Effects of Drive Assembly-Overhung Loads on Belt Conveyor and Pulley Design

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The author has seen an increased use of shaft-mounted, right angle drive assemblies. These designs use the pulley shaft as its "foundation" and can create significant overhung loads. To reliably perform, the pulley and shaft must be designed for these special requirements. This paper will focus on the development of overhung loads and their design effects. Special considerations to maximize performance will be presented and comparisons with other drive options will be made. Experience in heavy mining applications will be used throughout.

INTRODUCTION

Most bulk solids conveyors use electric motor(s) to deliver and/or remove power from the system. Typically the motor is connected to a speed reducer, which transmits power to its drive pulley and eventually to the conveyor belt. In this power transmission system each drive pulley, and associated shaft, must redirect the belt, resist the belt tensions, transmit the driving torque, and resist the reactions from power transmission attachments.

Belt redirection, belt load, and driving torque effects on pulley stresses have been sufficiently detailed by others (Lange 1963; Schmoltzi, 1974; Qiu and Sethi, 1993; Reicks, 1996). This discussion focuses on the reactions from power transmission attachments on a pulley shaft. A representation of these reactions, typically called "Overhung Loads," is shown in Figure 1.

A conveyor motor's power can be transmitted to a drive pulley shaft in many ways. Usually some type of speed reducer is attached between the pulley shaft and motor; although, chain and sprocket, direct mount motors, tires, bull gears, and other shaft attachments are used.

Figure 2 shows a typical parallel shaft mount reducer configuration. Our experience suggests this configuration is frequently used on smaller conveyors requiring 150–250 horsepower or less.

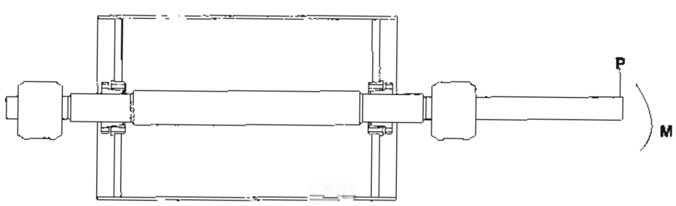

FIGURE 1 Basic definition of overhung shear (P) and over hung moment loads (M)

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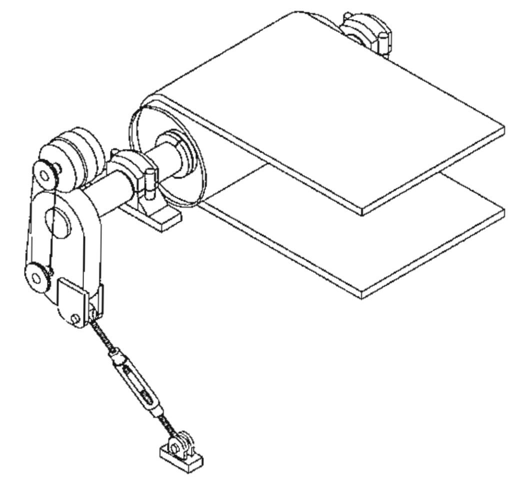

FIGURE 2 Representation of a typical parallel, shaft-mounted reducer drive assembly

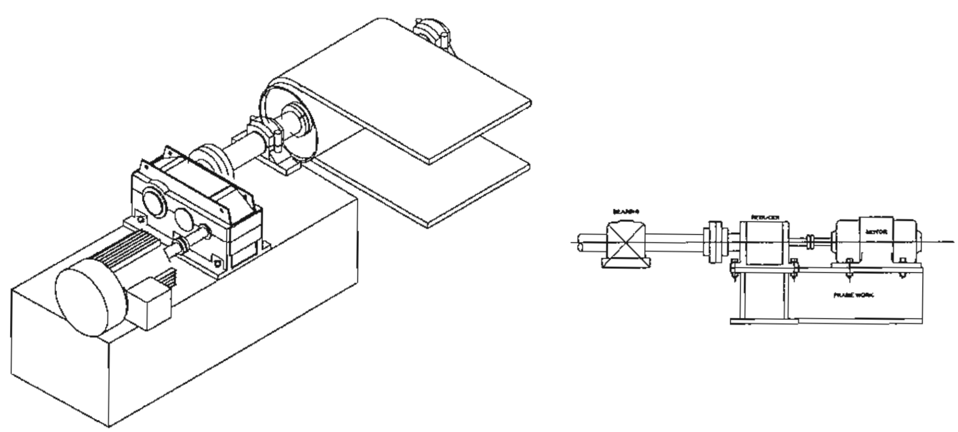

FIGURE 3 Representation of a typical parallel, fixed-base reducer flexibly coupled to a pulley shaft

This drive option creates overhung loads on the pulley shaft from its weight and the torque arm reaction. These loads can be important to the design, but due to its frequent use and successful history the author has chosen to omit design details.

When conveyors require more than 150–250 horsepower, our field experience suggests that foundation mounted parallel shaft reducers, as shown in Figure 3, have historically been the design of choice in the United States. With this option, the reducer is rigidly mounted to a foundation and flexibly coupled to the pulley shaft. Due to the flexibility of the low speed coupling, overhung loads are minimized. The author has seen cases where significant overhung loads were created when the coupling was severely misaligned.

A third option, the shaft-mounted, right angle reducer arrangement, is shown in Figure 4. The author has seen evidence this option has been popular outside the United States for some time and appears to be gaining popularity in the United States. We have observed a number of the highest horsepower conveyors of our day being designed with this arrangement. A typical right angle drive package is made up of a reducer, motor, and high speed coupling rigidly mounted to its own floating foundation, as shown in Figure 4. A variant on this design is to surround the high speed coupling with a structural torque tube that takes the place of the steel base, reference Figure 5. In either case, the right angle drive package is mounted to the pulley shaft and supported by a torque arm attachment. The shaft mounting is usually accomplished with

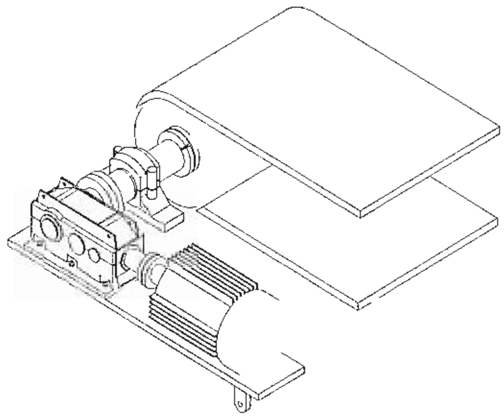

FIGURE 4 Representation of a typical right angle, shaft-mount reducer assembly rigidly coupled to a pulley shaft

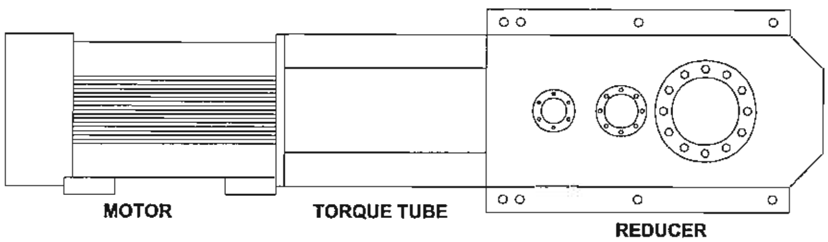

FIGURE 5 Side view of a typical torque tube variation to the right angle, shaft-mount drive assembly

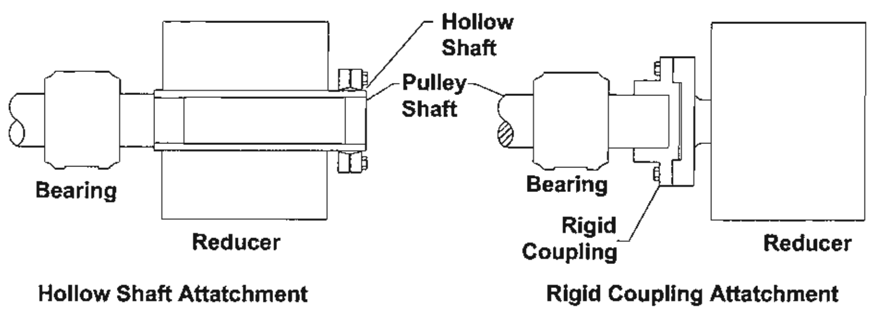

FIGURE 6 Typical pulley shaft to reducer mounting options

either a hollow shaft or rigid coupling attachment as shown in Figure 6. This drive arrangement can create significant "overhung loads."

When properly designed, right angle, shaft-mounted drive assemblies have proven to be a reliable, economical, and maintenance friendly power transmission option. The author has seen instances of pulley shaft and reducer failures due to improper consideration of overhung loads, use of fatigue sensitive attachment details, and improper torque arm designs. As the right angle, shaft-mounted drive assembly gains popularity, the author believes there is an increasing need for the understanding of its specific design considerations.

OVERHUNG LOAD ANALYSIS OF RIGHT ANGLE DRIVE ARRANGEMENT

Figure 7 shows the side view of a typical right angle drive assembly with each major component's center of gravity. Accessories such as cooling equipment, brakes, and backstops have been omitted. If these accessories are present, they can

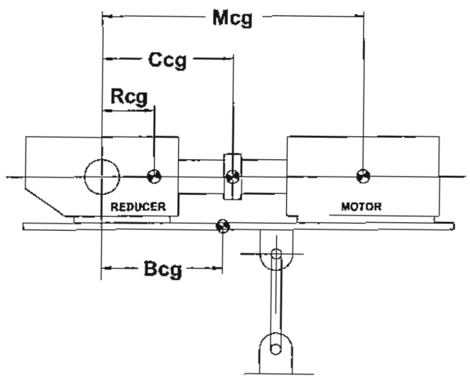

FIGURE 7 Sketch showing the side view of major drive assembly components and their associated centers of gravity

be included as additional components in the calculations. It is recommended that components having a center of gravity (CG) coinciding with the low speed shaft axis be excluded from this calculation, i.e., the low speed coupling. This protocol for total drive assembly CG calculation simplifies subsequent calculations. The total center of gravity is the summation of the product of each component's mass and center of gravity divided by the total mass.

The low speed shaft and torque arm must support the drive assembly weight and torsional reactions. Figure 8 shows a side view, free body diagram representing the drive assembly loads. The total mass (WT) and its center of gravity (CG), which were calculated earlier, are represented. Reactions at the low speed shaft (P), the torque arm (TR), and the torsional reducer output (M) are also shown. As will become evident in later calculations, the direction of the torsional moment (M) is quite important. In this free body diagram, it is assumed the direction of the torsional moment is that which acts upon the hollow shaft inside diameter or the rigid coupling half nearest the reducer.

In Figure 8 the torque arm reaction is assumed vertical, which is valid when the preferred torque arm design is used, reference Figure 12. In certain cases it is possible to get a horizontal component to the torque arm reaction. These conditions typically occur when "Z" approaches the magnitude of "T", in Figure 8, and the torque arm design creates restraint in more than one degree of freedom. An example of this situation is a torque tube design, as shown in Figure 5, with a rubber bushing and pin through the reducer flange acting as the torque arm. In these cases a horizontal component to the torque arm reactions should be calculated.

To solve for "TR" in Figure 8, it is recommended that one sums the moments about the low speed shaft axis, equates them to zero, and solves for "TR". This results in the following equation for the torque arm reaction:

$$TR = (M + \text{Total CG} \times WT)/T.$$

"TR" can now be used in the end view, free body diagram of Figure 8, which represents the drive assembly, low speed coupling, and pillow block bearing. It is important to note that dimension "C" may not coincide with the reducer/motor rotational axis. Backstops on intermediate stage reducer shafts and electrical enclosures are examples of items that may cause "C" to move from the axis of rotation. To isolate this part of the analysis from the belt loads, a fixed boundary condition is assumed at the bearing centerline, which is valid for shear and moment calculations from the bearing centerline to the reducer. Using the end view, free body diagram one can pick a location of interest and calculate its shear force and/or bending moment.

Overhung loads cause reactions between the bearing centers, which are important to pulley and shaft design. Interaction between overhung and belt loads is complicated by the fact that they are often out of phase. Using vector summation, these loads can be superimposed. The author defers to explanation of these methods in other works (AS1403, 1985; CEMA B105.1, 1990).

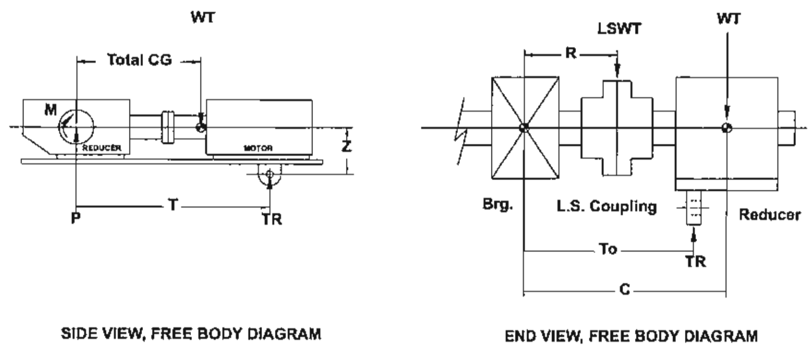

FIGURE 8 Free body diagrams for overhung load analysis

OPTIMIZATION OF A RIGHT ANGLE DRIVE ASSEMBLY

There are a number of critical design considerations specific to the right angle drive assembly shown in Figure 4. An example will be used to illustrate drive assembly considerations having significant impact on the conveyor pulley and its shaft design.

This example, which is similar to a South American hard rock mine application, considers a head discharge drive assembly as shown in Figure 4. At normal running conditions the drive is designed to deliver 480,000 in-lb of torque. A fluid coupling is used which limits the peak torque to 150% of normal running. For the first three production years the conveyor is expected to draw only 50% of normal running torque. As defined earlier, the torsional moment's sign will be positive in this case. All other dimensional and weight specifics are as follows:

Drive Assembly Weight (WT) = 17,000 lbs.

Drive Assembly CG (Total CG) = 65 in.

Low Speed Coupling Weight (LSWT) = 500 lbs.

Low Speed Coupling Centerline to Bearing Centerline (R) = 15 in.

Reducer Centerline to Bearing Centerline (C) = 43 in.

Torque Arm to Bearing Centerline (To) = 43 in.

Torque Arm to Low Speed Shaft Axis (T) = 182 in.

Figure 9 gives the overhung load analysis results at the bearing centerline. One important concept illustrated in Figure 9 is that the highest torsional moment, i.e., the highest horsepower, may not coincide with the worst case overhung loads. In this case the bending moment drops as the horsepower increases. To be sure the worst case scenario is used, the author recommends analyzing the full torsional moment range.

The bending moment results, shown in Figure 9, can be used as inputs into a rotating shaft design code (CEMA B105.1, 1990; ANSI/ASME B106.1M, 1985; AS 1403, 1985) to calculate the minimum shaft diameter for the location. When stress concentration effects (Pilkey, 1997; Peterson, 1974) at bearing attachments, coupling attachments, changes in diameter, and key slots are considered; the required shaft can become quite large. The author has seen a number of cases where overhung loads have created additional expense by requiring larger bearings than would otherwise be required. In the author's opinion, many of these additional expenses can be avoided by attention to torque arm location (T) and consideration of stress concentrations created by shaft attachment details.

Careful selection of the torque arm location (T) can significantly lower cost and increase reliability of a conveyor pulley shaft. To illustrate, consider the example presented earlier with torque arm location at 100 inches rather than the original 182 inches. Figure 10 shows results for this new scenario. Comparison of Figures 9 and 10 reveals that the new torque arm location significantly reduced the overhung loads. Using this method to minimize the overhung loads can help keep the pulley, shaft, bearings, and coupling costs to a minimum. It can be argued this method can increase the torque arm and its associated structures' cost.

Torsional Moment (M) (in-lb)	Torque Arm Reaction (TR) (lb)	Bending Moment at Bearing Centerline (in-lb)	Overhung Load Shear Component at Bearing Centerline (lb)
0	6,071	477,429	11,429
240,000	7,390	420,725	10,110
480,000	8,709	364,022	8,791
720,000	10,027	307,319	7,473

FIGURE 9 Shear and bending moment results at the bearing center line for an overhung load example

Torsional Moment (M) (in-lb)	Torque Arm Reaction (TR) (lb)	Bending Moment at Bearing Centerline (in-lb)	Overhung Load Shear Component at Bearing Centerline (lb)
0	11,050	263,350	6,450
240,000	13,450	160,150	4,050
480,000	15,850	56,950	1,650
720,000	18,250	-46,250	-750

FIGURE 10 Overhung load results at bearing centerline. Results illustrate reduced bending moment by optimizing the torque arm location

Since fatigue failures often cause unexpected reliability problems, often the most reliable and cost effective solution is one that reduces dynamic loads. In this case the pulley shaft's bending load is dynamic, due to its rotation, while the torque arm reaction is relatively static.

Careful design of shaft attachment details can also lower cost and increase reliability. Figure 11 shows an example of a detail sensitive to overhung loads. In this case the coupling's interference fit and the small radii used for the diameter change both cause relatively high shaft stress concentrations. According to Pilkey (1997) and Peterson (1974), when two stress concentration features exist at the same location, the actual stress concentration is significantly higher than the individual values. Since these two features are in the same location, the sensitivity to a shaft fatigue failure, from the overhung loading, is significantly increased. A suggested redesign of this detail, as shown in Figure 11, reduces the fatigue sensitivity by using a larger radius at the diameter change and moving it away from the interference fit. According to Pilkey (1997) and Peterson (1974), use of a shouldered shaft and back relieved coupling design will reduce the interference fit stress concentration.

TORQUE ARM DESIGN

A proper torque arm design is critical to the reliability of a right angle drive assembly. The preferred torque arm design, as shown in Figure 12, remains rigid in the vertical axis to carry its load, while maintaining flexibility in the other five degrees of freedom. This is important because no matter how accurately the drive components are manufactured there will be some relative motion between the drive assembly and the surrounding structure, i.e., the motor will wobble slightly when operating. The preferred torque arm design allows this small amount of motion to take place unrestrained, thus eliminating any unexpected restraining loads. If the drive assembly motion becomes restrained, an indeterminate amount of load will be created and may cause premature component failures. The author is aware of one drive component failure due to improper torque arm design on an ore handling conveyor operating in the Western United States.

It is typical for a right angle drive assembly to wobble slightly in operation. Analysis indicates this motion, when left unrestrained, doesn't create significant loads in the drive components. These results are primarily due to the relatively slow speed of the motion, and are supported by field experience. With this being presented, the author admits this was a challenge to remember when he was standing behind a 15,000-pound drive assembly oscillating 0.375 inch at the back of the motor.

It is believed the drive assembly movement is primarily due to component manufacturing tolerances and design clearances. If the conveyor user perceives this wobble as undesirable, they will need to specify manufacturing tolerances that can achieve the desired results and expect added cost for added manufacturing processes. One must keep in mind the drive motion can't be totally eliminated, but it can be minimized.

To understand the pulley and shafts contribution to the drive assemblies wobble, an explanation of the assembly process is helpful. During installation the pulley shaft is deflected slightly due to fabrication tolerances. When the shaft is rotated in the pillow block bearings, this deflection can be seen as a radial run out of the shaft extension past the bearing. Since the rest of the

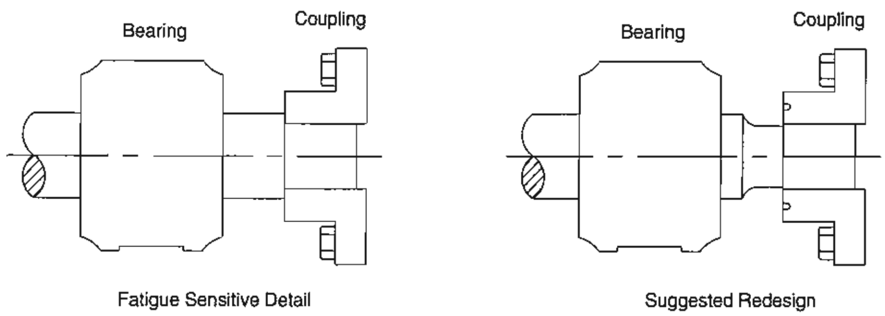

FIGURE 11 Example of fatigue sensitive features occasionally observed and suggestions on ways to reduce the fatigue sensitivity

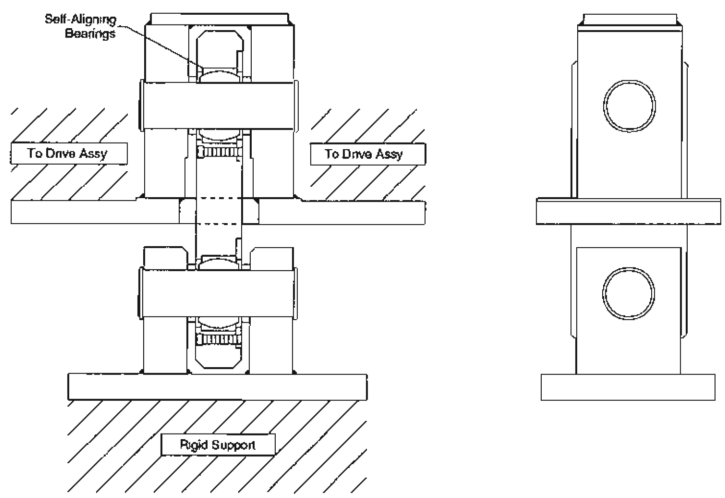

FIGURE 12 Example of a preferred torque arm design

drive components are rigidly attached to the pulley shaft, they tend to move with it. This small run out can contribute to the motion at the back of the motor. To illustrate, take the example assembly previously detailed and assume the pulley shaft is manufactured with a 0.001 inch of run out per inch of distance from the bearing center. This means that the pulley shaft's contribution to the wobble would be 0.043 inch at the reducer's low speed shaft and 0.182 inch at the torque arm.

The run out phenomenon is present, to varying degrees, with all pulley assemblies regardless of manufacturer. Most manufacturers have processes to minimize shaft run out, and they tend to be unique to each supplier. Without an industry standard for shaft run out it is difficult to know what each manufacturer's standard is, or if they have one. Until a standard is created, the best the author can recommend is to communicate your desire for minimized shaft run out, require your pulley supplier to guarantee the tolerance they state, and consider this in your pulley selection decision.

CONCLUSION

The right angle drive assembly appears to be gaining popularity in our industry. It has proven to be a safe, economical, and easy to install power transmission option in many of the largest mines around the world. Due to its unique design the following application considerations have an impact on the pulley and shaft design.

- Because the design uses the pulley shaft as its foundation, pulley shaft overhung loads can be created and should be considered in the design analysis.

- It is suggested that all running conditions be analyzed since less than peak power usage may create the worst case overhung loads.
- Careful selection of the torque arm position can significantly reduce the overhung loads, which can increase the reliability of the drive system.
- Attention to torque arm design will reduce the likelihood of unexpected restraining loads, which can increase reliability of the drive system.
- Some relative motion of the drive assembly is expected due to manufacturing tolerances. This movement doesn't induce significant loads, but can be perceived as undesirable. Negotiating tighter than standard tolerances with component manufacturers can reduce this perception.

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