane capacity of the facility or pouring capacity for cast segments becomes an issue.

Initial Data

The purpose of a grinding mill is to make large rocks into small rocks. To accomplish this task involves significant calculations on the part of the mill builder. These include reviewing the size of the incoming and outgoing product, the rate of production, the size of the mill in diameter and length, the grinding media, the theoretical critical speed of rotation, and the interior configuration of the mill. Unfortunately, to get what is required, this information needs translation into something that the gear designer can input into the rating calculations. The calculation of actual contact stress s_c does not have an input for tons/hour of mineral produced.

A theoretical relation of mill diameter to power is ~ mill diameter$^{2.5}$. To get torque, we also need the speed of the drum. This is based on the concept of a "theoretical critical speed of rotation (CS)." The critical speed of rotation is the speed (in rpm) at which an infinitely small particle will cling to the inside of the liners of the mill for a complete revolution.

$$CS = \frac{43.305}{\sqrt{\text{Mill Diameter}}} \tag{1}$$

where

CS is the theoretical critical speed of rotation, and is the mill speed, rpm;

Mill diameter is the nominal inside diameter of the mill, m.

Since we actually need the particles to come off the inside diameter of the mill to be processed, the typical mill speed is ~75% of the theoretical critical speed of that mill. Using the above formulas, significant experience of how the grinding process works, and material properties of the ore being ground, the mill builder can provide the gear designer with input power and output speed.

The next step is the interface dimensions. Since the gear needs to turn the mill, it needs to have a bore larger than the mill outside diameter. The mill outside diameter is a function of the grinding process selected. Autogenous mills are the largest in diameter since the feed grinds itself. A semi-autogenous mill uses some metallic or ceramic balls to assist the grinding process and can be slightly smaller. Ball mills are smaller still and use a larger percentage of balls to perform most of the work. Large-diameter mills allow for use of gear ratios not normally thought possible in single-reduction applications — namely, 8:1 to 20:1.

If the gear is to replace an existing gear, then manufacturing drawings or installation drawings complete with gear attributes, center distance and dimensions are required. Although budgetary pricing can be made without dimensional data, once an order is present, full data is required. These gears are made custom for each installation so there are no catalogs available to provide this information.

Site-specific considerations also need to be disclosed. If the gear is expected to operate outdoors or in unheated structures, a minimum and maximum temperature range should be given to assist in lubricant selection and method of application. Transportation limitations can also affect the design. If crane capacity or size limitations exist, the gear designer can increase the number of segments of the gear to allow for reduced handling weights.

Rating Standards

Once mill diameter has been established, the largest cost driver is the actual size of the gear. Gear power capacity is a function of how the ratings are calculated. There are two basic rating practices in use in gearing: ISO 6336 (Ref. 8) and ANSI/AGMA 2001-D04 (Ref. 5). Both exist to provide a common basis for comparing the power capacities of various designs. By their nature, these are general standards in that they apply for fine pitch gears of 4 mm in diameter as well as 13,000 mm gears, made from various materials and accuracy grades. Given that range, we run the risk of missing significant size effects — either large or small — or client expectations that will affect the performance of a gear set. This is why the general standards suggest use of an application-based standard when designing gears for a specific purpose.

The rating committee uses the fundamental standard as a criteria and method source for rating gears and adjusts the component factors to match experience and field performance for existing designs. AGMA and, to a lesser extent, ISO, have developed application standards for a variety of applications such as enclosed drives, high-speed units, drives for wind turbines, marine, automotive, and steel mill rolling applications to narrow the scope of the general rating practice and fine tune it for the nature of service.

For grinding mill service, an early application standard was AGMA 321.05 (Ref. 3); it was first approved for use in October 1943. Various iterations occurred with the last major re-write in 1968, when the standard was updated to use the formulations of AGMA 211.02 (Ref. 9) and AGMA 221.02 (Ref. 10). The last editorial corrections were issued in March 1970.

This rating practice uses concepts that predate our current AGMA 2001 thinking. The rating formulas for gearing in this standard are:

$$P_{ac} = \frac{n_p d^2 C_v}{12600} \frac{F}{C_m} I \left(\frac{s_{ac}}{C_p}\right)^2 C_H^2 \qquad (2)$$

$$P_{at} = \frac{n_p d K_v}{12600} \frac{F}{C_m} s_{at} \frac{J}{P_d} \qquad (3)$$

where

P_{ac} is allowable transmitted power for pitting, HP;
n_p is pinion speed, rpm;
d is operating pitch diameter, in;
C_v is dynamic factor pitting;
F is face width, in;
C_m is load distribution factor;
I is I factor;
s_{ac} is allowable contact stress number, lbs/in^2;
C_p is elastic coefficient, (lbs/in^2)$^{0.5}$;
C_H is hardness ratio factor;
P_{at} is allowable transmitted power for bending, HP;
K_v is dynamic factor bending;
s_{at} is allowable bending stress number, lbs/in^2;
J is J factor;
P_d is transverse diametral pitch, in^{-1}.

The major influence factors were assigned specific values based on the size and experience of the industry with this type of gearing. Two dynamic factors were used, but both were a function of the pitch line velocity of the set. Load distribution factor was a function of face width only, covering the range of 50 to 1,016 mm (2 inches to 40 inches) with modification factors to adjust its value when teeth were hardened after completion. The allowable contact stress was reduced by the standard; however no metallurgical properties other than hardness were discussed. The hardness ratio factor was expanded to cover a ratio range of 1:1 to 20:1. The allowable bending stress was also reduced by the standard but it also remained only a function of hardness. Service factors ranged from 1.5 to 1.65 for grinding mill service.

ANSI/AGMA 6004-F88 (Ref. 11) was the first attempt to reflect ratings based on tooth attribute quality for mill and kiln gearing; it was released in 1988. After its limited acceptance in the industry, the AGMA Mill Gearing Committee developed the current standard ANSI/AGMA 6014-A06 (Ref. 4), released in 2006. It is currently in its five-year review cycle with the committee.

The rating formulas used in AGMA 6014 are as follows:

$$P_{acm} = \frac{\pi n_p F}{396000} \frac{I}{K_{vm} K_m} \left(\frac{d\, s_{ac} Z_n C_h}{C_p}\right)^2 \qquad (4)$$

$$P_{acm} = \frac{\pi n_p d}{396000} \frac{F}{P_d} \frac{J s_{at} Y_n}{K_m K_{Bm}} \qquad (5)$$

where

P_{acm} is allowable transmitted power for pitting, HP;
K_{vm} is dynamic factor bending;
Z_n is stress cycle factor for pitting;
P_{atm} is allowable transmitted power for bending, HP;
Y_n is stress cycle factor for bending;
K_{Bm} is rim thickness factor.

This formula now includes the effect of stress cycle factors as well as making adjustments to the base evaluations of dynamic and rim thickness factor.

The critical changes that the committee made to the standard addressed the fact that these gears mesh through the use of independent bearing support. The gearing is not mounted in a

Table 1 Gear geometry and application data

Attribute	Pinion	Set	Gear
Number of teeth	21		314
Ratio		14.95:1	
Normal diametral pitch, in^{-1}		1.0154	
Normal module, mm		25.01	
Face width, in		27.25	
Face width, mm		692	
Axial overlap		1.100	
Outside diameter, in	23.043		313.670
Outside diameter, mm	585.3		8043.4
Tooth accuracy	Q12/A5		Q10/A7
Bore diameter, in			247.000
Bore diameter, mm			6273.8
Material hardness	55 HRC		335 HBW
Application			
Power, HP		5000	
Power, kW		3729	
Speed, rpm	202.42		13.538
Durability service factor		1.75	
Strength service factor		2.50	

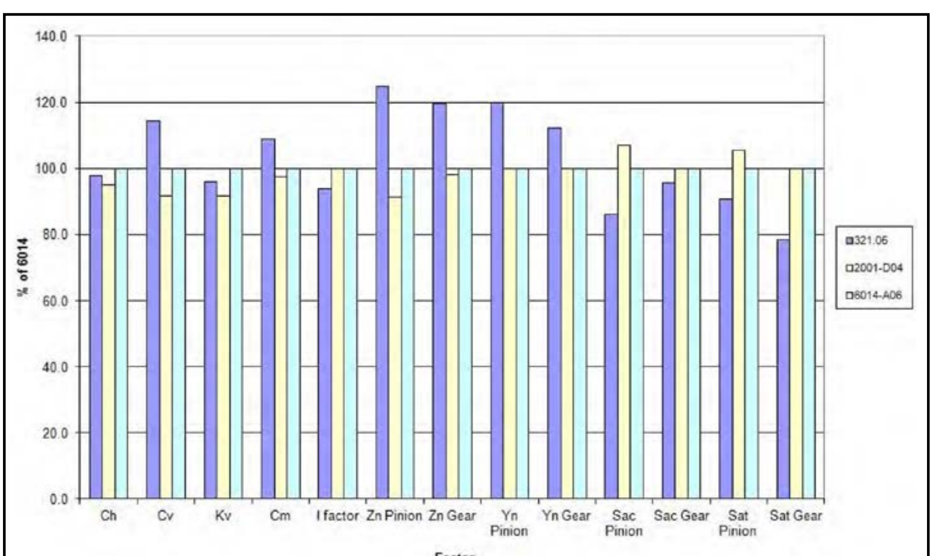

Figure 3 Normalized rating factors for base set.

housing where all bearing supports are aligned by machining. Based on the ratios, modules (pitches), and face widths (approaching 1.2 meters, 50 inches), the effect of size and material usage, cast or forged steel or ductile iron needed to be included in the standard. Achievable and measurable accuracy grades limit the values of the dynamic factor. Client expectations of long life indicate values of the stress cycle factor Z_N and Y_N be based on 25 years. Durability service factors were also increased from AGMA 321.05 to $C_{SF}=1.75$ for high-power mills over 3,350 kW (4,500 HP) in size. Strength service factors K_{SF} were also specified.

Given two standards designed to rate gears for this service, others occasionally use general standards or their own in-house-developed calculations. When this path is chosen, there can be significant risk that may not be realized by the user of the rating practice. As noted above, an application standard takes into account the narrower range of gear size, operating experience, typical materials, and mounting conditions of the process. A general standard, needing to be "all things to all people" can set requirements or allow mounting practices that are easily achievable when working with 100 kg (220 lb) size gear sets, but are problematic with 118,000 kg (130 ton) designs.

To illustrate this, an existing gear set was selected and rated per various standards (Table 1). Using the data in Table 1, this set was rated to the various AGMA rating practices to illustrate differences in specific rating factors. Each rating factor was normalized to its corresponding AGMA 6014 component (results shown in Figure 3).

The hardness factor C_H is more conservative in AGMA 321.05 and AGMA 2001. The dynamic factor $C_V\ K_V$ for AGMA 321.05 is not a function of accuracy, so it has a greater de-rating effect than the Q10/A7 values computed with the other standards. Also note that AGMA 2001 is more aggressive than AGMA 6014 for this factor. This was the intent of the mill gearing committee based on their experience with ANSI/AGMA 6004-F88 that adopted dynamic factor from the base standard without modification.

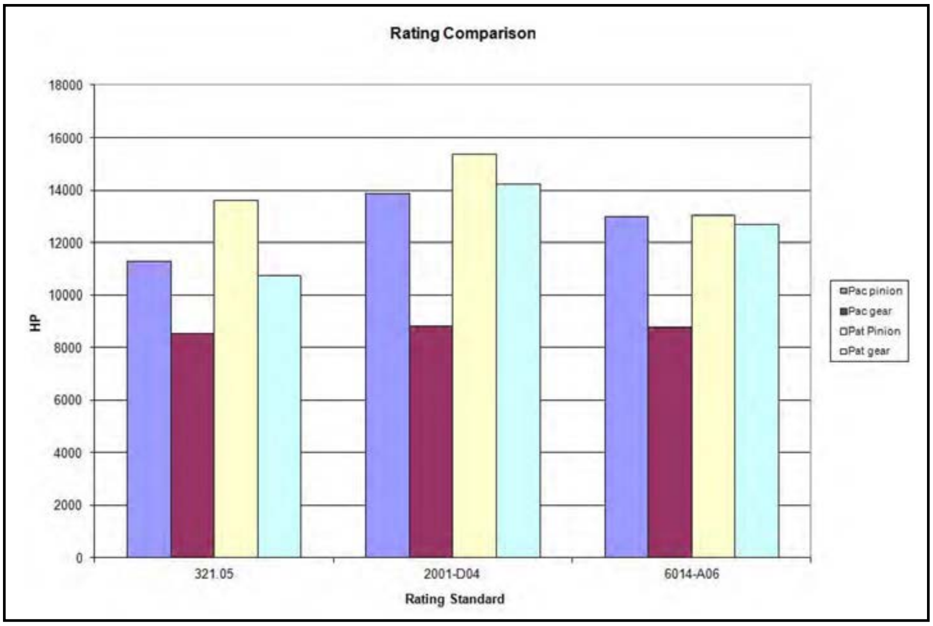

Figure 4 Rating comparison for the base set.

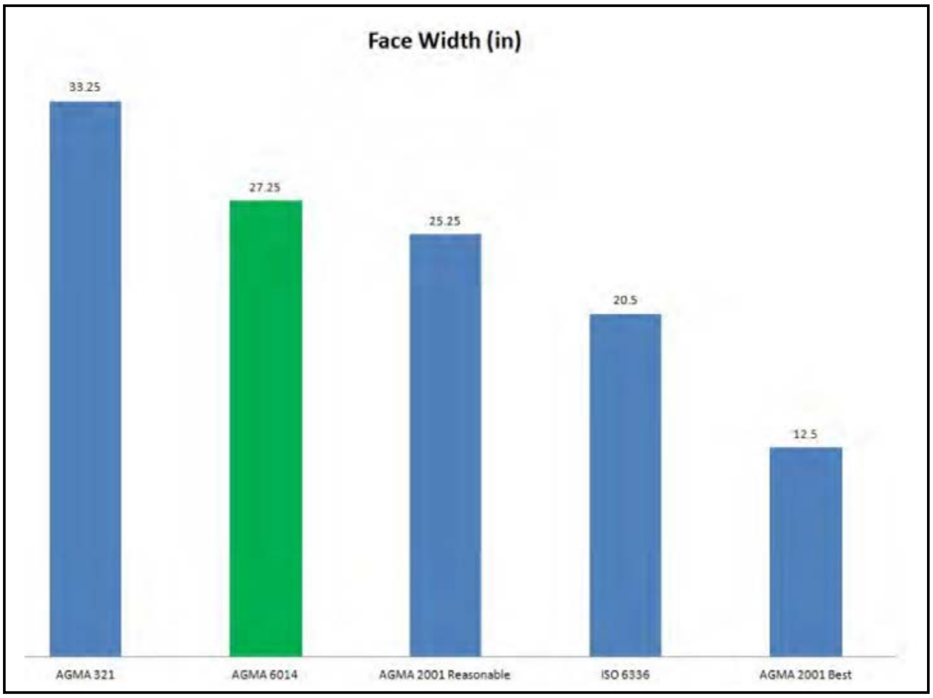

Figure 5 Face width as function of standard selection.

Load distribution $C_M\ K_M$ follows the same trend as dynamic factor for the same reasons. The change in I factor was caused by the release of the information sheet for its calculation. Stress cycle factors $Z_N\ Y_N$ were unknown in AGMA 321.05, and AGMA 6014 uses more conservative values than AGMA 2001 to control the power capacity of the set. The expansion of metallurgical specifications in AGMA 2001 and AGMA 6014 over the AGMA 321.05 requirements of hardness and "steel" affected the allowable stress numbers s_{ac} and s_{at}. The use of 55 HRC pinions also lowers s_{ac} and s_{at} in AGMA 6014 over the 58 HRC values in AGMA 2001. Reference 1 further outlines the differences and history of gear rating practice for mill and kiln drives in AGMA.

Given the interaction of the above factors, Figure 4 illustrates the resultant rating. The durability service factors based on transmitted power are 1.71, 1.76, and 1.75 for 321, 2001, and 6014, respectively. The strength service factors are 2.14, 2.84, and 2.53, respectively. We note the lack of strength rating in the AGMA 321.05 rating and the excess strength rating in the 2001 ratings highlighting the problem of rating a set optimized to a different rating practice. The actual power able to be transmitted is 3,204 kW, 3,758 kW, and 3,734 kW (4,297 HP, 5,040 HP, and 5,010 HP). The ~16.6% difference in power capacity between the AGMA 321.05 and AGMA 6014 ratings meets the goal of the Mill Gearing Committee to achieve a rating difference of 15% between the two standards.

To look at the effect of changing the base standard, Figure 5 compares the 6014 base design to sets designed to other standards utilizing face width adjustment. The axial overlap and heat treatment was kept constant as the face width was increased or decreased to meet the service factor requirement of 1.75/2.50.

The more conservative AGMA 321.05 increased the face width by six inches. Use of a general rating practice (AGMA 2001 and ISO 6336) with similar attributes to AGMA 6014 reduced face width by two and 6.75 inches, respectively. Aggressive use of the rating practice, termed "AGMA 2001 Best," enabled a 54% face width reduction. However, with extra precision mounting requirements, high tooth accuracy requirements, and reduced stress cycle performance, it is unlikely that the predicted, optimistic performance of this gear set in this demanding application would meet client expectations.

All gear rating standards stress the need for an experienced gear designer capable of selecting reasonable values for rating factors and who is aware of the performance of similar designs through test results or operating experience. When this is removed from the equation, through the use of an inappropriate rating standard or in combination with OEM or end user in-house practice, a valuable reality check is lost. In most cases, when the gear designer faces such a request, they also check the proposed design under AGMA 6014 or AGMA 321.05 to make sure it satisfies the standard requirements. In cases when the design sufficiently deviates, concerns of suitability or fitness of purpose need to be raised with the client.

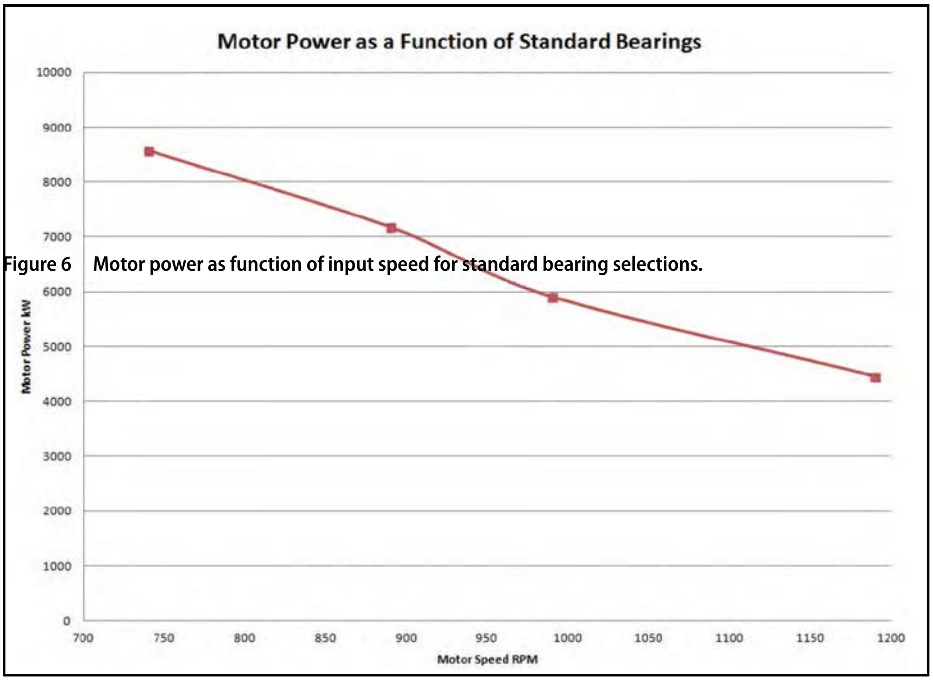

Figure 6 Motor power as function of input speed for standard bearing selections.

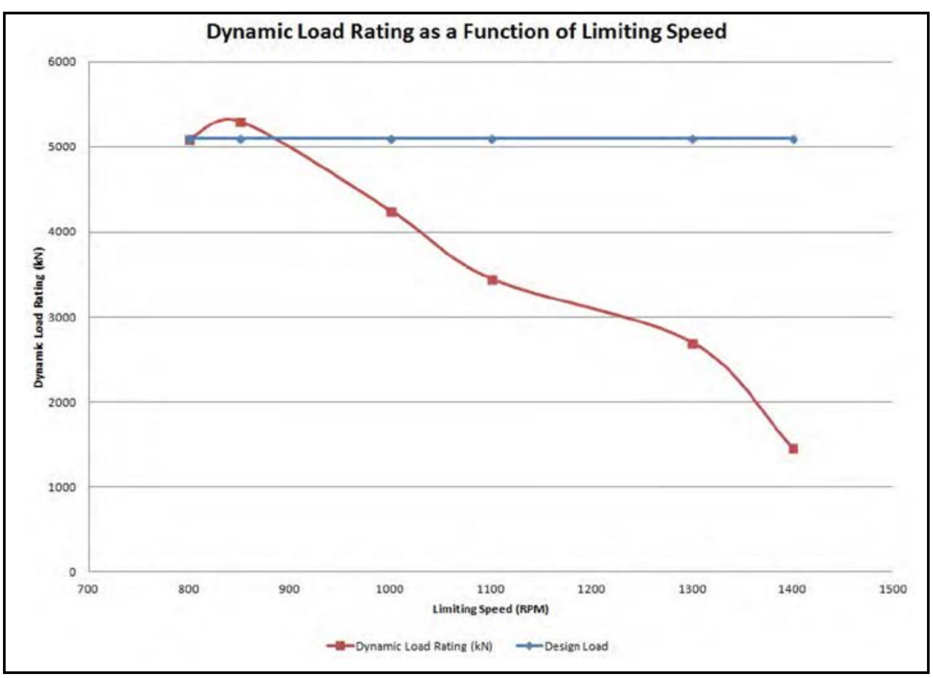

Figure 7 Dynamic load rating as a function of limiting speed.

Prime Mover Selection

Having resolved the output speed and input power, the next decision point is to determine the input speed to the system. Low speed synchronous motors in the range of 250–150 rpm are one option. This eliminates a source of power loss by removing the gear drive and coupling from the drive train. However there is a cost premium to multi pole (20 – 40) motors over the more conventional kind. Another option is to insert a gear drive between the motor and the mill pinion. If one is trying to minimize motor cost, the tendency is to maximize motor speed (1,500 – 1,200 rpm).

Figure 6 illustrates that as input speed increases, the amount of allowable power in a gear drive decreases. This is mainly due to cage velocity of the gear drive's input shaft bearings. Many bearing manufacturers publish speed limits in their catalogs as a function of thermal loading, above which some method of supplying cool oil to the bearing is required, as well as a limiting speed.

Table 2 Typical load history for mills

	Base load, hp	Speed	Start factor	Actual load, hp	Time per year, seconds	Number of starts per year	Time per year, hrs	Years	Time for 25 years of operation, hrs
Starting load	10806	180	1.5	16209	7	12	0.02	25	0.583
Inching load	123	1.4631	1	123	1800	12	6.00	25	150.000
Running load	10806	180	1	10806	31514316	1	8753.98	25	218849.417
							8760		219000

Table 3 Expected life of mill given a typical duty cycle for 1.0 overload factor

Overload factor K_o = 1.0 all cases				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	7.29597E-06	0.001922728	1.40726E-14	6.7759E-15
Sums	7.30141E-06	0.001929854	2.4685E-14	1.86478E-14
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life hours	29994206923.06	113480102.36	8871779508117490000.00	11744009610536000000.00
Life years	3423996.22	12954.35	1012760217821630.00	1340640366499550.00

Table 4 Expected life of mill given a typical duty cycle for 1.13 overload factor

Overload factor Ko = 1.13 for running loads				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	3.42739E-05	0.006479422	6.18972E-13	2.98033E-13
Sums	3.42793E-05	0.006486547	6.29584E-13	3.09905E-13
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life Hours	6388689342.00	33762184.86	347848597742773000.00	706667465221281000.00
Life Years	729302.44	3854.13	39708744034563.20	80669801965899.70

Table 5 Expected life of mill given a typical duty cycle for 1.25 overload factor

Overload factor K_o = 1.25 for running loads				
Duty	**Ratio wear pinion**	**Ratio wear gear**	**Ratio strength pinion**	**Ratio strength gear**
Starting	5.23024E-09	6.6996E-06	1.06083E-14	1.18697E-14
Inching	2.06011E-10	4.25751E-07	4.1522E-18	2.2215E-18
Running	0.000122969	0.017670034	1.40828E-11	6.78081E-12
Sums	0.000122974	0.017677159	1.40934E-11	6.79268E-12
	Pinion wear	**Gear wear**	**Pinion strength**	**Gear strength**
Life Hours	1780862895.23	12388868.30	15539229428435500.00	32240579740876500.00
Life Years	203294.85	1414.25	1773884637949.25	3680431477269.00

Limiting speed is a function of the form, stability, or strength of the bearing cage, lubrication, forces, precision, and other effects. Exceeding the limiting speed of a standard bearing forces the designer into high-precision, limited-production-run bearings that may not be readily available — or feasible.

Figure 7 illustrates the drop-off in load carrying capacity as a function of limiting speed for a 340 mm spherical roller bearing series. This illustrates the problem of increasing shaft speeds, thus limiting bearing selection to cause the rating element in the gear drive to be the high-speed bearing in place of the more typical and more expensive low-speed gear.

Gear Drive Considerations

The next selection point — given use of a high-speed motor — is how to distribute the ratio between the gear drive and the mill set. An initial conjecture is to wrap the gear as closely as possible around the mill or kiln and place the remaining ratio in the gear drive, based on the assumption that a carburized, hardened and ground enclosed drive is more cost-efficient in torque transmittal capabilities than the open set. This needs to be balanced by the loss in efficiency if a multiple-stage reduction drive is necessary for the ratio required. Typical single-reduction drives achieved efficiencies of 98.5 – 99%, whereas double-reduction drives are in the 97 – 98% range. If one is using a line of catalog gear drives, the steps in torque transmittal capacity as a function of unit size will also drive the selection. Forcing a mill pinion speed in a reducer drive train or selecting too fast of a motor speed can lead to low-cost items — such as input shaft bearings in the gear drive — constraining the entire design of the drive train. An example of this is the combination of high power (over 5,000 kW 6,700 HP) high-speed motors with L10 bearing requirements greater than the design amount based on the service factor of the drive. Requesting 100,000 hours of L10 life with a 2.0 service factor that implies 50,000 hours of life in a catalog-designed drive requires the drive designer to increase the size of the input shaft bearings to achieve such a life requirement. This may lead to going to the next unit size to achieve the L10 life requested. Not allowing the ratio in the drive to increase to use more of the excess torque capacity of the gear drive by slowing down the pinion speed causes an uneven distribution of torque generation between the drive and the gear set, thus increasing costs. It is best to advise the gear supplier of either the direct-driven or reducer-driven option and let them work out the most cost-efficient solution to size the gear/gear drive combination.

Duty cycle. A key parameter in gear train selection is the frequency of use. Since these sets are designed for 25 years of life, one needs to review the load cases to ensure that all modes of operation are addressed. ills experience starting loads; bringing the mill from rest to full operation; inching loads; where the mill is slowly turned at ~ 0.1 rpm for inspection or maintenance purposes; and the normal running load during operation. A typical load history is shown in Table 2.

Given this load spectrum, a Miner's rule analysis can be performed to determine expected life. Although the starting loads at 1.5 times and the inching loads are 1.4 times base motor power, they have a minuscule impact on life of the mill. Tables 3–5 list the expected lives of a mill set for selected values of overload factor K_O.

Service factor is made up of overload capacity, life expectation, reliability of stress number data, and economic risk of failure. For this type of service, the major component of service factor is *economic risk of failure*.

Design Considerations

The last items to consider are the arrangement and structure of the gear train. Gear material choices are a large cost driver to the overall design. These gears are typically made from cast steel, fabricated-forged and rolled steel rim with welded steel web, or ductile iron. Each material has its sweet spot in terms of cost-per-inch/pound of torque. Figure 8 illustrates torque capacity as a function of price index, with the most expensive design normalized to a value of 100. Reference 2 further outlines the cost considerations for large gears. As with items outlined above, since one is purchasing torque, it is usually best to allow the gear supplier to determine the optimal material for gear construction.

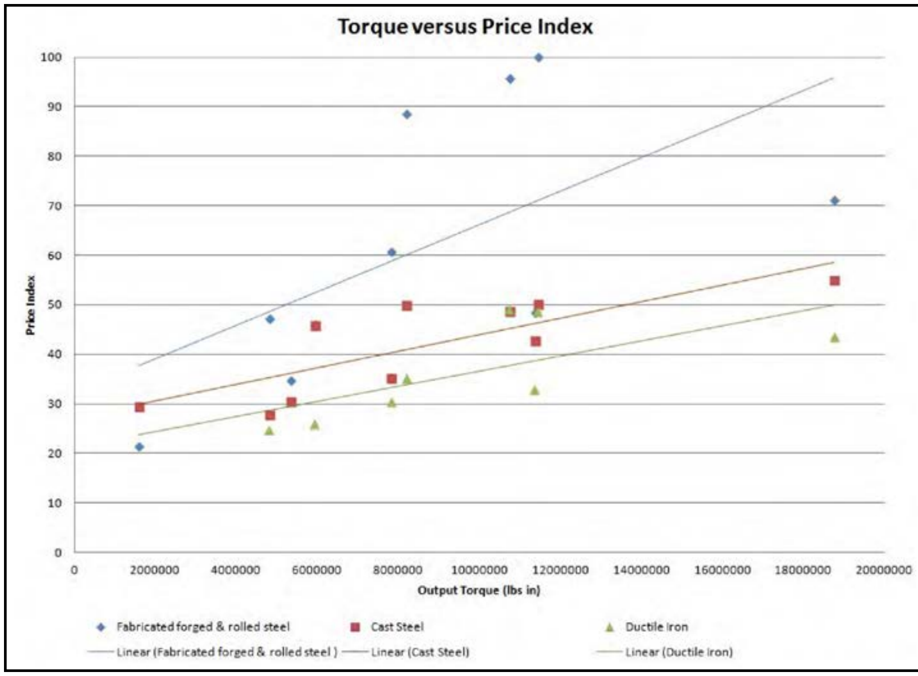

Figure 8 Cost comparison of blank construction as a function of torque.

Ambient conditions play a role, usually in the form of thermal considerations. Gear drives of this size are usually cooled by heat exchangers that need either a source of water or air at a reasonable temperature.

Lubrication systems are used to keep a constant flow of oil to the bearings and gear meshes. They need to function across the wide temperature range to ensure that the drive is not starved for lubricant at cold temperatures. In many cases successful oil pumping becomes an issue below 14°C (57°F) for VG320 mineral oils and 9°C (48°F) for synthetic. Immersion heaters, and/or bypass filtration lines may be necessary to ensure an adequate supply of oil. These considerations are avoided in the direct-connect, low-speed motor design.

The mill set typically requires much higher oil viscosities than a gear drive requiring either the use of diluents or heat-traced pipes to ensure flow of lubricant to the mesh. Another consideration is the altitude of the mine site where heat transfer to air may be reduced. Therefore minimum and maximum expected temperatures, altitude, and the availability of cooling methods need to be specified.

Support equipment can also influence drive train size. Pillow blocks are typically used to support the mill pinion. This gives the flexibility to adjust center distance and pinion orientation to optimize load contact.

The economic cost of downtime typically leads to large-diameter pinion extensions to reduce torsional stress. This — combined with a helix angle range of 5 to 11 degrees — usually results in shaft diameter in place of L10 life requirements determining the size of pillow blocks required. Coupling selection will influence the length of the shaft extension on the drive and driven equipment, as well as the torsional resilience of the drive train.

Required Data

Given all the above, the following data is necessary to successfully specify a mill drive set:

- Motor power
- Number of motors
- Mill speed
- Motor speed
- Design standard (for gear set and gear drive if required)
- Service factors based on above standard
- Gear interface dimensions (e.g., bore, minimum center distances, and drive train arrangement, weight limitations if any)
- Inching requirements (% of full load torque, mill speed in inching, desired connection point – mill pinion or gear drive/electric motor shaft)
- Duty cycle (if not continuous)
- Ambient temperature range (low and high)
- Altitude
- Specification requirements (e.g., nondestructive testing such as ultrasonic or magnetic particle, material properties)
- Inspection and witness requirements (on site visits to manufacturing location)
- Documentation requirements
- New or existing installation (if existing, need tooth geometry)

Conclusions

To resolve the two previously cited problems in life, one needs to clearly understand what one wants to do and communicate that to the people who can accomplish the task. Writing a gear train specification requires attention to detail and a realization of the impact that those choices can make. This paper reviewed the drive train outlining the information necessary for the gear designer to successfully develop a gear for this application. It noted various items that can play a dramatic role in the size and cost of a selection and indicated where creative freedom is necessary for an optimized drive that considers capacity, cost and lead time. **PTE**

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10. AGMA Standard 221.02. *Rating the Strength of Helical and Herringbone Gear Teeth.*
11. ANSI/AGMA Standard 6004-F88. *Gear Power Rating for Cylindrical Grinding Mills, Kilns, Coolers and Dryers.*

Frank Uherek, principal engineer with Rexnord in Milwaukee WI, received a BSME from the Illinois Institute of Technology in 1981 and a MBA from the same institution in 1985. He has been involved in the gear engineering field for over 33 years, holding various positions in design engineering and quality management covering enclosed drives, wind turbine drives, and open gearing for mill and kiln applications. AGMA activities include chairman of the Helical Gear Rating Committee; membership of numerous technical committees; and editor of AGMA 2001, AGMA 2121, AGMA 6014, and AGMA 6015. He received the AGMA TDEC award in 1997 for his outstanding contributions to the art of gear design and utilization. He has previously presented three papers at AGMA Fall Technical meetings and co-wrote two papers for IEEE cement industry conferences. In 2011 he was honored with the AGMA Distinguished Service Award for his work in developing AGMA gear rating standards.