Acoustic Properties

OF INDUSTRIAL MACHINERY

Understanding noise—its origins and why it's occurring—can help avoid problems and lead to better outcomes in the manufacturing environment.

Provided by Philadelphia Gear

Introduction:

In the early 1970s the Occupational Safety and Health Act (OSHA) generated more awareness, concern, and a need to control industrial noise (sound levels). This in turn created a need to advance technical knowledge on industrial product sound generation and to set sound standards on all types of equipment. However, the complexity of the subject made most sound standards difficult to apply or interpret properly. Therefore, some of the major gear manufacturers developed literature to disseminate knowledge to their product users on acoustical measurements, level control, and sound attenuation for gear driven equipment.

Also at this time, the AGMA Acoustical Technology Committee was formed to standardize methods for sound measurement and geared product acceptance testing. It also developed a Gear Sound Manual to provide improved communication between project engineers, gear manufacturers, and users in the areas of Fundamentals of Sound as Related to Gears (Part I), Sources, Specifications and Levels of Gear Sound (Part II), and Gear Noise Control (Part III). The work of two gear engineers—Bill Bradley at Philadelphia Gear Corporation and Richard Schunck at the Falk Corporation—formed a majority of the information that went into this AGMA Gear Sound Manual that was first published in 1978. The information was originally issued as three separate documents. In 1987 the information was combined and published as one document. The latest version of this AGMA Information Sheet was updated between 2002 and 2004 and published as AGMA 914-B04. It contains additional references and information such as typical levels generated and analysis methods for those who wish more on the subject.

This white paper—the second in a two-part series—is to inform you generally on the topic of noise as related to Philadelphia Gear Corporation industrial products. Realizing that you lack the time, background or inclination to read texts on acoustics, this outline will be kept as brief as possible. The first installment appeared in the January issue of *Gear Solutions* magazine, and can be found at [www.gearsolutionsonline.com].

Section II: Designing Noise Out of Gears

The best time to solve a gear-noise problem is at the design stage. "Retrofixing" is usually expensive and unsatisfactory. Many factors play a role in ensuring a quiet gear mesh. Adjusting one or more of them before the hardware is manufactured adds little to the cost, but may make a significant difference in noise level. However, to complicate matters, there is always the need for a balance between low noise and satisfactory performance. The most important design factors that influence the noise level of a gear set are:

- Type of gearing
- Quality level
- Tooth profile
- Surface finish
- Pitch
- Gear runout and unbalance
- Pressure angle
- Gear ratio
- Recess action
- Resonance
- Profile modification
- Lubricant viscosity
- Overlap ratio
- Type of bearings
- Backlash
- Material
- Tooth loading
- Housing

Type of Gearing

For quiet operation, parallel-shaft gears such as spur, helical, and double helical are preferred to right angle and crossed-axis gearing. The primary reason is that spur and helical gears make it possible to maintain tight tolerances and to operate with minimum friction.

Helical gear types have the added advantage of maintaining more than one tooth in contact during operation (helical overlap). Because of this, it is possible to get as much as 12 dbA reduction in noise by using them instead of spur gears.

With double helical gears, there is the problem of correctly manufacturing the two helices with exactly the same phase and accuracy; in other words, without apex runout for weave. Any slight deviations in manufacturing, coupled with the axial mass inertia of the gear, will prevent equal load sharing and smooth operation; this will contribute to vibration and noise.

Thus, for smooth, noise-free operation, the optimum type of gear throughout all speed ranges is the single helical gear. Any thrust or overturning loads that single helical gears may induce are easily offset by modern design techniques.

Tooth Profile

Tooth profiles other than the common involute type have been developed over the years. None of them have been widely accepted, mainly because they lack sufficient advantages and manufacturing history to justify deviating from the involute form.

Gears with circular-arc tooth profiles, for example, can retain more lubricant between mating teeth, thus tending to reduce noise and wear. However, the circular-arc form is still largely experimental, and has not

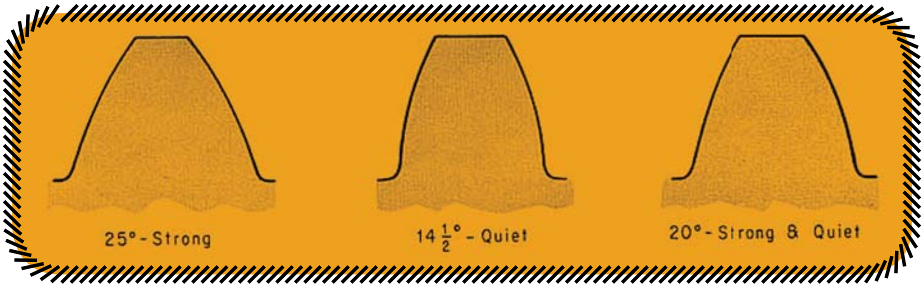


FIGURE 1: A 20-DEGREE PRESSURE ANGLE IS THE BEST COMPROMISE BETWEEN STRENGTH AND QUIETNESS. OTHER TOOTH FORMS MAY BE USED WHERE ONE FACTOR OR THE OTHER IS MORE IMPORTANT. A 14-1/2 DEGREE TOOTH GIVES LESS NOISE AND 25 DEGREE GREATER STRENGTH.

proved to be inherently quieter than involute gears. Additionally, manufacturers are generally tooled for involute types, which is one of the main reasons why other tooth forms have not been adopted.

Other forms have been designed specifically to carry much higher loads than equivalent involute types. These may be just as noisy, if not noisier, than conventional involute types. Extensive experience with involute gears has led to profile modification for quieter operation. This experience is not available for other tooth forms.

Pitch

For quiet, smooth operation, the finest possible pitch should be selected for the given load conditions. The finer the pitch, the higher the number of teeth in contact. This increases the amount of tooth overlap, both transverse and helical. The higher tooth overlap produces a smoother transfer of load, reducing dynamic oscillation of the gear mesh. This will result in a quieter gear train even tough the finer pitch will produce a higher mesh frequency.

Of course, the finer the pitch, the smaller the tooth size and the lower the strength rating of the gear set. To compensate for this, there are several choices: enlarge the pitch diameters, make the gears wider, or use the higher hardness material and precision-grind.

One drawback to the use of finer-pitch ground-tooth gears is that more time is needed to manufacture the teeth, which adds to the cost of the gearing.

Pressure Angle

Where a low noise level is a key requirement, the lowest possible pressure angle should be chosen. With the ever-increasing need for higher-capacity gears, many designers have tended to the 25° pressure angle in place of the more common 14 1/4° and 20° pressure angles (Fig. 1). The 25° pressure angle gears do indeed provide greater load capacity; but such gears tend to be noisier, because the contact angle and transverse overlap ratio are lower. In general, a 20°-pressure angle is a good compromise between quiet operation and high load capacity.

Recess Action

In cases where a gear drive will be transmitting load in only one direction of rotation, recess-action gears can provide a further reduction in noise. In gears of the "full-recess" type, the teeth of one of the gears in mesh have all-addendum profiles, while the mating gear has all-dedendum teeth (Fig. 2). In these gears, contact occurs during the recess portion of the tooth's engagement and disengagement cycle. Contact during the approach action, and some of the resulting detrimental scraping effects, are avoided.

The reason for the undesirable effects of approach-action contact can be understood by looking at a standard involute gear pair, as shown in Fig. 2a. Initial contact occurs at A when the lower gear is rotating counterclockwise. The approach action takes place from A to B, and recess action from B to C, when contact ceases.

During contact, the action is a combination of sliding and rolling. Approach-action friction tends to increase scuffing, wear, and noise. There is a change in the direction of sliding at the rolling point of the pitch line that tends to break down the oil film, causing increased friction. The recess phase of the meshing, on the other hand, is a "sliding-out" action. Friction is lower and in a direction to help rotating (Fig. 2b). Thus, the effect of recess action on the teeth is less severe than the approach action.

It is possible to design a semi-recess action system that combines the benefits of full recess action with those of standard action. Such gears are more common than the full-recess types.

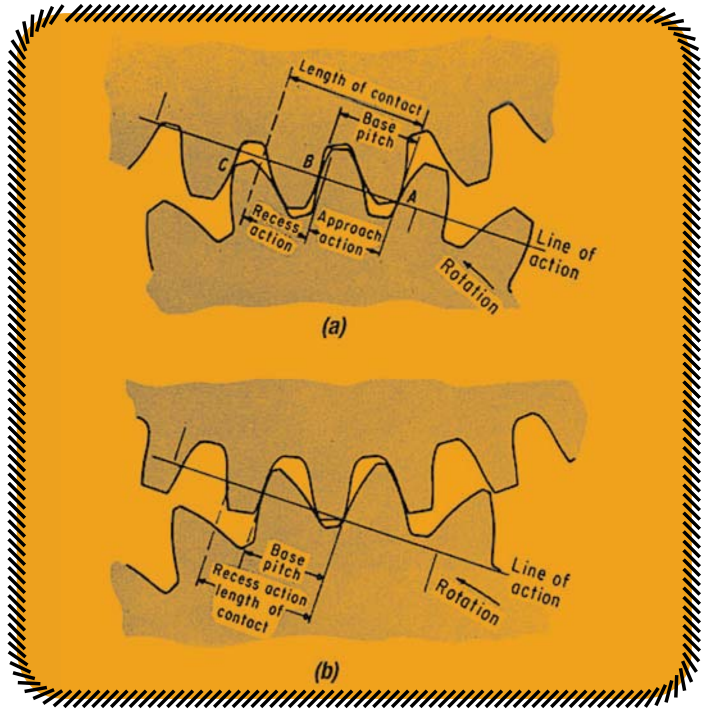


FIGURE 2: COMPARISON OF INVOLUTE GEARS (A) AND FULL-RECESS GEARS (B). IN FULL-RECESS GEAR SETS, THE TEETH OF ONE OF THE GEARS HAVE ALL-ADDENDUM PROFILES, WHILE THOSE OF THE MATING GEAR ARE ALL DEDENDUM.

Tooth Modification

Often overlooked, proper tooth modification is very important if the gear is to be quiet and long-lived. Dynamic studies have shown that most high-powered gears, as well as many lightly loaded ones, require modification of the tooth profile. This ensures smooth sliding of the teeth into and out of contact without knocking and compensates for misalignment, errors in manufacture, and deflections under load.

Tip and root relief are important. The tips of the pinion teeth are thinned slightly, beginning at a point halfway up the addendum (Fig. 3a). This modification prevents the tooth tip from striking one of the opposite teeth before it begins to pick up the load, or striking the trailing flank of the preceding teeth when rolling out of mesh. The root of the pinion is also thinned to allow for the tips of the gear teeth. In some cases, even the hydrodynamic effect of the lubricating oil can cause the teeth to emit a rapping sound during operation, even though there is no physical contact at or near the tips.

Crowning of the tooth flank is another extremely important tooth modification. Load deflections and slight errors in alignment of the gear housing in the bearing bores can cause hard contact on the flank ends of the tooth. It is therefore desirable to crown the gear (form a barrel shape, Fig. 3b) to center the contact area. As the load increases, a smooth "spreading" of the contact occurs until the entire flank is loaded.

Overlap Ratio

Designing can reduce gear noise at the mesh so that the total overlap ratio is an integral number of teeth. Tests have shown that if the ratio is exactly 2.0—so that exactly two teeth are always in contact—the smoothest transfer of load is obtained.

Overlap ratio expresses the average number of teeth in contact; in spur gears, it is obtained by dividing the length of the line of action by the normal pitch. With helical gears, the total overlap is the combination of the transverse overlap and the helical overlap.

As an example, assume that a gear pair has an overlap ratio of 1.5. At first, there will be two pairs of teeth in contact (Fig. 4a) until a point is reached (Fig. 4b), where the leading pair disengages at C. After disengagement, the trailing pair will be the only pair in contact until another pair engages. Thus, at some point, two teeth are in contact sharing the load, and at another point one tooth will assume the entire load. This sets up an oscillation about the deflection of the tooth that, although small, adds to sound generation.

With a higher overlap ratio, more teeth are in contact to share the load, providing increased capacity. An overlap ratio of 3.0 is often best for maximum load capacity. However, a value of 2.0 is generally best for reducing noise and dynamic loads.

Backlash

Adequate tooth clearance (backlash) must be provided for thermal and centrifugal expansion. A "tight" backlash is usually required only with reversing drives, where lost motion and impacts during reversing may be a problem.

The operating temperature of a steel gear or gear rim might be 100° F or more above ambient conditions; also, at high speeds, centrifugal forces can become considerable. Expansion results, and the "too-tight" gear set begins to howl. It is relatively simple to calculate fairly accurately the total anticipated expansion, and the resulting amount of backlash needed.

Tooth Loading

As shown in Fig. 5 the higher the tooth loading factor, K, the lower the total dynamic load and, consequently, the lower the amplitude of vibration and sound. Factor K is proportional to the tangential driving force, W, on the gears:

$$K = (W/F\ d)\ [(m_G + 1)/m_G]$$

where F = face width, d = pinion pitch diameter, and m_G = speed ratio

Quieter operation results from higher loadings because there are always some inaccuracies in gears, regardless of the quality level to which they are manufactured. With higher loadings, tooth deflections are greater and errors have less effect on uniform transmission of power through the gear set.

Quality Level

One of the easiest ways to reduce noise is to specify higher AGMA quality levels. As shown in Fig. 6, at higher quality numbers, the total dynamic load is reduced. Generally, AGMA Quality 12 or better is needed for smooth operation.

Of course, higher quality levels cost more. For example, going from Quality Number 8 to 12 means an increase in cost of 25 to 50 percent, mostly because of the need for special, high-accuracy finishing and grinding equipment and techniques.

Surface Finish

Surface finish should be as fine as possible. Several methods—shaving, for example—can be used to obtain a good finish. However, where a superior finish is needed, grinding is usually chosen.

Machining can cause periodic undulations on the gear teeth; these should be kept to less than 50 to 100μ in. Such undulations usually

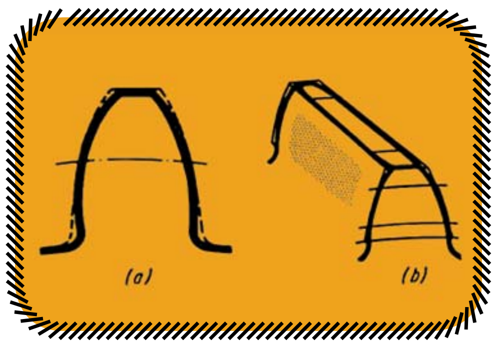


FIGURE 3: TOOTH MODIFICATION IS IMPORTANT IF A MESH IS TO BE QUIET. SHOWN HERE ARE TIP AND ROOT RELIEF (A) AND BARRELED TOOTH FLANK (B). IN EACH CASE, THE SOLID LINES REPRESENT THE MODIFIED FORM.

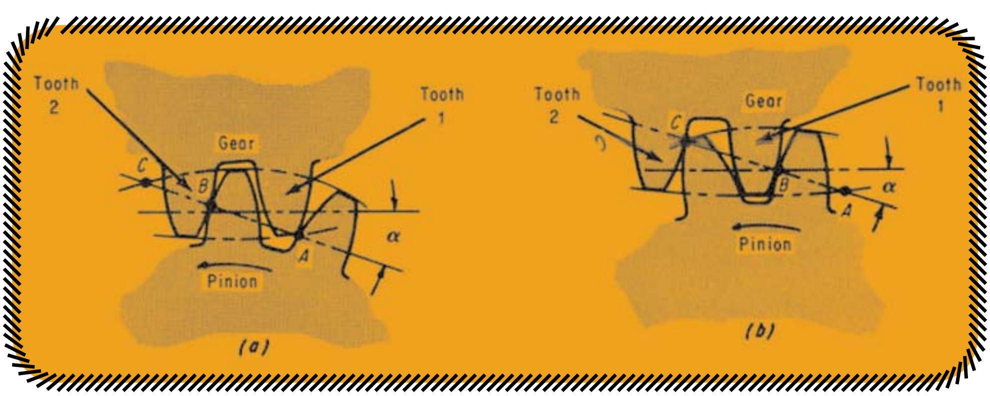


FIGURE 4: ACTION OF TEETH WITH CONTACT RATIO OF 1.5. IN (A), TWO TEETH ARE IN CONTACT, AT A AND B. HOWEVER, AS THE MESH TURNS (B) THE LEADING PAIR OF TEETH DISENGAGES AT C AND ONLY ONE SET IS IN CONTACT UNTIL THE NEXT SET ENGAGES. SUCH ACTION CAN SET UP OSCILLATIONS THAT CAN ADD TO NOISE.

result from cyclic runouts and inaccuracies in the master wheel of hobbing machines used to machine the gears. They can cause such odd phenomena as, say, a mesh noise frequency indicating 132 teeth when the bull gear actually has 215 teeth. Upon investigation, the 132 frequency will turn out to be the number of teeth in the master wheel on the hobbing machine.

Gear Runout and Imbalance

There is little sense in having Quality Number 14 gears, for example, with tooth tolerances held to within a few ten-thousandths of an inch, and then specifying two or three thousandths tolerances on the alignment and parallelism of the bores. All alignments and runouts should be in keeping with the quality level used.

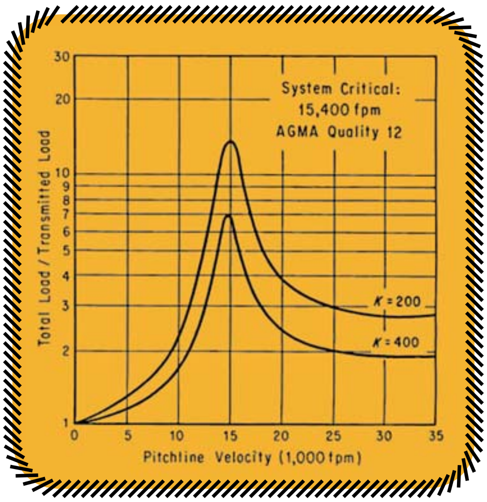


FIGURE 5: EFFECT OF TOOTH LOADING FACTOR, K, ON DYNAMIC LOAD. AS LOAD DECREASES, SO DOES NOISE.

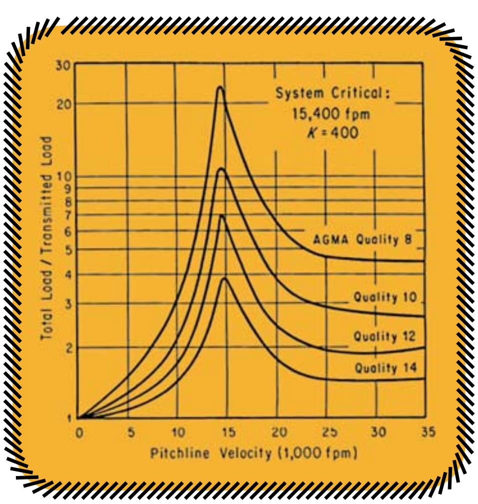
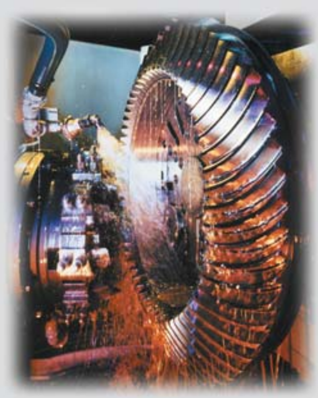


FIGURE 6: EFFECT OF QUALITY LEVEL ON TOTAL DYNAMIC NOISE. ALTHOUGH, MORE COSTLY, A HIGHER QUALITY LEVEL CAN OFFER A SIGNIFICANT REDUCTION IN NOISE.



For optimum quiet operation, the unbalance in the rotors should be limited to less than $U = 3W/N$

where U = rotor unbalance, oz-in.; W = weight of rotor, lb; N = speed, rpm

Gear Ratio

A non-integral gear ratio should be selected to prevent a tooth on the pinion from contacting periodically the same teeth on the mating gear. In most applications, an exact gear ratio is not necessary; by juggling the ratios slightly, it is possible to obtain an odd tooth, called the "hunting tooth."

As an example, a gear pair with 30 teeth on the pinion and 120 teeth on the gear has a speed reduction ratio of 4:1. A particular tooth on the pinion will contact teeth 1, 31, 61, 91, and continue to repeat the cycle, contacting the same four teeth. If a 30/121 pair is used instead, the reduction ratio becomes 4.03: 1, and the odd tooth on the 121-tooth gear will not contact the same tooth on the pinion every four cycles. For example, a tooth on the pinion will contact teeth 1, 31, 61, 91, 121, 30 60, 90, etc. and will not repeat the cycle until all teeth on the gear have been contacted.

Resonance

Any resonating member is a source of vibration and sound. There are two types of resonances to be concerned about. One is caused by the rotating parts and is associated with critical speeds; the second involves the support cases, foundation, and structures.

The critical speeds (resonances) of the rotating parts should be at least 20 percent from the operating speeds, from their multiples (harmonics) and from the mesh frequencies of the gear teeth. The farther the operating speed is from the critical speeds, the less chance there will be of detrimental effects.

Resonances of gear cases and other supporting members should also be 20 percent from operating speeds, multiples, and tooth-mesh frequencies. This may be difficult to obtain at times, so that a more practical minimum may be 10 to 15 percent, depending on how accurately the resonant frequencies can be calculated.

Lubricant Viscosity

The higher the viscosity of the gear lubricant, the greater the damping action at the mesh, and the quieter the operation. However, higher viscosities will result in some loss in operating efficiency and power. Most designers of high-speed gears choose a light grade of turbine oil (ISO 32 or ISO 46) because it is the same oil used in the turbine or compressor. But from the standpoint of the gear box itself, oils of ISO 68 or ISO 100 are better suited, provided that the power loss is not a problem. Besides quieting the gears, the oils with a higher viscosity will cut down gear tooth scoring and wear.

Type of Bearings

Both hydrodynamic (sleeve type) bearings and rolling element (anti-friction type) bearings can be used to support gear shafting. Hydrodynamic bearings, which operate on an oil film, provide for quieter operation. Hydrodynamic bearings typically have higher friction losses and require more lubrication oil than rolling element bearings. For standard applications, rolling element bearings are usually less expensive than hydrodynamic bearings. The particular application often dictates the choices of bearing types.

Material

The type of material chosen for the gear can have a significant effect on noise. Certain metals, such as the manganese-copper alloys, can dampen vibrations considerably. However, such alloys lose their damping characteristics at about 150° F. Unfortunately; many enclosed gears operate above this temperature in industrial applications. Non-metallics, which also have good damping characteristics, have relatively low load carrying capabilities, and are more difficult to manufacture accurately.

In general, alloy steels are still the best choice. A surface hardened precision tooth ground gear set today is the preferred tooth treatment method to maximize gear-set ratings. Superior ground tooth accuracy and fine tooth surface finishes made possible by current state of the art tooth grinding machinery also provide quieter operating characteristics.

Housing

Proper housing design can go a long way toward blocking the transmission of sound to the environment. Generally, a cast-iron housing will be "deader," or quieter, than a welded steel housing. However, welded steel has many advantages. It is stronger and more easily designed to meet custom requirements.

To improve the damping characteristics of welded steel housings; thin, flat plates should be avoided, because they can be excited by rotating components and mesh frequencies. Additional mass can be added, as well as stiffeners, and reinforcing ribs to break up plate-section resonances. If further noise damping is required, a layer of synthetic material such as felt, synthetic rubber, or polymer can be sandwiched between two steel plates to form a constrained layer of damping material.

To cut down the transmission of noise through the floor, the housing should be mounted on a vibration-isolating material placed directly under the gear box. A relatively stiff laminated rubber or other type of material should be used to absorb the higher frequencies.

ABOUT THE AUTHORS:

Bill Bradley developed this material when he was chief test engineer for the Philadelphia Gear Co. He is currently vice president of the AGMA's technical division [www.agma.org]. Richard Schunck is a staff engineer with Rexnord Industries, LLC [www.rexnord.com]. This material was compiled by Jules DeBaecke, vice president of engineering at Philadelphia Gear, and Sudhakar Rao, gear analyst. Go online to [www.philagear.com].