With compliments Precision, Inc.
phone: (515)628-3115
fax : (515)628-3658

WELD NOTCH AFFECTS ON PULLEY AND BELT CONVEYOR RELIABILITY

Allen V. Reicks PE
Precision, Inc., 300 SE 14th, Pella, IA 50219

Modern conveyor pulleys are welded steel structures subjected to dynamic loads. In this paper, fatigue notch and fracture mechanics methods are applied to several weld joints on typical pulley constructions. These methods are used to analyze field applications and are correlated with laboratory testing. Generalized approaches are presented which provide high confidence in pulley design with corresponding high reliability.

INTRODUCTION

Conveyor pulley reliability is an integral part of conveyor availability since pulleys maintain belt tensions essential for load support and belt movement. This paper will address weld design methods that help provide this reliability.

Overall conveyor reliability can be considered as the product of the individual components reliabilities, such as structure, belt, load support components, and belt tension support components including pulleys and drives. Less clear are the difficulties in managing and minimizing unique issues of pulley failure such as:

- Pulleys are buried within the conveyor structure making them difficult to inspect or repair.
- To repair or replace pulleys, belt tension must be removed which often involves breaking the belt.
- The brittle nature of fatigue cracking gives little warning and causes immediate conveyor shut down and possible belt damage.

These issues make pulleys among the most important components in a conveyor, and show why reliability is best addressed at the design stage rather than through maintenance or spares. Pulleys can, and for the most part do, provide 'infinite' operating life. (It should be noted here that this discussion excludes wear or maintenance items such as bearings, lagging or shell abrasion, and general corrosion.) The majority of conveyor pulleys are of designs and constructions which have proven themselves in hundreds of successful applications.

Historically, these designs have often been extrapolated

for use on larger conveyors, often with misunderstood risks. Compounding the problem, these larger conveyors, with their increased tonnages and distances, are frequently crucial to the overall reliability of the mine and its processes. This paper will review modern design and analysis tools available to pulley designers and manufacturers. These can provide a high level of confidence in meeting the reliability demands of modern mines. Design methods related to material properties are essential because the growing range of conveyor sizes, speeds, belts and applications make long term, full scale testing of each design impractical and/or impossibly slow.

PULLEY CONSTRUCTION

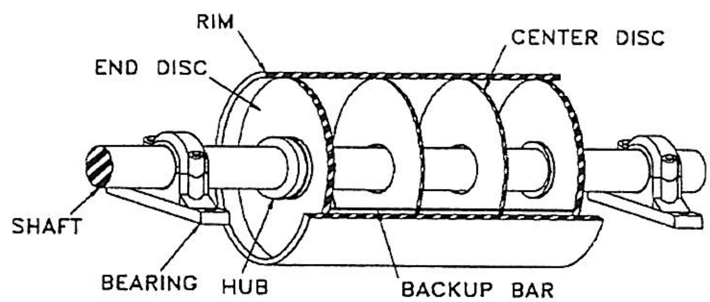

Figure 1. Common Weld Steel Pulley Construction

Figure 1 shows the important structural elements of welded steel conveyor pulleys in common use today. It is important to understand that many of the construction features have developed due to the need for pulley manufacturers to provide an extremely wide range of pulley sizes and capacities which can be produced with short lead times. This can be accomplished with a range of mild steel plate stock and an inventory of shafting and hubs. Welding and other flexible fabricating methods are integral to providing consistent assemblies without time consuming machining operations on all but the largest sizes. Welded fabrications become impractical when designs require weld joint thicknesses in the range of 75mm.

Machined profiles and weld joints are often used in these cases to keep welds smaller (commonly called 'Turbine' pulleys, these constructions are often considered more reliable since they allow design redundancies, comply with weld codes, and may be inspected more easily). For the vast majority of conveyors the above material inventories, when fabricated with capable processes to meet design assumptions, provide all that is needed for reliable pulleys when the designer uses the tools available. Structural analysis, especially weld joint design, is among these and will be discussed in principle and examples.

PULLEY LOADING

Various load effects and resulting stresses develop due to the belt pressure on the rim shell and the bending moments developed due to outboard bearing supports. Bending and direct, or in-plane, stresses result from these loads. The pressure distribution from the belt is often assumed to be somewhat uniform. History has shown this to be safe practice though this is an area that justifies better understanding, especially as affected by conveyor and belt construction.

Consistent belt tensions and continued rotation combine to produce fatigue sensitive conditions. A given point on a pulley is constantly cycled through a repeating pattern of stress magnitude and direction. The stress extremes during each revolution constitute the stress range or twice the alternating stress, σ_a. Safe design requires an understanding of the stresses and the material limits. Though many approaches to fatigue analysis have been developed, it is commonly understood that fatigue life (N) is composed of crack initiation (Ni) and crack propagation (Np) phases as follows;

N=Ni + Np (1) (Lawrence et al, 1978)

Where N represents life in stress cycles (typically 1 cycle per revolution).

In the models that follow we chose to either prevent a crack from initiating or prevent it from propagating in order to provide long life. To prevent a crack from initiating in sound metal, a representation of the alternating stress must be kept below a design endurance limit, σ_e, of the steel. Endurance limit is commonly accepted as the maximum stress which will result in 2 x 10^6 cycles. This is called 'infinite' life because testing with mild steels does not show failures at lives greater than 2 x 10^6 cycles. To prevent existing crack growth (propagation), Fracture Mechanics methods are applied to limit the stress intensity to less than the crack propagation threshold. Fracture Mechanics also provides a method to evaluate the effect of impact loads, as described later.

Other important fatigue issues include:

- fatigue is a statistical phenomenon so that failure probability can be treated as a design input. (BS 5400: Part 10:1980)
- Minor's Rule of Cumulative Damage and finite life estimates for stresses greater than σ_e provide a way to anticipate the effects of startup and shut down dynamics; (Fuchs & Stephens, 1980); for example, 9 rev./day @ 200% load and 120 rev/day @ 150% load can increase the alternating stress affect by 1.06.
- Operating environment, part thickness and steel surface finish affect the endurance limit of a given material.(Radaj, 1990)

In addition to cyclic fatigue loads from the belt, various assembly stresses need to be acknowledged, especially that from expansion of the hub as it clamps the shaft. These mean stress effects will be addressed later.

STRUCTURAL ANALYSIS

This paper addresses general methods of analyzing the pulley structure. Though examples that follow will address specific areas for demonstration, a full understanding of various stresses is beyond the scope of this work.

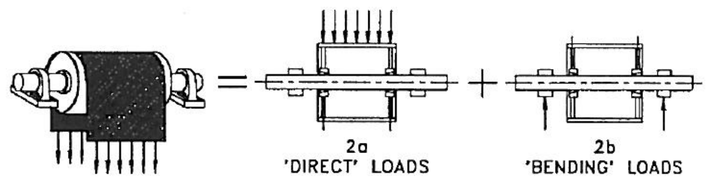

Figure 2. Belt Loading Model

Figure 2 illustrates the most common method of separating the load effects in the process of predicting the operating stresses in a pulley. Figure 2a illustrates direct loads. These primarily develop bending and membrane stresses in the rim and membrane stresses in the disc. Figure 2b shows bending moments induced through the structure. They produce bending stress in the disc and direct stresses in the rim plate. Both sources contribute to stress in joints. Historical methods utilize plate, shell and beam theory with simple boundary conditions to calculate nominal stresses and the effect of varying geometries at particular critical locations on the pulley. To allow closed form calculation, they typically ignore load transfer at transitions such as that between the disc and rim. Fundamental to this approach, the design stress limits themselves were developed as a function of the calculation methods rather than actual material limits. Design parameters, such as weld size, were assigned as a rule of thumb rather than by actual strength.

Nonetheless, these designs methods developed to generally produce safe designs when properly applied. Model studies with strain gages and with destructive methods allowed refinement of general formulae to extend the range of applicable pulley sizes. Unfortunately, due to the high cost of exhaustive, large scale testing, mines and plants often became test sites, with the miners suffering the consequences. These failures were then addressed with new formulas or lower limits, though these corrections were often not justified or were insufficient in other applications (example: fictitious free shaft deflection limits reduced hub and end disc failures in the

1970's. Though leading in some cases to more practical constructions, the author believes this practice needlessly adds to pulley cost in other cases).

With the availability of computers, pulley designs became more throughly analyzed, using practical numerical methods with large matrices and Fourier summations for differential equation solutions. Just as important, the finite element method (FEM) and boundary element (BE) analysis allow a much clearer understanding of the actual stresses, especially as affected by hub deflection, disc to rim transition and center discs. These methods and Conveyor Dynamics Inc's hybred analysis program PSTRESS 3.0 (Qui and Sethi, 1993), with parametric inputs, allow routine analysis and model studies to understand the detailed 'structural' stresses. Structural stresses are those that reflect results of forces and moments that exist at <u>each</u> point in a pulley. Knowledge of these specific stresses allows correlation to specific material properties in the plate and weld materials.

It is important to distinguish the differences among nominal stress, structural stress, and local stress (Radaj, 1990) in order to understand the process of assigning actual material limits to various parts of the pulley body. "Nominal" stresses result from classical analysis of global or gross deflections, forces, and moments in individual components of a structure. "Structural" stresses incorporate the shared deflections and force vs. moment conversions at transitions between components so that one understands the loading along the full length of the various components. "Local stresses" are those most relevant to material properties. These vary the most within the pulley structure and become the most difficult to analyze. Radii, notches, and various surface finishes disturb the stress distribution through the plate or beam thickness so the actual local stresses may vary from classical stress distributions indicated as structural stresses. Internal or residual stresses are also local stresses.

Actual analysis compares the stress at worst case locations (10-20 points usually suffice) and properties associated with material conditions (as-rolled, machined surface, weld heat affected zone, weld notch, etc...)

WELD LIMITS

As-rolled or machined surfaces have well established fatigue properties with surface finish corrections. (Radaj, 1990) They allow for simple analysis because structural stress and local stress values will be very similar due to consistent surfaces. In contrast, pulley locations on or near welds and weld joints have widely varying local stresses or require special treatment to know their fatigue characteristics. This is due to a wide range of factors including:

- Metallurgical changes, varying hardness, and grain size due to weld metal dilution and thermal cycles.
- Shrinkage or residual stresses from weld metal contraction as it cools. These may be 'yield strength' level (3 to 10 times the cyclic fatigue stresses).
- Abrupt geometrical discontinuities causing high stress concentrations or local stress magnifications as forces 'flow' between components.
- Variability in surface roughnesses of the weld, on prepared plate edges, and in the degree of weld penetration into the joint.

It should be noted that weld quality issues of porosity, cracking, and improper penetration have been addressed by most reputable pulley manufacturers. Non- destructive testing (NDT) and, in particular, ultrasonic testing (UT), give welders important feedback on the quality of their welds and the weld processes.

A large amount of research has gone into understanding the fatigue capacities of welds. Several of these from the literature will be described and some beneficial applications will be described.

Nominal Stress Methods

Blodgett (1963) describes methods to represent a weld as a line and to calculate a 'section modulus' of the cross section through the weld. Force per unit weld length and required weld size for this unit load could then be determined. This method lacks in accuracy and applicability to plate structures in similar ways that classical beam stress calculations can be in error. That is, beam stress calculation accuracy near boundaries vary from those at center span due to neglecting the role of shear stresses in transferring force from the beam body to its support.

Various welding codes including AWS D1.1 (1990), CSA W59 (1989), BS5400 (1980) etc., address weld fatigue capacity of various joints but fall short as design tools since only a limited number of joints are addressed. They are necessarily limited to those commonly used in buildings and other structures. In addition, codes ignore the common situation of a plate size being selected due to stress at one point while much lower stresses exist at another point. A code will require an unnecessary and difficult weld nonetheless. For example, the bending stress in an end disc near the hub is 1.5 to 10 times greater then the stress near the rim, so that the weld joint near the hub must be much stronger than the weld near the rim. Codes are, however, useful in some analyses, since allowable stress limits incorporate metal differences and residual stress effects. These limits can be applied to "structural" stress calculations when joints without local variations are being evaluated or if the particular joint and its loads are addressed in the code.

Another fatigue method that serves a useful role is an extension of Basquin's relationship.

$\Delta S\, K_f = (\sigma'_f - \sigma_o)\,[2N_f]^b$ (2) (Yung & Lawrence, 1985)

Where:

ΔS = Stress range=2 σ_a
K_f = fatigue notch factor
σ_o = mean stress
σ'_f = fatigue strength coefficient
b = fatigue strength exponent
$2N$ = crack initiation life, stress reversals.

The worst case effects of weld residual stress on allowable stress range can be seen by setting σ_0=yield strength (S_y) of the base metal using material properties σ_f and b from the literature (Lawrence et al, 1978), and with N=2 x 10^6 where infinite life is accepted for mild steels. (Goodman, Gerber, Haigh and others have developed similar representations for the effect of mean stress, all of which infer similar conclusions regarding high residual stresses.) This conservative practice also means that no allowance for additional mean stresses from assembly, torque, etc. are needed.

This equation also explains why no consistent benefit is seen in weld fatigue life with higher strength steels. As the yield strength rises, higher residual tensile stresses σ_0 are trapped near the weld, negating the benefit of higher tensile steel. Tests show that stress relief does not consistently nor predictably eliminate these harmful internal stresses. (Maddox, 1987)

Local Stress Methods

Weld joints pose problems not addressed by nominal or structural stresses due to the variability of the surfaces and to the sharpness of the notches developed. Numerical analysis of local geometry shows a clear and dominating influence of assumed transition radius. This sensitivity makes these methods unusable unless a design methodology can be proven. Two approaches to this problem will be presented here. The first addresses K_f from above while the second approaches the issue using Fracture Mechanics methods.

Identifying a notch factor multiplier, K_f, in Basquin's relationship is the remaining issue to establishing the cyclic stress limits necessary to prevent cracks up to and therefore beyond 2 x 10^6 revolutions. This K_f is similar to stress concentration, K_t, used in static design. In fact, K_f, for smooth surfaces and radii can be modified from K_t through a notch sensitivity correction (q).(Peterson, 1974)

The magnitude of stress concentration, K_t, *increases* as the notch radius becomes smaller. This applies to the transition from the weld plate to the toe of a fillet or butt weld or at the hidden root of a weld where the penetration stopped. Obviously, these surface and buried radii are impossible to control when welding. Fortunately, notch sensitivity, the relationship of K_f to K_f *decreases* as the notch radius (r) becomes smaller.

Lawrence (Lawrence et al, 1978) has combined the counteracting trends of K_t and notch sensitivity versus notch radius to develop the concept of the 'worst case' notch. He combined the equations for K_t =f(r) and q=f(r) to produce K_f=f(r). By setting $dK_f'/dr=0$, he was able to produce a value for a worst case design value for r. Radaj (Radaj, 1990) describes a similar approach in which r=1 mm is used as a fixed value. His K_f is multiplied by σ_a and compared to as rolled plate fatigue strength. Both methods showed good correlation with large amounts of laboratory test data. This design approach allows FEM or BE models to produce relevant results by using the 'gross' geometry of the weld combined with specific radii at transitions.

These models can be 2D cross sections since they are used to convert structural stresses to local stress. General 3D methods described above produce the structural stresses while the smaller, more detailed 2D model produces K_f.

Fracture Mechanics is an engineering field that addresses the effects of stress on preexisting cracks, (Barson & Rolfe, 1978). Recognizing the possibility of crack-like elements near welds suggests that Fracture Mechanics may apply to pulley welds. The brittle 'fracture' appearance of failures in otherwise ductile materials also indicates that at least a part of the fatigue fracture mode is crack related and therefore applicable to Fracture Mechanics (FM). A fundamental FM relationship is:

$K= \sigma \cdot \sqrt{a} \cdot f(g)$ (3) (Barson & Rolfe, 1987)

Where:

K = "stress intensity", a FM parameter
σ = structural stress
a = crack length
f(g) = dimensionless function which varies with geometry, loading, etc.

K will vary with the direction of loading relative to the direction of crack orientation. K_I, where the crack is opened in-plane, is the most damaging. It is commonly compared to 'critical' material properties of K_{Ic} and K_{Id}, for static and dynamic loading respectively. Above these limits, the crack will quickly grow through the plate. (If the crack is too short, so that K_I is low, high levels of s will cause yielding or ductile failure rather than brittle fracture.)

Fracture Mechanics also applies to fatigue as a crack growth mechanism until the crack is long enough to develop K_{Ic}. Crack growth or propagation is described by:

$\frac{da}{dN} = A (\Delta K)m$ (4) (Barson & Rolfe, 1987)

Where

da/dN indicates change in crack length per cycle (varies as crack grows)

ΔK= stress intensity range (varies as the crack grows)

A and m are constants relating to the specific case.

A FM concept more relevant to long life is the crack propagation threshold, ΔK_{th}, the stress intensity range below which a crack will not grow. The design goal is to develop joints so that ΔK_I from fatigue stress is less than ΔK_{th}. Conveniently, testing has shown ΔK_{th} to be independent of steel type. ΔK_{th}, K_{Ic}, and K_{Id} data are developed under consistent ASTM methods and are available in the literature for design purposes.

Fracture Mechanics itself has been applied to welds

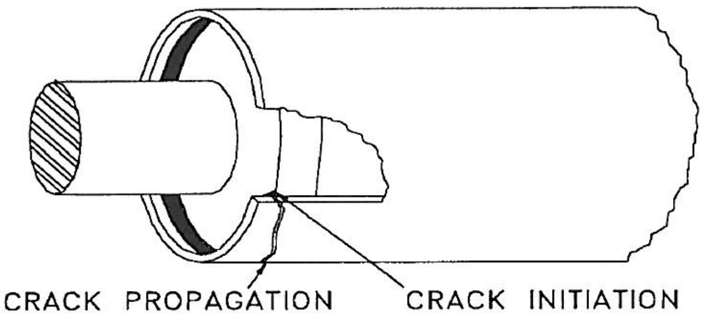

Figure 4. Example 2 Mode

Several months later, 3 pulley failures were reported on this conveyor, including the bend pulley described above. The rim showed brittle type cracking through the rim next to the fillet weld all around the pulley (Fig. 4). After replacements were provided and installed, the mine commissioned Conveyor Technologies LTD to review the application (Harrison & Hustrilid, 1993). Measurements pointed to high braking tensions as a possible source of failure. CTL measured the conveyor deceleration during fully loaded braking and estimated tensions at 965KN; 27 times running tensions. (It was later determined that incorrect brake orifices had been installed at the time of conveyor commissioning.)

Though it was clear that these stresses are near yield strength, it was not clear why a 'brittle' fatigue failure pattern was observed when only a small number of the braking stress cycles occurred. A structural stress analysis using CDI's PSTRESS 3.0 was conducted to further investigate the cause of failure. The result indicated cyclic loading of 3.6 Mpa membrane stress and 1.2 Mpa bending stress. A BEASY™ analysis showed that these converted to 1.06 Mpa $m^{1/2}$ and .2MPa $m^{1/2}$ respectively or 28.6 Mpa $m^{1/2}$ and 5.4 Mpa $m^{1/2}$ at braking impact. (Note that direct rim stress developed higher stress intensities than bending stresses.)

FM data (Barson and Rolfe, 1987) for ASTM A-36 steel shows that, at slow loading rates above -73°C, one would not expect a brittle failure (cracking) mode, no matter what the stress or stress intensity. The same data shows that with higher loading rates, the critical stress intensity curve shifts to higher allowable temperatures. With a maximum strain of 6.3 x 10^{-4} applied at pulley rotational speed of 72,000 rad/sec we find a peak strain rate of 45/sec. Available data shows that this is well above the 10/sec required for K_{Id} (critical stress intensity under dynamic load) levels to apply. K_{Id} at 10/sec and -17°C is approximately 40 MPa $m^{1/2}$ applied. Since this mine is in a mountain climate, and the loading rate is 4 times that of K_{Id} we can see how a single braking incident could cause a crack to develop in the rim (total failure would be delayed due to restraint from adjacent lower stressed material). Cyclic running stresses would be sufficient to cause propagation to failure.

	'Load'	vs	'Capacity'
Notch Stress analysis (Radaj, 1990)	$K_f \cdot \sigma_a$	vs	σ_e
-K_f from FEA with (r)=1mm corporates correction to 10% probability (Radaj, 1990)	6.1 · 15.9MPa =97 MPa		
-a)from σ_f'= 725 Mpa b= -.132, σ_o=0, N=2x10^6 for ASTM A-36 HAZ (Lawrence et al, 1978) Corrected to 10% prob. (x.715)(BS 5400,1980)		vs	95Mpa(a)
-b) AWS code (D1.1 -90, 1990)			82.8Mpa(b)
Fracture Mechanics	K_I	vs	K_{th}
-K_I from BEASY™	8.1 Mpa $m^{1/2}$	vs	
-K_{th} (Barsom and Rolfe, 1987)			6.5MPa $m^{1/2}$
Uniaxial Fatigue Test-7 tests (Banás and Lawrence et al, 1994) -longitudinal butt weld with discontinuous backup bar			σ_{test} @ 2x10^6 cycles
-R=-1corrected -corrected for induced bending stresses -lower bounded value			=17.9Mpa
Estimated Structural Stress Range -18.6 Mpa to 13 Mpa; σ_a=15.9 Mpa caused field falure.			

Table 1

under the worst case assumption that the weld process causes small cracks in and near the weld. These methods have been used to develop structural codes discussed above. Their relevance may be debated for machinery such as pulleys where weld size, materials, and processes are closely controlled and inspection is easily done. They are more clearly applicable to hidden notches at the root of fillet welds and to butt welds into backup bars. In addition, Fracture Mechanics is the predominate method to address impact effects. (See example 2)

A more difficult issue has been developing f(g) for equation (3). Though design manuals are available with a wide variety of geometries, they cannot address all of the possible geometries a designer needs to evaluate. Commercial computer packages, such as BEASY™ from Computational Mechanics, provides a solution to the need for a general design procedure. It incorporates BE numerical modeling including methods for assigning a crack location to solve for K as a functional σ. As with the K_f for the worst case notch concept, BEASY™ allows an understanding of local geometry effects from structural stress inputs.

Design Example 1: K_f vs K_I

The following example results from applying the above concepts to a pulley failure mode as influenced by common industry practice. This limitation has been seen in products from several manufacturers. One installation will be described to illustrate the use of the above design analysis methods.

A number of pulleys with 500 mm diameter, 1500 mm width and 150 mm shafts were supplied to a coal mine in the Midwest United States. They were designed to operate with 73KN belt tension with 90° of belt wrap in take-up locations. After failures were reported, tensions over 95KN (30% overload) were measured. These over-tensions were explained by wire rope reaving friction that did not allow the take-up weight to rise after start up. The locked-in belt stretch caused higher than predicted belt tensions.

Inspection of the failed pulleys showed cracks initiated in the rim at the intersection of the center disc and rim seam weld. The cracks were radial through the thickness of the rim and had propagated circumferentially (Fig. 3). This cracking pattern implicates high axial rim stresses.

'Nominal' stress analysis showed that bending stresses were very low but that in-plane or membrane stresses were significant. These axial stresses developed from the small diameter, wide face width pulley acting as a beam.

This unfortunate episode provided an opportunity to compare the use of the local stress methods with field experience. To do so required a better understanding of the failure mechanism. A FEM analysis of the pulley provided more accurate 'structural' stresses. These showed that the center disc did little to affect the axial stresses (they primarily act as radial stiffeners to limit circumferential deformation and stress). The discontinuity of the seam weld at this center disc was also considered (Fig. 3). Rim seam welds use backup bars to allow welding from the outside of the pulley. These backup bars are interrupted at the center discs, creating a stress concentration of the axial stresses. The notch effect of the discontinuous backup bar was modeled. (Rudolphi & Rogge, 1995)

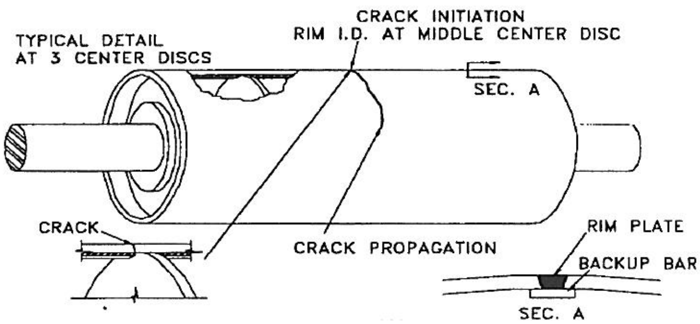

Figure 3. Example 1 Mode

In addition, uniaxial fatigue tests were performed (Banás & Lawrence, 1994) to evaluate the accuracy of these methods. Table I summaries assessments of the weld notch effects of the discontinuous backup bar. Several observations can be made.

- The discontinuous backup bar provides a severe notch. Normally, low structural stress due to a large section modulus (large pulley diameter) make this weakening unimportant.
- A consistent design stress approach can be expected to prevent this rim failure mode by reducing structural stress (thicker rim) and/or design with continuous backup bars.
- Notch stress and Fracture Mechanics approaches gave similar results; both are compatible with actual field performance. Model laboratory testing predicted fatigue capacity close to that of the local stress methods.

Design Example 2: Impact Loading

A pulley of 450 mm diameter and 1675 mm width with a 200 mm shaft was provided for a downhill conveyor for a major coal mine in western Colorado, USA. This pulley was in a bend location next to the brake pulley. Since it was between the brake and the tail, it saw braking tensions, including occasional fully loaded emergency stops. Design tension of 71.2 KN were provided to anticipate braking tensions even though the gravity take up developed running tensions of only 35.6 KN. An integral hub design was used for this pulley due to the large shaft (Fig. 5). This resulted in a 85 mm thick disc being fillet welded to a 13mm rim plate, a seemingly safe design for the tensions provided.

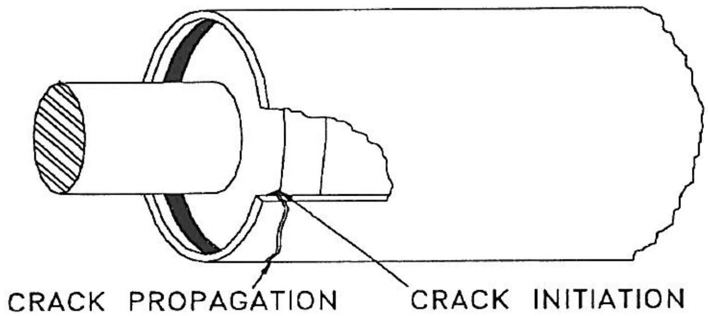

Figure 4. Example 2 Mode

Several months later, 3 pulley failures were reported on this conveyor, including the bend pulley described above. The rim showed brittle type cracking through the rim next to the fillet weld all around the pulley (Fig. 4). After replacements were provided and installed, the mine commissioned Conveyor Technologies LTD to review the application (Harrison & Hustrilid, 1993). Measurements pointed to high braking tensions as a possible source of failure. CTL measured the conveyor deceleration during fully loaded braking and estimated tensions at 965KN; 27 times running tensions. (It was later determined that incorrect brake orifices had been installed at the time of conveyor commissioning.)

Though it was clear that these stresses are near yield strength, it was not clear why a 'brittle' fatigue failure pattern was observed when only a small number of the braking stress cycles occurred. A structural stress analysis using CDI's PSTRESS 3.0 was conducted to further investigate the cause of failure. The result indicated cyclic loading of 3.6 Mpa membrane stress and 1.2 Mpa bending stress. A BEASY™ analysis showed that these converted to 1.06 Mpa $m^{1/2}$ and .2MPa $m^{1/2}$ respectively or 28.6 Mpa $m^{1/2}$ and 5.4 Mpa $m^{1/2}$ at braking impact. (Note that direct rim stress developed higher stress intensities than bending stresses.)

FM data (Barson and Rolfe, 1987) for ASTM A-36 steel shows that, at slow loading rates above -73°C, one would not expect a brittle failure (cracking) mode, no matter what the stress or stress intensity. The same data shows that with higher loading rates, the critical stress intensity curve shifts to higher allowable temperatures. With a maximum strain of 6.3 x 10^{-4} applied at pulley rotational speed of 72,000 rad/sec we find a peak strain rate of 45/sec. Available data shows that this is well above the 10/sec required for K_{Id} (critical stress intensity under dynamic load) levels to apply. K_{Id} at 10/sec and -17°C is approximately 40 MPa $m^{1/2}$ applied. Since this mine is in a mountain climate, and the loading rate is 4 times that of K_{Id} we can see how a single braking incident could cause a crack to develop in the rim (total failure would be delayed due to restraint from adjacent lower stressed material). Cyclic running stresses would be sufficient to cause propagation to failure.

	'Load'	vs	'Capacity'
Notch Stress analysis (Radaj, 1990) -K_f from FEA with (r)=1mm corporates correction to 10% probability (Radaj, 1990) -a)from σ_f'= 725 Mpa b= -.132, σ_o=0, N=2x10^6 for ASTM A-36 HAZ (Lawrence et al, 1978) Corrected to 10% prob. (x.715)(BS 5400,1980) -b) AWS code (D1.1 -90, 1990)	$K_f \cdot \sigma_a$ 6.1 · 15.9MPa =97 MPa	vs vs	σ_e 95Mpa(a) 82.8Mpa(b)
Fracture Mechanics -K_I from BEASY™ -K_{th} (Barsom and Rolfe, 1987)	K_I 8.1 Mpa $m^{1/2}$	vs vs	K_{th} 6.5MPa $m^{1/2}$
Uniaxial Fatigue Test-7 tests (Banás and Lawrence et al, 1994) -longitudinal butt weld with discontinuous backup bar -R=-1corrected -corrected for induced bending stresses -lower bounded value			σ_{test} @ 2x10^6 cycles =17.9Mpa
Estimated Structural Stress Range -18.6 Mpa to 13 Mpa; σ_a =15.9 Mpa caused field falure.			

Table 1

Though the purpose of this is to show the nature of the calculations, several observations may be made regarding this example:

- Thick, rigid end discs develop higher shear transfer to the rim. This loading develops higher stress intensities and is more vulnerable to dynamic loads.
- The load needed to cause this cracking failure was 7 times the running loads necessary to cause a crack at the weld root to propagate (K_{TH}) over long life.
- Impact loading temperature, material toughness and joint design can be addressed in pulley design.

CONCLUSION

This paper has described methods for understanding the capacities of weld joints in conveyor pulley constructions. These methods take advantage of modern numerical developments in structural analysis and materials-based weld capacities. Therefore, they can provide quantitative insight into component interactions and potential failure modes. Though additional development is necessary, these tools provide a means to infinite design life and high reliability.

Conveyor designers have provided tools to better understand the interaction of and effects on different components. Design methods such as those described above allow pulley manufacturers to complement these efforts. Together, miners, engineers, and manufacturers can develop highly reliable conveyors through understanding of details.

REFERENCES

ANSI/AWS D1.1-90, 1990, *Structural Welding Code*, American Welding Society, Miami, Florida, Chapter 9.

Banás, Grzegorz, and Lawrence, Frederick V., 1994, "Fatigue Report of Longitudinal Weldments", Technical Report to Precision, Inc.

Barson, John M., and Rolfe, Stanley T. 1987, *Fracture and Fatigue Control in Structures*, Prentice-Hall, Englewood Cliffs, NJ.

Blodgett, Omer W., 1963, *Design of Weldments*, James F. Lincoln Arc Welding Foundation, Cleveland, Ohio, Sec 6.3.

British Standard 5400, 1980: *"Steel, Concrete and Composite Bridges, Part D. Code of Practice for Fatigue"*.

CSA W59-1989, 1989, *Welded Steel Construction (Metal Arc Welding)*, Canadian Standards Association, Rexdale, Ontario, Chapter 12.

Fuchs, H.O., and Stephens, R.I., 1980, *"Metal Fatigue in Engineering"*, Wiley, New York.

Harrison, A., and Hustrulid, A., 1993, Private report.

Lawrence, F.V., Jr. Mattos, R.J., Higashida, Y., and Burk, J.D., 1978, "Estimating the Fatigue Crack Initiation Life of Welds," *Fatigue Testing of Weldments, ASTM STP 648*, D. W. Hoeppner, Ed., American Society for Testing and Materials, pp. 134-158.

Maddox, S.J., 1987, *The Effect of Plate Thickness on the Fatigue Strength of Fillet Welded Joints*, The Welding Institute, Abington, Cambridge, Great Britain.

Peterson, R.E., 1974, *Stress concentration Factors*, Wiley, New York.

Qui, Xiangjun, and Sethi, Vinit, 1993, "A New Pulley Stress Analysis Method Based on Modified Transfer Matrix", *Bulk Solids Handling* Vol. 13, No. 4, November, p713-724.

Radaj, Dieter, 1990, *"Design and analysis of Fatigue Resistant Welded Structures"* Halsted Press New York.

Rudolphi, Thomas J., and Rogge, Thomas R, 1995, "Stress Analysis of Backup Bar/Pulley Rim Weld", Technical Report to Precision, Inc.

Yung, J.-Y. And Lawrence, F.V., 1985, Analytical and Graphical Aids for the Fatigue Design of Weldments", *Fatigue and Fracture of Engineering Materials and Structures*, Vol. 8, No. 3, p223-241.