

Introduction to Worm Gearing

Worm gears are among the oldest types of gearing, but that does not mean they are obsolete, antiquated technology. The main reasons for the bad experiences some engineers have with worm gearing are misapplication and misuse. No form of gearing works for every application. Strengths and weaknesses versus the application must be weighed to decide which form of gearing to use. For proper application and operation of worm gears, certain areas that may differ from other types of gearing need to be addressed.

The Basics

Worm gear reducers are quiet, compact, and can have large reduction ratios in a single stage. The ideal ratio range for worm gearing is 5:1 to 75:1. This is the general range for most catalog reducers. Ratios of 3:1 to 120:1 are practical and have applications that are very successful. For ratios below 3:1, worm gearing is not a practical solution for most applications, and other forms of gearing should be considered. Worm gearing for ratios above the ranges mentioned are generally more practical as part of a multistage reduction.

In service, worm gears survive large overloads and high shocks. When properly applied, worm gearing can offer excellent performance and cost savings. Worm gearing has an inherent 200% overload (i.e., 3x rating) capacity in its rating. Other forms of gearing do not have this built-in service factor. Therefore a lower service factor can be used when sizing a worm gear set.

Explanation of Hand

The purpose of left-and right-hand gearing is to change the relative rotation of the worm to the gear (Figure 2). Hand refers to the direction of axial thread movement as the worm is rotated. If you point your thumb in the direction of axial movement and curl your fingers in the direction of rotation, the hand that corresponds to the worm is the hand of the gear set (Figure 1). Bolts are a simple example. Normally they are right-handed, and experimenting with a nut and bolt will help to clarify this description.

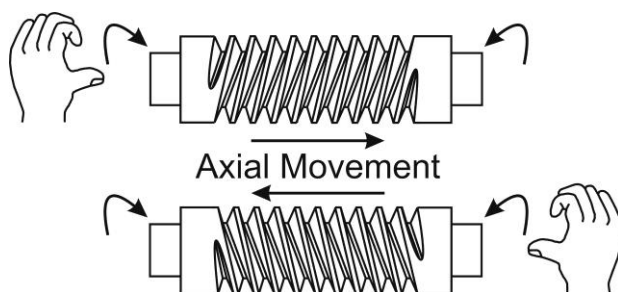


Figure 1 Hand of Worm

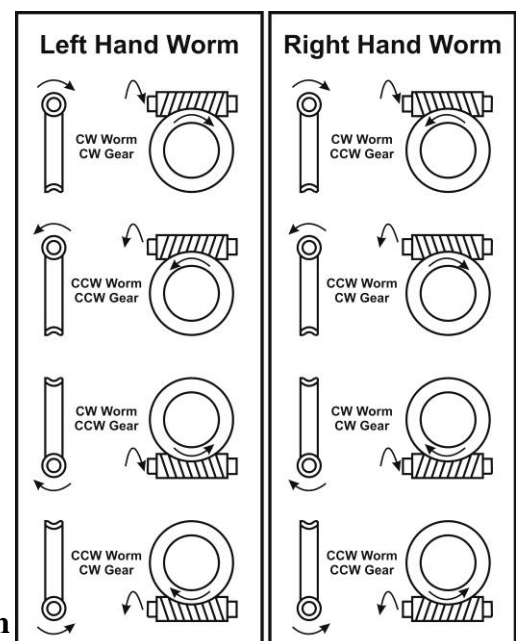


Figure 2 Relative Rotation

Right-hand gear sets, like bolts, are the industry standard. More right-hand gear ratios are available as standard items, and most manufacturers will supply right-hand gearing unless

otherwise specified. This does not mean there is a flaw in left-hand gearing, but left-hand ratios may not be as readily available.

Back Driving

Running a worm gear set with the gear (worm wheel) as the input member is commonly called *back driving*. Back drive efficiency of a worm gear set is lower than its forward drive efficiency. By varying design, the back drive efficiency can be reduced to zero, as in a *self-locking* or *irreversible* gear set. If the gear tries to drive the worm, internal friction causes the mesh to lock. No matter how much torque is applied to the gear shaft, mesh friction increases proportionally, preventing rotation. This is the same principle that keeps a nut and bolt from unscrewing under an applied tension load.

Back driving can occur in many applications. A worm gear speed increaser is the most obvious, but it is rarely used because of its low efficiency. It also occurs in lifting applications, such as cranes, hoists, and crank arms. When lowering the load, the gear is the input member. Worm rotation controls the rate of descent. Also, during braking or coast-down, the momentum of a device will back drive a worm.

A self-locking worm gear can be designed by making the lead angle less than the *friction angle*, which is defined as the arc tangent of the coefficient of friction. The static coefficient of friction is .20 to .15, equating to a friction angle of 11.3° to 8.5°. Vibration in a non-rotating gear set can induce motion in the tooth contact. The mesh velocity is zero, but the tooth contact is dynamic. At a mesh velocity of zero, the theoretical dynamic coefficient of friction is .124, or a friction angle of 7.0°. To provide a safety factor, a 5.0° lead angle is recommended as the upper limit of self-locking, and a 15.0° lead angle is recommended as the lower limit to assume a worm gear will back drive.

Contrary to popular belief a self-locking worm gear set once in motion will coast due to dynamic effects. Back drive efficiency decreases with decreasing speed. The slope of this curve is exponential and is affected by the lead angle (Figure 3).

This factor should be considered when sizing a brake and its rate of application. Often a brake placed on the worm can be smaller than normally anticipated. Using a brake on self-locking designs must be thoroughly analyzed. Most brakes have an increasing torque rate when applied. Also, the efficiency will be decreasing during slow down. This double effect can cause the

$$[\text{eq. 1}] \quad \text{Efficiency} \approx \frac{\tan(\lambda - \eta)}{\tan(\lambda)}$$

$$[\text{eq. 2}] \quad \eta = \tan^{-1}(\mu)$$

$$[\text{eq. 3a}] \quad \text{for } v \leq 10$$

$$\mu = 0.124e^{-0.074v^{0.645}}$$

$$[\text{eq. 3b}] \quad \text{for } v > 10$$

$$\mu = 0.103e^{-0.110v^{0.450}} + 0.012$$

$$[\text{eq. 4}] \quad v = \frac{\pi \times D_p \times \omega}{12 \cos(\lambda)}$$

λ = Lead Angle

η = Friction Angle

μ = Coefficient of Friction

e = Natural Log (i.e. 2.7183)

v = Mesh Rubbing Velocity (ft/min)

D_p = Pitch Diameter of Worm (inch)

ω = Rotational Speed of Worm (rpm)

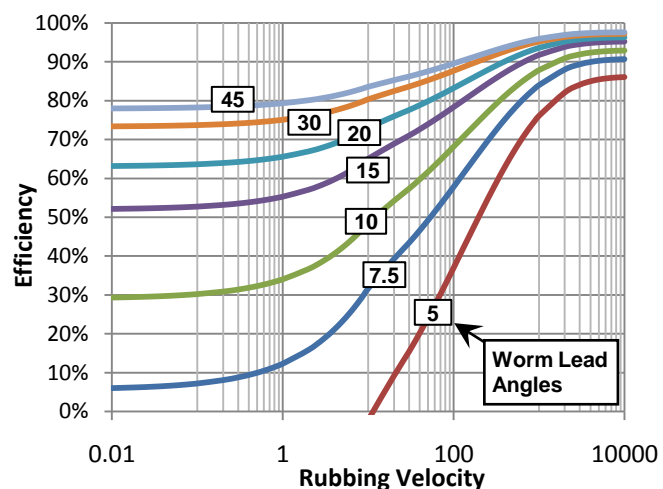


Figure 3 Efficiency versus Rubbing Velocity

braking torque to rise at a surprising rate, causing a sudden stop. High inertial loads with self-locking designs should have controlled motor speed ramp down for braking.

On the other hand, back drive efficiency increases with increasing speed. Therefore a constant back driving torque restrained only by a worm gear will have a rate of acceleration that increases exponentially. This is a very important point to remember when designing hoists. Unless it is properly designed, relying solely on self-locking to suspend a load may be dangerous. The load may stay suspended until an outside influence starts a vibration in the gear mesh. Initially, the load will creep slowly. As it falls, it accelerates at an exponentially increasing rate.

Since many factors influence the coefficient of friction, gear set designs should be tested for their back drive suitability. Break-in of a gear set will reduce the coefficient of friction. This may make a gear set self-locking when it is new and not self-locking after use. Also, synthetic lubricants can have an effect on the coefficient of friction and may be used in the field without the knowledge of the gear designer.

If self-locking is critical to safety, a brake or “*back stop*” should be used. A backstop is a clutch device that permits rotation in one direction only. It is sometimes referred to as a “Sprag” or “cam clutch” and is commonly used on conveyors that lift material to prevent reversal if power is lost.

Pitting

Gear tooth-pitting results from the combination of several forces. Normal force (referring to a direction 90° to the tooth surface) at the contact point produces Hertzian stress. Friction produces a tangential force, also inducing stress. Friction also generates heat. Temperature at the contact point is much higher than the surrounding area. Differential thermal expansion (the phenomenon that can cause a glass to break when a hot liquid is poured into it) is the third factor inducing stress.

Constant cycling as the tooth goes through the mesh can cause a surface fatigue crack. Oil in the gear mesh is under extreme pressure from contact forces. The oil is forced into the fatigue crack, and hydrostatic pressure tries to lift a piece out. Continuing cycles cause the crack to encircle the high stress area. The crack grows deeper, until a piece literally pops out, leaving a pit.

In most gearing, a pitted tooth surface signals impending failure. For worm gearing, pitting is part of normal operation. Corrective pitting is a break-in process. In manufacturing a worm, the thread is generated by a continuous line that can be described by the grinding wheel. It produces a continuous (i.e., smooth and uniform) surface curved in all three planes. The gear is hobbled by a gashed cutting tool that is in effect a worm having a discontinuous or interrupted surface. It produces gear teeth that have a series of short flats or discontinuous surfaces that approximate the desired tooth form. Because of the flats, the gear tooth form is imperfect. Where two flats join there must be a peak. At such a point, the contact stress would be infinite if deformation did not occur.

The break-in process is the gear tooth form being improved by the worm. This is done by elastic or inelastic deformation, wearing away the high spots or pitting them away. After the many high spots are either worn or pitted away. The worm rides on the larger flat areas. The pit areas retain pools of oil, which help support the load by hydrostatic pressure and aid in lubrication. Corrective pitting ceases after a sufficient area has been developed to sustain the load and normal wear takes over. A new worm gear will pit at an alarming rate, and then quickly stop. No additional pitting will occur for a long time. Then the surface will again pit rapidly and quickly stop, the cycle recurring throughout the life of the gear.

Destructive pitting is a case of the gear not being able to correct itself enough to support the imposed load. It is the result of overload, improper gear adjustment, improper tooth profile, or improper lubrication. In this case, pitting continues until the gear tooth surfaces are completely destroyed. This is not a common problem, because most errors large enough to cause failure will normally show up as the gearbox overheating.

Materials

Worm gearing has a high sliding component in its tooth meshing action. Sliding contact materials are selected to make one member hard and strong and the other soft and ductile. Friction is generally proportional to the combined hardness of the mating surfaces. Two hard surfaces cannot deform to broaden the contact area and distribute the contact stress. By hardening only one part and having the other ductile, the combined hardness is increased, while still being able to distribute stress. Also, using dissimilar hardness reduces the chance for galling. Steel and bronze have been the materials of choice because they balance strength, ductility, lubricity, and heat dissipation. Shaft bushings are common examples of sliding components using this arrangement.

The worm is the hard member, and the gear is the ductile member. There are several reasons for this arrangement. Contact stress in both members is equal. The worm goes through more contact cycles because of the ratio of the gear set. Compared to steel, bronze has a lower strength, a lower endurance ratio, and a higher number of cycles required for infinite life. Figure 4 uses these factors in a generalized, theoretical S-N curve. Stress levels that have a finite life for bronze would have an infinite life for steel. Since the bronze will fail at a fewer cycles, it is used for the member requiring the fewest cycles.

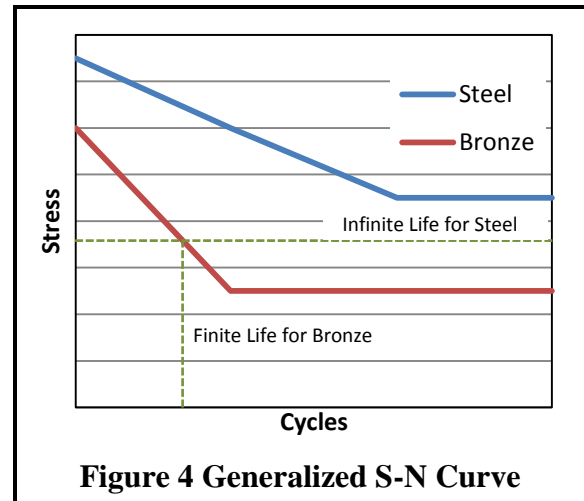
Gear mesh reaction forces are equal and opposite in both members. The worm is much smaller in diameter than the gear and has a greater span between supports. Therefore bending stress is greater in the worm, requiring it to be made from the stronger material.

Manufacturing methods also play a part in material choice. Grinding is generally used for accurate finish of high-hardness, heat-treated steels. Grinding the worm is a simple process, using the flank of a straight-sided grinding wheel. Grinding the gear requires a complicated process using a form dressed grinding wheel and a three-axis grinder.

Tin bronze has proved to be the most successful alloy for worm gears. It has a low coefficient of friction and a low rate of wear. Good heat conduction carries away heat generated in the mesh and dissipates it throughout the gear. Aluminum bronzes have higher strength, but also a higher coefficient of friction. A less obvious disadvantage of the higher strength alloys is lower ductility. Theoretical contact between a worm and gear is a line. In practice, the bronze deflects under load broadening the contact line to an area. The material deflects until the contact area broadens enough to support the load. A low-ductility material may have localized failure before reaching a large enough area. Small contact areas of a lower ductility material have higher localized contact temperatures, which further increase the sub-surface stresses.

The unique properties of tin bronze can be traced to its grain structure. When the bronze solidifies partial segregation of the copper and tin occurs. High tin areas or grains are commonly called the delta phase. Hardness of the delta phase is approximately 320 Brinell. The high copper matrix supporting it is approximately 145 Brinell. The hard grains provide wear resistance and help reduce friction. The softer matrix allows surface deformation to distribute stress. A simple model would be to picture marbles (delta phase) imbedded in clay (matrix).

Alternate gear materials may increase certain properties, but losses in others will tend to make them unsuitable for general use. For special applications bronze alloys other than tin bronze may perform better. Gear materials, such as cast iron, plastic, and even steel, have worked very well in



certain applications each application must be thoroughly analyzed by a gear engineer before selecting alternate materials.

Worms are generally made from an alloy steel. Steel worms can be divided into hardened and non-hardened. Hardened worms are superior in most applications. When surface hardness of approximately 58R_c is used, several benefits are gained. Material strength is increased, friction is lowered, and wear is reduced. Often a worm can be reused after the gear has worn out.

Non-hardened refers to the surface being lower than the typical 58 R_c. Non-hardened worms may actually have a heat treatment to bring up the core hardness for increased strength. In industrial applications, a core hardness of 300 Brinell is typical. Non-hardened worms are useful in applications with low continuous power and very high peak or shock loads. These applications are most often machine adjustments or mechanisms that are infrequently activated. Heat treating for increased surface hardness may be eliminated in low power applications to decrease cost. If a worm is used with a cast iron or steel gear, it should be non-hardened.

Backlash Measurement

Backlash is the measure of the free clearance between the worm and the gear teeth. Locking the worm against rotation, setting a dial indicator on a gear tooth at the pitch radius, and rocking the gear back and forth does measurement. The total indicator reading is the measurement of backlash. Locking the gear and measuring worm rotation does not measure backlash. In an assembled unit where the gear teeth are not readily accessible, backlash can be approximated by placing the indicator on any convenient point that is fixed to the gear, such as a shaft keyway or coupling. This measurement must be multiplied by the ratio of the gear pitch radius to the measurement radius. Note that if the selected point is on the radius smaller than the pitch radius a multiplication of measurement error will occur.

$$\text{Backlash} = \text{Measurement} \times \frac{\text{Gear Pitch Radius}}{\text{Measurement Radius}}$$

Lubrication

Worm gearing has a high slide-to-roll ratio when compared to other types of gearing. Because of a high sliding component, it relies heavily on the generation of an oil film between the worm and gear. The oil film produces an effect similar to what happens when a speeding car hits a rain puddle. The car tire has a tendency to float on a wedge of water. In a car this is called hydroplaning; in gears it is called elasto-hydrodynamic lubrication (EHL). This is a simplistic description with other modes of lubrication coming into play, depending on conditions, but it gives the general idea.

For EHL to be the only lubrication mode, it must generate a film thickness greater than the surface roughness of the contacting parts. Film thickness is proportional to the sliding velocity and lubricant viscosity and inversely proportional to the unit load. High unit loads possible at the relatively low speed of worm gearing requires a very high viscosity lubricant. Viscosities of over 400 cSt at 40°C are normally used to prevent premature wear and high contact temperatures. Under high loads the film can collapse, causing the surfaces to contact. This is called “boundary lubrication.” In this lubrication mode, other properties (i.e., lubricity or slipperiness) of the lubricant become more important than the viscosity. In a worm gear set, a mixture of EHL and boundary lubrication is at work.

A satisfactory lubricant for most average applications is an AGMA 7 compounded oil. Low speeds require the higher viscosity of AGMA 8 compounded oil. Both are petroleum-based mineral oils compounded with 3% to 10% fatty oils. These lubricants are sometimes referred to as steam cylinder oils. The compounded oil provides lower friction and better wear characteristics than a

straight mineral oil. At the high pressures and temperatures in the contact area, a chemical reaction occurs on the tooth surface, forming a protective skin.

Extreme pressure oils (EP oils) are another type of lubricant that uses a surface acting chemistry. Most EP oils use sulfur, phosphorus, and/or chlorine additives, and are designed to work in steel-on-steel applications. When these oils are used with bronze under high temperature and pressure, conditions common in the mesh contact, the chemical reaction can go awry. The surface of the bronze can begin to flake off, causing massive wear, and inter-granular stress corrosion can cause the teeth to break. There are EP oils designed for use with bronze that use a different additive package, and in certain applications a standard EP oil may work very well. When selecting an EP oil for bronze gearing make sure it was carefully reviewed.

Synthetic lubricants are also very common. They are more viscosity-temperature stable than mineral oils. This allows one lubricant to provide adequate service over a broader temperature range. They have longer service life, reducing the number of oil changes required. They reduce wear and friction, increasing gearbox life. Efficiency increases of 20% of the lost power are possible. Under severe conditions properly selected synthetic oils are outstanding. Many companies have found cost advantages using the more expensive synthetic oil for normal applications.

Contact Pattern

The area of contact between the worm and the gear is the *contact pattern*. The ideal contact pattern uses 90% of the full face with the remaining 10% open on the side where the worm enters contact with the gear. This gives maximum contact while allowing oil to be dragged in for lubrication. If the entering side has contact the oil will be wiped off the worm as it rotates into mesh. Without oil being drawn in the gear set will not generate an oil film. This will cause overheating and rapidly fail (see [Lubrication](#) section).

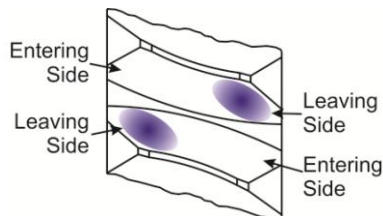


Figure 5 No Load Pattern

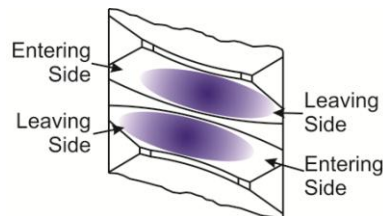


Figure 6 Full Load Pattern

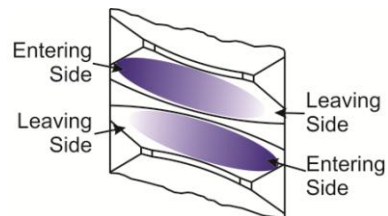


Figure 7 Overload Pattern

Under load the gearbox, worm and gear will deflect. These deflections cause the contact pattern to spread across the gear face toward the entering side. To compensate for contact pattern spread the gear can be moved axially in relation to the worm. A no-load contact pattern of approximately 30% of full face biased toward the leaving side is desirable.

Since deflections occur in opposite directions for opposite rotations the profiles of two opposing flanks of the gear are not directly in line. They are shifted axially with respect to each other when the gear is cut. Axial movement of the gear from 90% full face to 90% full face on the opposing flank is the *total shift*. Total shift anticipates deflections that will occur from full load forward to full load reverse.

The no load contact pattern is determined by lightly coating the worm with high spot blue (i.e. Prussian Blue). This transfers to the gear teeth when rotated by hand. Although not required, coating the gear with a mixture of grease and powdered orange paint pigment improves contrast with the blue making it easier to see. It also adds some lubrication when checking very large gear sets that may be difficult to rotate by hand. To determine the full load pattern clean all oil from the gear teeth and paint them with layout blue. Under load the paint will quickly wear off. Do not run

the gearbox too long before checking the loaded pattern. Lubricating oil tends to wash off the layout blue, especially when the oil warms up in operation.

In severe applications the fully loaded pattern should be checked. The amount of shift cut into the gear may not be enough to compensate for overly flexible housings or loads higher than anticipated. For one rotation applications or where loading is always on one side (ex. hoists) shift is not of major concern as the gear and typically be shifted to favor one flank. Take care that the worm does not lose backlash and contact both flanks under no load. That will cause excessive heat and cause premature failure.

In most reducer designs axial gear adjustment is accomplished by shims between the gear shaft bearing caps and the housing (Figure 8). First determine the total shims for proper bearing end play. Then transfer shims between the bearing caps until optimum no load gear pattern is achieved. In heavy duty applications where it is desirable to set the shaft based on the fully loaded pattern this would be very complex. An alternate design is to have the bearings opposite the output shaft in a carrier that can be moved without accessing the drive side (Figure 9).

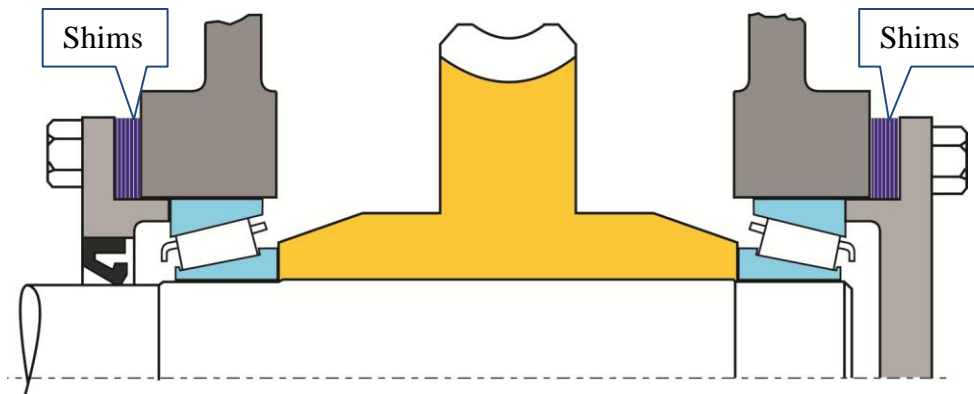


Figure 8 Shift Adjustment at Both Sides

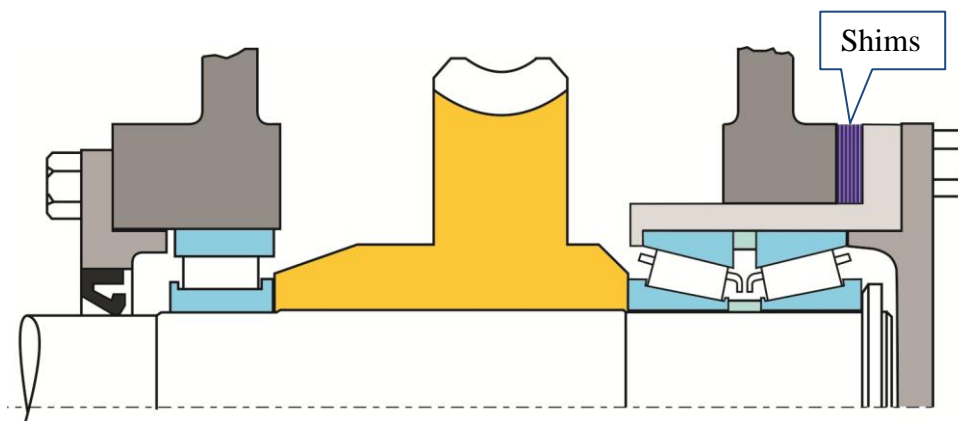


Figure 9 Shift Adjustment at One Side

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James K. Simonelli is a Licensed Professional Engineer with 30 years experience designing and troubleshooting machine automation, heavy duty equipment and industrial products. He has a broad background with department head roles in engineering, quality and business development in companies varying from startups, turnarounds to Fortune 100 corporations. Mr. Simonelli has served on committees developing industrial standards for the American Gear Manufacturers Association and the Hydraulics Institute.

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