APPLICATION OF WELDMENT FATIGUE METHODS TO CONVEYOR PULLEY DESIGN

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ABSTRACT

The paper "Weld Notch Affects on Pulley and Belt Conveyor Reliability" has done an excellent job of describing theoretical fatigue and fracture mechanics methods used in modern weld design. From this foundation, we intend to further detail areas previously discussed and present new research in the application of these principles to pulley weldment life prediction. Affects of changes in alternating stress range, mean stress, and residual stress have been investigated using controlled laboratory fatigue tests on a pulley weldment. Comparisons between these results, weld codes, and theoretical fatigue predictions will be made and discussed.

INTRODUCTION

Production demands continue to increase conveyor belt speeds, loads, and usage. Advances in conveyor belt technologies continue to allow system designers the use of smaller diameter pulleys, which operate at faster revolutions per minute. As haulage systems get larger, often dependence on the conveyor(s) increases. These conditions translate into pulleys designed to reliably operate under ever increasing loads for many more fatigue cycles. It is not uncommon for a pulley to have a design life of two hundred million(2x10^8) stress cycles. With demands such as these it's important for pulley manufacturers to have the ability to estimate the suitability of their designs for high cycle fatigue service.

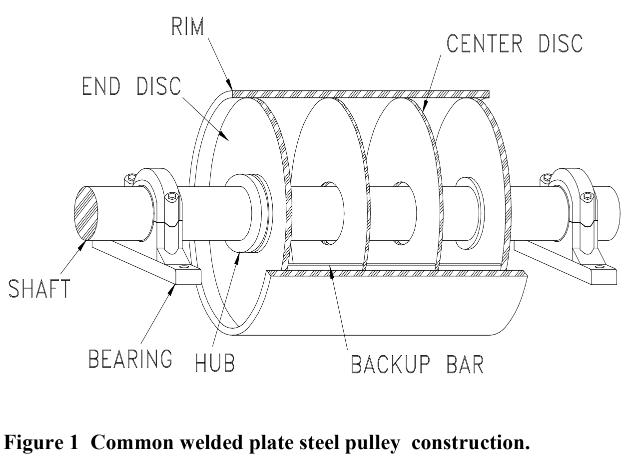

Figure 1 Common welded plate steel pulley construction.

Most pulleys manufactured in North America are constructed using a welded plate construction as shown in Figure 1. This construction uses a hub and bushing system, which clamps the pulley to the shaft while allowing for removal if necessary. Each hub is connected to an end disc with a continuous cruciform (fillet) weld on either side of the plate. An example of this weldment is shown in figure 2. The pulley rim is a rolled plate, which is wrapped around the end discs as shown in Figure 1. Often center discs are used in the construction to help support the belt load and improve rim roundness on the finished pulley.

Welded plate designs have been successfully used in thousands of conveyor applications in all types of industries. Often they are the preferred design since they usually give the best balance between reliability, delivery, and cost. To meet delivery and cost constraints, welded plate designs emphasize the use of structural hot rolled plate, simple shapes, and weld joints that adapt well to high production semi-automatic methods. The welded plate design, as in all structures, has areas where a potential failure could occur. The challenge for the pulley designer is to recognize these

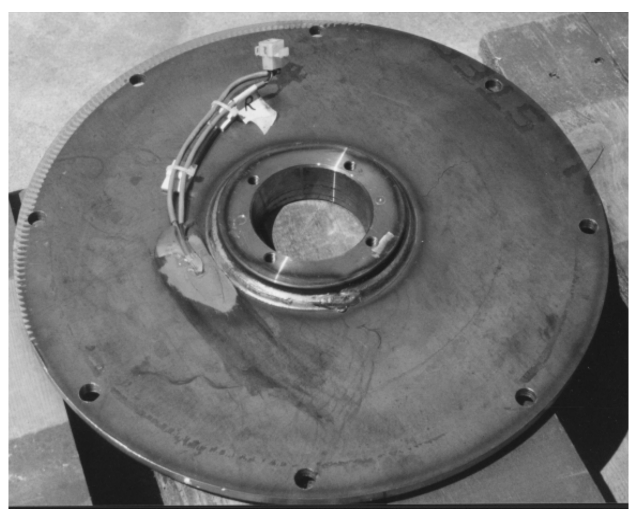

Figure 2 Test end disc showing cruciform (fillet) weld between end disc and hub. Strain gages attached to monitor stresses induced during set-up and loading.

areas, and design against fatigue failure at these sites. It should be mentioned there are many pulley designs emphasizing construction features deemed more fatigue resistant or lending themselves to simpler stress analysis (i.e. turbine discs, no center discs, etc.). Due to economic and delivery factors, these constructions tend to have limited usage in only very high tension applications. Today the sciences of fatigue and fracture mechanics give us the ability to develop high reliability while reducing the use of costly design features.

To properly predict the fatigue characteristics of a pulley design, many factors must be identified and understood. Some of the most important factors include the :

- type of pulley construction selected;
- type of materials used;
- welding processes selected;
- load scenario of the application;
- mean stresses created during fabrication;
- type of post weld treatment performed;
- fabrication workmanship; and
- welding workmanship.

To try to detail the effects of all these variables on the fatigue life prediction of a pulley is beyond the scope of this paper. The scope of this investigation is to investigate the effects of changes in alternating, mean, and residual stresses in the cruciform welds used at the hub to end disc attachment shown in Figure 2. The goal is to illustrate an application of state of the art weld fatigue life prediction techniques and compare these results to laboratory results. To properly estimate the fatigue life of a pulley, this process must be repeated for each potential failure site.

DESCRIPTION OF FAILURE MODE

Figure 3 Hub portion of a fatigue sample after testing to failure. Failure initiated at toe of cruciform weld and propagated around weld circumference and through end disc plate.

Figure 2 shows an end disc and hub assembly prepared for destructive fatigue testing. The assembly has a cruciform weld on either side of the disc to hub connection. The failure most often observed is a crack initiating at the weld toe on the end disc plate surface. Upon initiation, the crack begins to propagate around the weld circumference and into the end disc plate toward the weld toe on the opposite side. If propagation is allowed to continue, the hub and disc eventually completely separated as shown in the failed specimen of figure 3. In addition, radial cracks will occasionally be observed propagating toward the rim.

WELD JOINT STRESS ANALYSIS

Pulley belt loads, and the stresses they induce, have been sufficiently detailed by others (Lange 1963; Qiu and Sethi, 1993: Reicks, 1996). For this discussion we will concentrate on detailing the stress state near the hub to disc weld failure area.

To understand the stresses in the disc it’s helpful to start with an analysis of the shaft. The shaft acts as a beam, simply supported at the bearings, being bent by the belt forces. The pulley hub and bushing provide a relatively rigid connection to the shaft, which causes the pulley end disc to bend with the shaft. An example is shown in figure 4. A close inspection of the end disc in figure 4 shows the curvature changes between the upper and lower parts of the cut-away. This change causes the stresses in the disc to reverse between tension and compression during each revolution. These alternating stresses are the primary consideration when making a fatigue analysis at this weld joint.

In addition to the alternating stresses, there are a number of mean stresses, which can effect the life prediction of this weld. Some of the most common are:

- shaft clamping systems with mechanical interferences on tapered surfaces that create clamping pressures between the shaft and pulley;
- shaft clamping systems which have a tendency to pull out the end disc when tightened;
- end disc bending during shaft installation due to manufacturing tolerances for hub parallelism;
- residual stresses from fabrication and welding;
- and reduction of residual stresses from post weld heat

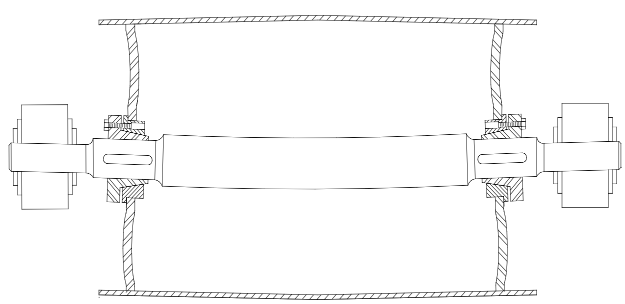

Figure 4 Sketch illustrating how shaft deflection, created by belt tensions, causes the pulley end discs to deflect.

treatments such as thermal stress relieve.

At present most major pulley manufacturers understand the importance of the alternating radial end disc stress on pulley life. Using in-house testing, and field experience, each manufacturer has developed allowable limits for this stress with their product. When asked to design a pulley, the most common practice is to vary the end disc thickness to keep the alternating radial end disc stress below the manufacturers allowable. The mean stresses previously listed are occasionally discussed during marketing presentations, but in the authors experience are rarely considered in the design analysis of welded plate pulleys. It should be mentioned that this statement may not hold true when discussing other pulley designs (i.e. turbines).

This design method has proven safe for the majority of cases, but we feel has some shortcomings. This method depends on the manufacturer having experience in the size of product being designed. An uncertain amount of risk is assumed when an extrapolation from experience is made. Increases in conveyor speeds and up time have significantly increased the number of cycles needed for a safe design. These demands may not be consistent with stress allowables based on historical experience. In addition, manufacturer's usually don't incorporate a thickness effect into their stress allowables. This has the potential to significantly reduce, or eliminate, safety factors as thicker materials are used. For these reasons, we believe a more rigorous analysis of this weld connection, and all pulley connections, is necessary for the appropriate reliability.

CODE WELD STRESS ALLOWABLES

A number of engineered pulley specifications in the industry make reference to various welding codes for fatigue stress allowables. Some common references are to specifications such as CSA W59, ISO 5049/1, BS5400, BS7608, and AWS D1.1. We believe this is done, at least in part, to establish some level of control over the design practices mentioned earlier. Unfortunately, complete control may not be realized due to a number of difficulties in applying codes to a pulley design.

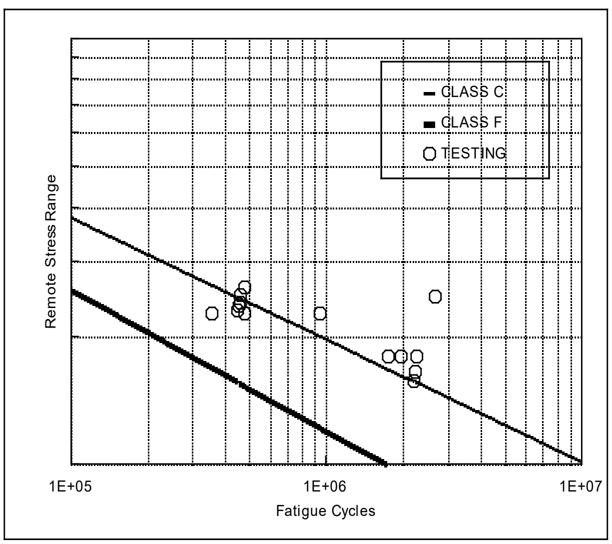

Figure 5 Comparison of two weld joint classification selections from BS7608/5400 with fatigue test data of a pulley hub to disc weld. Classification "C" represents code allowables for an ideal full penetration butt weld with surfaces ground smooth. Classification "F" represents code allowables for a cruciform joint of similar geometry to the hub to disc pulley weld. Note the values shown are mean curves compared to raw data, which shouldn't be taken as safe design values.

Most codes have originated from bridge or structural design, which causes the joint selections and load cases found in them to be oriented to joints used in those structures. Often codes don't address welds used in pulley constructions or, if the weld is documented, the failure mode seen in practice may not be covered. An excellent example of this is a crack initiating at the root of a fillet weld (Reicks, 1996). Few codes give the designer direction when working with this failure mode of a cruciform weld. One code which does (BS7608, 1993) addresses a failure mode through the weld throat, but doesn't speak to a failure mode through the base metal.

For the case of the cruciform weld used on the hub to disc weld we see greater coverage in codes. Unfortunately, some codes don't apply to cruciform welds with an alternating stress perpendicular to the joint and failure mode at the weld toe in the base metal. One code, BS7608, does address this case and for this reason we will continue our discussion specific to it. In BS7608, our joint would most likely be given an "F" classification. A line representing the mean S-N curve for the "F" weld classification is plotted in figure 5. It should be noted this curve isn't intended to be used as a design curve. The mean curve represents the average life expectancy of the population. Presented within the code is a statistical method for deriving a design stress based upon the standard deviation of the sample population and the reliability desired.

For comparison, we also plotted the curve for a "C" classification weld joint. The "C" class represents a full penetration butt weld with surfaces ground smooth and NDT inspection for flaws required. This curve is being presented as an "ideal" weld joint. Plotted over these curves, are results from end disc fatigue testing. As can be seen in figure 5, the testing indicates the cruciform joint significantly outperforms its "F" classification and is actually quite similar in performance to the ideal "C" class weld joint. The authors of BS7608 have provided for this situation, and give the designer the ability to use the results of controlled fatigue testing to move to a classification more representative of their situation.

A closer look at the loading on this cruciform weld can help provide a possible explanation for its better than expected performance as compared to the weld code classification. An important variable affecting fatigue life is the stress ratio (Barsom and Rolfe, 1987). Using the minimum alternating stress (Smin) and the maximum alternating stress (Smax) the stress ratio is calculated by the following equation:

$$\text{StressRatio } (R) = S_{min} / S_{max}$$

The stress ratio gives an indication of how much of the stress cycle is in compression and how much is in tension. Since fatigue failures rarely, if ever, occur in compression it has been observed that as more of the stress cycle becomes compressive the fatigue life increases (Barsom and Rolfe, 1987; Lawrence, 1996). The tests shown in figure 5 had stress ratio's with a significant portion in compression. In comparison, the tests used to create the code weld classifications are usually conducted at a stress ratio of zero (BS7608, 1993). A stress ratio of zero is a loading condition where the entire stress cycle is tensile and the minimum stress is zero. Due to the difference in stress ratios, we would expect the pulley hub to disc weld to have a longer fatigue life than its code classification indicates.

Most weld codes don't include stress ratio effects on the premise that residual weld mean stresses are already at the base materials yield strength and are considered when setting the classification allowables. This premise is contrary to the argument previously presented as a possible explanation of hub to disc weld test results. The effects of mean stress have been included in the initiation-propagation (IP) model (Lawrence, Dimitrakis, and Munse, 1996), which will be discussed in a later section. Our investigation continued with a test designed to see how changes in the mean stress, on the cruciform hub to disc weld, would affect fatigue life when the alternating stress range was held constant. If the IP model could be applied to our case, we would expect a beneficial effect from a reduction in weld mean stress.

Test results, as presented in figure 6, indicate the fatigue life of the hub to disc weld can be increased by a decrease in the weld mean stress. In addition, predicted results from the IP model are plotted. The IP line shows that the weld code's treatment of mean stress is valid for mean stresses greater than zero. When mean stresses are less than zero the IP model, and test results, indicate an improvement in the fatigue life can be realized.

INITIATION-PROPAGATION MODEL

The initiation-propagation (IP) model (Lawrence, Dimitrakis, and Munse, 1996) is a method of applying modern fatigue and fracture mechanics methods to predict the total life of a weld joint. The model estimates a total life (Nt) by adding an estimate of the initiation (Ni) and propagation (Np) lives. Initiation life is estimated from the Basquin-Morrow equation including the concept of stress relaxation during the initial set-up stress cycle. Propagation life is estimated from the Paris Power Law.

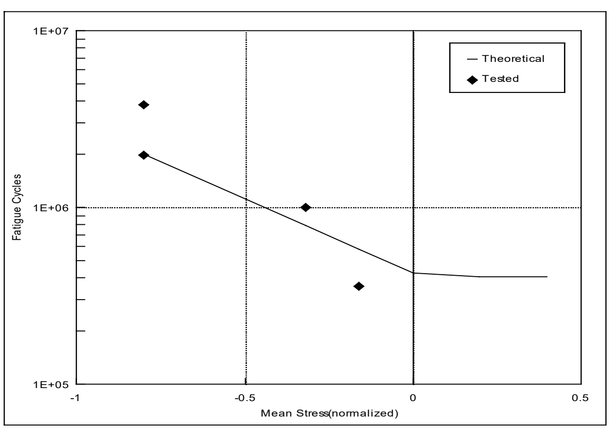

Figure 6 Results of fatigue test on hub to disc cruciform weld with alternating stress range constant while varying mean stress. Theoretical line is results from initiation-propagation model (Lawrence, 1996).

Fatigue life predictions are partially dependent upon geometric stress concentrations, material properties, and type of loading at the initiation site. The IP model uses methods for estimating the geometric stress concentration and converting it into a notch factor (Kf) through the use of Peterson's equation (Peterson, 1974). Since weld failures often initiate in the heat effected zone (HAZ), the IP model includes material properties for the HAZ. A particular joint will have different fatigue lives with axial and bending stresses so the IP model has a method of combining the effects of both.

A comparison of IP model results and test data is made on figure 7. By estimating the weld toe stress concentration of the test samples, and using it in the IP model, a total life (Nt) prediction falling near the test data was made. Total life predictions include initiation (Ni) and propagation (Np) predictions.

When looking at weld joint lives up to approximately ten million cycles, it's important to look at initiation and propagation lives. For fatigue lives greater than approximately ten million cycles, initiation life dominates the total life prediction. Since pulleys are typically designed for lives greater than ten million cycles, the initiation life becomes the dominate contributor to total life.

Accurate fatigue life prediction from the IP model requires accurate stress predictions at the failure location. An inaccuracy in stress prediction of 10 percent can reduce the life prediction by an order of magnitude (Lawrence, 1996). For our case of the hub to disc joint this presents a problem since the classical stress calculations have to make assumptions about the boundary conditions on the inside and outside diameters of the disc. This problem can be over come through the use of finite element analysis (FEA) or Conveyor Dynamics PSTRESS software. For failures in the rim, shown in figure 1, the stress analysis becomes more complex. Classical methods for rim stress calculation of a pulley with center discs usually don't predict an actual stress near a failure site. To make the closed form equations

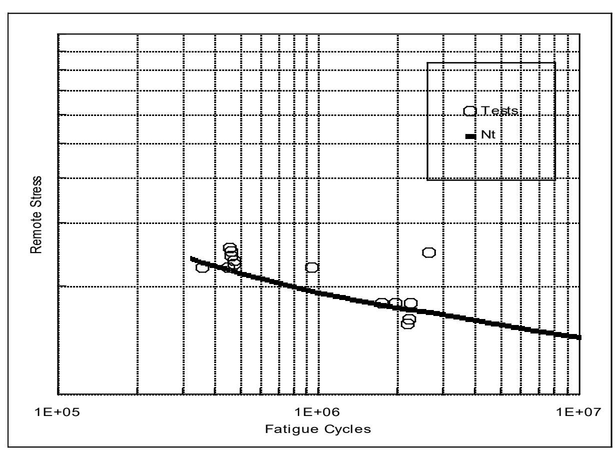

Figure 7 Comparison of initiation-propagation model to cruciform weld joint test results. Total life (Nt) estimate includes initiation (Ni) and propagation (Np) predictions.

solvable, these methods tend to make simplifying assumptions that can cause serious inaccuracies under certain conditions. In addition, these methods usually estimate a stress in only one location. This location may not be close to many of the potential failure initiation sites. In our experience, closed form solutions for rim stresses of center disc pulleys don't provide the accuracy required to use the IP model. When analyzing pulleys with center discs it is recommended that some type of finite element analysis be used to calculate stresses used for IP model predictions.

CONSTRUCTION EFFECTS ON LIFE

Using the IP model we are able to get an understanding of the effects of weld variables on the life of our hub to disc weld joint. As discussed earlier, research shows that changes in the mean stress at a weld failure site can significantly change its fatigue life. For the hub to disc weld, there are a number of fabrication conditions which can shift the fatigue life curves shown in figure 8. The near zero mean stress curve represents the fatigue life curve for a hub to disc weld joint used in a CEMA type pulley assembly including only the effect of hub clamping pressure. If the designer includes the effects of actual fabrication stresses, the positive mean stress fatigue curve would be representative of the fatigue life. Two of the primary contributors to these positive mean stresses are:

- hub pullout from clamping of single taper bushing systems;
- and deflection of hub and end disc during shaft installation.

On figure 8, a fatigue curve is given for a negative mean fabrication stress. This situation can occur when using multi-taper keyless locking elements. These devices compensate for the lack of a key by significantly increasing the clamping pressure on the hub and shaft. The high radial pressure on the hub can cause a beneficial compressive mean stress at the weld toe. When using this effect the designer must be certain the hub to disc weld toe compressive mean stress gets high enough to shift the fatigue curve. This is usually accomplished by reducing the hub radial stiffness, which causes a higher mean radial stress in the disc. A word of caution though, for the locking elements to perform properly sufficient hub stiffness must be maintained to prevent frettage fatigue failures at the locking element contact surfaces.

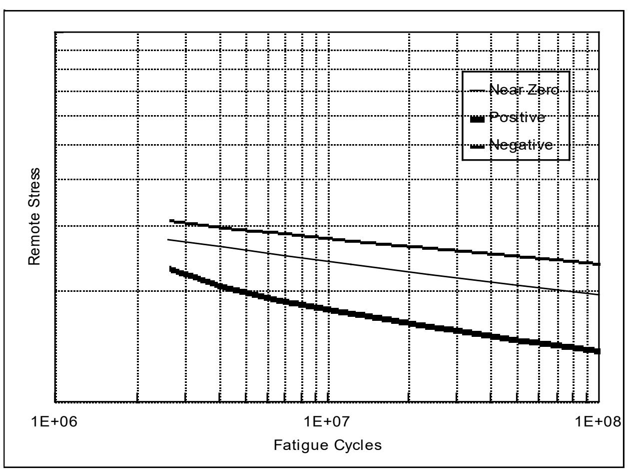

Figure 8 Total life (Nt) predictions from IP model for an ideal cruciform hub to disc weld. The "Near Zero" curve represents a pulley and shaft assembly without fabrication stresses included. The "Positive" curve represents the fatigue life including detrimental positive fabrication stresses. The "negative" curve represents beneficial compressive mean stresses from keyless locking assembly expansion.

The IP model also gives the designer the ability to include material thickness in the fatigue life prediction. Figure 9 shows total fatigue life predictions of the cruciform hub to disc weld for different thicknesses. From these curves it can be seen that total fatigue life is inversely proportional to plate thickness. This inverse proportionality can be a dangerous situation when using a nominal stress design approach. As stated earlier, nominal stress allowables are usually based on testing or experience. If design thicknesses are extrapolated beyond the thickness the nominal stress allowable is based upon, the potential for a reduction or loss of safety factor could be quite likely.

Often pulleys are thermal stress relieved to reduce weld residual stresses. This practice has been believed to improve the fatigue life of a pulley and the hub to disc weld joint, but little has been published to quantify the effect. The IP model gives the ability to include the effect of thermal stress relieve in life predictions. Fatigue life prediction results from the IP model are given including the effect of thermal stress relieve in figure 10. For comparison, the as welded fatigue life prediction is also included. As expected the life is significantly higher after thermal stress relief. Laboratory tests have shown a significant increase in fatigue life after thermal stress relief.

CONCLUSION

As pulley reliability demands continue to increase, the ability to predict the fatigue life of a pulley will become of

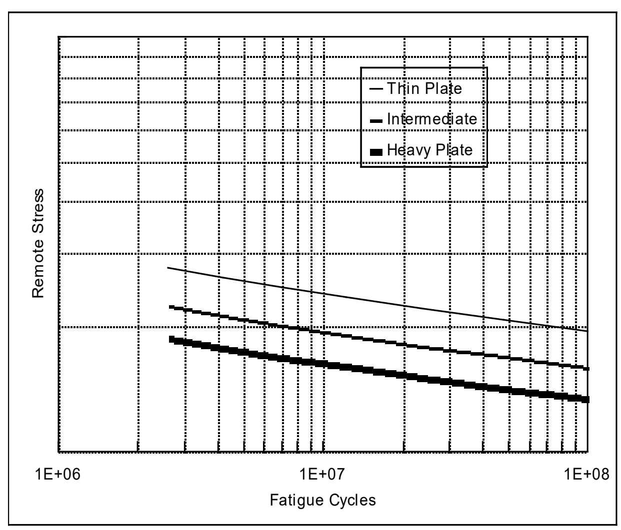

Figure 9 Total life (Nt) predictions from the IP model for an ideal cruciform hub to disc weld. Each curve represents the life

greater importance. Nominal stress methods have served

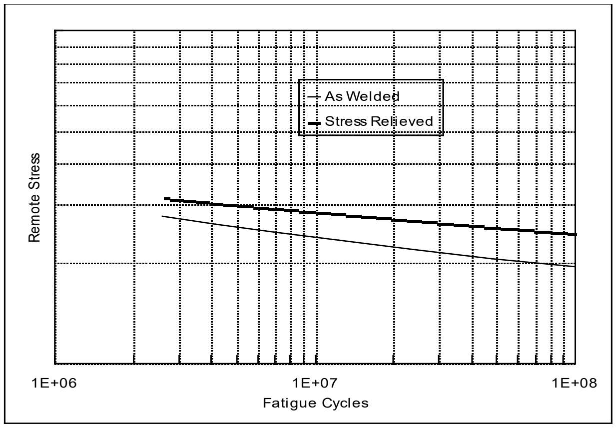

Figure 10 Total life (Nt) predictions from IP model for an ideal cruciform hub to disc weld in the "As Welded" and "Thermal Stress Relieved" states.

the industry well, but have limitations as conveyor speeds and belt tensions increase. Application of welding codes with the nominal stress design method can create many ambiguities. These codes may not apply to the weld joints, loading conditions, and failure modes typical of welded plate pulley constructions. Today's modern fatigue and fracture mechanics methods give pulley engineers the ability to predict the life of a weld joint.

Modern weld fatigue life methods have been incorporated into Dr. Lawrence's initiation-propagation model. As an example, the IP model has been applied to the pulley hub to disc weld. By thoroughly analyzing each weld joint in this manner, the IP model can be used to predict the fatigue life at each potential failure location. The IP model gives the pulley designer the ability to create a more balanced design by matching life predictions for each failure mode on a particular construction. Use of the IP model, utilizing modern FEA methods for accurate stress analysis, gives pulley engineer's the ability to confidently create more reliable and practical designs for all applications.

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