A Comparison of Welded Steel Hub/End Disc Design to Profile Hub/End Disc

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ABSTRACT:

For years Conveyor Pulleys have been constructed of an end disc welded to a hub. Heavy mining applications have put an increasing demand on these Conveyor Pulleys. While this has resulted in the increasing usage of Turbine Pulleys, it is not always cost effective to use Turbine Pulleys on every conveyor. Profile disc represents a cost effective alternative to Turbine Pulleys, and an improvement on the conventional welded hub/end disc. This paper will focus on comparing pulleys built with welded hub/end disc to one with a profile hub/end disc using FEA (Finite Element Analysis).

INTRODUCTION:

Many of the conveyor pulleys used in heavy mining applications utilize an end disc welded to a hub. This method of pulley construction is quite versatile as the manufacturer will burn an end disc from steel plate and weld it to a standard hub. This method of construction has its limitations as the weld that joins the hub to the end disc is the highest stressed point in the pulley.

Figure 1 shows an FEA (Finite Element Analysis) stress output of the radial stresses in a welded hub-end disc pulley. The concentration of stresses in the end disc adjacent to the weld shows the highest stresses and is cause for the majority of pulley failures, such as shown in Figure 2. However, high stress is only one aspect of the problem. The shape of the weld plays an important part, Reicks (1996). The profile or blending of the weld to the end disc can dramatically affect the stress concentration factor at this critical junction.

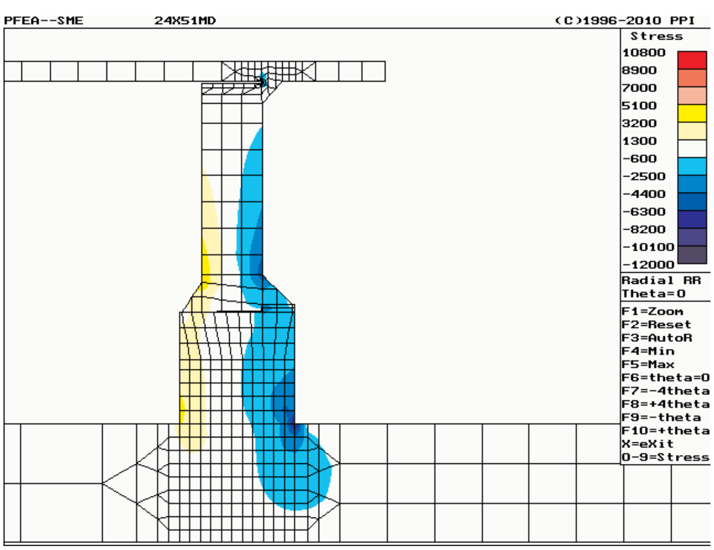

Figure 1 – FEA Radial Stress of A Welded End Disc Drum Pulley

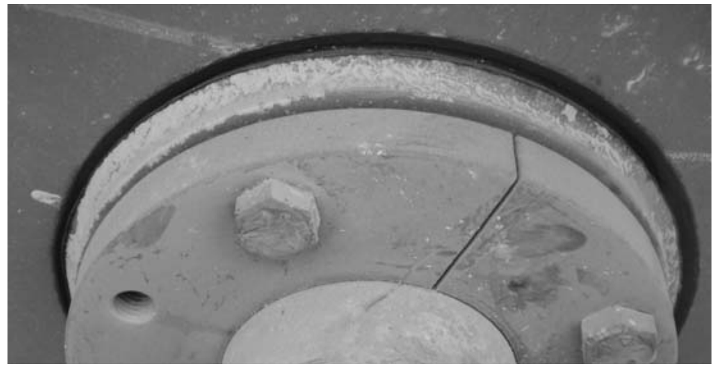

Figure 2 – Example of End Disc failure at the Hub Weld

The welding of this joint will also leave residual weld stresses in the joint. As the weld cools it will shrink and pull at the joint. It has been shown, Wolf (1998), that Thermal Stress Relieving will relieve these stresses and increase the life of the pulley by a factor of approximately five. Thermal Stress Relieving has become a standard addition to Turbine and large welded steel pulleys, however it would be a challenge to do on a large scale to all conveyor pulleys and it would do nothing to change the heat affected zone in the end disc.

Re-welding of the crack does not work, as the end disc will crack next to the new weld as shown in Figure 3.

Figure 3 – Example of Re-Weld Failure at the Hub Weld

PROBLEM:

Pulley geometry is a key factor in designing a pulley. Pulley diameters are often determined by the belt manufacturer and the conveyor designer and not the pulley designer. In the 1970's pulley diameters were 6 to 8 times the shaft diameter. As illustrated by Figure 4, these pulleys did not present a problem as

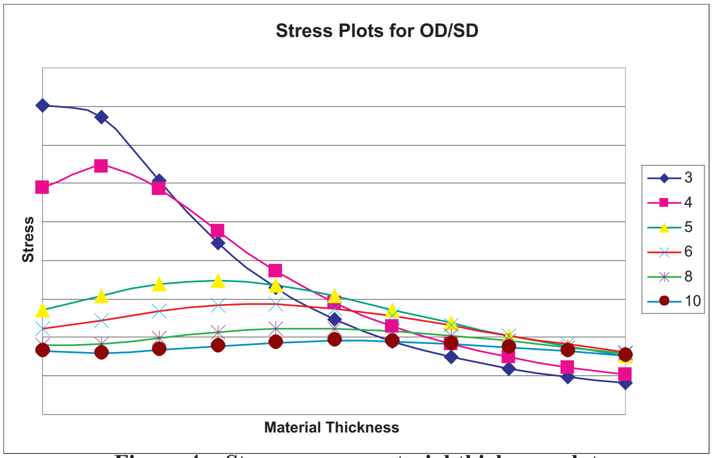

Figure 4 – Stress versus material thickness plot for various values of the Pulley Outside Diameter (OD) divided by the Shaft Diameter (SD)

most end discs worked well. However, today's belts have higher ratings and can wrap around smaller pulleys. As a result, today's pulley diameters are usually 3 to 5 times the shaft diameter. This adds to the design dilemma. Those pulleys with a diameter of less than 3 times the shaft diameter present a problem in manufacturing and will typically be of an integral hub design, where there is no end disc.

With advances in design, we can use FEA and Life programs, Wolf (1998), Laughlin (2006), to design a pulley that will handle these limitations. Even then, overloads can be a problem. A recent mine visit revealed that the customer applied a service factor to their counterweight. Rather than applying it to the tensions they applied the service factor to the counterweight increasing the counterweight beyond the design load of the pulleys. Other times the weight box is left open allowing material to fall into the box resulting in higher loads over time. While others think that more weight is better, a simple 10% increase in the running tensions can lead to a 10 fold reduction in the pulley life, Wolf (1998).

Some have been increasing the end disc thickness beyond what is needed to address the issues of overloads. Heavier end disc will lower the stresses in the end disc and increase the life of the pulley. This will also result in the pulley becoming more rigid. The problems involved with pulleys using keyless locking assemblies that are too rigid are well documented, Laughlin (2002).

Similar problems also exist with tapered adapter mount systems. Even though these systems were designed to handle high bending moments, they do have limits. Early bushing problems caused by a very rigid end disc can manifest themselves as a squeak which is a prelude to fretting corrosion. More often the pulley will walk, as shown in Figure 5, or wallow out

Figure 5 – Example of a Pulley that has "walked" on the shaft.

Figure 6 – Example of a Bushing that has been "wallowed".

the inside of the bushing as shown in Figure 6. In the extreme case, i.e. the bushing is driven in and locked into place without changing the design of the assembly, the shaft can break as shown in Figure 7.

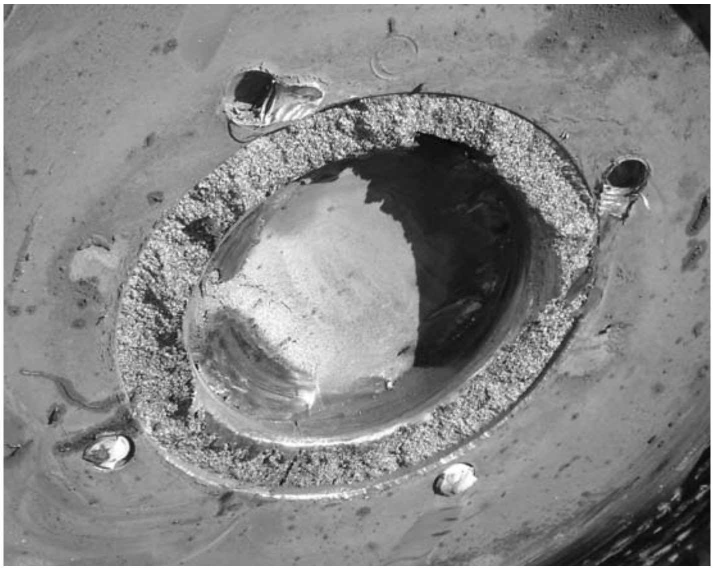

Figure 7 – Example of a Pulley with a Failed Shaft..

ANALYSIS:

While the pulley analysis would suggest that pulley diameters be limited to a minimum of 6 times the shaft diameter, this is not practical. It would increase the cost of the reducer as it would require a larger reduction and it would cause problems with underground mining as space is a premium.

Staying with a welded end disc pulley design presents a dilemma for the engineer. If the end disc is too thin there is a significant risk of end disc fracturing next to the hub weld. If the end disc is too thick then it will put too much load on the shaft connection which can cause problems. Often there is a fairly narrow window of end disc thickness that will meet all conditions.

One of the challenges faced is knowing that the running tensions are accurate. While momentary overloads are included in the design, the design has limited allowance for a steady state overload. If this overload factor is included in the end disc design only and does not include the shaft, the pulley will often become too rigid for the shaft connection. The option is to apply the service factor to the whole pulley assembly including the shaft by increasing the diameter at and/or between the hubs. However, this is not always acceptable as it does add cost.

There are two inherent problems with a welded end disc design, first the weld, second the disc. The weld adds weld stresses, heat affected zone, and a notch effect. The best improvement for the hub to end disc weld is to not weld. Next, the disc is cut out of plate steel and it is constant thickness, as shown in Figure 1, but the stresses are not constant and decrease as the disc radiates outward to the rim. If the cross section (thickness) of the disc is reduced along the radius as the disc radiates out away from the hub, we could do so without any increase in stress. Then the end disc would be able to flex and not overload the shaft connection. Figure 8 is a FEA plot of the radial stresses for a Profile end disc design.

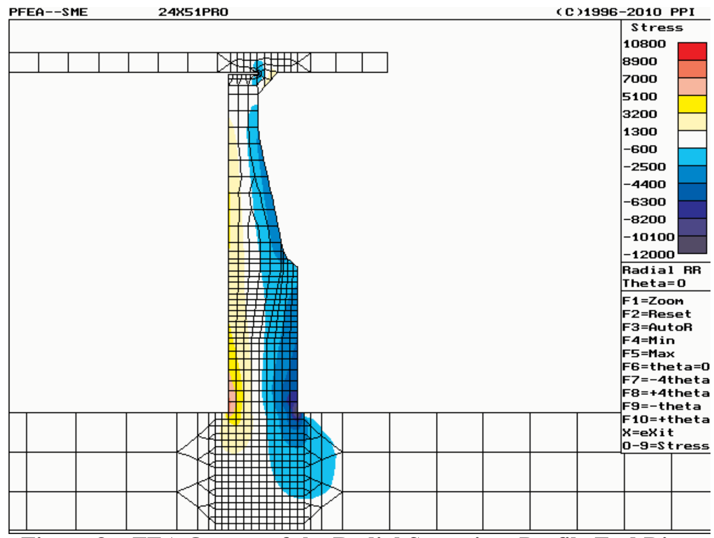

Figure 8 – FEA Output of the Radial Stress in a Profile End Disc.

DEFINITIONS:

Turbine *– end disc thickness varies with the diameter.*

Profile *(turbine) – the end disc variation is on one side*

Turbine-T *– the section that joins the rim to the disc is machined into the disc to form a T.*

COMPARISON:

FEA runs were performed on a 610 mm x 1295 mm pulley (24” x 51”) with a 152 mm shaft (6”). This represents an OD/SD ratio of 4 and is typical of a mining conveyor pulley. FEA was performed with 38 mm end disc (1.5”) and a profiled end disc that was tapered from 38 mm down to 19 mm (1.5” to 3/4”). A load of 107,000 N (24,000 lb) was applied. Figure 1 and 8 are the radial stress output from the runs. Table 1 is a summation of the results of the 2 runs.

Table 1 – Welded versus Profile comparison

	Welded	Profile	%
Hub Moment	10,525	7,775	-26
Disc Stress	12,050	7,894	-34
Shaft Deflection	0.0142	0.0213	50
Pulley Weight	888	765	-14

The Profile design allows the end disc to deflect with the shaft reducing the stress in the end disc and the bending moment that is transmitted through the shaft connection. Figure 9 compares the stress in the end disc as a function of the disc diameter.

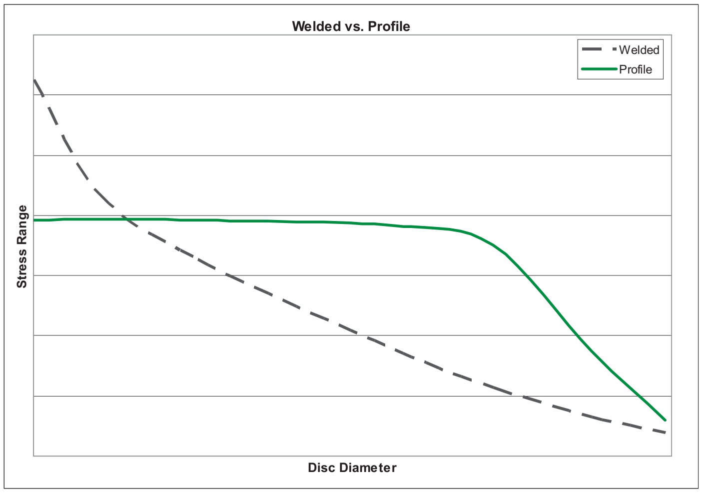

Figure 9 – Comparison of the radial stress in welded vs. profile disc.

Figure 10 is a FEA deflection plot of a profiled end disc. While the stresses in the Profile end disc are less, the deflection is more. This design allows the end disc to deflect and absorb the load, reducing the load on the shaft connection.

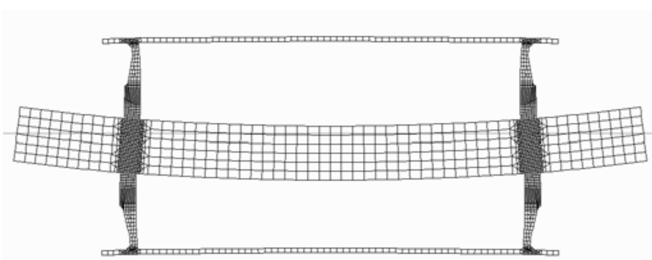

Figure 10 – FEA Deflection Plot of a Profile End Disc.

Figure 11 is a FEA deflection plot of a welded end disc. The pulley dimensions and loading are the same. The difference is the end disc construction.

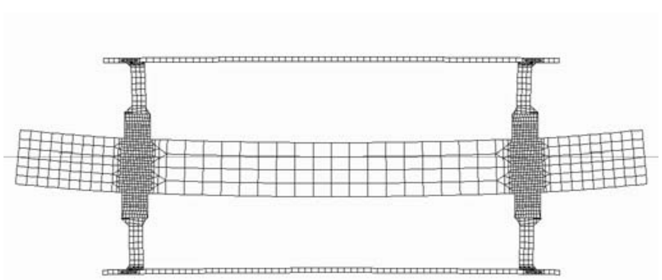

Figure 11 – FEA Deflection Plot of a Welded End Disc.

Profile end discs will reduce the weight of the assembly, while improving the performance. The reduction in weight of 14% is typical.

Figure 12 – Shaft Failure at 380% of Design Load

FIELD CASE:

In March, 2010 a Profile Pulley 762 mm x 1980 mm (30" x 78") with a 138 mm (5.4375") shaft was supplied to a mine. The pulley was designated as a drive snub with 30 degree of wrap. The shaft broke in April, 2011, shown in Figure 12 & 13. A review of the conveyor layout showed that the actual belt wrap was 194 degrees. That meant that the pulley was overloaded by 380%.

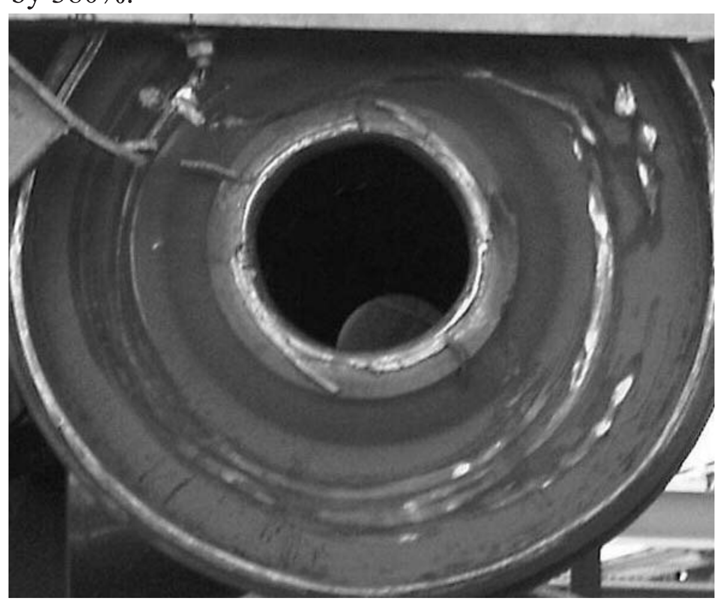

Figure 13 – Pulley after Shaft Failure at 380% of Design Load

If this pulley had been a welded end disc construction, it would have only lasted hours before the end disc would have failed. Instead it was a profile design that lasted months and even then it was the shaft that failed and not the end disc.

CONCLUSION:

The amount of hub moment reduction varies and is usually seen from 25 to 50%. While any size pulley can see a reduction in hub moment an OD/SD ratio greater than 8 can achieve a satisfactory life with a welded steel end disc. Those with an OD/SD ratio of 4 to 6 see the greatest life improvement. In addition, any pulley with an OD/SD ratio of 3 to 8 will see an improvement in product life from a profile end disc.

As the loads and shaft increase, the stress will rise in the rim to disc weld. This area will become an increasing concern and will need FEA to check for proper design. This is normally not an issue for profiled disc in shafts up through 250 mm (10"). Shafts over 360 mm (14") usually require a Turbine-T Design to handle the rim to disc connection. For pulleys with shafts in 250 mm to 360 mm (10" to 14") range will require a detailed analysis of the rim to end disc connection.

Stress is not the only factor in designing a conveyor pulley. The deflection and balance of the design is important. An increase in rigidity of the pulley can lead to problems in other aspects. A Profile end disc design brings Turbine technology to all pulleys. It is a cost effective alternative to welded end disc design that can improve the service life of conventional pulleys and provide a stronger than shaft design.

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