TITLE:

Performance Review of Keyless Locking Assemblies in a High Capacity Slope Belt Application

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ABSTRACT:

Mine site managers reported occurrences of locking assembly bolt failures on the Enlow Fork Coal Mine's slope belt conveyor. The observations were made after a number of years of operation and raised questions as to the status of the assemblies in operation. These questions prompted a detailed review of the situation by independent consultants. Using operational history of the conveyor and confirmation of belt loads by static and dynamic methods, a locking assembly fatigue curve was created to suggest expected service life. From this work suggestions were made to modify the pulley assemblies for extended service life.

INTRODUCTION:

Enlow Fork is a Consol Energy underground coal mine located in southwestern Pennsylvania. In 1990 Consol Energy created a team of experts and began design engineering on a new slope conveyor, which at the time was one of the largest slope conveyor installations of its kind. The latest technologies available at the time were used to design the drive components and conveyor pulley systems. This work lead to the conveyor's startup in March of 1991 with continuing service to this day. Installation of the slope conveyor has contributed to Enlow Fork becoming the world's most productive underground coal mine on a tons per man-hour basis. The conveyor has transported over 95 million metric tons (105 million tons) of raw coal since March 1991.

Since the slope belt is the critical conveyor for all of Enlow Fork's production, preventive maintenance has been a priority. Over the years some problems have appeared causing short periods of downtime. Maintenance managers have reported multiple incidents of problems with bearings and locking assemblies on the low-tension pulley assemblies. Thus far these problems have been detected and corrected without major incident. The main concern has been that as the system aged would these problems become a significant risk of catastrophic failure.

Concern for the risk of a catastrophic failure prompted Consol to reexamine the slope conveyor components with today's technology. To accomplish this task, a design review team from Consol, Precision Pulley and Idler, and Overland Conveyor was formed. This team was charged with reviewing the slope conveyors initial design, its operating status, and its maintenance history to create a recommendation on expected service going forward.

BASIC CONVEYOR DESCRIPTION:

The slope conveyor uses a 1.8 meter (72 in) wide 35 degree troughed steel cable belt to transport bituminous coal over an approximate horizontal length of 686 meters (2,250 ft) with a vertical rise of 199 meters (652 ft) at 3.2 meters per second (630 fpm). Typically coal is transported at a rate of 5,000 mtph (5,500 tph) with a maximum of approximately 6,350 mtph (7,000 tph). The belt is driven by two 1,865 KW (2,500 hp) motors, one each on pulleys P1 and P2. Pulley P5 has a mass attached and is free to slide on vertical tubes to create a vertical gravity take-up. The conveyor layout is as shown in figure 1. Both high tension pulley assemblies were designed identical and are represented by positions P1 and P2. All low tension pulley assemblies were

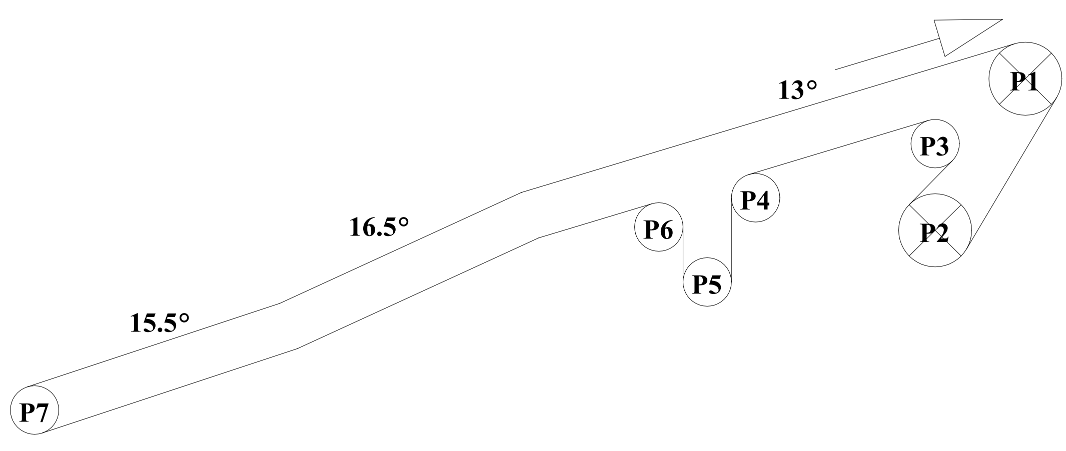

Figure 1: Slope belt conveyor layout.

designed identical and are represented by positions P3 through P7.

Figure 2: Locking assembly bolt fatigue failure.

DESCRIPTION OF PROBLEMS ENCOUNTERED:

Two problems with the low tension pulley assemblies have been encountered. One problem has been premature bearing failures in the P3 position. The second problem has been locking assembly bolt breakage and shaft frettage under the locking assemblies in pulley positions P3 and P5. The teams primary focus was on the locking assembly problems, although the investigation did provide information useful in the bearing analysis. Figure 2 shows a bolt that failed in one of the low tension pulleys. Inspection indicates it to be a fatigue failure initiating in the transition area from head to shank. Figure 3 shows the shaft surface frettage in the area where it contacts the locking assembly.

BELT TENSION ANALYSIS:

Prior to performing tension calculations the gravity take-up weight was isolated and weighed. This measurement indicated a static belt tension of 33,560 Kg (74,000 lb) was present at the take-up position P5. Review of initial design data indicated the system had been designed using a value of 27,483 Kg (60,500 lb). This variance was noted and the present value was used in all subsequent calculations.

The Overland Conveyor Company performed a complete dynamic analysis of the conveyor under starting, stopping, and normal running conditions. These analyses were also performed with the belt empty, partially loaded, at design load, and at peak loading conditions. For this work to be completed, inertia values of moving components were acquired and motor control characteristics were modeled. The motor control is accomplished by changing the resistor values as the conveyor

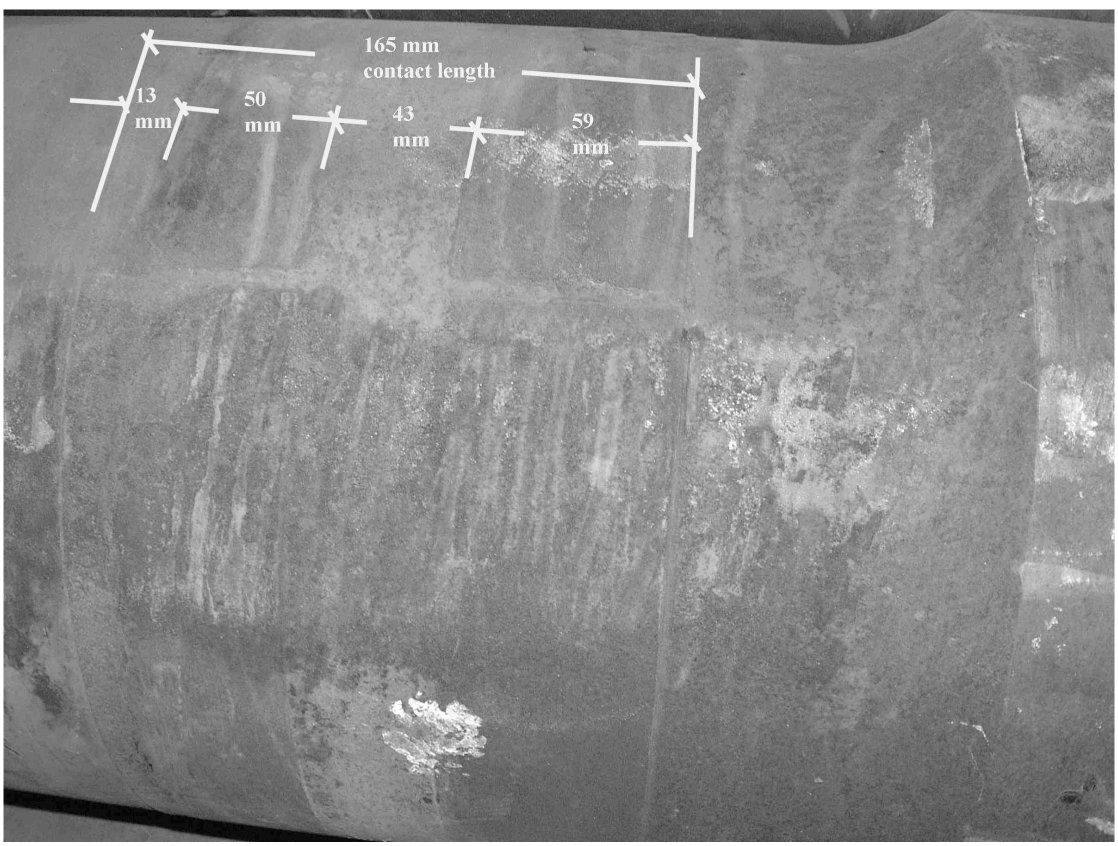

Figure 3: Example of shaft frettage in area of locking assembly contact. Thirteen millimeter (0.5 in) wide area on the left side of contact area is severely fretted. In this area the shaft diameter was reduced 0.6 millimeters (0.025 in) from the original diameter.

progresses through a start. This unique design required the modeling of three different motor controls and coordination of the time each was applied to the system. The model was verified by comparing take-up movement and motor current readings under various conditions.

Review of the dynamic analysis results indicated the slope conveyor's operating characteristics were controlled in most operating conditions. If the conveyor were to be started under peak load a slip condition could occur at the secondary drive, position P2. If the secondary drive slipped, it is possible a tension spike of up to 2.7 times normal running could be seen at the P3 position. It is possible this condition could be contributing to the bearing problem at P3.

One way of improving the bearing and locking assembly problems would be to reduce the counter weight mass. This idea was investigated and rejected due to the potential of secondary

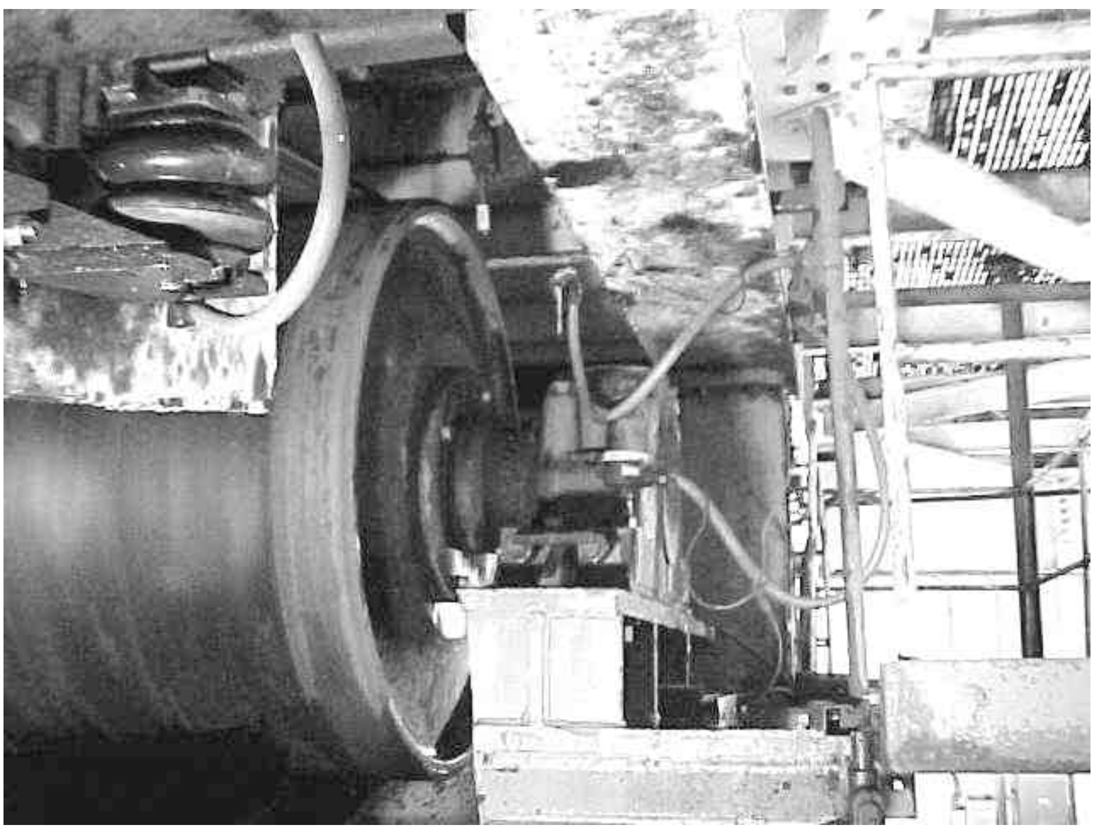

Figure 4: Low tension pulley in P3 position.

drive slip and the creation of excessive belt sag near the tail, position P7.

LOCKING ASSEMBLY LOADS:

The low tension pulleys P3 through P7 were designed identical and the basic pulley information is as follows:

- The outside diameter of the rim is 1,219 mm (48 in).
- The pulley face width is 2,083 mm (82 in).
- No internal discs were used with a rim thickness of approximately 28.5 mm (1.1 in).
- End disc design is a turbine "T" style with a width of 165 mm (6.5 in).
- Locking assemblies are a heavy duty series for a shaft diameter of 300mm (11.81 in).
- The shaft is designed with a 165 mm (6.5 in) wide stress controlling land at the locking assembly.
- Bearings are SAF type 22544 series with auxiliary seals on 2,794 mm (110 in) centers.

For a non-drive pulley application, the primary locking assembly design load is the bending moment transmitted through each locking assembly. Using belt loads calculated earlier, each of the low tension pulleys were analyzed using finite element analysis (FEA). Results of the FEA

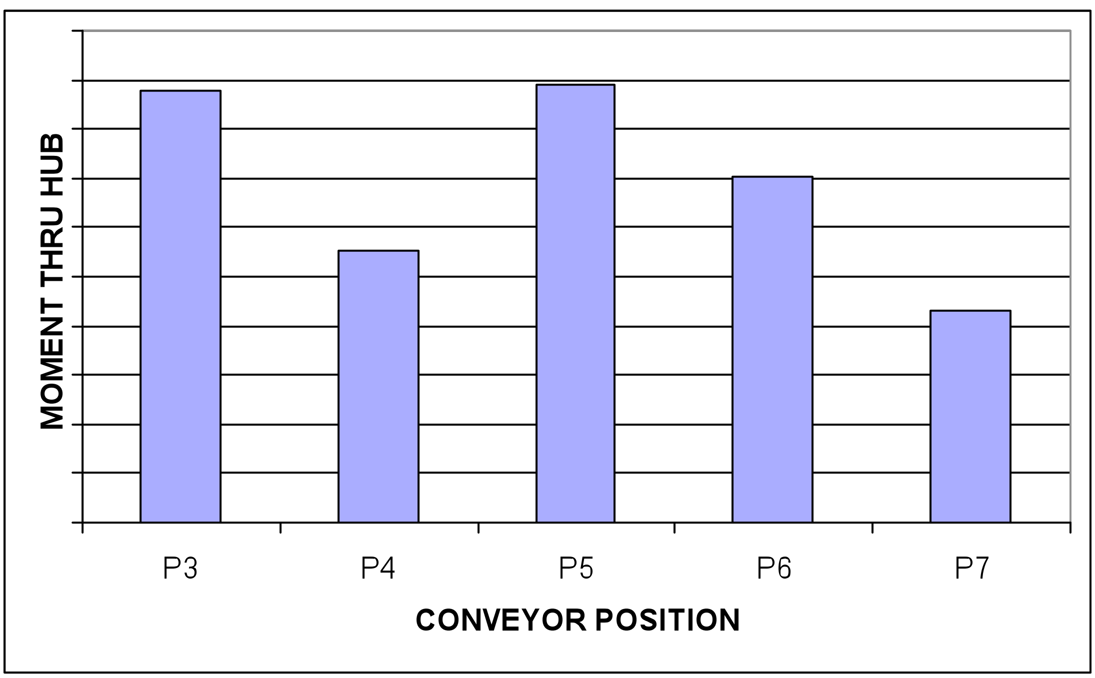

Figure 5: Bending moment through the locking assembly by conveyor position.

produced bending moments through the locking assembly as shown in figure 5. Site maintenance records showed that problems had occurred at P3 and P5 positions, which correspond with the positions of highest bending moment in figure 5.

BOLT FAILURE ANALYSIS:

The graph in figure 6 is a S-N type curve comparing actual field data with the locking assemblies load rating. Individual points on the graph are results observed in service. Points designated "RUNNING" represent locking assemblies in use that have not shown signs of failure and those removed from service for unrelated reasons. Points designated "FAILURES" represent locking assemblies where bolt breakage and frettage have occurred. The two design lines shown represent a manufacturer's locking assembly load rating and a proposed design line based on fatigue data.

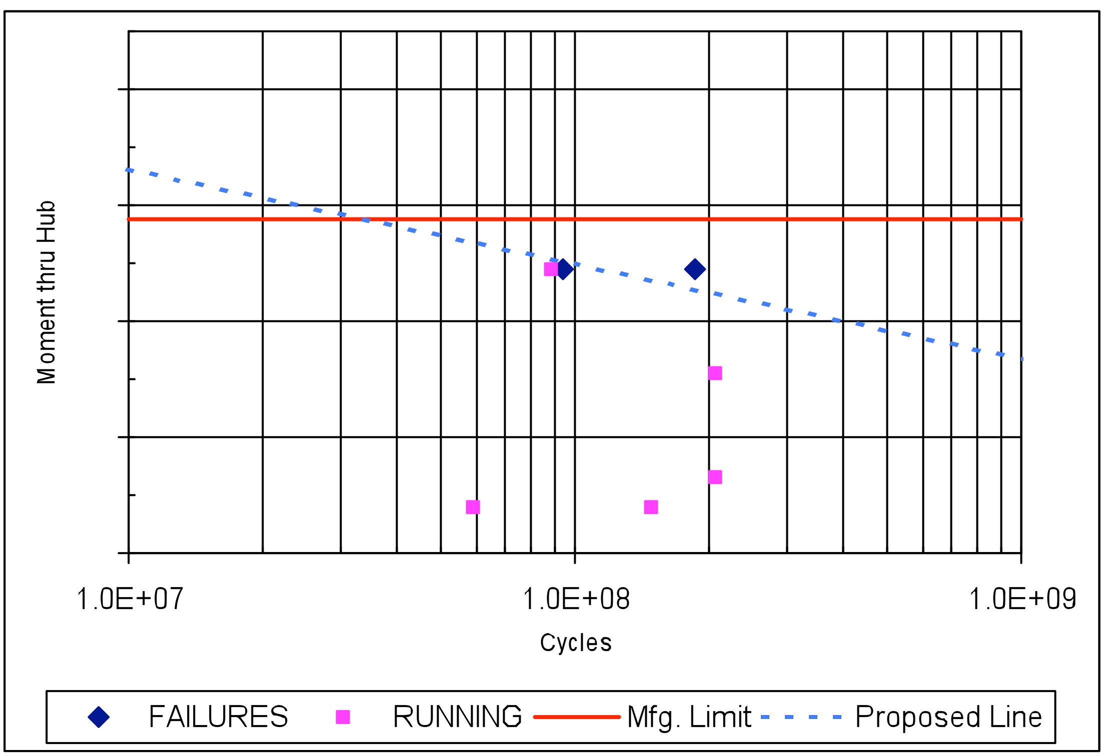

Figure 6: Locking assembly S-N curve.

Of the two reported failures, one can be considered premature due to installation discrepancies. When disassembled, it was observed that the locking assembly had been mounted over 30 mm (1.18 in) off center with the shaft lands and pulley hub. This offset caused a significant portion of the inner contact ring to have no shaft contact and reduced the hub contact with the outer ring the same amount. By all indications the second pulley was properly assembled, and since the failure's occurred at a load level less than the rated value further investigation is warranted.

The manufacturers locking assembly load rating presented has been reduced 10 percent from normally published values due to original pulley design specifications. The original design called for locking assembly bolt torques to be reduced by 10 percent. When using moderately reduced bolt torques, some locking assembly manufacturers recommend lowering published load ratings by the same percentage. They also qualify this as an oversimplification of the situation, since the relationship between bolt torque and load rating is not linear. This simplification is used because the relationship is nearly linear for a moderate reduction in bolt torque. The authors recommend consultation with the locking assembly manufacturer before attempting this modification.

An attempt at correlating the field data with analytical models of the locking assemblies behavior were made based on previous work, Schmoltzi (1974). Through manipulation of friction factors some correlation with the shaft frettage problem could be made, while predictions of the bolt failure and onset of frettage with the hub didn't correlate well. In addition, some locking assembly finite element work was performed. After starting this work, it was determined that trying to develop a universal locking assembly model was beyond this teams scope. The authors do see a need for additional work in this area and hope this topic will be discussed in future work.

The suggested safe running line shown on figure 6 is an estimate of the fatigue life expectations of the locking assembly bolts in this application. The bolts used in locking assemblies are typically made from heat treated carbon alloy steel consistent with SAE grade 8 specifications. Since this applications life expectations are greater than 10^8 cycles the bolts high cycle fatigue properties were of primary concern. For this reason only the elastic strain portion of the S-N curve is presented. It also should be noted that the line is sloped, which goes against the commonly accepted practice of considering 10^7 cycles as a transition to infinite life. A number of works referenced discuss this concept, and its acceptance appears to be increasing. One authors experience and research in high cycle weld fatigue supports the theory. The proposed design line uses a material fatigue strength exponent of –0.08 for the slope, which is then empirically superimposed over the field data.

SHAFT FRETTAGE ANALYSIS:

As stated earlier, only marginal success was realized in correlating analytical models with field observations. Existence of the frettage in itself wasn't of primary concern. The major concerns were that as the frettage progressed it effectively narrowed the locking assembly width and that the surface damage could initiate a fatigue crack.

Inspection of the FEA results showed stresses on the front ring of the locking assembly were higher than on the inner ring. This matched with observations that the frettage was primarily found under the front ring, with the inner ring being nearly free of frettage. Consultation with the locking assembly manufacturer indicated that a variance in stiffness between the outer and inner ring could be a contributing factor. Each ring had twenty holes for the bolts. In addition, the outer ring had nineteen holes providing access for removal. Ten of these holes were for removal of the front ring and the remaining nine were for removal of the back ring. The locking assembly manufacturer recommended using a variant to the standard product. This variant changes the removal process slightly and reduces the number of outer ring holes from nineteen to ten. This locking assembly design change significantly increased the outer ring stiffness and should reduce the onset of frettage.

Nondestructive testing of the shaft was performed in the area of locking assembly contact to determine if any fatigue cracks existed. These inspections did not show any signs of crack development. The frettage area had a rough surface finish with smooth transitions between variances in finish rather than sharp notches that would tend to initiate fatigue cracks.

TEAM RECOMMENDATIONS:

The team concluded the project with recommendations to mine maintenance managers on ways to reduce the occurrence of these problems in the future. A number of options were presented with the intent being to lower the locking assembly bending moment load at positions P3, P5, and P6. These included narrowing the bearing centers, increasing the shaft diameter inside the pulley, modifying the end disc profile to add flexibility, and installation of the new locking assembly design. In addition, installation of ceramic lagging on the secondary drive was recommended to reduce the possibility of slip at peak loads. At the time of publishing, these recommendations are still under review by maintenance managers so information on implementation isn't available.

CONCLUSIONS:

This paper presents the path the team followed in analyzing these occurrences of locking assembly bolt breakage and shaft frettage on the Enlow Fork slope conveyor. The investigation, and implementation of the recommendations, has the potential to reduce the risk of further occurrences of the problems seen. It should be noted that this conveyor, and its components, have been quite successful and continues to provide satisfactory service. The goal of this paper is to raise awareness that all pulley components are controlled by fatigue, and that the concept of a load rating threshold may be an oversimplification when high fatigue life is expected. It has been presented that further research in predictive models would be beneficial, and it is hoped that future work will be performed to address this.

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