Gear Material Selection and Construction for Large Gears

Frank C. Uherek

A road map is presented listing critical considerations and optimal use of materials and methods in the construction of large gears.

Introduction

The purpose of any gear mesh is to transmit rotary motion and torque from one location to another at a consistent rate. Various rating practices from AGMA, ISO and others go into great detail about the tooth proportions, accuracy requirements, material selection and cutting methods to produce a tooth that satisfies the requirements of the application. However, standards do not provide all the information necessary to ensure the torque at the gear tooth is actually moved to the piece of driven equipment, i.e.—gear blank design. In most enclosed drive applications, a disk of the same face with a bore and keyway is sufficient. Yet, in the realm of large gears—defined as three meters (10 feet) in diameter and above—a solid blank fulfills the design engineer's maxim of making the part difficult to manufacture and impossible to install. Blank design needs to be driven by the application and the range of materials available to ensure that sufficient stress capacity is available at the teeth—as well as the ability to connect with the driven equipment. This paper covers these issues in a specific area of use: gearing for cylindrical grinding mills and kilns.

Background

Grinding mill and kiln services are unusual installations for gearing when compared to traditional enclosed gear drive installations, yet these applications have been utilized for more than 85 years. The grinding process—more accurately, a tumbling process—uses horizontal rotating cylinders that contain the material to be broken, potentially augmented by grinding media. The material moves up the wall of the drum until gravity overcomes centrifugal forces; it then drops to the bottom of the drum to collide with the remaining material. This breaks up the particles and reduces their size. Kilns rotate at far slower speeds to enable even firing of their contents. 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. For mill applications the gear is mounted on the mill using a flange bolted connection (Fig. 1). For a kiln, various types of spring plates are used; 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 rolled steel or spheroidal graphitic iron. Pinions are carburized, induction-hardened or through-hardened steels. 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 weight or pouring capacity becomes an issue.

Structure Requirements

Based on the application, these gears need to have large bores to accommodate the mill or kiln shell. This enables use of reduction ratios not normally thought of as reasonable (i.e., 8:1 to 20:1) in a single stage. The gears are bolted to the mill through a flange connection or mounted on tangential spring plates to allow for thermal growth (Fig. 2).

The next step is to connect the bore of the gear to the teeth; this is done by either using a "box,"—also known as a delta- or Y-shape—or T-shape structure (Fig. 3).

A typical ring gear has a series of windows cut into the material for handling and weight considerations (Fig. 4).

Figure 1   Grinding mill installation.

Over time, design rules have been developed to address the material shape distribution of the various elements of the ring gear structure. Rexnord has over 5,000 gears in service with design lives exceeding 25 years that confirm these rules and calculations reflect field requirements. The purpose of the structure is to provide stability at the tooth location to ensure the assumptions made at the rating phase of gear development are supported by the actual blank design. Annex C of ANSI/AGMA 6014–A06 discusses the following considerations for blank design:

- Reduction of strength rating by moving the location of bending fatigue failure into the gear rim from the tooth root (K_{Bm} factor)
- Effect of rim deflection on the load distribution factor K_m
- Influence of the mating element on load distribution factor K_m
- Definition of dynamic alignment techniques to achieve correct mesh patterns

Rim thickness is a significant parameter in the design. There is a minimum value of the thickness specified by the rating standards to ensure any bending strength failure of teeth would travel through the base of the tooth and not through the rim of the blank. Based on field experience, AGMA 6014 suggests designs having a backup ratio $mB > 1.0$

$$m_B = \frac{t_R}{h_t}$$

(1)

where:

m_B is back-up ratio
t_R is gear rim thickness below the tooth root, in.
h_t is gear tooth whole depth, in.

This avoids the need to derate the gear to move the failure mode to a more conventional area. Other standards, such as ANSI/AGMA 2001–D04, feel a value of 1.2 is more appropriate. A point of debate is what is considered the inside rim of a gear. Conservative thinking would require that any missing material below the tooth root is the start of the inside rim diameter. Many designs feature a groove in the side of the gear for mounting of a dust shield. This groove is located to generate a backup ratio of ~ 0.6– to 0.80. The loss of support, typically 13 mm (0.5 inch) is not considered significant when working with face widths of 380 to
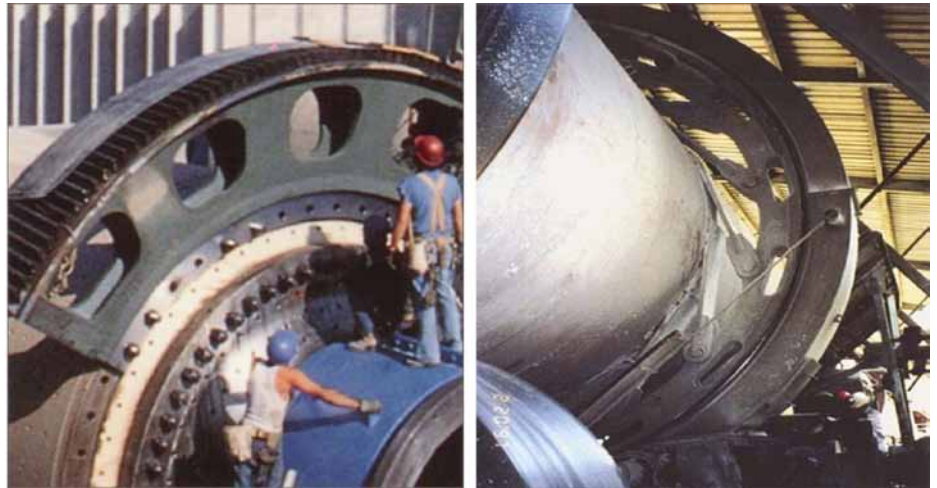

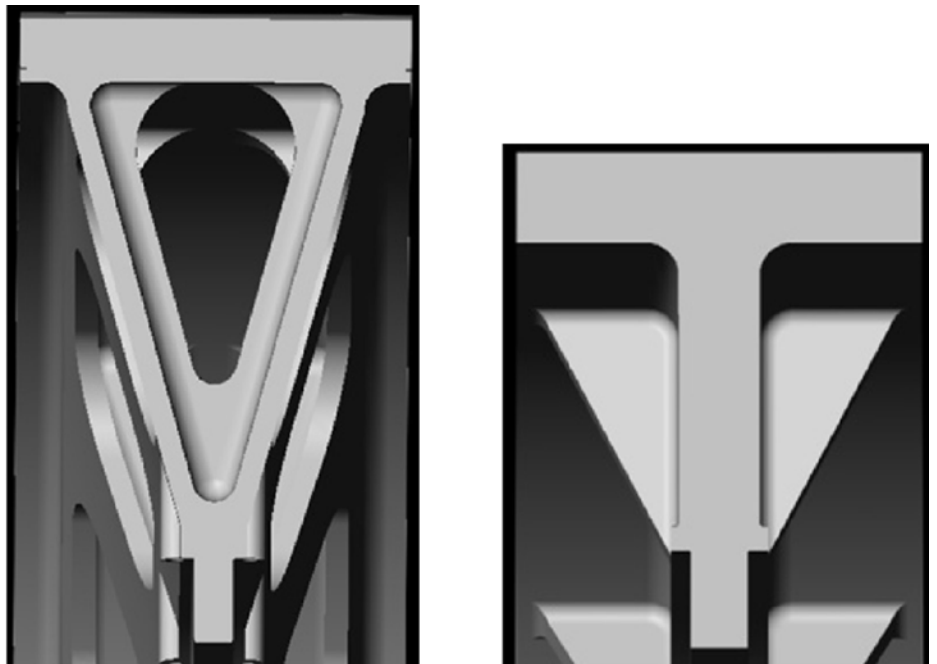Figure 2   Flange mounting and spring mounting options.

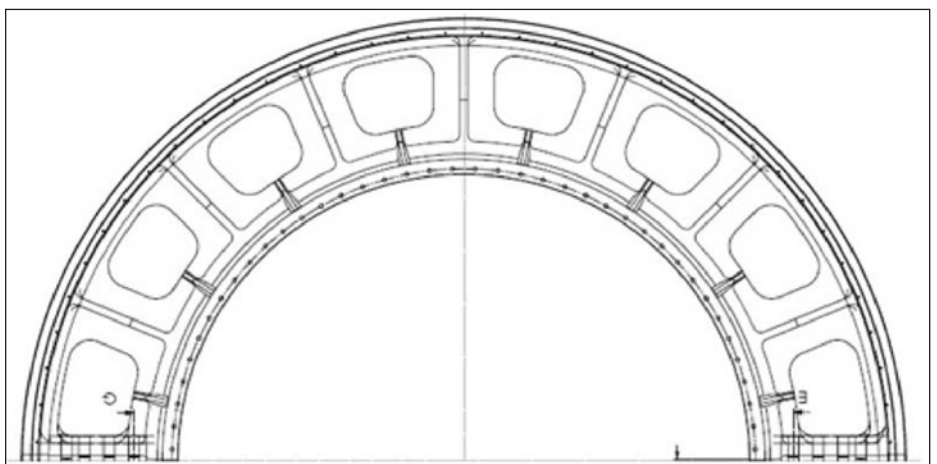Figure 3   Box/Y/delta and tee shape cross section.

Figure 4   Side view of ring gear.
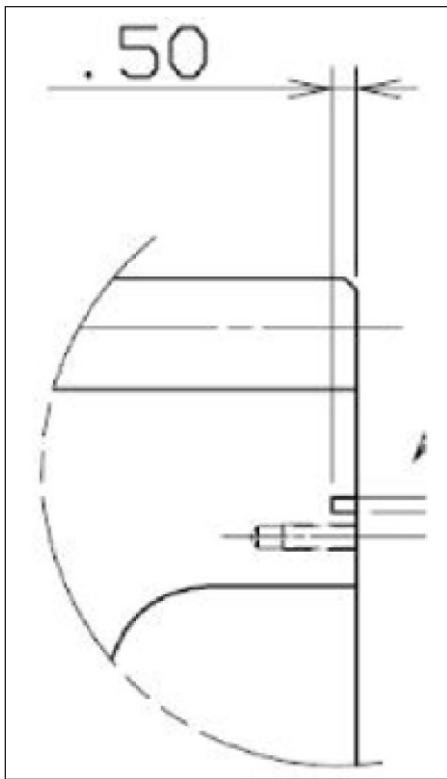
Figure 5   Locations of dust guard groove and indicating band.

Figure 6   Overhang deflection modes.

1,015 mm (15 to 40 inches). The next loss of material underneath the tooth root is an indicating band typically turned on the inside of the rim diameter to facilitate installation (Fig. 5). This loss of support is typically 25 mm (1 inch) wide. Finally there is the true inside rim diameter that is typically 13 mm (0.5 inch) beyond this value. Reasonable design practice tends to use the machined indicating band as the location of the inside diameter for the purposes of determining a value for the rim thickness factor, KBm.

Since 60 percent of the weight—and, therefore, cost— is tied to rim size and thickness, optimization pays large dividends. A start point for rim thickness values are a backup ratio of 1.10 for box section gears and 1.25 for tee section gears. As gears move toward finer pitches (i.e., $< 25\ m_n, > 1.0\,\text{DP}$) what tends to drive rim thickness is the tapped hole beneath the guard groove for support of the external dust guard. At larger modules, deflection tends to be the controlling factor.

Achieving calculated values of load distribution, K_m, is a function of tooth generation accuracy and rim support. Based on the rating practice, these types of gears are typically A9 to A7 (Q8 to Q10) for helix accuracy. Typical verification methods are
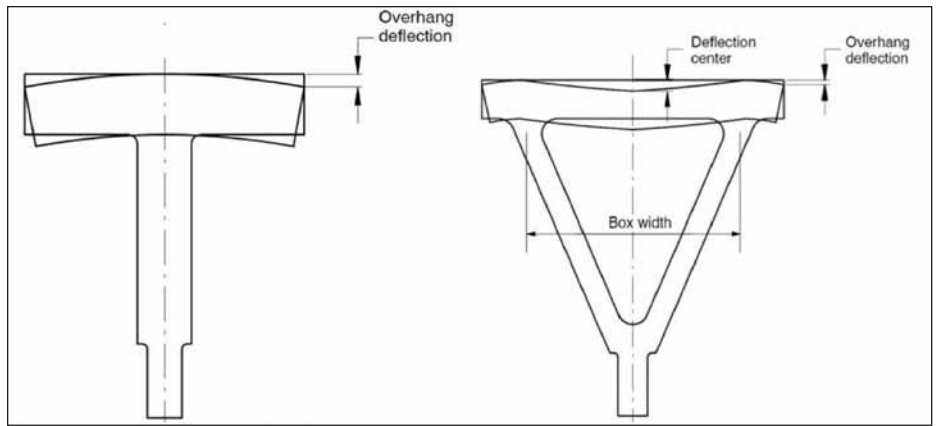
a helix check of the pinion and a contact check with the gear to confirm mesh compatibility. One typical deflection source comes from face movement away from the pinion either in the center portion of box Y rims or the end portions of T rims.

Two other deflection modes are rim deflection and face deflection (Fig. 6). Rim deflection occurs when the rim sags between the arms of the gear. Face deflection arises when the entire gear bends from the mounting flange due to thrust force of the teeth. A good design practice is to limit maximum deflections of these three modes to be less than 25 μm (0.001 inches).

The other two parameters affecting K_m are influence of the mating element and dynamic alignment techniques to achieve correct mesh patterns. These are beyond the scope of this paper.

When designing large gear blanks, the major factors to be considered are:
• Load
• Face width
• Rim thickness
• Stiffener spacing and number of windows
• Window size
• Support web thickness
• Material

Loading on these blanks comes from three sources: the amount of power being transmitted though mesh, handling as horizontal rings during manufacturing, and handling as vertical segments or semi rings during installation.

Typically the requirement for maintaining tooth alignment is the chief driver for dimensional selection. Wider face widths tend to require additional rim thickness to manage overhang deflection. For cast steel designs, the crossover point

between tee and box Y section designs is 760 mm (30 inches) of face width. In a specific example, a 6,250 kW ball mill gear at 16.76 rpm output speed has a required rim thickness value of 210 mm (8.26 in) in a tee configuration whereas the box Y gear has 165 mm (6.51 in). This reduces the overall weight of the gear to 61,600 kg (135,700 lbs.) in a box Y configuration but 67,500 kg (148,700 lbs.) as a tee configuration.

Fabricated steel and ductile iron designs cannot take advantage of a box Y design due to cost of construction and material flow during the production process and as a result will have thicker rims with these face widths.

As noted above, rim thickness is driven mainly by requirements for failure through the tooth root and not the blank. Locations of customer supplied guarding and deflection also drive this parameter.

The number of stiffeners is a function of the web height of the gear (i.e., distance between the bore and the inside rim diameter), the number of windows, and the amount of helix angle of the gear to prevent face deflection. For gears of tight cross section (< 150 mm, 6.0 inches) they may not be needed due to the stiffness of the web and are not practical from the construction standpoint. As the diameter of the gear increases, the distance between stiffeners becomes a greater influence factor on rim deflection over the windows in the blank. To provide adequate support, stiffeners should be placed ~1 m (40 inches) apart for tee

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designs and 1.2 m (50 inches) for box Y designs. This allows for a reasonable balance between deflection and rim thickness. This will also drive the number of windows between the stiffeners.

Window sizing is a parameter that is more critical for cast steel and ductile iron designs than for fabricated gears. All three gears share the same need for windows from the standpoint of openings for lifting slings and chains. Since the window is a sunk cost for a fabricated design in that the web plate is sold by weight and window cutouts are unlikely to be used in other portions of the fabrication, the size of the window should be dictated by handling only. Whereas for cast designs, material usage can be optimized allowing for larger windows having a weight savings. In addition, they are necessary to support the top portion of the mold during pouring. From the deflection standpoint, window size plays a role in supporting the rim between stiffeners. In bending deflection, the moment of inertia is in direct proportion to width but a cubic function of height. Therefore, when dealing with rim deflections issues, lowering the window outside diameter to stiffen the web and therefore reduce the deflection may be a better use of material than adding thickness to the rim area.

Support webs come in two types for these gears. Box width for cast steel gears is a function of allowing sufficient space for the split joint hardware as well as stiffness to manage sag between the side supports. Web width for tee sections adds stiffness and transmits torque between the rim and the bore of the gear. In both cases, thickness and location are driven by rim deflection considerations.

Material is the last parameter to consider in gear blank design and has a significant impact on cost. An advantage of fabricated designs includes the ability to assemble gears with high alloy at the tooth locations, necessary for torque transmission, while using a lower grade alloy for the structure of the blank. This cost savings may be offset by the welding assembly cost. Contrary to fabricated gears, cast gears are constructed of uniform isotropic material, avoiding any issues with performance variation as a function of alloy. Optimizing window size is an option cast gears can utilize to offset a portion of the material cost.
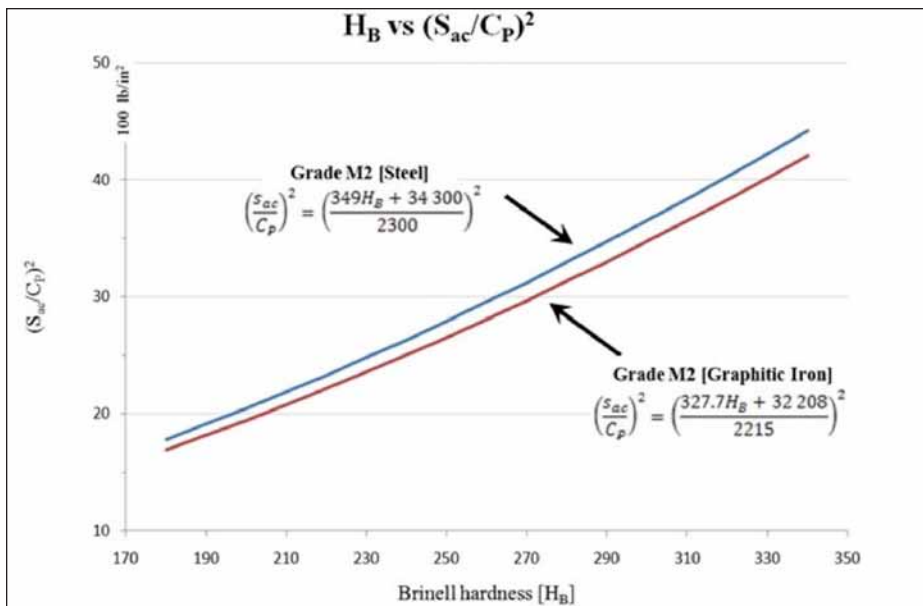

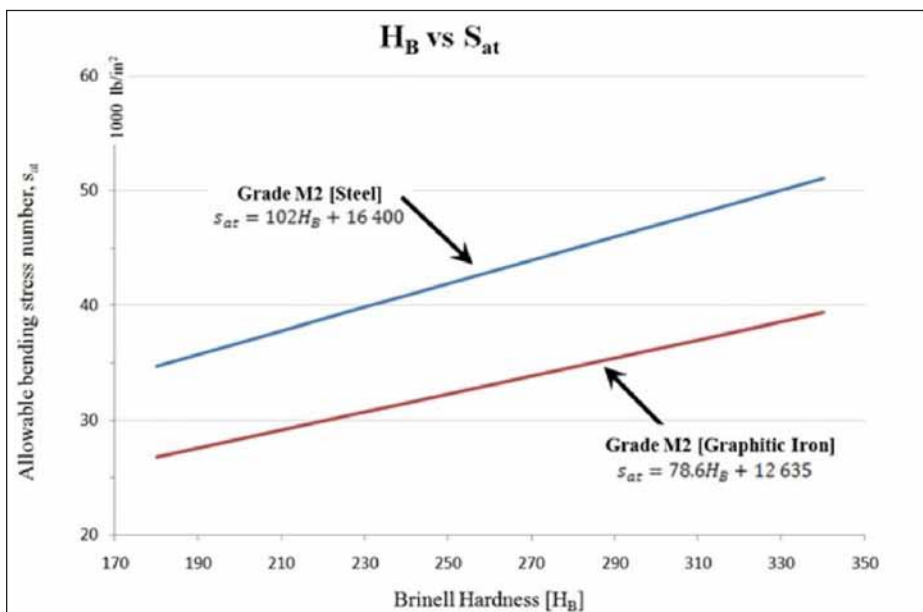**Figure 7   Comparison of material related factors for pitting resistance.**

Figure 8   Comparison of material related factors for bending strength.

Rim Material Choices
The selection of rim material is driven by the blank manufacture method. For large gear design there are currently two choices in use today: fabricated and cast structures. Steel can be used in both options, while ductile iron is only available as a cast option.

Fabricated structures consist of a rolled rim SAE 1045 or rolled ring forging SAE4340 plate that is welded to an ASTM A36 web plate with stiffeners. Design hardness is 180 HBW for the 1045 material and up to 265 HBW for 4340 plate.

Cast steel is the traditional material used for large gear blank designs. Typically proprietary alloys are used to enable sufficient hardenability through the rim area to ensure design hardness at the root diameter. Design hardness ranges from 180 HBW to 335 HBW.

Ductile iron is an alternate cast material; it offers similar weight optimization attributes as cast steel, with the additional benefit of absorbing noise vibration due to the precipitated graphite particles. As noted below, this comes at the cost of reduced power capacity. Design hardness ranges from 180 HBW to 335 HBW.

From the power capacity standpoint, there is no difference in allowable transmitted power between fabricated and cast steel gears of the same hardness. But the same *cannot* be said for ductile iron.

Due to the lower values of the modulus of elasticity, Poisson's ratio and different fatigue life performance in contact and bending stress, the power capacity of a ductile iron gear is typically lower than the same gear made from fabricated or cast steel. Pitting resistance changes by 5 percent in this case (Fig. 7).

A larger difference occurs in bending strength, resulting in a 23 percent difference in power capacity (Fig. 8). These values are based on the rating formulas in AGMA 6014 using grade-two material. Grade-one material requires less material certification and therefore has lower power capacity. Other standards may indicate a different comparison.

Typical service factors for grinding mills require higher values for bending strength than pitting resistance. Kilns at 1.5 rpm output speed require values of 1.00 and 1.75 for C_{sf} and K_{sf}, respectively. Ball and SAG mills require 1.75 and 2.50, respectively in higher power applications ($> 3{,}350\,kW$ or $4{,}500\,hp$). When using ductile iron gears in these applications, the reduction in bending strength requires either wider face widths or larger modules (coarser pitches).

Another consideration is the yield strength of the material. Figure 9 illustrates that ductile iron has ~60 to 70 percent of the yield strength of its steel counterpart for the same material hardness.

This becomes an issue when reviewing the performance of mill gears in low cycle inching or maintenance drive usage when the number of load cycles is expected to be less than 10,000. For cycles greater than 10,000, there is no fatigue life performance difference between steel and ductile iron, per AGMA 6014.

Construction Considerations

Each of the three methods of ring gear fabrication offers significant benefits, as well as noteworthy disadvantages that can be used as a guideline for the selection process.

The initial consideration is the client interface dimensions, as gear designers have little control over the bore of the gear and the connection interface to the structure. For applications that feature a gear reducer in addition to the gear set, the distribution-of-ratio between the gear drive and the final-stage reduction will have a significant impact on cost. 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. 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. The final consideration is the overall cost of providing torque to the mill or kiln in terms of selecting a low-speed (200 rpm) motor and directly connecting it to the mill pinion in place of a higher speed motor (1,170–740 rpm) and including a gear drive in the train. 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. Forcing a mill pinion speed in a reducer drivetrain 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 drivetrain. An example of this is the combination of high- power (over 5,000 kW or 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 requires the drive designer to increase the size of the input shaft bearings to achieve the life requirement. This may lead to an increase in drive 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 between the drive and the gear set—thus increasing costs.

Cast Ductile Iron

The next consideration is to select the material for construction. When designing with ductile iron, the first consideration, as noted above, is the reduced bending strength rating. This will tend to drive the design to larger modules that require greater rim thickness due to the requirements of the rim thickness factor, K_B; having a thicker rim will also help in controlling overhang deflection of the rim. Having a modulus of elasticity ~11 percent less than steel will result in ductile iron moving more under the same load. To control this, rim and web sections tend to be larger than on a comparable-sized steel gear. For successful casting, abrupt section size changes should be avoided with this material due to solidification dynamics. In addition, to achieve
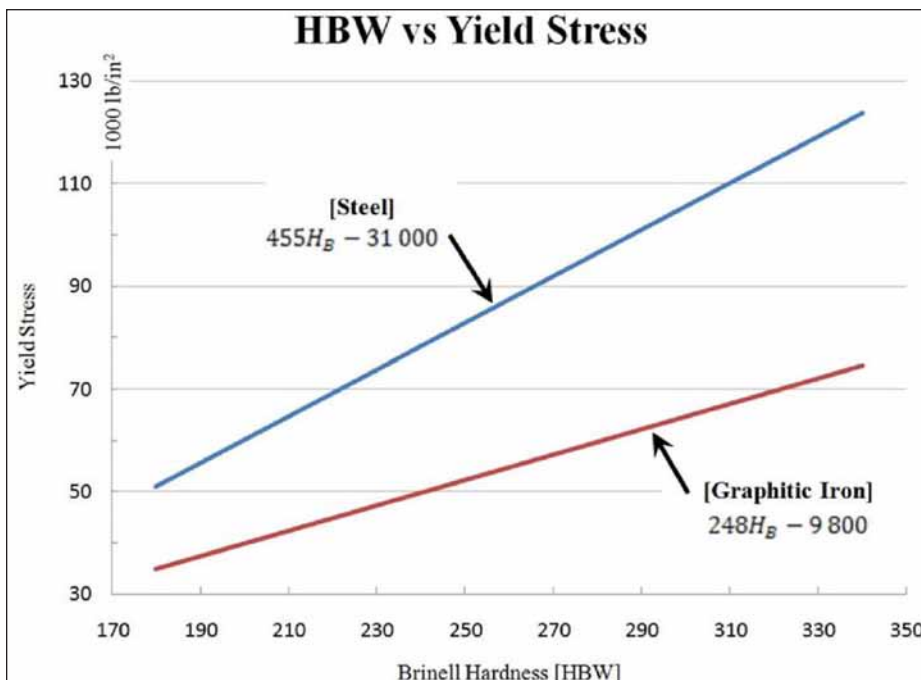

Figure 9   Comparison of material related factors for yield strength.

uniform cooling, mold chills are required in the rim and bore area; this tends to prevent adoption of the box Y design for these types of gears. With the use of a T section, the pattern cost of a ductile iron gear is reduced.

Steel Gears

The next option in material is the use of steel for the rim material. This eliminates the loss of rating present in ductile iron designs. Therefore, the choice between fabrication and cast steel designs is dependent on material performance issues, construction options, lead time and cost.

Material performance of wrought versus cast steels. Wrought rim material consists of a steel plate that is rolled into a ring shape. Rolled metal develops and retains a fiber-like grain structure aligned in the principal direction of working. These fibers are the result of the elongation of the microstructure constituents of the metal in the direction of working. Due to the directionality, the mechanical properties of the plate are not uniform in the three principle directions of longitudinal, long transverse and short transverse—causing anisotropic performance. Variations in property performance are attributed to this elongation, as well as the stress concentration effect of loading normal to the major axis of an essentially elliptical void.

Cast rim material is poured into its final shape by the molding process. On a microstructure level, there are no directional property variations in the material since the micro-discontinuities generally have no preferred orientation. This random orientation allows for isotropic performance, avoiding changes in material performance based on load direction.

Construction options. Fabricated gears, in some applications, can reduce costs, since no pattern is required. Manufacturers may be limited by the ability to obtain a rolled, forged ring for the rim of the gear as well as oven or furnace capacity to stress relieve the gear after welding.

For fabricated gear blanks, the process starts with flame cutting of the rim plate material to required width. After stress relieving, the plate is rolled to shape and stress-relieved again. The center portion of the gear is also flame-cut and stress-relieved prior to welding to the rim mate-

rial. The various parts of the gear are then welded, stress-relieved and, finally, the assembly is normalized and tempered to specified hardness. The large number of stress-relieving operations is necessary to prevent subsequent movement during machining and in-service operations. Selection of the proper weld rod material—typically heat-treatable electrodes—and pre-heat temperatures are necessary to ensure a successful fabrication. Pedersen (Ref. 9) indicates that it is well known that welded joints have low fatigue strength compared to the base material. This is mainly caused by local stress concentrations due to the presence of notches and high-tensile, residual stresses. Notches occur both because of the geometry of the joint and weld imperfections, such as undercuts and slag inclusions. Tensile-residual stresses arise from the contraction of the weld metal during cooling and solidification. Therefore, on mill gears, the weld joint tends to fail first from fatigue, given no installation, alignment or lubrication issues are present.

Generally speaking, castings are versatile and economical. The casting process utilizes the liquid metal's ability to flow into extremely complex shapes—even those with internal pockets and external projections. As a result castings produce a seamless, one-piece component that offers uniform strength and toughness. For cast gear blanks, the process starts with the construction of the pattern to make the casting. After pouring, the blank is allowed to cool in the mold. The risers and casting gate system are then removed and the remaining sand is cleaned off the blank. Magnetic particle inspection is conducted to indicate areas needing process welding. After process welding—preferably with heat-treatable electrode material—and a final magnetic particle inspection—the piece is normalized and tempered to achieve specification hardness. The location and volume of risers are critical to achieve a casting free of macro-porosity. Based on solidification rates, the risers feed additional material while cooling and collect slag and other material impurities. It is critical to have this impure material out of the tooth rim location to ensure uniform performance.

Lead time. A mill gear, whether constructed as a cast structure or fabricated rim, is a product that requires weeks of manufacturing before the client receives the end product. The main factor that determines the lead time is the construction process of the blank. For a cast gear these include building a pattern, melting and pouring the raw material, and solidification and extraction of the gear; this takes about 14 to 16 weeks for a cast gear. Since a fabricated gear is shaped from rolled steel plate, the construction time is shortened compared to a cast design—assuming stocked plate material. A fabricated blank must be hot- or cold-rolled from flat plate and then welded into shape; these two processes usually take about three to four weeks. Beyond construction of the blank, the rest of the gear manufacturing—milling, boring, turning, tooth cutting and drilling—is the same, regardless of blank construction. Construction time of the blank, whether cast or fabricated, seems to be minimal compared to the overall process from start to finish that historically tends to average around 50 weeks.

Due to the need for rapid response for field issues, some business interruption policies require either storage of the blank pattern to reduce turnaround time for replacement, or having a spare blank available for tooth cutting. Transportation and storage costs need to be reviewed when this option is selected.

Cost. Fabricated gears have a perceived cost advantage over castings due to the lack of pattern construction. Molding cost is not an issue since all steel starts out as cast in either ingot or melt form. Specification for plate steel must include low sulfur requirements to avoid issues with rim laminations. Cast designs must be produced with sufficient pouring capacity for manufacturing both the gear and filling the required risers to ensure material integrity at the rim, split joint and bore flange of the gear. Sixty to 65 percent of the steel used in the pouring of a mill gear is consumed in the riser and gating system. This is recycled for the next gear so this additional material is not considered part of the gear cost.

To identify a crossover point between the three options, a selection of sets was developed and the gear blanks were priced reflecting the three types of mate-
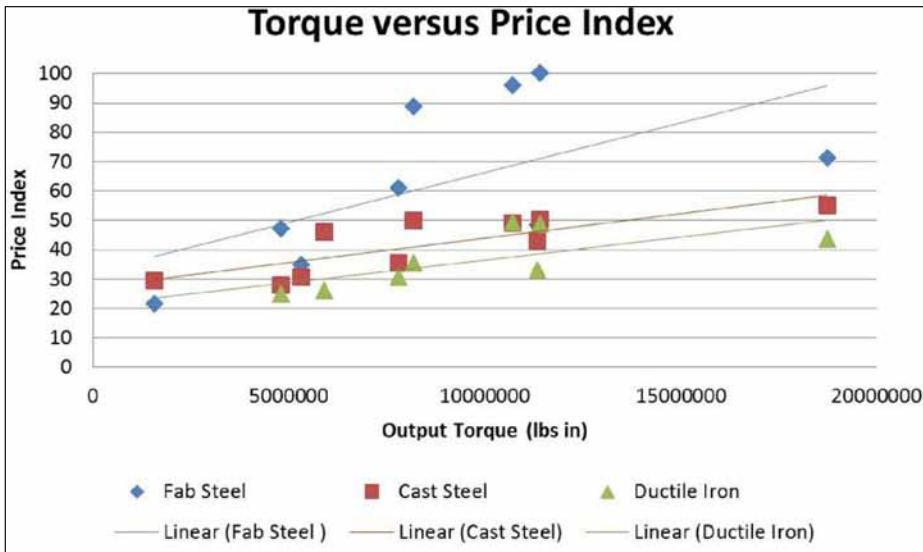
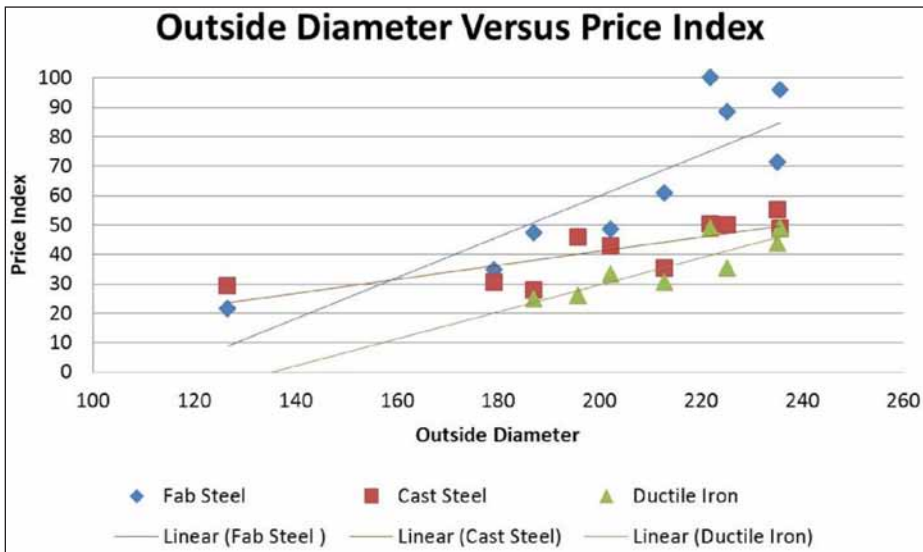**Figure 10   Cost comparison of blank construction as a function of torque.**

Figure 11   Cost comparison of blank construction as a function of diameter.

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rial. The price index is normalized to the most expensive blank at 100; the results are plotted in Figure 10.

Looking at the data as a function of outside diameter (Fig. 11) illustrates a crossover point of ~ 4 m (160 inches), where cast steel gears overcome the pattern cost. As gears become larger the ability to shape the material to match loading requirements begins to pay significant dividends.

Conclusions
There are a variety of options for manufacturing methods for large gear designs. Fabricated steel, cast steel and ductile iron designs offer advantages as a function of output torque requirements. Voice-of-the-client data indicates that the key drivers

in selection are lead time and cost, based on the expectation that high quality and compliance to specification are provided. Based on that, we see a transition from fabricated to cast designs at about 4 m in diameter; further work is required on the influence of bore size and hardness on cost. Ensuring that good design and material selection criteria are followed, each material type and construction method has its place in providing torque transmission for the application. ⚙

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Frank C. Uherek is principal engineer/mill products for Rexnord Geared Products in Milwaukee WI. He received both a BSME (1981) and MBA (1985) from the Illinois Institute of Technology. Over 32 years, he has held various positions in design engineering and quality management, covering enclosed drives, wind turbine drives and open gearing for mill and kiln applications. He has served on numerous AGMA technical committees, including as vice chairman (and editor) of the Helical Gear Rating Committee, and received the AGMA TDEC award in 1997. In 2011, Uherek received the AGMA Distinguished Service Award for his work in developing AGMA gear rating standards.