CONVEYOR PULLEY DESIGN

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INTRODUCTION

Conveyor pulleys are essential components in all conveyor belt systems, which are used throughout the mining industry. The larger conveyor belts are normally major arteries of the mining process. Conveyor pulleys and belts are critical items, the failure of which can result in substantial downtime costs, damage to local equipment, or injury to personnel in the area.

A conveyor pulley is a relatively simple structure, the type dealt with here consisting essentially of a drum mounted on a shaft. The shaft has bearings at each end, and the drum has ends that are fixed to the shaft by means of locking mechanisms. Figure 1 shows a typical conveyor pulley.

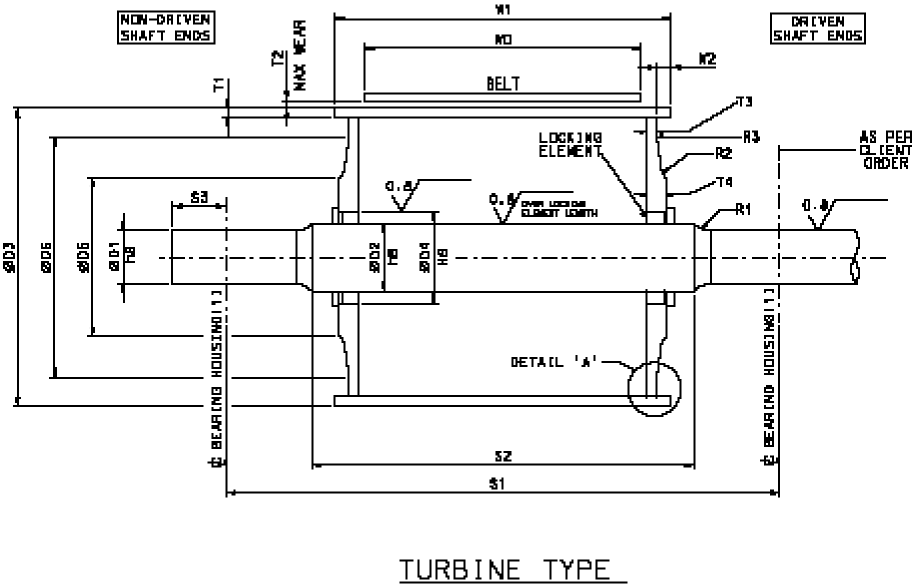

Figure 1. Typical Pulley

There are many variations possible in the details, in particular the end disc to drum weld details, the means of fixing the end discs to the shaft, as well as the bearing arrangements. Conveyor belts have a number of pulleys and have different functions: such as drive pulleys, return pulleys and deflection pulleys. The different pulleys on a given conveyor will typically carry different loads.

DESIGN BY FINITE ELEMENT ANALYSIS

Despite having a relatively simple structure, pulleys are problematic to design because it is not possible to reliably predict the stresses throughout the pulley other than by finite element analysis[1] (FEA). Most finite element programs can only model a pulley by means of a three-dimensional construction which is fairly time consuming to construct, and may also be time consuming to run (i.e. solve) once the model is constructed. Once the results are available it is then often necessary to make some changes to the design because, for example, stresses

[1] Finite element modelling and analysis is a technique requiring specialised software that can be used to build geometric models of equipment, input the material properties, describe the loads applied, and then have the software determine the resulting deflections and stresses occurring throughout the model.

may be too high in some region. The whole procedure then has to be repeated over again until a satisfactory solution is found.

Pulleys are three-dimensional axi-symmetric structures. This means their geometry can be described by a two-dimensional profile, lying on an axis of rotation. This is shown in Figure 2, but only half the length of the pulley need be represented because of symmetry (the other half being a mirror image).

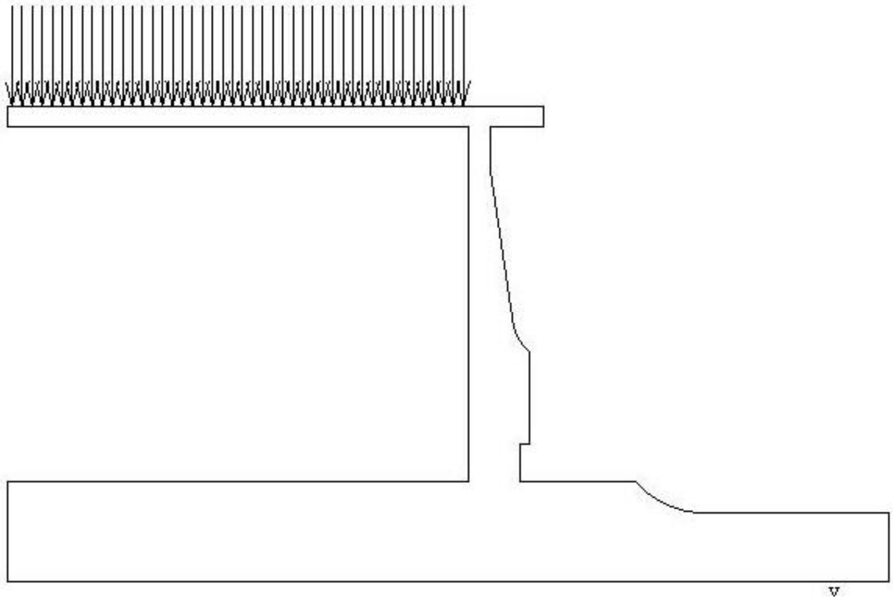

Figure 2. Profile for ½ Pulley

A few FEA programs have the capability of allowing three-dimensional axi-symmetric structures to be modelled using only a two dimensional profile such as above, but also providing for loads that vary around the circumference. Most FEA programs do not have this capability. This is what occurs on pulleys, as the belt pressure only acts over part of the pulley circumference. Modelling a pulley in this way, rather than in full 3-D, represents a large time saving in creating the model and in solution run-time. For the same reasons further time saving is effected if alterations to the model need to be made, for example in order to reduce stress levels.

Finite element stress contours in a typical cross-section of a T-Bottom pulley are shown in Figure 3. Higher stresses can be seen on the inside fillet radius. The level of stress in the fillet area can be controlled by the radius used, but the stress is also dependent on the shell and end disc thickness and belt pressure loading distribution on the drum.

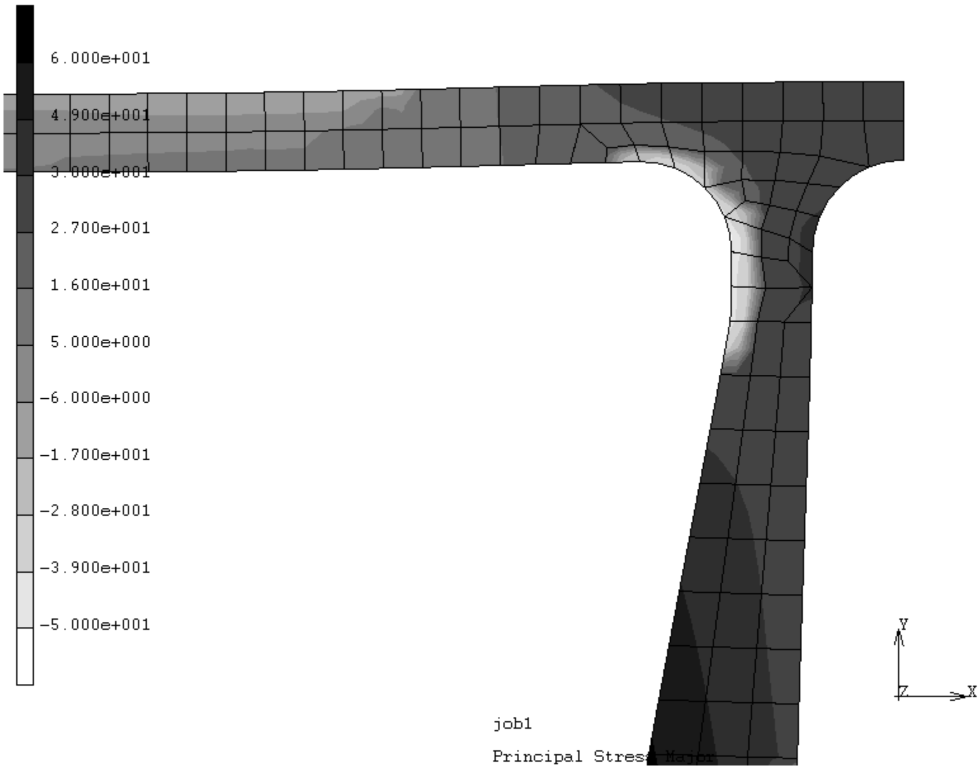

Figure 3. Typical stress distribution in a T-Bottom type pulley

DESIGN BY CONVENTIONAL METHODS

When pulleys are procured the purchaser typically specifies only the belt load and a basic size required, and relies on an experienced pulley manufacturer to design the details of the pulley. Herein lies a potential problem. Pulley manufacturers have competition. To win orders they cannot afford to over-design the pulley, but they also typically don't have finite element analysis software, or the expertise to operate it. Furthermore, doing an FEA adds to the cost. Pulley manufacturers have, over a lengthy period and at considerable effort, evolved in-house methods for designing conveyor pulleys, and do not have FEA done on pulley designs on a regular basis. The methods they use are based largely; it appears, on research done at academic institutions. Most of this work was done in the 1970's and 80's, before FEA became widely accessible, and the methods have mostly served the manufacturers satisfactorily.

Anglo American's Technical Division has recently found that the basis of the in-house methods is flawed. As part of an investigation into a recent pulley failure the design method used by the manufacturer was reviewed, and it was found that the stresses in the pulley calculated by the manufacturer are far higher than those actually occurring in practice (approximately double). This has been verified by finite element analysis. It was also found that the allowable stresses used by the manufacturer are far higher than those allowed by recognised and proven International Standards (also approximately double). It was further found during the failure investigation that the stresses in the pulley, determined by FEA, were too high at the welds, exceeding recommended stress range by 60% at the weld in one area.

The implication of this is that, despite the low incidence of pulley failures occurring, there is an unknown element of risk in pulleys designed by the pulley manufacturers. In essence, although the methods have been largely successful, they are not set on a fundamentally sound engineering basis. There is therefore no assurance that a pulley designed by these non-FEA methods will have stresses, in particular weld stresses, which are within safe recommended limits, only the likelihood that they are, based on past operational experience.

In the pulley failure investigation referred to above it was found that the pulley failed prematurely because of a weld flaw. This does not however change the findings regarding the design method used by the manufacturer. Had the weld been sound, the pulley would no doubt not have failed at that point in time, but the fact remains that the stress levels in the welds were too high and an unpredicted failure may well have occurred at a later time as a result.

FATIGUE AT WELDS

The stresses at the pulley welds referred to above are not of constant magnitude for a given pulley and loading. The stress at any point on a weld, and in the pulley in general, changes in magnitude as the pulley rotates. The magnitude of the stress change is directly related to the magnitude of the belt load. The changes in stress are the cause of fatigue, particularly at welds; the greater the stress change the sooner fatigue failure will occur. The stress range at any point is simply the difference between the maximum and minimum stress that occurs at that point as the pulley rotates. There is however, for each type of weld, a stress range below which fatigue failure is extremely unlikely to occur (the fatigue endurance limit), and for which indefinitely long life can be expected. One of the objectives in the design of a pulley should therefore be to ensure that the stress range at all welds does not exceed the recommended allowable stress range.

BS 7608 *Fatigue design and assessment of steel structures* [1] Is a well established International Standard that gives the allowable stress range for different types of welds. If the stress range levels given in BS 7608 are not exceeded, the weld is extremely unlikely to fail by fatigue, provided the weld is sound. By running an FEA on a pulley design one can find the stress range occurring in operation under a given load at any point or weld in the structure. It is therefore only necessary to identify the type of weld in BS 7608 that is being used in a pulley design, read off the allowable stress, and then to ensure that the calculated stress range occurring in operation does not exceed this value. The problem however is that currently the operating stress can only be determined by FEA, or by strain gauge measurement. As described above, the non-FEA methods typically used by pulley manufacturers give calculated stresses that are far higher than actually occur on pulleys. These incorrect predicted stresses therefore cannot be used to assess a weld against BS 7608 allowable values.

Typical welds on pulley drums only fall into two types in terms of BS 7608, namely:

1. Full penetration butt welds, as typically occur on all drum longitudinal seam welds, and on T-type end disc welds. Figure 4.
2. Double filleted full penetration welds, as typically occur on Turbine type end disc welds. Figure 5.

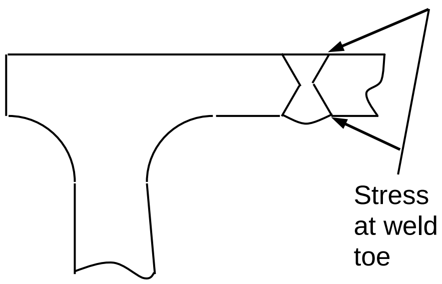

Figure 4. Full Penetration Butt Weld

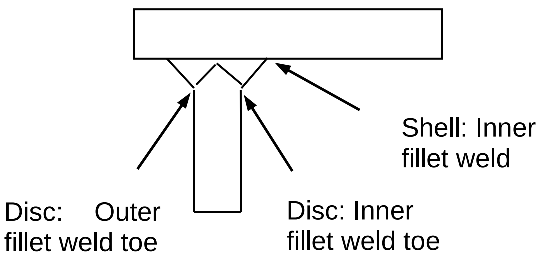

Figure 5. Full Penetration Double Fillet Weld

Allowable stresses given in BS 7608 for the welds in Figures 4 and 5 are shown in Table 1:

Table 1.

Weld Location	Weld Class	Type Number	Allowable Stress Range (MPa)
Stress along weld direction in: 1. Barrel longitudinal full pen weld 2. T-type end disc to barrel circ weld (With or without machining or grinding the weld reinforcements flat).	D	4.3	53
Stress perpendicular to weld in: 1. Barrel longitudinal full pen weld 2. T-type end disc to barrel circ weld (Class E upgraded to Class D if top & bottom weld reinforcement machined or ground flat.)	D	6.3	53
Stress perpendicular to weld in: 1. Barrel longitudinal full pen weld 2. T-type end disc to barrel circ weld (Top or bottom weld reinforcement NOT machined or ground flat)	E	6.3	47
Stress in barrel perpendicular to weld in: 1. Turbine-type end disc to barrel circ weld a) with full penetration plus fillet welds b) with partial penetration or fillet welds	F F2	5.2 5.3	40 35
Stress in end disc perpendicular to weld in: 1. Turbine-type end disc to barrel circ weld a) with full penetration plus fillet welds b) with partial penetration or fillet welds	F F2	8.1 8.2	40 35

The stress ranges that occur at welds under fully loaded operational conditions should be compared against the above allowable stress ranges, because it is the stresses under operational running conditions that can lead to fatigue failure. Start-up loads result in higher stresses but these have a negligible effect on fatigue life because of the relatively low number of start-up cycles. Under start-up conditions it is only necessary to ensure that yield stresses are not exceeded, with a suitable safety margin allowed.

Anglo American's Technical Division prefers not to use the Class F2 welds listed in the above table. Although easier for the manufacturer to fabricate Turbine type end discs with Class F2 welds, inspection for weld defects is more difficult than for Class F welds (i.e. with full penetration fillet welds).

For all of the weld types in Table 1 it is also necessary to ensure that the weld produced is sound, otherwise early fatigue failure may occur. Quality control and inspection of welds is therefore essential.

LOADS ON PULLEYS

Plant operators are often guilty of overloading belts and pulleys. This can also lead to early pulley fatigue failure and pulley detail drawings should contain a note stating the design running load. Conveyor general arrangement drawings and operating instructions should contain clear instructions regarding belt tensioning loads and maximum belt capacity loads.

Belts and pulleys can also be overloaded on one side due to poor belt tracking, and again drawings and manuals should clearly state targets within which belt alignment should be maintained in order to prevent this.

LOADS ON LOCKING ELEMENTS

Locking element manufacturers, such as Bikon, specify the allowable transmissible bending moment and torque for each locking element size and type. An FEA allows the bending moment transmitted by the locking element to be determined and directly compared with the manufacturer's allowable bending moment. Loads occurring during start-up should be used for comparison against the manufacturer's allowable loads.

The conventional rule of limiting the shaft deflection angle at the end disc hub to 5 minutes has no fundamental analytical basis, and is purely a rule-of-thumb. Anglo American's Technical Division recently designed a series of 108 pulleys using the two dimensional FEA approach described above. From the results it appears that the 5 minutes rule-of-thumb often leads to a conservative design at the locking element/shaft connection, but frequently does not, and so can lead to the locking element being overloaded. It was also observed that when bi-conical locking elements are used this rule-of-thumb is far too conservative, leading to oversized and therefore expensive shafts and locking elements.

STANDARDISATION

The risks involved in pulley design can only be removed by the application of finite element analysis. However this is impractical, due to cost and time constraints, unless a standardised range of pulleys is designed. This was is fact done recently by Anglo American's Technical Division and it is the intention that Anglo's operational divisions will, when applicable, ask pulley manufacturer's to quote and manufacture pre-designed pulleys detailed on a supplied drawing. There will no doubt sometimes be exceptions when sizes required do not fall within the standardised range, and a decision will then be made by Anglo on what approach to take.

Apart from the above, standardisation also results in a number of other benefits:

- The cost of pulleys should be reduced. Manufacturers will not have to cover the cost of designing pulleys and producing drawings.
- The design of each pulley is optimised, with direct cost benefits.
- Minimum technical and quality assurance requirements can be prescribed for the manufacture of pulleys. These will be the same for all suppliers, resulting in equitable adjudication of tenders.
- Manufacturers will not be under pressure to trim designs in order to win tenders.
- It ensures that pulleys are designed to a level of structural integrity acceptable to Anglo American Corporation.
- It allows Anglo American Corporation to draw up specific manufacturing quality assurance procedures, which can be controlled.
- It allows reduction of stock holdings due to the rationalising, standardising and minimising of the number of pulley sizes, thereby providing improved inter-changeability of spares between various conveyors and mines.

CONCLUSION

Traditional methods and rules-of thumb used for design by pulley manufacturers can and sometimes does lead to over-stressing of welds and pulley components. Finite element analysis can provide more reliable results, but is time consuming and costly to implement, and also requires knowledgeable interpretation. Standardised pulley designs, engineered by finite element analysis, provides a solution to the above problems, and also provides other benefits.

REFERENCES

1 BS 7608:1993 Code of practice for *Fatigue design and assessment of steel structures*.

AUTHORS CV

Allan Lill obtained a BSc from the University of the Witwatersrand majoring in Applied Mathematics, and also an MSc in Engineering from the same institution. Allan has 35 years of work experience, mainly involving stress analysis, and has worked for Industrial Research & Development, Siemens, Atlas Aircraft, Control Data, Winder Controls and Anglo American, where he is currently employed as a senior mechanical engineer. He has extensive experience in the application of the finite element method.