AGENDA OF THE CEMA ENGINEERING CONFERENCE
CONVEYOR PULLEY COMMITTEE MEETING
Tuesday, June 26, 2012 – 8:30 AM

1. Call to order

2. Attendance and introductions – any new committee members?

3. Review and approval of previous minutes (attached)

4. Old Business
   a. Update of 501, “Welded Steel Wing Pulleys.” We will work on this after we complete the changes for Chapter 8 of the Belt Book.
   b. An Engineered Pulley “Best Practices” Manual. Should this be added to the Engineered pulley section of Chapter 8?
      i. Need copies of the DIN and ISO standards.
      i. Stress concentration for keyless locking assemblies
      ii. Review first draft from Todd Swinderman.
      iii. Review draft of new Bearing Section
   d. Continue discussion on lagging recommendations for ultra-high tension applications.

5. New Business
   a. Election of new Chairman
   b. Election of new Vice-Chairman

6. Next Meeting

7. Adjourn

Attachments:

   EC 2011 – Conveyor Pulley Committee Meeting Minutes
   Chapter 8 First Draft
   Draft of section on Bearings
MINUTES OF THE CEMA ENGINEERING CONFERENCE
CONVEYOR PULLEY COMMITTEE MEETING
Tuesday, June 28, 2010

1. The meeting was called to order at 10:45 am.

2. Roll Call – Thirty-three (33) attendees from twenty-seven (27) companies were present.

3. The previous minutes were approved.

4. Old Business
   a. Update of 501, “Welded Steel Wing Pulleys.” We will work on this after we complete the changes for Chapter 8 of the Belt Book.
      i. Instead of creating a Best Practices we will add this to the Engineered pulley section of Chapter 8
      ii. Anyone that has information on the DIN and ISO standards for pulleys is requested to send it to David Keech.
      i. Reviewed and approved the changes made by the subcommittee for metrification.
      ii. It was proposed and approved that we add a section on bearings for conveyor pulleys.

5. New Business
   a. Discussed determining the stress concentration for keyless locking assemblies. It was proposed that we add it as an additional factor under the $k_f$ in the shaft stress equation. Fenner Drives agreed to work on this.
   b. It was requested that we review the wording on Drum and Wing Pulley standards about the use in steel cable belt applications.
   c. There was discussion about lagging in ultra-high tension applications. Conveyor belting is now available up to 10,000 PIW. There are no recommendations on lagging for these applications. Some of the concerns are the compressive pressures and the drive pulley lagging’s ability to transmit the torque. This is multi discipline problem that will probably require some testing. The consensus is that this needs to go to the ORs for them to review and provide direction to the Engineering Conference.

6. The next meeting of the CEMA Pulley Committee will be Tuesday June 26, 2012 at the LaPlaya in Naples, FL.

7. The meeting was adjourned at 11:45AM.

Submitted by

David Keech, Pulley Committee Chairman

Attachment: Roll Call

THE VOICE OF THE NORTH AMERICAN CONVEYOR INDUSTRY
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THE VOICE OF THE NORTH AMERICAN CONVEYOR INDUSTRY
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   Standard Steel Wing Pulleys
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INTRODUCTION

Accepted engineering practice is to consider pulleys and shafts together because they form a composite structure whose operating characteristics are mutually related. Therefore, they are discussed as one topic of belt conveyor design and construction in this chapter.

CONVEYOR PULLEYS

Conveyor pulley construction has progressed from fabricated wood, through cast iron, to present welded steel fabrication. Increased use of belt conveyors has led industry away from custom-made pulleys to the development of Standard Steel Pulleys with universally accepted size ranges, construction similarities, and substantially uniform load-carrying capacity for use with belts having a carcass composed of plies or layers of fabric. “Standard” Drum and Wing Pulleys are suitable for these applications. The present trend, however, is to use higher tonnage conveyor systems with wider, stronger belts that incorporate a carcass of either steel cables or high-strength tensile members. In these applications, where high tensions are encountered, the use of custom-made “Engineered” welded steel pulleys is dictated. See Chapter 7 for a description of the various types of conveyor belts.

Pulley Types

The most commonly used conveyor pulley is the Standard Steel Pulley as shown in Figure 8.1. They are manufactured in a wide range of sizes and consist of a continuous rim and two end discs fitted with compression type hubs. In most wide-faced conveyor pulleys, intermediate stiffening discs are welded inside the rim. Other pulleys available are self-cleaning wing types, which are used at the tail, take-up or elevator boot locations where material tends to build up on the pulley face. Figures 8.1 through 8.6 illustrate the more common types of conveyor pulleys now in use.

Standard Steel Drum Pulleys

Standard welded steel drum pulleys are defined by CEMA B105.1. The standard establishes load ratings, allowable variation from nominal dimensions, permissible crown dimensions, and overall dimensions normally necessary to establish clearances for location of adjacent parts. The standard applies to a series of straight and crown-faced welded steel conveyor pulleys that have a continuous rim and two end discs, each with a compression type hub to provide a clamp fit on the shaft.

The tabulated ratings for drum pulleys and shaft combinations are based on using non-journaled shafting with pulleys centrally located between two bearings. High strength shafting may be required with drive pulleys to withstand the added shaft stresses from torsional loads, overhung loads or turndowns for bearings. Belt tension limits are also provided and must be checked, especially for pulleys with low arc of contacts such as snubs or bends.

CEMA B105.1 is not applicable to single disc pulleys, wing pulleys, cast pulleys, or pulleys not utilizing compression hubs. The standard is not intended to specify construction details other than those listed above. The standard covers pulleys up to 60 inch diameter for shaft diameters up to 10 inches and face widths up to 66 inches and therefore encompasses the majority of combinations of welded steel pulleys with compression type hubs that are normally used in current belt conveyor and elevator practice. It is not recommended to operate Standard Drum Pulleys above a belt speed of 800 feet per minute. For higher speeds, manufacturers should be consulted.

Welded steel conveyor pulleys covered by CEMA B105.1 should not be used with steel cable or other high
modulus conveyor belts because such belts create tension concentrations and demand manufacturing tolerances beyond the capacities of these pulleys. Such conveyor belts require Engineered conveyor pulleys.

**Standard Steel Wing Pulleys**

Standard welded steel wing pulleys are defined by CEMA 501.1. Like the drum pulley standard, this standard establishes load ratings, allowable variations from nominal dimensions, permissible crown dimensions, and overall dimensions normally necessary to establish clearances for location of adjacent parts. The standard covers pulleys up to 36 inches diameter for shaft diameters up to 8 inches and face widths up to 66 inches and therefore encompasses the majority of combinations of welded steel wing pulleys with compression type hubs that are normally used in current belt conveyor and elevator practice.

The standard applies to a series of straight and crown-faced welded steel wing pulleys that have a number of steel wing plates that extend radially from the longitudinal axis of two compression hub assemblies and are equally spaced about the pulley circumference. The purpose of the compression hubs is to provide a clamp fit on the shaft. The wings are supported or joined by welded steel plates so arranged as to form the shape of two frustums of cones or regular pyramids joined at their bases. A contact bar is attached to the outer longitudinal edge of each wing to provide contact area with the belt.

The tabulated ratings for wing pulleys and shaft combinations are based on using non-journaled shafting through the pulley hubs, with pulleys centrally located between two bearings. Belt tension limits are also
PULLEYS AND SHAFTS

provided.

Welded steel wing pulleys covered by CEMA B501.1 should not be used with steel cable or other high modulus conveyor belts because of the eccentricity inherent in the construction of a wing pulley. It is not recommended to operate Standard Wing Pulleys above a belt speed of 450 feet per minute. For higher speeds, manufacturers should be consulted. Wing pulleys are not designed to be used in locations that transmit torque.

The use of wing pulleys should be limited because they will shorten the life of the belt and the mechanical belt splice. Wing pulleys should only be used after all other means of keeping material from getting between the belt and the pulley have been tried and failed. This would include skirt boards, belt cleaners, belt plows, and belt scrapers. Then wing pulleys should only be used for take-ups, tails and elevator boots.

Advantages Of Using CEMA Standards

The Standard Drum and Wing Pulleys discussed above can be used in most conveyor applications. The designer of belt conveyors will find the CEMA standards listed above invaluable in determining specifications for pulleys and shafting and in finding information that will permit the framework and supporting bearings to be detailed into the design. Suitable pulleys conforming to these standards can be obtained readily from the principal pulley manufacturers. Many of the sizes are considered stock sizes and are in their inventory of finished goods.

Mine Duty Pulleys

Standard size drum and wing pulleys are available in mine duty construction. Typically, a mine duty pulley is one whose material thicknesses have been increased for a rigid, conservative design. Mine duty pulleys were originally specified and used for underground mining operations where the abusive environment and high cost of installation demanded a more conservative design. Mine duty pulleys can be appropriate for conveyors with frequent starting and stopping, overloads exceeding 150% of running tensions, or where increased reliability is necessary.

Mine Duty Pulleys can be considered in a conveyor application requiring heavier construction and more conservative design to give greater service life where abrasion is a factor; or there are longer running conveyor hours to consider. Mine duty pulleys are pre-engineered, not to a specific application or for a particular purpose but will have lower stress and deflection on the various components and offer greater
service factors over standard CEMA rated pulleys. These increased ratings can be achieved by design and manufacturing considerations including heavier rim and end disc material thicknesses, increased rigidity of shafts and end discs and use of manufacturing processes that increase the endurance strength of the pulley. No ANSI standard governs the load ratings or material thicknesses of mine duty pulleys. Each pulley manufacturer should be contacted for specific details on their mine duty pulley design and manufacturing process.

**Engineered Pulleys**

Engineered Pulleys are specifically designed to meet load conditions of a particular conveyor. Specific information is required for proper and economical design, since the designer must allow for sufficient strength in the rim, end disc, shaft, and mounting system to carry the belt loads and to assure proper pulley to shaft connection.

Common reasons for using Engineered Pulleys are:

- Belt tensions and resultant loads exceed CEMA standards B105.1 and B501.1
- Pulley diameter, face width, and shaft diameter combination falls out of size ranges defined in CEMA standards B105.1 and B501.1
- Conveyor belt uses steel cable, steel mesh, aramid, or other high modulus carcass
- A desire to control project costs by optimizing pulleys for their intended use
- Conveyors operate at high speeds with nearly continuous service
- Transient belt tensions greater than 50% more than steady state running tensions

Typical Engineered Pulley applications are overland conveyors, high tonnage mining conveyors, large coal power plants, ore processing facilities, and large capital projects.

High modulus belts are defined as those having operating tension ratings greater than 800 PIW (XX kN/m) or a modulus greater than 80,000 PIW (XX kN/m). These belts require pulley design considerations and dimensional tolerances exceeding CEMA standards. Startup, braking, and other dynamic loads may be more directly transmitted to the pulleys. Reduced belt stretch requires straight faced pulleys with rim and lagging concentricity improved to 0.030 inches (0.76 mm). Special emphasis should be placed on aligning structural supports and pulleys with the beltline to prevent damage from load concentrations. See Chapter 7 and contact your belt and pulley manufacturers for more information on high modulus belts.

Construction details vary, ranging from flexible end discs and compression type hubs with plain shafting to heavy rigid end discs fitted to specially machined shafts. Special manufacturing specifications are common and often include accurately machined surfaces, stress relieving, special welding controls, and non-destructive testing.

System designers may specify special design requirements, some of the most common being stress limits, deflection limits, materials, and construction techniques. CEMA recommends using 0.0015 in./in. (0.0015 mm/mm), i.e. 5 minutes, shaft deflection limit for Engineered Pulleys used in critical applications, which results in shafts consistent with components often used on high tension pulleys. Specifying deflection limits less than CEMA standard 0.0023 in/in (0.0023 mm/mm) for Standard Wing and Drum Pulleys is not intended to imply the difference is a safety factor on running loads. CEMA recommends system designers apply appropriate safety factors to belt tensions communicated to assure they represent maximum steady state running conditions. Reference pulley overloads section of this chapter for more information.

All components making up the pulley and shaft assembly must be integrated to provide a fully capable power transmission system. Several factors must be considered to design reliable and economical conveyor
pulleys for the intended operating conditions, which require the following information:

- Diameter, face width, straight or crowned
- Bearing centers
- Location of pulley: head, bend, snub, take-up, etc.
- Type of belt take-up: gravity, screw, etc.
- Type of conveyor belt
- Transient belt tensions on pulley
- Worst case running belt tensions on pulley
- Belt wrap angle on pulley
- Shaft diameter (at hub, bearing, drive) if predetermined
- Lagging specifications
- Overhung load with location on drive pulley shaft (if it exists)
- Drive Power
- Belt speed
- Starting mechanism (reduced voltage, fluid coupling, etc.).
- Special environmental and operating conditions

**Pulley Overloads**

Excessive belt tensions may result in premature failure of pulleys, shafting, or bearings. Differentiating between transient tension increases and steady state running tensions is important for proper pulley design.

Transient, or dynamic, tension increases happen for a short period and then subside. These periods generally last for a few minutes or less and represent less than 1 percent of operating time. Some examples are starting, stopping, and jam-ups. Transient loads should not exceed design loads by more than 50 percent. If greater than 50 percent or more than 1 percent of running time, Engineered Pulleys are recommended and this information provided to your pulley manufacturer.

Steady state running tensions happen for a significant period of time and represent the fundamental operating conditions. Conditions that can increase running tensions are excessive belt misalignment, excessive material loaded, excessive take-up weight, gravity take-up frictional increases, and over tightening of screw take-ups. Normal running tensions for Standard Pulleys should not exceed ratings in the CEMA B105.1 and 501.1 load tables. Normal running tensions for Engineered Pulleys should not exceed those used for design.

**Conveyor Take-Up Discussion**

From the pulleys point of view take-up systems are the most important part of the conveyor. Take-up belt tensions are the primary load for the majority of pulleys on typical conveyors. A high percentage of pulley problems can be traced to poor communication of take-up conditions, changes to the take-up after commissioning, or poor take-up maintenance.

Screw take-up conveyor belt tensions are dependent upon the judgment of the person turning the wrench. In many cases, the take-up is capable of creating excessive belt tensions. Methods in Chapter 6 are designed to anticipate higher tensions, but over tightening the take-up can still result in excessive tensions. Tightening beyond what is needed to control drive slip and belt sag is not recommended.

Larger conveyors often use a gravity take-up. In theory, the hanging weight should eliminate unexpectedly high take-up belt tensions. Testing of actual systems suggest noticeable belt tension variation from design
values exists. Common reasons for this are:

- Loads such as the box holding the weight, attachment structure, and pulley assemblies get omitted in the take-up belt tension calculation
- Weight is added after the initial commissioning
- Sheave friction and wire rope bending forces get omitted (testing has shown wire rope tensions up to 40% higher at the pulley than at the weight box)
- Material spillage builds up on the take-up
- Structural damage or poor alignment causes restrictions to take-up travel.

**Abrasive Environments**

Drum pulley rims are usually manufactured using low carbon steel plate, pipe, or tubing. Wing pulley contact bars are usually manufactured using low carbon steel bars or plate. The rim and contact bars are not designed to be a wear item. For some belt conveyor applications, these materials provide sufficient pulley life. For abrasive conditions, the pulleys should be lagged to prevent wear. The lagging should be monitored for wear and replaced before it wears down to the steel. Another consideration for increasing pulley life is to specify thicker rims or contact bars. Wing pulley contact bars can also be manufactured from abrasion resistant (AR) steel to increase the life.

**Pulley Diameters**

Standard Steel Drum and Wing Pulley diameters and permissible diameter runouts are shown in Table 8.7. All other pulley diameters are considered special. These nominal diameters apply to straight and crown-face pulleys and are for bare pulleys only; they do not include any increases in diameter due to the application of lagging. The nominal diameter is measured at the midpoint of the pulley face width. Pulley diameters should be selected per the CEMA standards and the belt manufacturer’s recommendations, as described in Chapter 7, as well as for other drive or space considerations.

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<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<tr>
<td>16 (406)</td>
<td>0.250 (6.35)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<tr>
<td>18 (457)</td>
<td>0.188 (4.78)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<tr>
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<td>0.125 (3.18)</td>
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<tr>
<td>24 (610)</td>
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<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<tr>
<td>36 (914)</td>
<td>0.188 (4.78)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<td>42 (1067)</td>
<td>0.250 (6.35)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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</tr>
<tr>
<td>48 (1219)</td>
<td>0.188 (4.78)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
<td></td>
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<tr>
<td>54 (1372)</td>
<td>0.250 (6.35)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
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<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>60 (1524)</td>
<td>0.188 (4.78)</td>
<td>0.125 (3.18)</td>
<td>0.125 (3.18)</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

Table 8.7

Standard pulley diameters and maximum permissible total indicator reading runout for common applications

Permissible variations from nominal diameters of Standard Steel Pulleys are based on face width and given in Table 8.8. All other face widths are considered special.

<table>
<thead>
<tr>
<th>Standard Pulley Face Widths in (mm)</th>
<th>Over Nominal Diameter</th>
<th>Under Nominal Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Drum</td>
<td>Wing</td>
</tr>
<tr>
<td>12 (305) up to 26 (660)</td>
<td>0.250 (6.35)</td>
<td>0.125 (3.18)</td>
</tr>
<tr>
<td>26 (660) through 66 (1676)</td>
<td>0.625 (15.88)</td>
<td>0.125 (3.18)</td>
</tr>
</tbody>
</table>

Table 8.8

Standard Pulley permissible diameter variations based on face width
These limitations apply equally to straight face and crown-face pulleys. The nominal diameter is measured at the midpoint of the pulley face width. The diameter is defined as the bare diameter exclusive of lagging. Listed variations may occur from one pulley to another. The permissible diameter variations listed are not to be construed as runout tolerance. Runout tolerance on diameter is measured at the midpoint of the bare pulley face and is given in Table 8.7. When the lagging is not machined, the runout tolerance over lagging is specified by the individual pulley manufacturers.

Engineered Pulleys to be used with steel cable or high modulus belts are usually machined with a straight face and have permissible total indicator reading runout tolerances as given in Table 8.9.

<table>
<thead>
<tr>
<th>Engineered Drum Pulley Description</th>
<th>Maximum Permissible TIR in (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unlagged Engineered Pulley</td>
<td></td>
</tr>
<tr>
<td>Lagged Engineered Pulley (under lagging)</td>
<td>0.030 (0.76)</td>
</tr>
<tr>
<td>Lagged Engineered Pulley (over lagging)</td>
<td></td>
</tr>
</tbody>
</table>

Table 8.9
Permissible total indicator reading runout for engineered pulleys for steel cable/high modulus belts

**Pulley Face Widths**

Pulley face width is the length of rim, wing or contact bar along the shaft axis. Standard pulley face width is normally equal to the belt width plus 2 inches (50.8 mm) for belt widths up to and including 42 inches (1067 mm) and belt width plus 3 inches (76 mm) for belt widths over 42 inches (1067 mm). Engineered pulley face widths are generally 6 to 12 inches (152 to 305 mm) greater than the belt width to provide greater clearance. The conveyor belt should not wander beyond the edge of the pulley face. Refer to Chapter 7 for more information on pulley face widths.

<table>
<thead>
<tr>
<th>Standard Steel Drum and Wing Pulley Nominal Standard Face Widths in (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12 (305)</td>
</tr>
</tbody>
</table>

Table 8.10
Standard Pulley face widths

The listed variations in face width may occur from one pulley to another. The permissible face width variations are not to be construed as an edge runout tolerance. Edge runout tolerance is specified by the individual pulley manufacturer.

**Pulley Crown**

See Chapter 7 for a discussion of the use of crowned conveyor pulleys. There are currently four types of pulley crowning available, straight face, taper crown, trapezoidal crown and curve crown.

**Straight Face**

Straight face pulleys have no crown and are favored by the belt manufacturers. They are recommended for all installations using reduced ply, high modulus, low stretch belts, such as those with a carcass of steel cables or high strength tensile members.
PULLEYS AND SHAFTS

Taper Crown
On taper crown pulleys, the face forms a “V” with the rotating axis larger in diameter in the center of the pulley. This crown is expressed in inches of crown per foot of total face width, by which the diameter at the center of the face exceeds the diameter at the edge. Normal crowns of this type vary from 1/16 to 1/8 inch per foot (5.2 to 10.4 mm per meter) of total face width.

Trapezoidal Crown
Trapezoidal (Trap) crown pulleys have a flat surface in the middle portion of the pulley face with the ends tapered. Trapezoidal crown pulleys may be appropriate for wider face width pulleys.

Curve Crown
Curve crown pulleys have a long, flat surface in the center of the pulley with the ends curved to a smaller diameter. Except on short pulleys, the curved surface extends in approximately 8 inches (203 mm) from the edge.

Pulley Weights
Pulley weights must be used to determine pulley and shaft selection. Average weights for Standard Steel Drum and Standard Wing Pulleys are available from the manufacturer. There are some variations in manufacturing practices that will affect the weight of the pulleys. Engineered Pulley weights are dependent on the tensions encountered and can vary widely.

Pulley Hub And Bushing Systems
Although some conveyor pulleys are manufactured with cylindrical bores, compression hub/bushing systems are more common. These systems consist of a hub and a bushing with tapered mating surfaces that cause the hub to expand and the bushing to contract onto the shaft when the screws are tightened. There are three major types of hub systems. Hubs can be welded into the end discs, expanded into the end discs or the end discs can have an integral hub. Hubs have a tapered bore that mates to the taper on the outside diameter of the bushing. Welded-in and integral hubs have no split, while hubs that expand into the end discs must be split completely through one side (axially therefore forming a ‘C’ shape) to develop the forces necessary to transmit the required torque via friction.

All systems use split tapered bushings. They can have normal or shallow tapers, and they can come with or without flanges. The bushings are installed by inserting cap screws through the bushing and engaging them into threaded holes in the hubs. Flangeless bushings are either installed in a similar fashion or installed with screws that engage in threaded half holes in the mating hub and non-threaded half holes in the bushing (Figure 8.12). Tightening these screws causes the hub and bushing to move axially relative to one another. The hub expands and the bushing contracts on the shaft. These radial forces develop enough friction to allow the hub/bushing systems to keep the pulley locked securely to the shaft axially and usually with the aid of a key, to transmit torque from/to the shaft on drive pulleys. Keys are generally not required on non-drive pulleys.

Flanged Systems
These types of systems have a taper in the order of 2 to 3 inches per foot (166 to 250 mm per meter) on the diameter. The advantage of the taper is that it reduces pulley end disc stress caused by the installation
of the second bushing. The less the amount of axial movement required to develop the expansion forces, the lower the stress induced in the end discs when installing the bushing on the other end.

Flanged Systems with a Shallow Taper Angle
These systems have a shallow taper, approximately 3/4 inch per foot (62.5 mm per meter) on the diameter. The advantage of a shallow taper is that they can develop higher expansion/contraction forces. These systems also are used where the hub is not welded to the end discs but expands into it producing a friction connection.

Flangeless Systems
Flangeless bushings can have either normal or shallow tapers. As mentioned previously, the bushings are installed by inserting cap screws through the bushing and engaging them into threaded holes in the hubs. In addition, there is a type of flangeless bushing, shown in Figure 8.12, that is installed with screws that engage in threaded half holes in the mating hub and non-threaded half holes in the bushing. Since there is no flange, the bushings require less axial space than flanged bushings. Also, because there is no flange, the bushing mounts flush or nearly flush with the hub and there are few if any protruding parts to collect dust or debris.
Advantages Of Compression Hub/Bushing Systems

There are several advantages to using compression hub/bushing systems. The tapered bushings are very quick and simple to install and dismount. The clamping force of the bushing on the shaft improves concentricity and minimizes the probability of fretting corrosion. No damage to the shaft occurs because no setscrews contact the shaft, and dismounting is quick and simple. Each hub size will accommodate one bushing size. In turn, each bushing size can accommodate a range of bore sizes. Therefore a pulley with a given hub size will accommodate a range of shaft diameters simply by changing the bushings. This results in a decreased inventory of spare parts required and the subsequent benefits. Since most tapered bushings are also available with a minimum plain bore, stocking these bushings, and boring as necessary to size, can further reduce inventory.

Keyless Locking Assemblies

Though not classified as a compression hub/bushing system, keyless locking assemblies are also used to connect shafts to pulleys. These are not to be confused with shallow taper type compression hub/bushing systems that do not require keys to lock non-drive pulleys and some small-bore drive pulleys to the shaft. These devices use many more screws than the other systems. They develop very high expansion/contraction forces and therefore do not utilize keys to lock the pulley to the shaft. A split inner ring contacts the shaft, and a split outer ring contacts the hub bore. There are single and double taper styles. They can have steep, normal, or shallow tapers. Tightening the screws slides the cones towards each other, forcing the inner ring against the shaft and the outer ring against the end disc or solid hub respectively. Due to the very high expansion/contraction forces, keyless locking assemblies require special design considerations to handle the resulting stresses in the shaft and end disc/solid hub.

**PULLEY LAGGING**

Conveyor pulleys can be covered with some form of rubber, fabric, ceramic, urethane or other material. Lagging is used on driving pulleys to increase the coefficient of friction between the belt and pulley. See Chapter 6 for details on the coefficient of friction between the lagging and the belt and $C_w$ factor chart. Lagging is also used to reduce abrasive wear on the pulley face and to effect a self-cleaning action on the surface of the pulley. Abrasive wear and material buildup can substantially decrease pulley life. Drive pulleys should always be lagged. Non-driving pulleys, especially on the carrying side of the belt, should be lagged whenever abrasive or buildup conditions exist. Also, see RPMA’s [RMA is now ARPM - Has
ARPM reissued the handbook or should I still say RMA? Roll Covering Handbook for additional information.

**Thickness And Attachment**

Lagging thickness can vary from a few thousandths of an inch [what is a few thousandths of an inch in mm? 0.031 ~ 0.8 mm so can I say 1 mm?], as with a sprayed-on coating, to thicknesses of 1 to 2 inches (25 to 50 mm), as with some solid rubber vulcanized coatings. Common methods of attachment are bolting, cold bonding, welding, and vulcanizing. Cold bonded and vulcanized lagging are the preferred methods for heavy duty or severe service applications. Of the preferred methods, vulcanized lagging is the most common and most economical. In applications where added rubber elasticity is needed, the cold bond method is preferred. Lagging can be obtained in various grooved and other specialized surface finish types. Bolted-on lagging usually consists of a rubber cover reinforced with multiple ply fabric construction similar to conveyor belting. The fabric plies are required to provide strength under the bolt heads. Welded lagging can be either a slide-on or weld-on lagging. Slide-on lagging is constructed of rubber or ceramic tiles embedded in rubber molded to a metal backing plate and slots welded onto the pulley face. Replacement of the pads can be accomplished by sliding the old strip out of the slots and the new strip in without removing the pulley from its conveyor location. Weld-on lagging is constructed of rubber or ceramic tiles embedded in rubber molded to a metal backing plate. The metal backing plate is welded to the pulley face. Replacement of weld on lagging can also be done without removing the pulley from its conveyor location.

**Rubber Lagging Hardness**

Rubber lagging used on drive pulleys normally has a durometer hardness of 60 on the Shore A scale. The lagging used on non-driving pulleys may have a hardness of 45 or to 60 Shore A, depending on the application. For snub and bend pulleys, which contact the carrying side of the belt, softer rubber tends to better resist buildup on the pulley face. With high tension belts, lagging of 70 Shore A hardness is sometimes used.

**Lagging Grooving**

Drive pulleys, which operate in wet or damp conditions, commonly incorporate grooved lagging. These grooves generally take a shape designed to shed water and slurry materials that can build up at the lagging interface and lower lagging drive efficiency. Common shapes used include diamond, herringbone, and chevron patterns. Grooves are usually spaced on 1-1/4 to 2-1/2 inch (30 to 63.5 mm) centers. In a chevron pattern, the grooves meet at the center of the pulley face, while in the herringbone pattern the grooves are offset by one-half the groove spacing. Figures 8.16 through 8.18 illustrate the grooving patterns. In the Herringbone and Chevron patterns, the apex points in the direction of belt travel. Generally, the dimensions of the grooves are 1/4 inch (6.35 mm) wide by 1/4 inch (6.35 mm) deep with a 1/8 inch (3.18 mm) minimum thickness of material under the bottom of the rubber. There are also groove configurations and sizes of rubber lagging, which can be furnished to enhance belt tracking or reduce the accumulation of material on the pulley face. Ceramic lagging also incorporates grooves, which are configured based upon the ceramic tile layout.
**Figure 8.17**  
Herringbone lagging grooves

**Figure 8.18**  
Chevron lagging grooves

**Figure 8.19**  
Diamond lagging grooves

**Figure 8.20**  
Pulley with diamond groove lagging

**Figure 8.21**  
Slide on lagging

**Figure 8.22**  
Ceramic lagging
Ceramic Lagging
In applications where belt slip or high wear are a concern, ceramic lagging may be used. Ceramic lagging commonly consists of a series of tiles embedded in a rubber substrate forming a bar profile. The tiles may be smooth or have raised surfaces on each tile. Those with raised surfaces tend to have better drive characteristics under wet, sloppy conditions. Due to the raised surface on the tile and the nature of ceramic, this type of lagging exhibits a superior coefficient of friction and greater wear resistance than rubber lagging. Depending upon application conditions, dimpled ceramic lagging can provide approximately 2 times higher traction than rubber lagging.

Wing Pulley Lagging
There are many types of lagging just for use on wing pulleys. Much of what has already been covered applies to lagging for wing pulleys. Each manufacturer has their own design so contact the pulley manufacturer for details.

High Tension Applications
With steel cable and other high modulus conveyor belts, lagging is always used on drive pulleys and is preferred on pulleys that contact the carrying side of the belt. For maximum pulley life and belt safety, lagging can be used on all non-drive pulleys. Lagging for high tension applications can be either cold bonded or solid vulcanized with either rubber or ceramic lagging.

SHAFTING
Suitable shafting to be used with a steel pulley cannot be selected independently of the pulley load rating. In fact, the load capacity of a given pulley is a function of the shaft that is installed in that pulley. The shaft and pulley must be treated as a composite structural assembly. This is because the structural rigidity of the assembly depends upon both the shaft and the pulley and their interaction.

The shaft diameter required for a pulley assembly is a function of two criteria, strength and deflection. Depending on the exact pulley assembly, either strength or deflection can be the determining factor for shaft diameter selection.

Shaft Materials
Pulley design is based upon the use of any commercial or standard shafting material, such as AISI C1018 or C1045 steel. Load ratings of standard or mine duty pulleys are not increased when higher strength shafting is used. High strength shafting is of value in cases where it permits the shaft ends to be turned down so that smaller diameter, high capacity antifriction bearings can be used. It is also sometimes of value on drive shafts to withstand the added torsional stresses. While the use of high strength steel increases the strength of the shaft, its use does not decrease deflection.

Resultant Radial Load
The resultant pulley radial load is the vector summation of the belt tensions, pulley weight, and weight of shaft. The force from weights always acts downward, and the forces from the belt act in the path of the belt and away from the pulley. Resultant radial load calculations for typical pulley arrangements and a general case are illustrated using trigonometric methods in Figure 8.23.
Figure 8.23
Resultant radial load diagrams

\[ R = \sqrt{\left( T_{ccw} \times \cos(\phi_1) + T_{cw} \times \cos(\phi_2) \right)^2 + \left( T_{ccw} \times \sin(\phi_1) + T_{cw} \times \sin(\phi_2) - W \right)^2} \]

Equation 8.24
\( R \), Resultant pulley load for general case

Where:
- \( T_{ccw} \) = Belt tension counter-clockwise
- \( T_{cw} \) = Belt tension clockwise
- \( \phi_n \) = Belt tension angle for case "n" in degrees (+\( \phi_n \) CCW from 0) (-\( \phi_n \) CW from 0)
- \( W \) = Weight of pulley and shaft

\[ R = \sqrt{(T_1 + T_2)^2 + W^2} \]

Equation 8.25
\( R \), resultant load for 180 degree wrap drive pulley, horizontal belt

\[ R = \sqrt{\left( (T_1 + T_2) \times \cos(\theta) \right)^2 + \left( (T_1 + T_2) \times \sin(\theta) + W \right)^2} \]

Equation 8.26
\( R \), resultant load for 180 degree wrap drive pulley, inclined belt
PULLEYS AND SHAFTS

\[ R = \sqrt{(T_1 + (T_2 \times \cos(\theta))^2 + (T_2 \times \sin(\theta) - W)^2} \]

Equation 8.27
R, resultant load for >180 wrap drive pulled, horizontal belt

\[ R = \sqrt{(T_1 \times \cos(\theta_1)) + (T_2 \times \cos(\theta_2))^2 + (T_1 \times \sin(\theta_1)) + (T_2 \times \sin(\theta_2) - W)^2} \]

Equation 8.28
R, resultant load for >180 wrap drive pulley, inclined belt

\[ R = \sqrt{(2 \times T_3)^2 + W^2} \]

Equation 8.29
R, resultant load for tail pulley

\[ R = (2 \times T_3) - W \]

Equation 8.30
R, resultant load for vertical gravity takeup pulley

\[ R = \sqrt{(T_3 \times \cos(\theta))^2 + (1 - \sin(\theta) \times T_3 + W)^2} \]

Equation 8.31
R, resultant load for vertical gravity takeup bend pulley

\[ R = \sqrt{(T_3 \times (1 - \cos(\theta))^2 + (T_3 \times \sin(\theta) + W)^2} \]

Equation 8.32
R, resultant load for snub pulley

Where:

- \( T_1 \) = Tight side belt tension [lbf (kgf)]
- \( T_2 \) = Slack side belt tension [lbf (kgf)]
- \( T_3 \) = Belt tension non-driven pulleys [lbf (kgf)]
- \( W \) = Pulley and shaft weight [lbf (kgf)]
- \( \theta_n \) = Belt tension angles in degrees (+ as shown in Figure 8.23)

Shaft Sizing
Shafts are sized using both a Stress Limit and Deflection Limit. If there is an overhung load, it needs to be included in the Stress Limit calculations. The shaft is sized using the Stress Limit and then the Deflection limit. Then whichever gives the larger shaft size governs. The diameter is then increased to the next standard shaft size.
Shaft Sizing By Stress Limit

Equation 8.33 is given in standard B105.1 for the diameter of a pulley shaft loaded in bending and torsion (drive pulley with no overhung load) is:

\[
D = \sqrt[3]{\frac{32 \times F.S.}{\pi} \times \left(\frac{M}{S_f}\right) + \frac{3}{4} x \left(\frac{T}{S_y}\right)^2}
\]

*Equation 8.33*

\(D\), shaft size based on stress

Where:

- \(D\) = Shaft Diameter [in (mm)]
- \(F.S.\) = Factor of Safety = 1.5 (dimensionless)
- \(S_f\) = Corrected shaft fatigue limit = \(k_a \times k_b \times k_c \times k_d \times k_t \times k_f \times k_g \times S_f^*\)
- \(k_a\) = Surface factor = 0.8 for machined shaft (dimensionless)
- \(k_b\) = Size factor = \((D)^{-0.19}\) (for \(D\) in inches) or \(1.85 \times (D)^{-0.19}\) (for \(D\) in mm) (used as dimensionless)
- \(k_c\) = Reliability factor = 0.897 (dimensionless)
- \(k_d\) = Temperature factor = 1.0 for \(-70^\circ F\) (-57°C) to \(+400^\circ F\) (+204°C) (dimensionless)
- \(k_t\) = Duty cycle factor = 1.0 provided cyclic stresses do not exceed \(S_f^*\) (dimensionless)
- \(k_f\) = Fatigue stress concentration factor due to keyway (dimensionless)
- \(M\) = Bending moment [in-lbf (N-mm)] In Chapter 6 we are using kgf rather than Newtons
- \(T\) = Torsional moment [lbf-in (N-mm)] In Chapter 6 we are using kgf rather than Newtons

### Fatigue Stress Concentration Factor, \(k_f\) (dimensionless)

<table>
<thead>
<tr>
<th>Steel</th>
<th>Profilled Keyway</th>
<th>Sled Runner Keyway</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annealed &lt;200 BHN</td>
<td>0.63</td>
<td>0.77</td>
</tr>
<tr>
<td>Quenched and drawn &gt;200 BHN</td>
<td>0.50</td>
<td>0.63</td>
</tr>
</tbody>
</table>

*Table 8.34*

\(k_f\), fatigue stress concentration factors for typical pulley keyway configurations

### 50% of Tabulated Ultimate Tensile Strength, \(S_f^*\)

<table>
<thead>
<tr>
<th>Steel</th>
<th>(S_f^*) psi (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAE 1018 &lt; 200 BHN</td>
<td>29,000 (200)</td>
</tr>
<tr>
<td>SAE 1045 &lt; 200 BHN</td>
<td>41,000 (283)</td>
</tr>
<tr>
<td>SAE 4140, 200 BHN annealed</td>
<td>47,500 (328)</td>
</tr>
</tbody>
</table>

*Table 8.35*

\(S_f^*\), 50% of ultimate tensile strengths for typical pulley shaft materials

### Yield Strength, \(S_y\)

<table>
<thead>
<tr>
<th>Steel</th>
<th>(S_y) psi (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SAE 1018 &lt; 200 BHN</td>
<td>32,000 (220)</td>
</tr>
<tr>
<td>SAE 1045 &lt; 200 BHN</td>
<td>45,000 (310)</td>
</tr>
<tr>
<td>SAE 4140, 200 BHN annealed</td>
<td>60,500 (417)</td>
</tr>
</tbody>
</table>

*Table 8.36*

\(S_y\), yield strengths for typical pulley shaft materials
For drive pulley shafts with overhung loads outboard of the pillow block bearings, such as a shaft mount reducer or chain drive, refer to the ANSI drum pulley standard B105.1 or vector methods for inclusion of the effects of the overhung load on MB. MB and MT are not defined in the text. Do you mean M & T from equation 8.33?

The method shown above can also be used to develop allowable bearing turndown sizes by adjusting MB for the correct moment arm and multiplying MB and MT by the stress concentration factors for shaft turndowns, available in many engineering texts.

**Shaft Sizing By Deflection Limit**

The pulley assembly is a structural unit, and strengths of different components are interdependent. The shaft diameter, the bearing centers, the thickness of the end disc, the resultant loads on the pulley, the hubs, and the pulley to shaft attachment method are all interconnected in this respect. A steel pulley with very thin end discs, or flexible shaft attachment devices, can have an actual shaft deflection similar to the free shaft calculations discussed below. A pulley made with very thick end discs and rigid shaft attachment devices can result in a shaft deflection significantly less than expected by free shaft calculations. Calculating the load sharing between pulley and shaft is important for reliable design. Pulley constructions and shaft attachment component characteristics vary between CEMA members, so only your pulley manufacturer can determine the actual interaction between components.

ANSI B105.1 and 501.1 are standards maintained by CEMA for selection of Standard Drum and Wing Pulleys. These standards are based on the use of welded plate constructions with rigid compression shaft attachment devices. They apply to specific ranges of fabric belt widths and belt tensions. As an aid to the conveyor designer, these standards provide a convenient method of selection based on free shaft deflection, with a limit of 0.0023 in/in (0.0023 mm/mm) or 8 minutes.

Pulley manufacturers frequently design Engineered Pulleys to the demands of particular applications. In these situations a deflection limit of 0.0015 in/in (0.0015 mm/mm), or 5 minutes, is commonly recommended. System designers often specify a particular deflection limit. Reference this chapter’s engineered pulley section for more details.

Constructions outside of the design assumptions built into the ANSI standards require design models and limits tailored to the constructions characteristics. Pulleys with flexible shaft attachment methods, high pressure keyless shaft attachments, and extremely wide belt widths are examples. Safe deflection and stress design limits may vary from those stated in the ANSI standards and should be based on the capabilities of the components. Equation 8.37 is given in standards B105.1 and 501.1 for the free shaft deflection for any pulley assembly. Equation 8.37 is derived from the more general equation by setting $I=I_2$ and therefore $D=D_2$. Changed so the general equation is first Then 8.38 is derived from the general equation.

\[
\tan(\alpha) = \frac{R}{2} \times \frac{A}{E_y} \times \frac{C}{1} \left[ 1 + \frac{B-2A-2C}{2I_2} \right]
\]

*Equation 8.37*

Shaft deflection equation for a two diameter pulley shaft

\[
\tan(\alpha) = \frac{R \times A \times (B - 2A)}{4 \times E_y \times I}
\]

*Equation 8.38*

Shaft deflection equation for a single diameter pulley shaft
Where:

\[
\tan(\alpha) = \frac{R \times A \times (B - 2 \times A)}{4 \times E_y \times I}
\]

- \(A\) = Moment arm for pulley [in (mm)]
- \(B\) = Bearing centers [in (mm)]
- \(C\) = Moment arm for shaft reinforcement [in-lbf (N-mm?)]
- \(R\) = Resultant pulley load [lbf (N)]
- \(E_y\) = Young's modulus [29 x 10^6 psi (MPa) for steel]

\[
I = \text{Area moment of inertia of shaft at hub} = 0.049087 \times D^4 \text{ [in}^4 (\text{mm}^4)]
\]

\[
I_2 = \text{Area moment of inertia of shaft inside pulley} = 0.049087 \times D_2^4 \text{ [in}^4 (\text{mm}^4)]
\]

\(\tan(\alpha)\) = Tangent of the angle of the deflected shaft and its neutral axis before bending at the pulley end disc (deg)

**Figure 8.39**
Pulley shaft deflection for a dual diameter shaft, reference Equation 8.37

**Figure 8.40**
Pulley shaft deflection for a single diameter shaft, reference Equation 8.38

**TERMINOLOGY**
The CEMA Pulley Committee has approved the following terminology to standardize Pulley Components, End Disc/Hub Configurations and Weld Configurations.
Pulley Components

The basic components of a drum pulley are identified in the pulley cross section in Figure 8.41.

![Basic pulley components](Figure 8.41)

End Disc/Hub Configurations

The common end disc / hub configurations are defined in Figure 8.38. The most common configuration used in CEMA pulleys is the Welded Hub. The other configurations are used more in Engineered Pulleys.

![Pulley end disc/hub configurations](Figure 8.42)
Weld Configurations

The common weld configurations are defined in Figure 8.39. The most common welds used in CEMA pulleys are the Fillet Weld for the end disc to rim connection and the Groove Weld with Back-up for the rim seam weld. The other configurations are used more in Engineered Pulleys. For more detailed information on welds, see AWS D1.1 or AWS D14.6.

![Figure 8.43 Pulley end disc and rim weld configurations](image-url)

SPECIAL PULLEYS

These special pulleys are not covered by any CEMA standards. They are included to let you know they are available and to give some general information on what they are and why they are used.

Dead Shaft Pulleys

Dead shaft conveyor pulleys are designed with non-rotating shafts. The construction typically uses a drum or wing pulley designed to accept bearings for shaft attachment, rather than the typical hub and bushing connection. Rigid mounting blocks are then used for conveyor frame attachment. These can include pillow block, manual take-up, and flange type arrangements. Dead shaft pulleys can be used in nearly all conveyor applications and pulley positions. The non-rotating shaft can make them a preferred choice where weight, safety, or dust control sealing is an important design consideration. Pulley locations requiring torque transmission are typically avoided, due to the non-rotating shaft, i.e. drive, brakes, backstops. Regulations in specialized industries should be considered in the selection, some grain enclosures require external bearings. Design of dead shaft pulleys is similar to the process detailed in CEMA B105.1 and 501.1. Modifications to these standards in the areas of shaft stress analysis and shaft deflection can provide safe designs. Consultation with your CEMA pulley manufacturer for proper dead shaft pulley design is recommended.

CEMA pulley specifications use a fatigue design method and stress limits for rotating shaft selection. Since a non-rotating conveyor pulley shaft is primarily under stationary load an appropriate stationary design method and stress limits may be substituted. Typically stationary stress limits can be higher than fatigue stress limits, which means that dead shaft designs may be safely designed with smaller diameter shafts.

CEMA pulley specifications use shaft deflection limits to control the bending moment transmitted to the pulley and to control the possibility of axial pulley movement along the shaft. Many ball or spherical roller bearings used in dead shaft pulleys have some ability to mis-align without transmitting a significant bending moment to the pulley. Tapered roller bearings have limited misalignment capability and should be considered differently. These misalignment capabilities can safely allow higher shaft deflection limits at the
pulley. In addition, since the bearings inner race to shaft connection is stationary the likelihood of axial pulley movement along the shaft is significantly reduced.

**Magnetic Pulleys**

Magnetic pulleys are a type of magnetic separator. They are drum pulleys located at the head or discharge of the conveyor. Magnetic pulleys are used in conveyors to automatically and continuously remove iron contamination and “tramp” ferrous materials like hammers, bolts, reinforcing bars and spikes from the pay load, be it wood chips, crushed stone, etc. This protects expensive process equipment like cone crushers and wood chippers. See Chapter 11 for more information on magnetic separators.

**Motorized Pulleys**

This chapter deals mainly with conventional conveyor pulleys. Normally the shaft from one of the pulleys of a conveyor is mounted on bearings and driven by an externally mounted motor through various power transmission components such as gear reducers, belt drives, chain drives, and couplings. This conventional arrangement is a live shaft configuration. That is, the shaft rotates.

It has been found advantageous in some applications to replace all of these components with a Motorized Conveyor Pulley. On the outside, a motorized conveyor pulley appears as a conventional pulley with wires coming out of one of the shaft ends. However, on the inside is a motor and gear set which drive the pulley at a specific speed. The shaft does not rotate and is normally supported by mounting blocks, which usually resemble bearing pillow block housings without the bearings.

Since the motor and gearing are assembled inside the drum of a motorized pulley, they must be purchased for a specific motor voltage and belt speed. Backstops and brakes are usually available as options, and some models may be disassembled for the repair of internal components. Motorized conveyor pulleys tend to be more expensive than the sum of components they replace, however they take up less space than conventional drives, and require much less assembly time and skill than their conventional counterparts. See Chapter 13 for more information.

**Spiral Pulleys**

Spirals can be added to drum pulleys and wing pulleys. It is more common to have a spiral wing pulley than a spiral drum pulley. Spacing between spiral bars creates intermittent support of the belt. Belt bottom cover damage has been observed when wide spacing is used. Maximum unsupported width is a function of the belt properties and belt tension. For most applications, limiting the space between the bars to 3 inches (75 mm) will not cause damage. Belt damage can also occur if the bars are too narrow. This can cause the contact pressure between the belt and the bar to be too high. Currently, there is no generally recommended minimum contact area. Contact your belt manufacturer for your specific application.

**Spiral Wing Pulleys**

Care must be used when applying a wing-type tail pulley, because the intermittent, on-again, off-again contact pattern between the “wings” and the belt can impart an up-and-down fluctuation to the belt line. This vibration makes it difficult to effectively seal the loading zone, resulting in additional material spillage. The effective diameter of a Wing-Type Pulley is often smaller than the nominal diameter and this must be taken into consideration when selecting the belt. Wrapping a band of steel in a spiral around the wing pulley allows the pulley to provide the self-cleaning benefit without creating oscillation in the belt line or reducing the effective diameter. See Figure 8.44.
**Spiral Drum Pulleys**

When the tensions are too high to use a spiral wing pulley and you still need the cleaning action that a spiral wing pulley provides, spirals can be added to drum pulleys to auger the material from the center of the pulley to the edges. The spirals on the drum pulleys are much thicker than the one used on wing pulleys to create an area for the material to travel through. See Figure 8.45.

![Figure 8.44](image1.png) Spiral wing pulley  
![Figure 8.45](image2.png) Spiral drum pulley

**STUB SHAFT PULLEYS**

Stub shaft conveyor pulleys are designed with a relatively short stub shaft on each end of the pulley. These stub shafts typically extend into the pulley a short distance and terminate with a significant portion of the pulleys internal length void of shafting. Shaft to pulley attachment is generally accomplished by welding in the shaft or bolted connections.

Stub shaft pulleys can be used in nearly all conveyor applications and pulley positions, and may be preferred when weight is a premium or when the pulley is relatively wide. Welded in stub shaft pulleys may be preferred when shaft removal is not necessary. Bolted connection stub shaft pulleys may be preferred when simplified shaft removal is desired and as a tool to minimize spares inventory.

Design of stub shaft pulleys can vary significantly with the methods detailed in CEMA B105.1 and 501.1. Some design concepts in these specifications do not apply to stub shafts. Other concepts need to design stub shafts are not addressed. A partial list of important concepts specific to stub shaft design would include: shaft deflection concept does not directly translate, welded shaft fatigue properties are not published, and bolted connection fatigue properties are not published. Consultation with your CEMA pulley manufacturer for proper stub shaft pulley design is recommended.

Seems to me that this statement, “*ANSI B105.1 and 501.1 are standards maintained by CEMA for selection of standard drum and wing pulleys.*” which first appears on page 18 should be made the first time the standard is mentioned on page 2. We are not consistent through out the chapter on the use of the names of these standards. Sometimes we call them CEMA 105.1 and other times it is CEMA B105.1 and sometimes just B105.1. We do the same thing with 501.1 and sometimes call it B501.1. I suggest we be consistent and following the statement “*ANSI B105.1 and 501.1 are standards maintained by CEMA for selection of standard drum and wing pulleys.*” we always say CEMA/ANSI B105.1 and/or CEMA/ANSI B501.1.

I also suggest we put in a table with all the applicable standards that CEMA members use to design pulleys. I see nothing in this chapter or CEMA/ANSI B105.1 and/or CEMA/ANSI B501.1 about shell thickness for example.
Mounted Bearings

Bearings are needed to allow free rotation and to support the weight of related machine components that are required to rotate. At the same time, maintaining machine component locations relative to other machine components.

Mounted Bearing Components

Rolling Elements
Bearings using rolling elements are considered either “ball” bearings or “roller” bearings. The name reflects the geometry of the rotating element within the bearing. Ball bearing rolling elements are round in shape allowing single point contact with the inner and outer raceway surfaces. Roller bearing rolling elements vary in shape - such as cylindrical, conical, tapered or barrel and have a larger contact area with the load carrying surfaces. See Figure 2.2. The rolling elements of a ball bearing make contact with the raceways at only a small area or point, which allows the bearing to move with little effort. For this reason, ball bearings generate less heat, allowing them to operate at high speeds. However, ball bearings cannot carry as heavy a load as a comparable sized roller bearing.

Seals
Multitudes of sealing arrangements are available for mounted units, with the seal selection being based on the bearing application. Seals can be made from a variety of materials and, in many cases, specialty seals can be incorporated if a standard seal is insufficient for the application. Seals supplied with grease lubricated mounted bearings are sometimes designed to allow excess grease to purge out of the bearing. This enables re-lubrication to occur safely under normal bearing maintenance. Not all bearing manufacturers use this type of seal. Contact seals typically incorporate a resilient flexible component that completely closes any path for contaminants to pass through to the bearing. The flexible element will always make direct contact and have relative movement with another component of the bearing. The flexible element will tend to return to its original shape if deformed. Usually, contact seals are arranged in single or multiple combinations and are best suited for wet or dry environments under slow to moderate speeds.

In contrast to a contact seal, a labyrinth seal protects a bearing without completely closing off the path to it. The seal is provided through a complex maze sometimes filled with grease. This maze blocks easy passage while at the same time reduces bearing operating temperatures due to the absence of any rubbing components. The lower operating temperature allows for high-speed operation without overheating the bearing. In addition, labyrinth seals are usually made from all metallic components and are suited for high temperature environments.

Combination seals blend elements or features of the above types of seals. Combination seals use contact seals toward the interior of the bearing and labyrinth seals at the outside of the bearing.
A basic clearance seal consists of multiple, stationary, thin rings or washers of rigid material. This seal does not make contact with the rotating surface of the bearing, but is very close to it. The absence of
rubbing components allows for reduced operating temperature and, therefore, high running speeds. It effectively blocks medium and large size contaminants. Clearance seals are usually entirely metallic and operate with the same benefits as a labyrinth seal – suitable for high temperature or high speed applications. Clearance seals are often preferred in high temperature environments. A flinger is a shield that is pressed over the inner ring or shaft and is used as a deflector as they rotate with the inner ring to “fling” material away from the bearing. Fingers are always installed to take advantage of centrifugal force imparted by the shaft rotation to deflect foreign material and/or to shield the primary seal from direct water sprays or larger particles.

**Mounting Methods**

Mounted ball and roller bearings are slip-fitted over a shaft and must be secured to the shaft to prevent rotation within the bearing bore. There are many methods of securing a mounted bearing to a shaft; each one is designed to carry out the same principal task. A basic setscrew lock, shown in Figure 3.2, is a very simple method that incorporates setscrews directly threaded into the inner ring. Setscrew locking bearings are easy to install and are relatively inexpensive. However, they impose limitations on high speed bearing operating due to the non-centered shaft, and rely on strict shaft tolerances for ideal performance.

![Setscrew Lock](image1)
![Eccentric Lock](image2)
![Concentric Collar Lock](image3)
![Tapered Adapter Sleeve Lock](image4)

When the set screws are tightened correctly, the clamp load causes the set screws to dig into the shaft. This clamp load provides for better holding power in axial (thrust) load applications when compared to
other locking methods. The deformation of the shaft that occurs from the clamp load can make it more difficult to remove the bearing from the shaft.

A concentric lock (see Figure 3.4) uses a split collar with one or more clamping screws. This clamping collar fits over the extended inner ring, which has radial slots to allow compression of the inner ring extension. When the bearing is in the proper location on the shaft, the lock collar cap screws are tightened to close the collar, compress the inner ring, and affix the bearing to the shaft. It locks the shaft in a 360 degree contact pattern, centers the shaft in the bore, and does not mar the shaft. A concentric collar lock is limited in axial (thrust) holding power.

The eccentric locking mechanism, shown in Figure 3.3, uses a cam design to create a wedge effect and secure the inner race to the shaft. The cam action is created by the eccentricity of the inside diameter of the locking collar in relation to the engaging extension of the inner ring of the bearing. A single setscrew in the lock collar assures the retention of the collar’s position by threading through the collar into the shaft. An eccentric collar locking mechanism has the same advantages and disadvantages as setscrew locking bearings, but is also limited to non-reversing applications.

A tapered adapter is designed to wedge a sleeve between an inner ring bore and a shaft. This wedging action is produced and retained by a lock collar that is used to draw the tapered sleeve into the bore of the bearing, or drive the bearing onto the outer diameter of the sleeve. This locking method excels at centering the shaft in the bearing bore, providing 360° locking, and reducing fretting corrosion between the bearing and shaft.

The shaft tolerance requirements are broadened, allowing use of commercial shafting. However, they are usually more complicated and time consuming to install compared to the previously discussed locking methods. Once experience is gained with the installation process, it can take the same amount of time to install as the other locking methods. Dismounting the bearing can be faster and easier.

Some applications require a bearing to be pressed onto a shaft for installation. This type of installation calls for an interference fit between the bearing and the shaft. The amount of interference is dependent on the type of bearing and the application. The press fit can be achieved by forcing the bearing onto an oversized shaft or by heating and expanding the bore until it can be easily installed. After cooling, the bearing will shrink around the shaft and a press fit will be achieved. One advantage of a press fit is its concentric 360° lock. In addition to a difficult installation process, the main disadvantage of a press fit is that it requires a precision machined shaft along with additional shaft collars or shaft shoulders to retain the axial location of the bearing.

There are other methods and variations of those discussed above, some of which are patented systems from various manufacturers. There are often advantages and disadvantages to using one locking system over another. The locking device selected for use is often determined by installation location, application or customer preference.

**Housing Materials**

Cast iron is the most common material. Other materials include ductile iron, steel, stainless steel and polymer.

- **Cast Iron** - A general purpose housing most commonly used for a variety of applications. This is the lowest cost housing and is easily machined and adapted. This housing can handle a wide temperature range and light to moderate loads. The most common grades have a 20,000 – 30,000 psi ultimate tensile strength.

- **Cast Steel and Ductile Iron** - Ideal for rugged heavy duty applications and accepts shock and heavy loads. They dissipate heat well, but are usually more costly than cast iron and not as readily available from stock. The most common grades have a 65,000 – 100,000 psi ultimate tensile strength.
Although cast iron, cast steel and ductile iron are the most common materials available for use, there are numerous other materials available for bearing housing (stainless steel, polymer, etc.) depending on the specific application requirements. Housings are designed as solid units or in two pieces. They are cast to shape and machined to size to accept a certain size bearing. The two-piece housings are cast and machined to ensure a proper mate between the two halves. An alignment dowel pin and hole can be used to help locate the two halves, ensuring a unified fit to accept the housing halves. Because they are machined together, housing halves cannot be mixed.

Fig. 3.6 Examples of Bearing Housings

**Housing Types**

Pillow Block: The most common housing type used. It encompasses the largest range of bore sizes and is easily adapted to most applications. Two- and four-bolt hole patterns are most common and as with the other mounted types, a variety of housing materials, shapes and sizes are available.

Tapped Base: A compact version of a pillow block when space constraints limit the size of the housing. The housing has bolt holes tapped into the base for securing the assembly to a mounting structure.

Flange Bearings: A flange housing mounts flush against a mounting structure and allows for assembly of mounting bolts in the same direction as the shaft. They are available in 2-bolt through 6-bolt designs, depending on bore size.

Piloted Flange: Uses a pilot on the back of the housing to mount into a frame or hole for added stability or precision locating.

Take-up Bearings: Available for numerous shapes but is always used where shaft positioning is required. This could be to add tension to a conveyor belt or move a bearing to increase ease of maintenance within an application.

**Expansion Mounted Bearings**

Expansion mounted bearings are recommended for all conveyor applications. If the mounted bearings on the shaft were rigidly fixed in position on both the shaft and the machine foundation, the changing length of the shaft could damage or even destroy the bearings.

**Anti-Friction Mounted Rolling Element Bearings**

Mounted anti-friction bearings can usually be classified into three groups:

1. Ball Bearings
2. Taper Roller Bearings
3. Spherical Roller Bearings
**Single Row Ball Bearing**

One of the most common anti-friction bearings is a ball bearing. Ball bearings are capable of handling pure radial load, pure thrust load, or a combination of both. They are suitable for low or high speed applications and, typically, are the least expensive type of mounted anti-friction bearing. Mounted bearings are generally supplied “shaft-ready” with a locking device, lubricant (usually grease), seals and housing.

Mounted roller bearings have many characteristics in common with mounted ball bearings. The primary advantage to this group of bearings is a greater load carrying capacity when compared to a similarly sized mounted ball bearing. However, the gain in load capacity comes with a sacrifice in the maximum allowable speed.

Tapered roller bearings consist of four basic components: the inner ring called the cone; the outer ring, called the cup; the rolling elements, which are cone-shaped rollers; and the retainer, called the cage.

The basic design of a spherical roller bearing, like other mounted, anti-friction bearings, consists of an outer ring, an inner ring, a double row of barrel-shaped rolling elements and a retainer Figure 3.9.

Spherical roller bearings are suitable for applications that require high radial loads and moderate thrust loads, but are not well-suited for pure thrust applications. They are designed for use in low-to-moderate speed applications with grease or occasionally oil as the standard lubricant.

![Fig. 3.8 Mounted Taper Roller Bearing Roller Bearing](image)

![Fig. 3.9 Mounted Spherical Roller Bearing](image)

**Bearing Loading and Life**

**Loading**

Bearing load determines the amount of stress on the bearing and is directly related to its life. Load can take many forms including radial loads, thrust loads, peak loads, shock loads and vibration. A complete analysis of the direction and magnitude of the load is necessary in order to make an accurate bearing selection. The majority of the time you will be working specifically with radial and thrust loads or a combination of both.

Radial Load – Loads that apply forces perpendicular to the shaft.

Thrust Load – Loads that apply forces parallel to the shaft.

Radial or thrust loads are primarily caused by the weight of the rotating component. However, they are also imposed by the magnitude and direction.

Another important aspect is speeds. Speed is generally measured in revolutions per minute (RPM).
It is also important to take temperature into account. This includes both maximum and minimum temperature. Keep in mind that bearing and shaft assemblies indirect sunlight will act as though the temperature is higher than the ambient temperature. The external environmental conditions will consider a gas or liquid environment, solid or liquid contaminant, corrosive, abrasive, flammable or explosive conditions or even ultra clean conditions of bearing location. The quality of maintenance and service expectations in terms of life and reliability can influence bearing selection.

The initial design considerations often have unknowns in the various conditions mentioned above. Assumptions and decisions can be made based on best judgments; however, the risks of making such decisions must also be evaluated on a worst-case scenario basis.

**Equivalent Radial Load**

The initial calculation is for determining the equivalent radial load on a bearing. When a bearing has no thrust loading, the actual radial load is used for the equivalent radial load. If the bearing has both radial and thrust loads, they both must be converted into an equivalent radial load.

The calculation for equivalent radial load is as follows:

\[ P = XFR + YFA \]

Where:

- \( P \) = Equivalent radial load
- \( FR \) = Radial load
- \( FA \) = Thrust load
- \( X \) = Radial load factor
- \( Y \) = Thrust load factor

The radial load and thrust load factors used in the above formula are available from bearing manufacturers catalogs and vary for the size and type of bearing.

**Bearing Life (L10)**

The second calculation is to determine the expected service life of a bearing in hours, referred to as the bearing L10 life.

L10 is the life attained by 90% of a statistically similar group of bearings operating under similar load and speed conditions.

This can also be described as the point in time when 10% of a group of like bearings under normal conditions has failed. This L10 life in hours is used as a design point for selecting a specific bearing for a known application.

For the installer and maintainer of the bearings, the L10 life is useful in determining the amount of time a bearing can be expected to last under ideal conditions. The failure mode expected at the calculated life is due to material fatigue resulting in spalling on the raceways. When selecting a bearing from a particular manufacturer’s catalog, the installer needs to first determine the equivalent radial load (as shown in the previous example), the RPM of the machine, and the dynamic capacity of the bearing. The dynamic capacity is based on the bearing bore diameter and can be found in the manufacturer’s catalog. The definition of dynamic capacity for spherical, ball, and tapered roller bearings can be found below.

Note that the dynamic capacity is denoted as \( C_r \) for spherical roller bearings and ball bearings, and as \( C_{90} \) for tapered roller bearings. The difference in notation is because the dynamic capacity \( C_r \) (spherical/ball) and \( C_{90} \) (tapered) are not to the same base. To compare basic dynamic capacities of spherical and tapered roller bearings, multiply \( C_r \) x 0.259 and compare to \( C_{90} \).

**Basic Dynamic Capacity Definitions:**

Spherical & Ball Bearings, \( C_r \) – The load at which 90% of a given group of bearings can meet or exceed 1,000,000 revolutions.
Tapered Bearings, C90 – The load at which 90% of a given group of bearings can meet or exceed 90,000,000 revolutions.

When consulting with a manufacturer, providing the equivalent radial load, RPM and diameter of the shaft will return a list of suitable bearings with varying expected service lives (see manufacturer’s catalog for formula). An installer or maintenance supervisor must consider that this L10 value is based on bearings in an ideal environment, correctly maintained.

A prudent estimate of the actual expected life of a bearing would be somewhat lower. In most cases, the customer will specify the desired L10 life. In cases where the customer does not specify a desired L10, then 60,000 hours is considered a proper design life for conveyor applications. When L10 is greater than 100,000 hours, the fatigue life of the bearing is considered non-existent. The formula for calculating the L10 value is shown below.

\[
\begin{align*}
\text{Ball Bearing} & : \left( \frac{C_r}{P} \right)^3 \times \left( \frac{1,665}{\text{rpm}} \right) \\
\text{Spherical Roller Bearing} & : \left( \frac{C_90}{P} \right) \% \times \left( \frac{1,665}{\text{rpm}} \right) \\
\text{Tapered Roller Bearing} & : \left( \frac{C_e}{P} \right) \times 1,500,000 \text{ rpm} \\
\end{align*}
\]

Where:
- \( C_r \) = dynamic capacity for ball or spherical bearings
- \( C_90 \) = dynamic capacity for tapered roller bearings
- \( P \) = equivalent radial load
- rpm = speed of the rotating ring

**Bearing Selection**

After a thorough analysis of a bearing’s end use, the type of bearing to be used in a specific application is often the first decision to be made.

Understanding the advantages and disadvantages of each bearing type is essential. Always keep in mind that the manufacturer’s literature is extensive and should be consulted in making final decisions. The following sections provide additional insight.

**Expansion Capability**

When selecting the proper bearing, a determination must be made as to whether the application requires the use of expansion type bearings. Adapter mounted bearings require expansion to allow for axial movement when tightening the unit to the shaft.

NOTE: Under no circumstance should all bearings on the shaft be expansion type bearings.

**Type of Locking Device**

Many types of locking devices are available. Each one has its own advantages and disadvantages. The most common types are: setscrew locking to the shaft; concentric locking to the shaft using tapered adapter sleeve or clamping collar; and eccentric locking to the shaft using a cam-like collar and equivalent machined inner ring.

**Environmental Operating Conditions**

Environmental conditions can play a role in the determination of which type of bearing, i.e., tapered, spherical or ball, but plays a much more significant role in the options that can be incorporated with each type of bearing. For example:
- Seal type and configuration
• Grease type and amount of grease fill
• Special bearing and housing coatings
• Special bearing hardware, i.e., stainless or other corrosion resistant materials

**Housing Style**
There are many different styles of housings available: Pillow Block, Flange and Take-Up, just to name a few. Each style has its place in the industry and has corresponding advantages and disadvantages. Criteria that must be considered when selecting the proper housing style are as follows:
• Housing Strength
• Housing Material
• Accessibility
• Environment
• Application

The proper bearing selection is key to obtaining the rated life of the bearing. Considerable time and care should be taken when making this critical decision. Decisions will have to be made for each application as to which selection criteria are most important.

**Bearing Handling and Installation**
**The Key to Long Bearing Life**

As with precision machine components, mounted bearings are sensitive to improper storage and handling. All bearings should be stored in a cool dry area. Often the manufacturer supplies a bearing in packaging designed to protect unused bearings from the effects of humidity, handling and exposure to contamination. Keep the bearings in their original packaging until ready for use.

Mounted bearings usually come with a factory packed lubrication, oil, or rust preventative that also will protect the bearing. If the bearing does not have any factory lubricant, covering the bearing with an additional light coating of a rust preventative may extend the storage life of the bearing.

**Bearing Storage Guidelines**
When a bearing remains in storage for an extended period of time, the manufacturer’s recommendations should be followed.

**Bearing Alignment**
Bearing alignment has a dramatic effect on bearing performance and service life. Misalignment can lead to premature bearing failure due to excessive loading, high bearing temperatures and vibration. Misalignment, normally measured in degrees, can originate from any one of several causes, but is usually attributed to improper installation.

Most mounted bearings are provided with allowances for misalignment and are termed “self-aligning.” The manufacturers’ literature will identify those bearings and their degree of acceptable misalignment.

The various forms of misalignment are identified below and are illustrated in Figures 6.1 and 6.2.

Static Misalignment Conditions - Bearing and shaft axis not co-linear before operation
• Bearing supports not on same plane
  - Vertical Axis
  - Horizontal Axis
• Deflection misalignment – shaft bending in a fixed direction due to load
• Dynamic Misalignment Conditions
  - Bent shaft
  - Unbalanced Rotating Load (Eccentric Load)

Most mounted bearings are designed with the ability to statically misalign. However, only spherical roller bearings can dynamically misalign. Spherical roller bearings misalign internally to the bearing, while most ball and tapered roller bearings (and some plain bearings) misalign externally.

**Bearing Foundations**

Bearing housings require a firm and fixed mounting location or foundation. The bearing housing must be secure or the bearing alignment may not be maintained. This can cause premature bearing failures.

Under heavy loads, proper housing bolt tightening torque is essential. In addition, housing shear bars may be used to keep the housing from shifting on the mounting surface and applying a shear load on the mounting bolts.

Housing foundations must also be flat and smooth. If a mounting surface is not true, the housing may be forced into an out-of-round condition that can “pinch” a bearing insert, or cause housing fracture during installation or later during bearing operation. A pinched bearing insert may cause the bearing rings to become elliptical (out of round) and may result in accelerated bearing wear, overheating, noise, and, ultimately, bearing failure.

![Fig. 6.1 Static Misalignment](image)

**Fig. 6.1 Static Misalignment**

![Fig. 6.2 Dynamic Misalignment](image)

**Fig. 6.2 Dynamic Misalignment**
Recommended Shaft Diameters
When selecting a bearing shaft diameter, it is best to use 15/16” and 7/16” for shaft sizes below 6”. For sizes 6” and above, increments should be 1/2”. Bearings in these sizes will have the best availability from bearing manufacturers and local power transmission component distributors.

Shaft Expansion
One of the prime considerations in bearing installation or replacement is the effect of thermal shaft expansion on a bearing. One of the ways to accommodate this expansion is to install a bearing that will allow for thermal growth, or shrink of the overall shaft length.

Setting Bearing Clearance
Most mounted bearings are shipped with a factory-set, internal radial clearance. However, a few bearing types require measuring and reduction of the internal radial clearance to properly install the bearing to the shaft. Check the instruction manual to determine if the internal clearance will need to be adjusted.

Cleanliness
As is true with all of the components of mechanical power transmission systems, keeping the machine components clean of foreign materials, debris and contamination will contribute to the expected service life of a mounted bearing.

Fastener Maintenance
The process of inspection, maintenance, and assembly of mounted bearings will invariably involve the use of threaded fasteners. Many of the fasteners used are specialized as to type, design and material. A manufacturer’s fastener selection is based largely on the application, with some consideration for cost or availability.

Fastener Torque
The various manuals and maintenance procedures for specific equipment will often contain cautions and notes specifying the degree of fastener tightening to be maintained on assembly of certain components. These should be specified by the manufacturer as necessary for the specific application.

In all cases where a manufacturer has specified a specific torque value, it is important that the personnel assembling the component adhere to that torque requirement. The tightening torque is directly related to the resultant fastener clamping force and is limited by the strength of the fastener and the mounted bearing housing.

Bearing Lubrication Reducing contact friction, wear, heat, and contamination

Lubrication Methods
Methods vary for applying a lubricant to the bearing contact surface area. The chosen method depends primarily on the type of lubricant used, but is also determined by the type and end use of the bearing, its environment, and its physical location. The following describes lubrication methods commonly in use based on lubricant type.

Oil Lubrication
Oil lubrication methods for mounted bearings include oil bath, oil circulation, oil mist and oil jet. The method selected is often dictated by the number of bearings, accessibility, type of oil used, etc.

Grease Lubrication
Anti-friction bearings are usually lubricated with grease because it is much easier to retain grease than oil in the housing over a long period, and because grease acts, to some extent, as a seal against dirt and other contaminants entering the bearings.

Rolling element bearings are often provided with an initial grease charge that allows installation with little or no addition of grease before operation. The typical installation of grease lubricated mounted bearings includes a grease fitting (see Figure 3.1). After a period of running, the operator may find it necessary to add grease using a hand grease gun or other pressurized grease injecting system.

Lubrication of unmounted bearings in storage will involve hand-packing the bearing. Before renewing the grease in a hand-packed bearing, the bearing assembly should be washed in clean kerosene, degreasing fluid or other solvents.

When replacing the grease, use your fingers to force the grease between the balls or rollers. The available space inside the bearing should be filled completely and the bearing should then be spun by hand. Wipe off any grease thrown out.

As a general rule, it is not necessary to grease-fill the free space in a ball or roller bearing greater than 1/3 of the void available. This is because, under normal operation, a bearing will expel or purge all grease in excess of about 1/3 of the volume available. The 1/3 rule, therefore, is essential for bearings with two shields or seals, since the excess grease cannot escape during normal operation. The result would be damage to the grease and seals as well as the bearing, because the churning of excess grease in the bearing during operation would create considerable heat.

The exception to this 1/3 rule is bearings that turn very slowly. The definition of a slow turning bearing is one that rotates at 20% or less of the maximum rated catalog speed. Bearings of this type are usually filled from 80% to 100% with grease.

Some manufacturers make bearings with seals that are capable of purging excess grease without causing damage to the seal or the bearing.

**Grease Re-Lubrication**

Re-greasing of a bearing depends on the type, size, revolution speed, operating temperatures, environment, sealing arrangements and types of grease. In all cases, refer to the manufacturer’s recommendations for when to re-lubrication.

**Bearing Failures**

Maintenance personnel often report bearing problems in relation to the resulting condition of the failure. Here is a sample list of the most commonly reported conditions:

- Bearing running hot
- Bearing running noisily
- Excessive vibration
- Bearing damaging shaft (spinning)
- Shaft difficult to turn
- Lubrication leakage
- Bearing life is too short
It is an established fact that roughly 80% of all bearings failures are lubrication-related. The following list shows the most frequently encountered lubrication failure modes, plus other common causes of bearing failure:

- Wrong type of lubricant
- Insufficient amount of lubricant
- Excessive amount of lubricant
- Contamination of lubricant
- Misalignment
- Inadequate bearing clearance
- Undersize shafting
- Imbalance in the system
- Improper bearing selection for application
- Mounting / Handling

Since the vast majority of bearing failures are lubrication related, it is imperative that proper lubrication techniques be incorporated into a preventative maintenance program. In order to do this, you need to determine the following 3 aspects:

1. Type of lubricant:
   a. Soap Base
   b. Required Viscosity
2. Lubrication schedule
3. Lubrication amount