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Holistic analysis approach and Design Improvement for Crane sheave early Failure

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Abstract

Even the best-designed engineering products can suffer from failures early in their lives either as a result of inadequacies in the design itself or caused by external conditions and unrestricted events. Failures are not easy to analysis and the precise mechanism of failure can often be difficult to identify. The objective of this study is to look briefly at what happens when crane sheave fails and to introduce the activity of redesign necessary to stop the failure recurring. For this a crane sheave is design as per standard BS 466 and shows how some simple design improvements are used, by following a design review and analysis of the failure. For the analysis purpose conventional FEM tool is used with load and boundary condition similar to reality and found that crane shave subjected to fail from the top edge due to excessive stress concentration and hence new modified design is suggested.
Keywords: Crane shave, Finite element analysis, Stress concentration

I. INTRODUCTION

Cranes use a system of pulley mechanisms to help lift the load. The pulleys may be single or ganged depending on the application. Pulleys are located within a fabricated steel sheaves an arrangement which holds the pulley spindles (or 'pins') so that the pulleys can revolve. The simple design shown in Fig 1 uses a 'hanger-type' sheave this is common on small utility cranes up to a safe working load (SWL) of 5 tones. The hanger is bent from low carbon steel sheet and fits over a machined pin which is supported by two frame plates. The 5 tone overhead crane had failed, dropping its 5 tone load of steel girders from a height of 2 meters on to the factory floor. It was clear that the failure was limited to the sheave hanger, the U-shaped bent steel bracket connecting the main rope pulley to the sheave pin and frame (Fig.1). Examination of the broken parts showed that the sheave hanger had fractured horizontally along the centerline of its bend, breaking cleanly into two parts. The sheave frame, pin and pulleys were undamaged. The fracture had happened without any warning- the load had not slipped, or jerked, and all the indications were that the crane was being operated correctly, within its safe working load capability, at the time of the failure. The crane was nineteen months old and had recently been inspected by an independent inspection authority (a statutory requirement) and its safety certificate renewed. The design of the crane is well proven, complying with British Standard BS 466 [1] and rated for a design lifetime of 2 x 10^6 cycles.

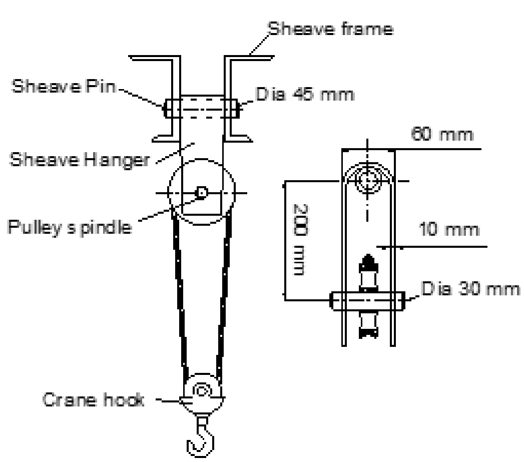

Fig. 1: The crane sheave assembly

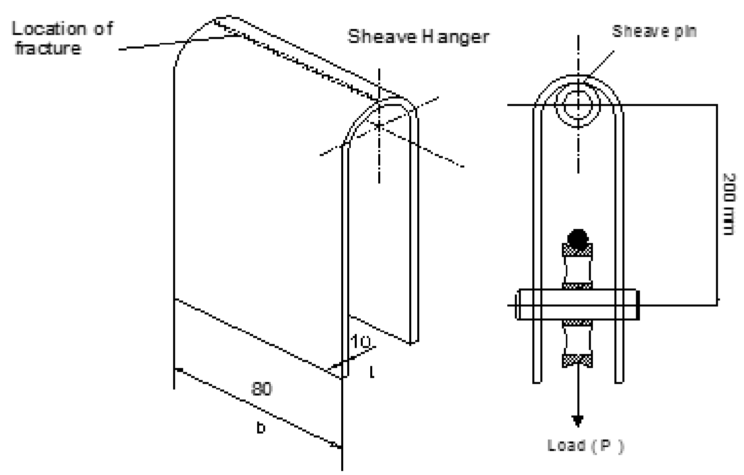

Fig. 2: The crane sheave hanger (Enlarge View)

II. CRANE SHEAVE EARLY FAILURE INVESTIGATION

The objective of analysis was to find what had caused the failure and to take any necessary actions to prevent a similar thing happening again. The investigators fell quickly that there are three things to consider during an operations review: working load, dynamic forces and fatigue life [4].

A. Working load

The crane was found to be correctly rated for a SWL of 5 tones, meaning that it was tested to 125 percent SWL during its pre-commissioning proof load test. Investigations showed that the weight being lifted when the failure happened was 4 tones, and that no heavier weights existed on the factory floor, so the crane could not have been overloaded.

B. Dynamic forces

An analysis was made to see if dynamic forces could have caused overstressing of the crane components. Two movements were considered: swinging and jerking. Figure 2 shows the situation; swinging of the load in either the x or y plane does not exert significantly more force as the assembly is free to move in either of these planes, a stress concentration of perhaps 1.2 or 1.3 being the maximum that could be expected. Jerking the load off the ground at the full hoisting speed would cause a significant dynamic effect. it is estimated that this could be up to a factor of 4. Combining these two, as a 'worst case' combination of events, leads to a minimum required factor of safety of about 5, to cater for dynamic forces.

C. Fatigue Life

The crane shave is design based on a 2 x 10^6 cycles fatigue life, one of the design life categories mentioned in BS 466. Calculation of expired fatigue life is as follows:
Design lifetime = 2 x 10^6 cycles over 25 years
= 220 cycles/day = 110 lifts/day
Expired lifetime = 570 days x 220 = 125 400 cycles
Estimated life expiry =125400 / (2 x 10^6) = 6.3%

From this approximate calculation it seems unlikely that the crane component failed directly as a result of the type of fatigue loading for which it was designed. This assumes of course that actual stresses experienced by the material are not higher than the 'fatigue limit' used in its design calculations.

III. Design Calculation

A design checks concentrates on the stresses experienced in the failed component the crane sheave hanger. The best way to approach this is to analysis the situation for an ideal stress case and then take a more pessimistic view, anticipating the effects of any worst case practical loading condition.

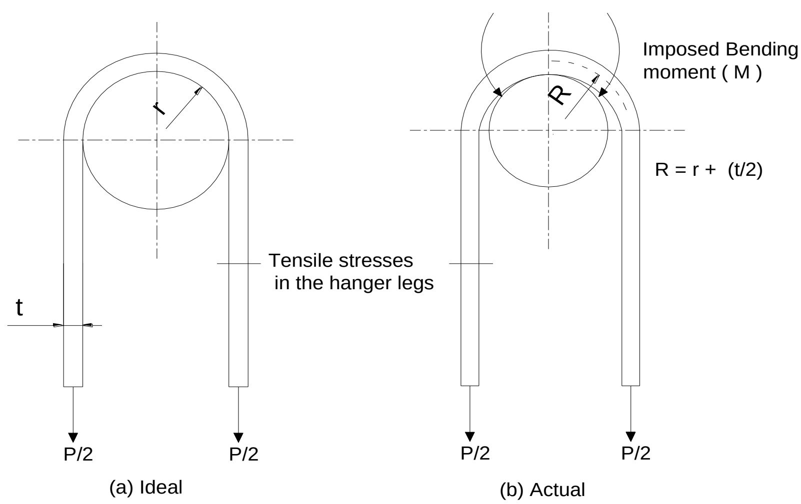

Fig. 3: Ideal and actual loading condition

Figure 3 shows the two anticipated states. Figure 3(a) is idealized loading condition in which the sheave hanger's internal radius fits snugly over the surface of the sheave pin. This produces pure tensile stress on the hanger legs as follows:

$$\sigma_{\text{tensile}} = \frac{\frac{P}{2}}{bt} = \frac{5000 \times 9.81}{2 \times 0.08 \times 0.01}$$

For a medium carbon steel with a yield stress of about 300 MN/m^2 this gives a nominal factor of safety of 12. This is higher than that needed to allow for dynamic loadings. So in theory, the failure should not have occurred. Figure 3(b) looks at an actual loading condition where the sheave hanger radius does not accurately fit the sheave pin; the main contact is over the top 20-30 leaving a clearance in the horizontal plane. This changes the loading condition significantly. The load (P) now causes a bending moment.
Maximum bending moment (M) [12,13]:

$$M = PR\left(\frac{R\,(2-\pi)}{a + 2\pi R} + \frac{1}{2}\right) \mathrm{Nm}$$
$$M = 5000 \times 9.81 \times 0.0275\left(\frac{0.0275\,(2-\pi)}{0.2 + 2\pi \times 0.0275} + \frac{1}{2}\right) \mathrm{Nm}$$
$$M = 560.87\ \mathrm{Nm}$$

Solving for maximum bending stress using [12,13],

$$\sigma_b = \frac{M}{Z}, \text{where } z = \frac{bt^2}{6}$$
$$\sigma_b = \frac{560.87}{0.08 \times 0.01^2/6} = 420.6\mathrm{MN/m^2}$$

This is higher than the nominal yield stress (YTS). So, even taking into account the uncertainties and assumptions of the above calculation, it is likely that the existence of such a bending moment will provide the conditions for failure, particularly when combined with the dynamic loads and fatigue effects.

IV. BULK MATERIAL PROPERTIES OF PLANE CARBON STEEL

In these study plane carbon steel is used as crane shave material. Mechanical properties of these material are listed in table 1.

Table - 1
Mechanical Properties

E	*v*	*UTS*	*YTS*
210 GPa	*0.28*	*399 MPa*	*220 MPa*

V. FINITE ELEMENT MODEL AND ANALYSIS

A crane shave of plane carbon steel of 80.0 mm width and 200.0 mm height and 10.0 mm thick is considered for simulation as shown in figure 2 and its FE model is shown in figure 4, To determining the stress of shave under the loading conventional FEM tool is used. For the analysis purpose object symmetry is used and applied boundary conditions were similar to experimental boundary conditions; bottom portion of shaves is subjected to load and upper portion of shave is encaster as shown in figure 5.

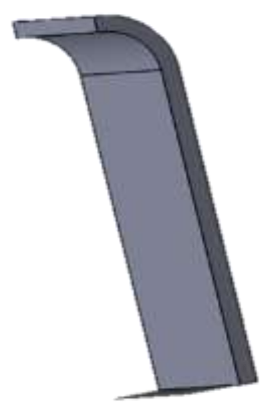

Fig. 4: FE Model of symmetric crane shave

From figure 6, it is clear that excessive bending stresses are likely, owing to the bending moments caused by radial clearances between the sheave hanger and its sheave pin. These bending stresses will almost certainly exceed the material's limit (i.e. yield stress (YTS)) under some conditions. This is what caused the failure and shave subjected to failure from top edge hence it is necessary to modified design of crane shave. Simple design change which will help eliminate this type of failure. The sheave hanger is fitted with a steel tube welded to its inside radius. The sheave pin is a good 'running fit' in the tube, thus eliminating the variation in radius of the contacting surfaces that caused the imposed large bending moments. The result of modified design is shown in figure 8 with applied load and boundary condition (i.e. inner radius is fixed and bottom portion of shave subjected to loading (figure 7)).

Fig. 5: Applied load and boundary condition

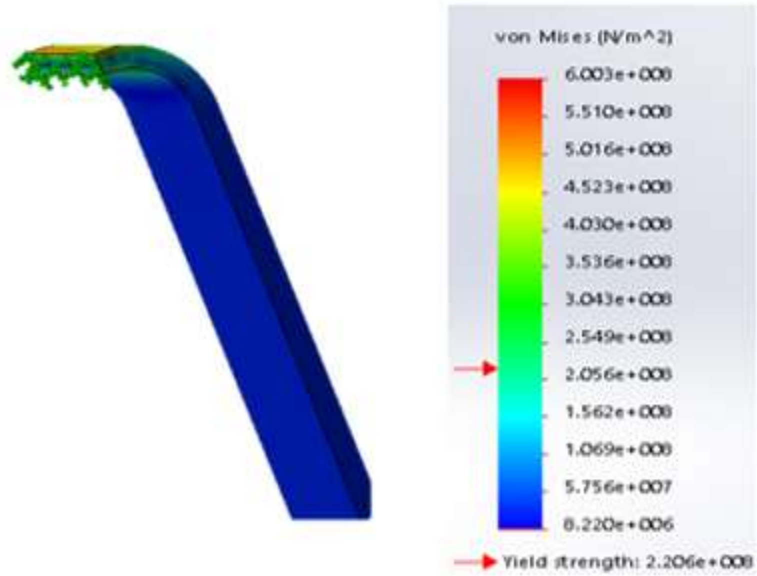

Fig. 6: FE results of conventional design

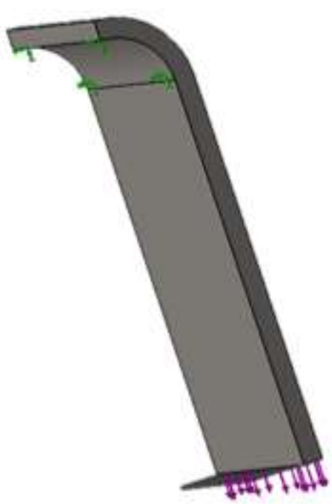

Fig. 7: Applied load and BC of modified design

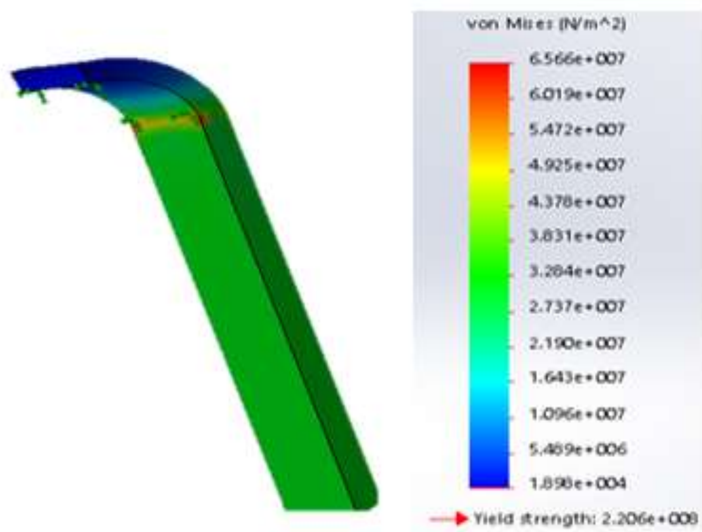

Fig. 8: FE results of modified design

VI. CONCLUSIONS AND RECOMMENDATIONS

The findings so far are quite typical for a simple mechanical failure event. The results seem clear indication of a fatigue mechanism having been in operation, affecting first the outer radius of the crane sheave hanger bend. The work hardened and coarsened microstructure are significant findings but are mainly consequential caused by the fatigue mechanism, rather than being a separate cause themselves. The operational review has proved useful because it has effectively eliminated operational factors from the cause of failure. Dynamic stress concentrations that will be caused by swinging or jerking of the load are well within the factors of safety built into the crane design. We have shown that excessive bending stresses are likely, owing to the bending moments caused by radial clearances between the sheave hanger and its sheave pin. These bending stresses will almost certainly exceed the material's fatigue limit and possibly also its yield stress (YTS) under some conditions. This is what caused the failure. A simple design change which will help eliminate this type of failure. The sheave hanger is fitted with a steel tube welded to its inside radius. The sheave pin is a good 'running fit' in the tube, thus eliminating the variation in radius of the contacting surfaces that caused the imposed large bending moments. The addition of the tube is a practical step- an equally good solution would be simply to bend the sheave bracket to the same radius as the sheave pin so it sits snugly over it. Practically, this is difficult and it is not easy to achieve close radial tolerances on the inside radius of bends, so the addition of the tube is a better solution

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