Comparison of shaft designs for conveyor pulleys and idler rolls

L.J. LAUGHLIN

L.J. Laughlin, member SME, is production marketing manager with Precision Pulley & Idler, Pella, IA. Preprint number 03-103, presented at the SME Annual Meeting, Feb. 24-26, 2003, Cincinnati, OH. Revised manuscript received and accepted for publication December 2003. Discussion of this peer-reviewed and approved paper is invited and must be submitted to SME Publications Dept. prior to Feb. 28, 2005.

Abstract

The majority of nondrive conveyor pulleys use rotating shafts, and typical idler rolls use nonrotating shafts, even though both products are used to support and guide the conveyor belt and its load. This paper explores and compares the requirements for rotating and nonrotating shaft designs and the impact of these designs on the product. Additionally, this paper describes how the "roller" bearing types (deep groove ball, tapered or spherical) affect these designs.

Stress

When designing shafts for pulleys and idlers, the first consideration is shaft stress. The ANSI equation for calculating the required shaft diameter due to stress is

$$D = \sqrt[3]{\frac{32Fs}{\pi}\sqrt{\left(\frac{M}{Sf}\right)^2 + 0.75\left(\frac{T}{Sy}\right)^2}} \quad (1)$$

For nondrive conveyor pulleys (where $T = 0$), this equation reduces down to the bending stress equation

$$D = \sqrt[3]{\left(\frac{RA}{2}\right)\left(\frac{32}{\pi 8,000}\right)} \quad (2)$$

ANSI B105.1 calls for the bending stress (Sf) to be limited to 55 MPa (8,000 psi) at the drum pulley hub or 42 MPa (6,000 psi) for wing pulleys.

To correctly size the pulley shaft, the first step is to make sure that it is properly sized for the pulley. First, the shaft must be sized as in Eq. (2). Then, with this initial size, the size factor, $kb = (D)^{-0.19}$, can be calculated. Sf (for use in Eq. (1)) is then calculated from ka, kb, kc, kd, ke, kf, kg and Sf^*. For conveyor pulleys, $ka = 0.8$, $kc = 0.897$, $kd = 1.0$, $ke = 1.0$, $kf = 0.63$ and $kg = 1.0$. Sf^* is equal to half of the ultimate tensile strength, while Sy is equal to the yield strength of the shaft material. This means that Sf becomes 0.226 $D^{-0.19}$ times the ultimate tensile strength of the shaft. If the shaft diameter needs to be increased due to fatigue consideration (Eq. (1)), then this cycle will have to be repeated (recalculating Sf each time) until the size used in kb is the size determined by Eq. (1).

While CEMA does not list a shaft limit for idlers, the manufacturers do use the bending stress equations. This is because the shaft of the idler is stationary and does not rotate. Therefore, it is not subject to the same fatigue analysis as a rotating pulley shaft. The other difference between pulleys and idlers is that the idler shaft is designed once and the same design is used repeatedly. However, pulley shafts will often be a unique design that is seldom repeated. In particular, the support centers, journal diameters or turndown radius will vary. Therefore, it is necessary for CEMA to publish detailed information on how to apply the product.

It is interesting to note that during the last 30 years, manufacturers have switched to using 1045 shafting for conveyor pulleys. While one might think that this would lead to a higher allowable shaft stress, no change has or will be coming. 1045 shafting costs are essentially the same in the sizes that are used in pulleys. However, sometimes 1018 is required for particular situations. By not chang-

FIGURE 1

Stress concentration factors.

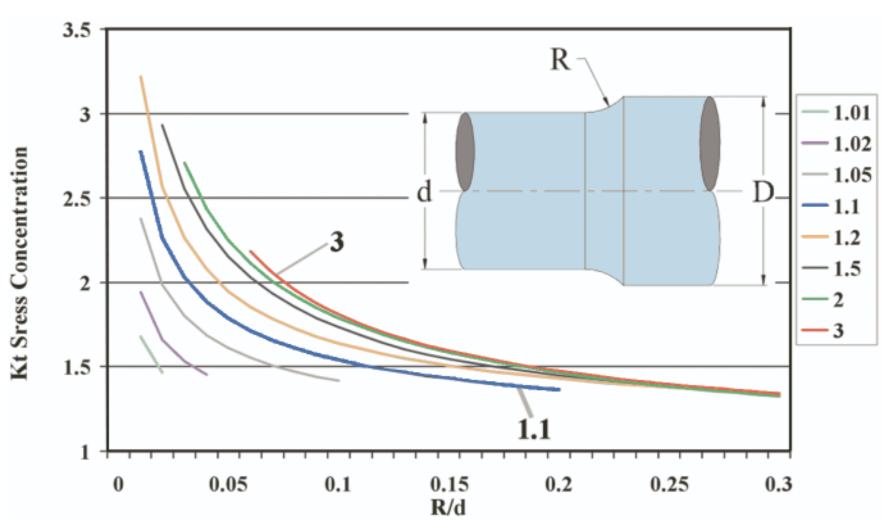
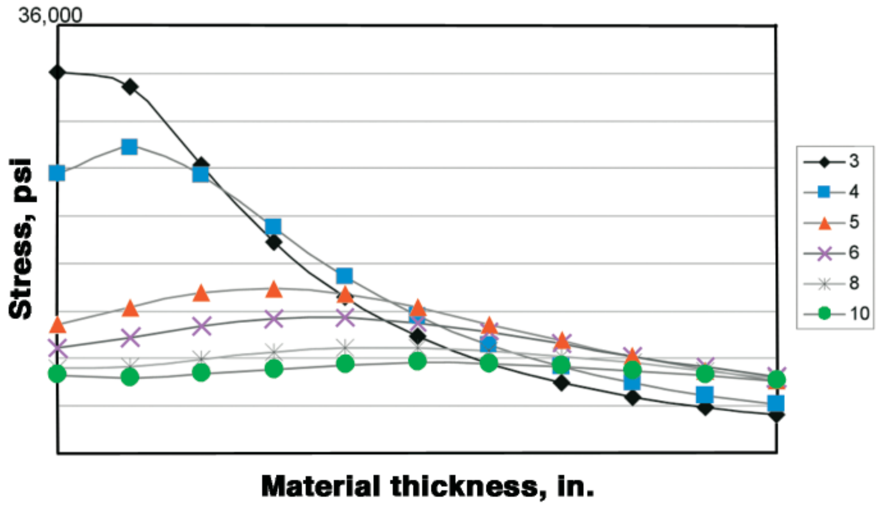


FIGURE 2

End disk stress as a function of OD/SD.



ing the limits, it allows the user to use either type of shafting. Another factor was that some would not pay attention to the stress concentration factors associated with the turndown radius. They would worry only about getting it to "fit."

Stress concentrations

To do a proper evaluation of the shaft design, the engineer needs to determine the stress concentration factor as in Fig. 1. Even with computers being common, some of the old rules of thumb work quite well — using turndown radius of at least one-fourth of the minor diameter. This rule keeps the stress concentration (Kt) from the journal to a minimum and keeps the stress concentration factor low enough that the local shop will not have to "check" it for design. If smaller radii are needed due to available room, etc., then this will have to be "checked" out by the engineer, as the actual stress will be equal to the calculated stress at the smaller diameter times Kt.

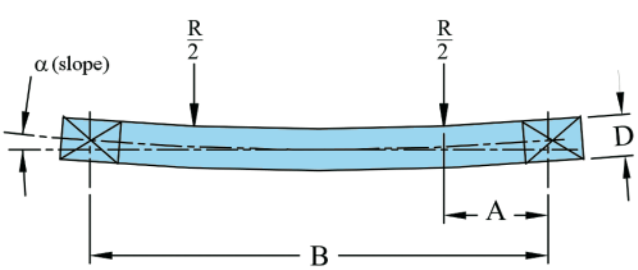
While the shafting commonly used in pulleys is 1045, the shafting used in idlers is 1018. To the casual observer this may seem unusual. On one hand the pulley shafting should be stronger because it is higher-carbon steel, but the idler shaft does not rotate. There is no reverse bending stress cycle. Therefore, the allowable bending stress can be significantly higher.

While idlers and pulleys are similar in nature, there are several things that differentiate them. One fact is that in idlers, the manufacturer controls the complete design of the roll and the framework that supports the roll. This means that the design of the shaft is under complete control of one design group. They can take a look at each shaft design. Because the design is done once for an entire series, it does not need to be looked at again until there is a design change. All that the user needs to know is the load capacity as listed by CEMA.

Meanwhile, in a pulley there is no control over the supports or even the journals. Each shaft is unique and needs to be looked at every time. Any turndowns must be examined for possible stress concentration factors.

FIGURE 3

Shaft slope.



Deflection

Stress is not the only consideration. Shafting for both idlers and pulleys must be designed to handle deflection as well as stress. Excessive deflection will lead to problems not in the shaft but in the part that the shaft supports. The problem is manifested through the connection to the shaft. In a pulley, the deflection of the shaft causes bending moment being applied to the shaft connection and the end disc. The significance of the bending moment is dependent on the pulley geometry, as shown in Fig. 2.

As belts advanced, they became stronger and more flexible. This resulted in the pulley diameter shrinking, while the shaft diameter increased. This lowered the ratio of OD/SD (outside diameter divided by the shaft diameter). Thirty years ago, this ratio was approximately six to eight. Today, it is three to five. This change in geometry puts more pressure on the shaft connection and the end disk and may lead to premature failures.

To limit the bending moment being applied to the pulley, manufacturers have used the slope of the unconstrained shaft deflection at the pulley hub as a published limit to the user. The typical limit is eight minutes of shaft slope for standard pulleys and five minutes of shaft deflection for more critical applications, as depicted in Fig. 3 and calculated by

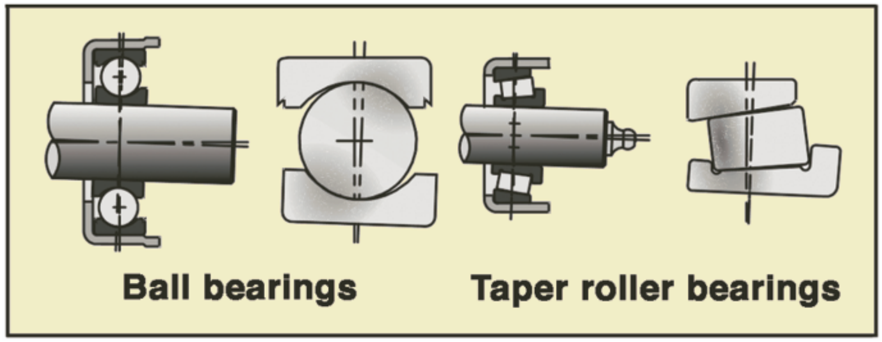
$$\mathrm{Tan}\alpha = RA\frac{(B-2A)}{4EI} \qquad (3)$$

At the same time, there is a deflection limit placed on idlers. However, the limit is not imposed by CEMA but by the bearing manufacturers. This is because excessive shaft deflection will lead to premature bearing failure in the idler roll. The limit for tapered roller bearings is two to three minutes; while for ball bearings the limit is ten to 15 minutes, depending on the exact geometry of a bearing (Fig. 4). The shaft may be larger due to the need to have "equally" sized bearings — equal in C rating, not shaft size. Another factor that can affect this design is the moment arm, the distance from the support to the loading point of the bearing. By keeping this distance small, the shaft stress will be reduced and so will the slope of the shaft at the bearing. One advantage of the tapered roller bearing is the location of the load point. In Fig. 5 one can see that the load point is not at the center of the bearing but is at a point towards one side. This dimension is given in the tapered roller bearing catalog, and the idler manufacturer will turn the bearing so that it faces out towards the supports. This will reduce the moment arm of the applied load, reducing the stress and the deflection of the shaft.

It might seem that tapered roller bearings have a distinct advantage. But each bearing has its own particular advantages and disadvantages — ball bearings are more forgiving on slope, tapered roller bearings can be designed with less moment arm, etc. It is up to the manufacturer to incorporate the advantages each bearing has into the design of the product.

FIGURE 4

Shaft deflection in ball and tapered roller bearings.

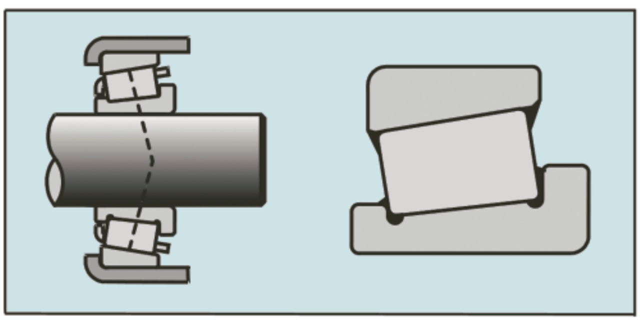


Conclusion

The shaft for pulley and the idler must be designed to handle stress and deflection criteria. The deflection criteria are needed to provide ample life for the product carried by the shaft, not necessarily the shaft itself. This is and will remain the case unless one uses a connection system that will align itself dynamically to the shaft. ■

FIGURE 5

Tapered roller bearing geometry.



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