A Comparison Of Turbine Pulley Design Philosophies With Historical Perspective

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ABSTRACT:

In recent years, heavy mining applications have put an increasing demand on Conveyor Pulleys. This has resulted in the increased usage of Turbine Pulleys. The designs of these pulleys are often based upon the failures seen in the conventional welded designs, overlooking the interaction of pulley-shaft system components as the profile of the disc is modified. This paper will focus on the development of balanced design philosophy using case studies of turbine pulleys in heavy mining applications, evaluated with FEA.

INTRODUCTION:

In the past, conveyor belting was made with a cotton fabric carcass. Because of the limitations of this belting, (requiring multiple plies) pulley diameters were quite large when compared to the diameter of the shaft. This gave the pulley end disc significant space to flex, thus allowing the hub to follow the deflection of the shaft. The ratio, OD/SD, the pulley outside diameter divided by the shaft diameter, is a measure of the flexible nature of the end disc. Looking back, this ratio was approximately 6 to 8.

Today's belting is more flexible and stronger due to high strength synthetic carcass materials that provide higher PIW per ply rating, requiring fewer plies for the same tension and allow the belt to conform to a smaller pulley diameter. This permits reduced pulley diameters, at the same time that shaft size is being increased, due to the higher tensions thus driving this ratio smaller. Economics of conveyor design are playing an important part of this, because with smaller pulley diameter, the designer can reduce the cost of the drive package, lowering the cost of the entire conveyor. However, this will put more pressure on the pulley and its design

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The typical diameter ratio, OD/SD, has gradually decreased to today's typical ratio of four. This tends to be a defining number for welded steel and turbine pulleys alike. As this ratio gets smaller, the pulley tends to be rigid. Specifically, any ratio four or under, the end disc will tend to be rigid. Any ratio over eight, the end disc will tend to be flexible. Ratios between four and eight vary, depending upon other factors in their geometry. Figure 1 shows a graphical representation of various stress curves for different ratios of pulley OD/SD. For any of these curves, anything to the right of the peak is considered rigid. Anything to the left would be considered flexible. Pulleys used to be in the range of five to eight, and their stress were fairly low. It was very easy to use standard plate materials and have a successful design. However, today we are dealing with pulleys that have a ratio of three to five. The effect is, regardless of the design philosophy, the pulleys are being driven towards being rigid. That is the paradox.

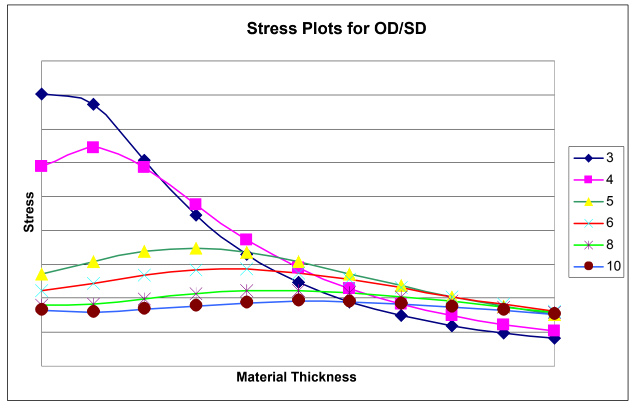

Figure 1

One of the features of turbine designs is to allow the disc to flex, reducing the transmitted bending moment through the locking assemblies. The keyless locking assemblies were originally designed to transmit torque without a key. Because they were not designed to handle a bending moment there were some problems encountered when they were used in a conveyor pulley. Since that time several companies have built keyless locking assemblies increasingly wider, in order to handle more bending moment. However this is still the suspect area of turbine pulley design, as we are unsure of the ultimate bending strength of these assemblies.

SAMPLE CASE:

The old adage that "More is Better" is another paradox. Because the thicker the end disc, the more rigid the end disc. While this may give the end disc a greater life, it can lead to premature failures in the locking assembly or any shaft clamping mechanism. One such example is 1220 mm X 2080 mm (48" x 82") pulley with RBH300 hub and 300 mm (11.811") shaft diameter. The end disc is 165 mm (6.5 inches) thick at the hub that tapers down to 76 mm (3") at the rim. The rim is 30 mm (1.1875") thick. The FEA mesh and cross section plot of the radial stress in the turbine disc is shown in Figure 2. This pulley has an OD/SD ratio of four. The load on this pulley is 33,500 Kg. (74,000 lb.) of tension with 180 degree of wrap. The bending moment being applied to the locking assembly is 120,600 N-m (88,900 ft-lb). This is within the prescribed limits Schmoltzi (1974). Reviewing Figure 2, we can note the maximum radial stress in the end disc is 12 Mpa (1,750 psi). This is quite low, several times lower than the allowed stress for this type of application. It is even lower than the principal stresses in the rim of 17 Mpa (2,500 psi). Which is opposite of conventional wisdom, because the rim undergoes three cycles of stress for every revolution, while the end disc undergoes one cycle of stress for each revolution. This would indicate for a balanced design, the rim stress should be lower than the cyclic end disc stress. (Note: The FEAs shown do not include the load from the hub pressure, only the cyclic load from the belt.)

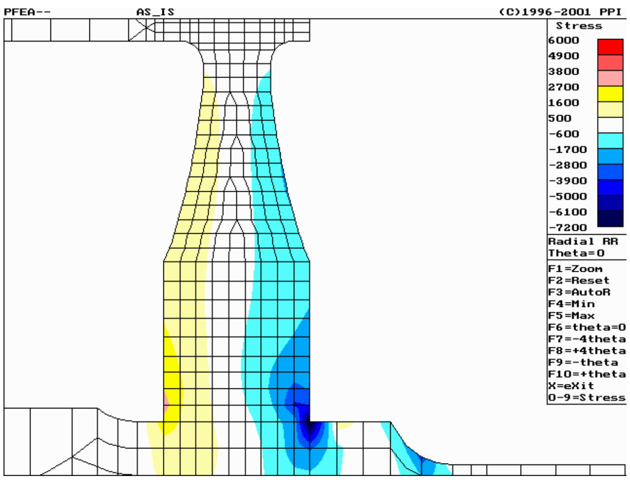

Figure 2

The bolts were starting to break after four years of service, even though the bending moment being transferred through the locking assembly was within acceptable standards, Schmoltzi (1974). This situation has two possible solutions, make the shaft more rigid, or make the pulley less rigid. Since the pulley is intact, and modifying it would be impractical, the problem is normally addressed by increasing the shaft size between the hubs and/or reducing the bearing centers.

REVIEWING THE TURBINE DESIGN:

The other option, reducing the stiffness of the pulley, involves a more detail analysis. Because, changing the basic parameters, such as OD is often impractical, due to the existing structure. However, if we could change the disc profile, we might find a practical solution. The disc, as it exists, is quite rigid, even though it does have a taper. One possibility is to reduce the end disc cross section from 165 to 120 mm (6.5 to 4.75") at the hub, and 76 to 57 mm (3 to 2.25") at the rim (Figure 3). This increases the end disc stress from 12 MPa (1,750 psi) to 15.5 MPa (2,250 psi). This would reduce the amount of bending moment from 120,600 N-m to 118,000 N-m.

If we take another step and drop this down to 89 mm (3.5") end disc tapered to 38 mm (1.5") at the rim (Figure 4), we increase the stress to 150 MPa (3,800 psi). Which is significantly more, but well within the parameters for a good design and a long life, Wolf (1998). The transmitted bending moment would drop to 110,300 N-m about a nine percent drop in load and a significant increase in life of the locking assembly.

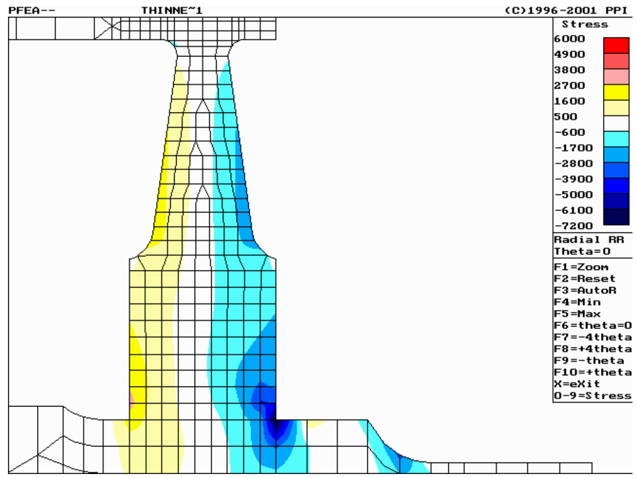

Figure 3

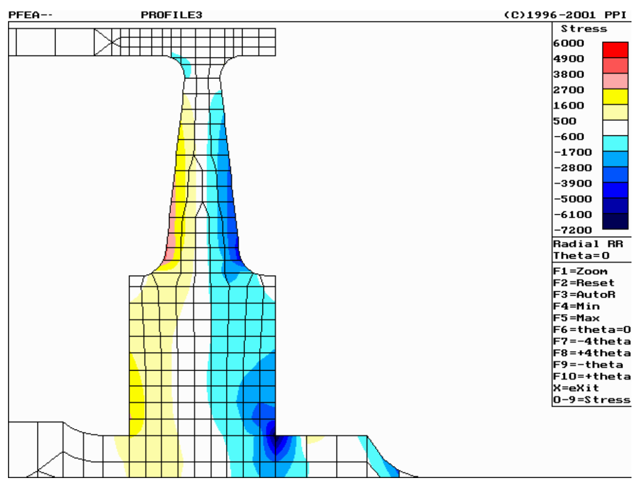

Figure 4

How does this effect the economics of manufacturing the pulley? Since the vast majority of turbine discs are machined from thick plate, the thinner the disc profile, the more it costs, e.g. more material to machine. This is opposite of the conventional drum pulley, where the thicker the end disc, the more it costs. One way to control the cost of the pulley is to use a thinner plate. This would reduce the cost of the raw material, as well as reducing the amount of steel that would have to be removed in the machining process.

However, this can cause a different problem. The Turbine-T is the preferred method for all turbine pulleys. This is due to the fact that it moves the rim to disc weld away from the stresses in the transition area between the disc and the rim. If a thinner plate is used to produce the Turbine, then it will move the circumferential weld closer to this transition region, and into a higher stressed area.

The effect of the different end disc profiles on the transition area can be see in Figure 5, 6, and 7. These figures show the axial stresses in the transition area for the three different disc profiles. This is the principal stress that is perpendicular to the weld section, Reicks (1996). In Figure 5 we can see that the original section causes little bending stress in the transition area.

As we reduce the thickness through the end disc section, the disc will deflect more, and cause higher stresses in the transition area. While these stresses are significant, they are not unreasonable, as long as the weld is not in the immediate area of the transition. If we reduce the thickness of the turbine disc, then it can make the design undesirable, which would move the weld into too high of a stress region.

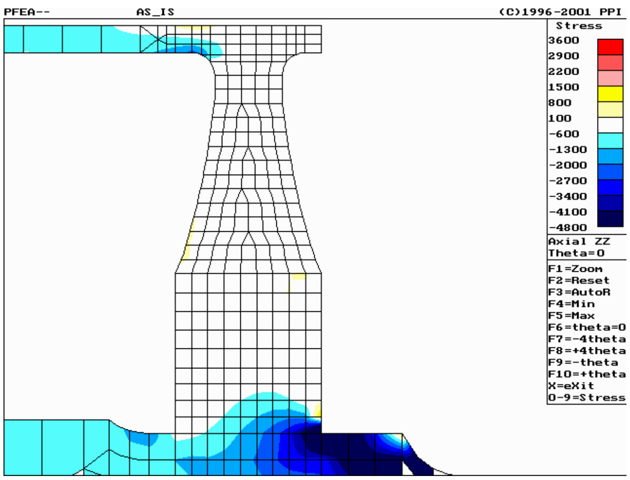

Figure 5

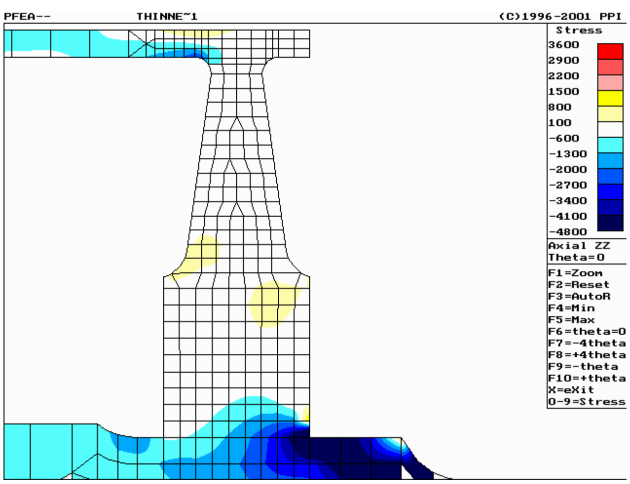

Figure 6

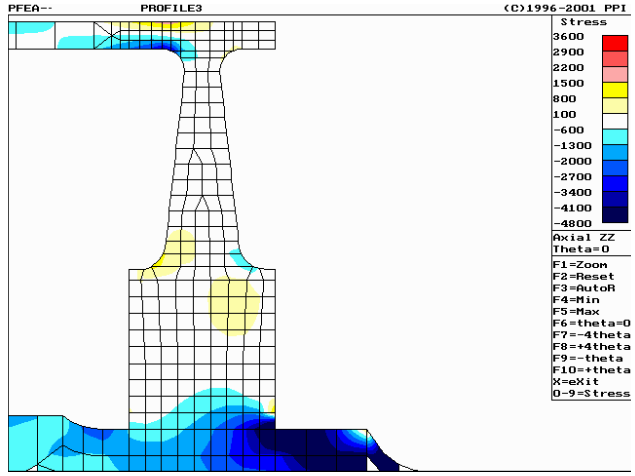

Figure 7

CONCLUSION:

Stress is not the only factor in designing a conveyor pulley. The deflection and balance of the design is important. This case shows how an increase in rigidity of the pulley can lead to problems in another aspect of the pulley. While the transmitted bending moment was within acceptable parameters, it was too high for a satisfactory service life.

For pulley designers, the economic pressure is to make the turbine thick, reducing the amount of machining done, and to increase the service factor of our product. However, we must look to the entire assembly and reduce the bending moment that the designs require the shaft clamping mechanisms to transfer. And look for better shaft clamping mechanisms or improve the current ones.

Customers need to realize the design limitations that their requirements are having on the designers and the product. Today's pulley designs require a different philosophy than what has been acceptable in the past. In some cases, the best design option may be to change the pulley diameter, however this may not always be practical.

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