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Failure Investigation of a Wheel Type Ride

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Abstract

A Rotating Giant wheel ride at an amusement Park in Lahore went through an accident and fell on the ground, resulting into three casualties and injuries of two school kids. It was noted that the ride failed just two weeks after its installation. The central shaft which was acting as a cantilever was broken into two pieces. Onsite visual inspection along with hardness measurements was carried out within few days after the accident. It was observed by examining the broken shaft that fatigue failure occurred due to the sudden step made on peripheral of the shaft during manufacturing. This step which was machined to fit the size of bearing caused enough stress concentration to develop on the outer surface of the shaft. It was also noted that the nominal load was exceeding the maximum allowable load. It was shown by detailed calculations that the shaft itself was not able to withstand full load even in the absence of any stress raiser, and the cross-section of the axle was 'just sufficient' to withstand the bending force applied on the axle (shaft) when the giant wheel ride was under full load. The shaft was modeled and analyzed later on by using ANSYS workbench to simulate and verify the theoretical calculations.

Key Words: Fatigue failure, Bending stresses, Stress concentration

1. Introduction

Joy Land rides and all such machines in which human life is involved are generally designed with a much higher safety factor but unfortunately the accidents still take place [1-3]. Worldwide most of the accidents are due to the failure of welded joints [4-6], while some of the mishaps are due to rust and ignorance in maintenance [6-10]. However, it is very unusual that some very simple errors in designing of critical sections cause the life-threatening incidents to happen.

At an amusement park in Lahore a giant wheel ride went through a failure and collapsed onto the ground causing the loss of three youngsters and wounding two others. The administration of the park informed that the ride was installed just two weeks before this unfortunate incident had happened. Fig. 1(a) shows the picture of this unhappy event, while in Fig. 1(b) the magnified view of the broken shaft can be seen, and Fig. 1(c) presents the geometrical drawing of the giant ride. The giant ride had a diameter of 42 feet that revolved in the upright-plane.

Rotating wheel was made up of 12 legs which were acting as cantilever beams and were bonded (through welding) to the middle hub. The free ends of the legs were attached to a cradle that had a capacity of 6 persons each. Therefore, the giant ride had the total capacity of 72 people at a time.

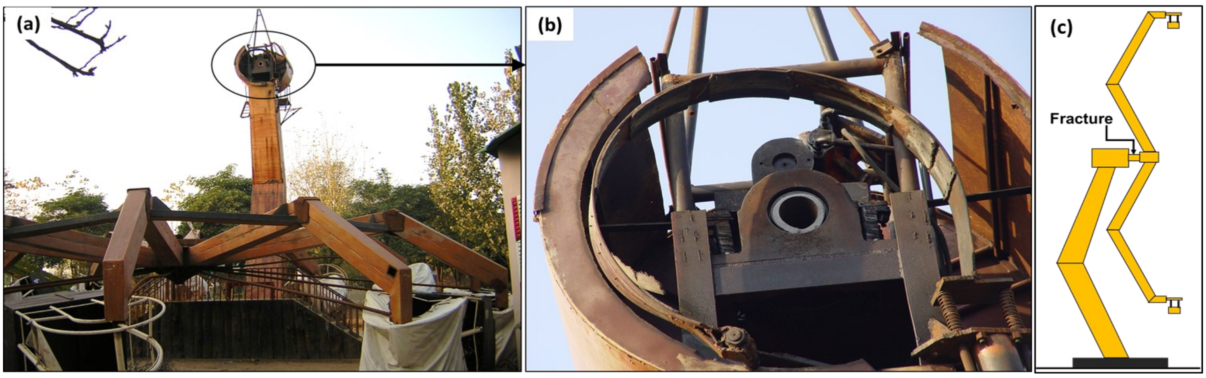

Fig. 1: (a) Photograph of the broken ride, taken few days after the event took place, (b) magnified view of the broken shaft, and (c) a drawing of the huge ride showing its main components

A cantilever shaft which was passing through the center of the giant wheel allowed the ride to remain in vertical plane. The other side of the circular shaft was welded with revolving mechanism. The rotating shaft, gear drives and the supports were placed in a horizontal position on the top of a column that was around 25.0 feet tall.

Visual inspection of the debris of the incident indicated that the accident might have happened due to a crack which could have eventually caused the failure of the shaft, while a substantial loss occurred to the cradles and the metal structure. Therefore, metallurgical inspection of the material of the shaft as well as the rupture mode became the prime considerations for this failure investigation.

2. Constraints of the Central Shaft

Shaft type: Hollow

External diameter of shaft: 8 in.

Nominal wall-thickness: 1.2 in.

A portable hardness tester was used to measure the hardness of the central shaft. The hardness was taken as the average of 12 readings. By using these values of hardness, the strength was calculated as follows:

Brinell Hardness: 220 BHN (average of 12 readings)

Estimated UTS (BHNx500): (~50 tons/in^2) 110,000 psi

Estimated Yield: 33 tons/in^2 (70 % of UTS)

3. Material Analysis

It was observed that the shaft of the Giant Wheel had failed due to rotational bending that caused fatigue fracture [11]. There was a crack 0.315–0.394 in. (8-10) mm deep on the circumference of the shaft. Photos shown in Fig. 2 illustrate the fracture on surface, wherein, the arrowheads point to the portion which has brittle fracture and fatigue regions. It was observed from the fractured surface that the fatigue-crack region was grey in color, whereas the brittle fracture was of reddish brown color, majorly due to metallic rust. Unlike the brittle region, the fatigued region was oily. It was a clear indication that the lubrication had seeped inside the fatigue crack, and this oil had acted as an anti-rusting agent.

Fig. 2 (a) shows that the shaft was machined with a step (sudden reduction in diameter). Fig. 2(b) shows a portion of the crack running through the inner side of the step. The disastrous failure of the middle shaft was due to the same crack. It was noticed that the sharp-step on the middle-shaft was present at the point where the bending moment was maximum i.e. at the most critical point. Fig. 3 shows that how the existence of the sharp step on the middle shaft had made it very dangerous.

Since the failure occurred after two weeks of the installation of the ride, it was essential to evaluate the amount of the total load applied on the shaft.

4. Structural Analysis

A trustworthy data about the weights of different segments of the rotating giant wheel was needed to determine the stresses in the middle shaft. On the contrary, it was very difficult to determine the actual weights of different parts of the giant wheel ride. Therefore, there was no other option except to estimate the weight of different sections and then add them all as below. The shaft was loaded in two parts:

1. 12 cradles full of people will make a total load of ~5 tons.
2. Total weight of 12 legs (which connect the central hub with the cradles) along with its linked assembly was taken to be ~2 tons.

The moment arm for 5 tons load was estimated to be 5.5 feet and the moment arm for the load of 2 tons was taken as 2 feet. Additionally, the section modulus for the middle shaft was evaluated to be 38 in^3.

As described in section 2 above, the projected ultimate tensile strength for the shaft is approximately 50 tons/in^2, and an allowable yield strength of approximately 33 tons/in^2. But the shaft failed at 10 tons/in^2 which is 20 % of the UTS (ultimate tensile strength) so it means that there must be some other reason for the failure of the shaft. So it can be said that the stress concentration regions were developed on the shaft where there was sudden decrease in diameter that caused the shaft to fail at much lesser stress than UTS. The sudden decrease in the diameter must have tripled the stress at the critical point. Therefore, the base of the sharp step was exposed to a tensile stress of around 20-30 tons/in^2 i.e. approximately 50 % of the UTS. This approximation of working stress is validating the fact that the middle-shaft was subjected to fatigue-loading.

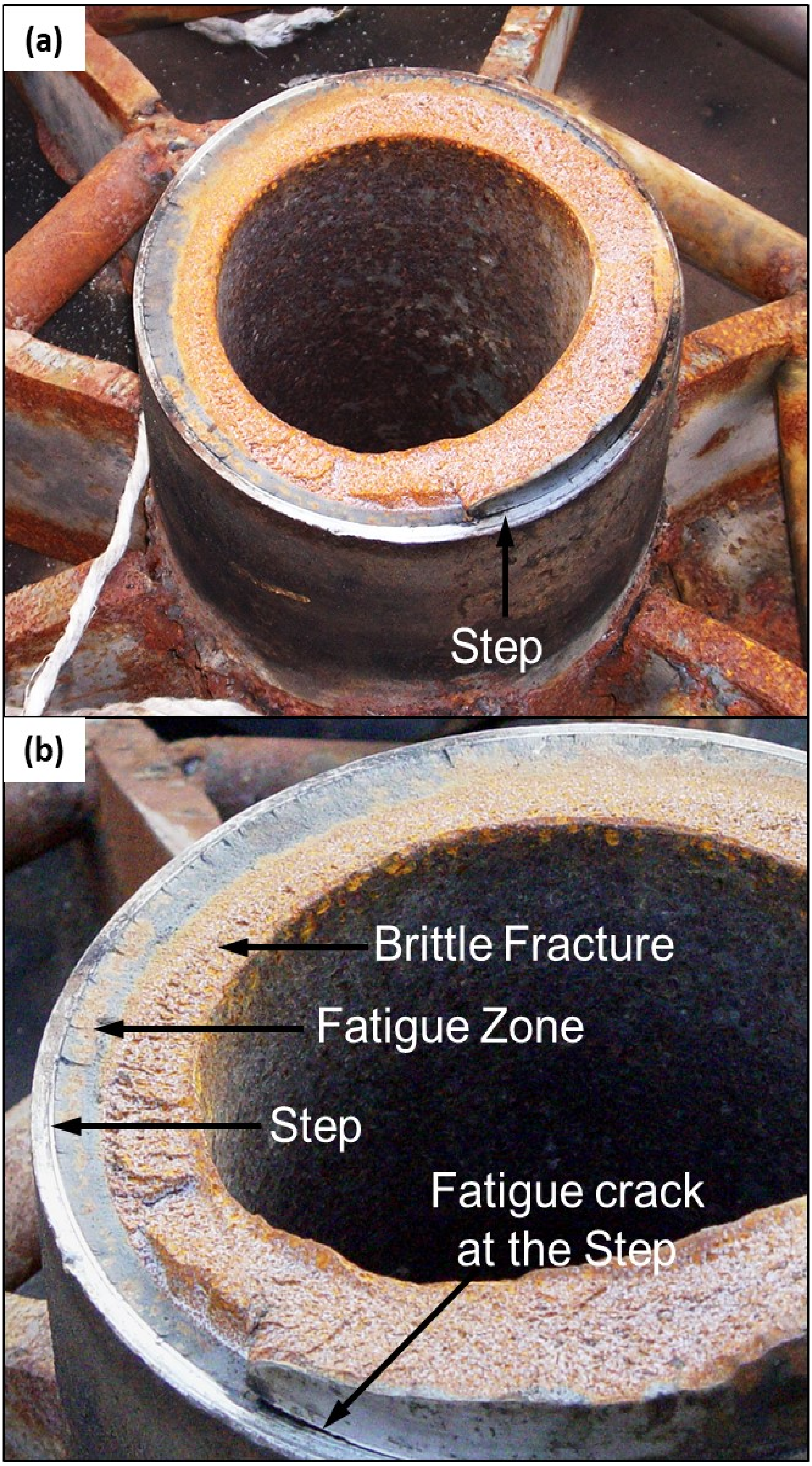

Fig. 2: Fractured face of middle-shaft showing: (a) the 'step' from where the fracture had started and (b) Various phases of crack-propagation

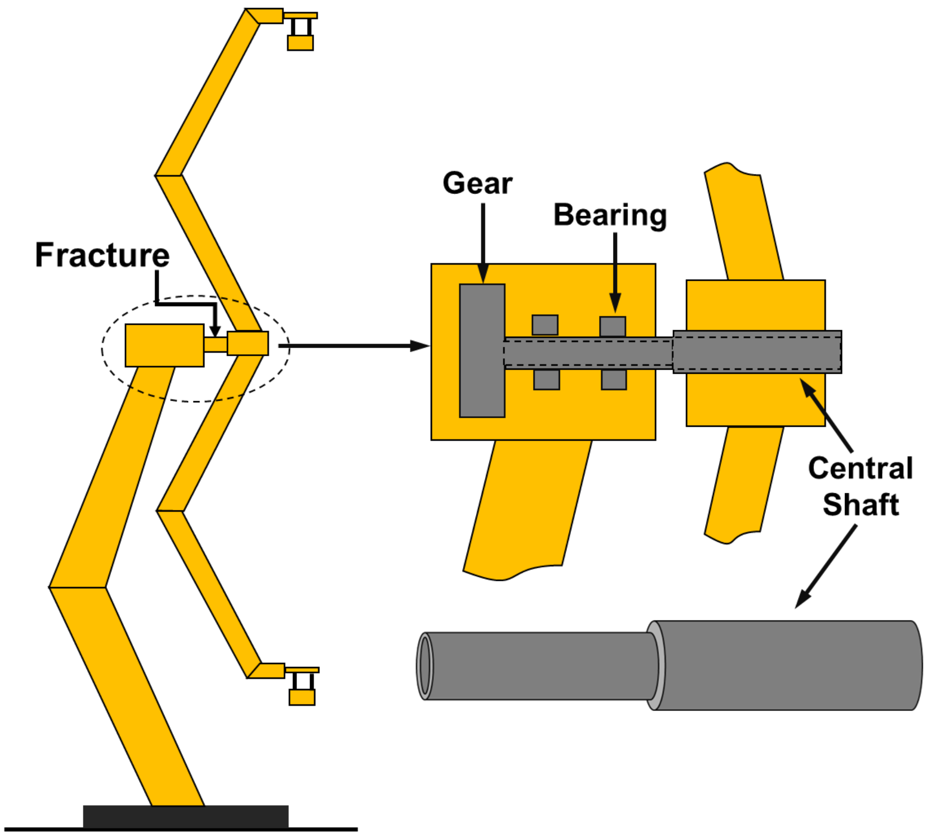

Fig. 3: Schematic diagram of the middle shaft and the components attached to it

The step was very sharp (as shown in Fig. 2) with a depth of 0.177-0.197 in. (4.5-5 mm). But for the approximation of stress concentration factor (with reference to Fig. 4), the radius of curvature 'r' was taken equal to 0.0394 in. (1 mm) approximately at the root of the shaft, and the external diameter of the shaft was used as 8 in. (D = 203 mm). Using these values;

$$\frac{r}{D} = 0.005$$

The smaller (reduced) diameter after step was taken as 7.48 in. (d = 190 mm), hence;

$$\frac{D}{d} = 1.05$$

Using above data, the stress-concentration factor was calculated using the graph given in Fig. 4. The graph shows that the stress-concentration factor increases exponentially with the decrease in r/D value. The graph shows that in the current case the stress-concentration factor was approximately 3.

As the middle shaft of the giant wheel was the utmost critical part, so the FOS (factor of safety) should be much more than the current value.

Sudden decrease in diameter of shafts is extremely objectionable and should be avoided lest there is no alternative, and sharp edges should be replaced by fillet radius or through proper chamfering. In this incident the shaft was machined to a smaller radius in order to fit the available ball bearings in it, which became a very serious mistake in the manufacturing process.

5. Computational Analysis

Stress analysis of the shaft was carried out in ANSYS workbench version 15.0. The results were very much similar to the theoretical calculations. A central hollow shaft of outside diameter of 8 in. and wall-thickness of 1.2 in. was modeled in Creo parametric 2.0. Shaft was divided in 908327 tetrahedral elements with 1248631 nodes, and element edge length was 0.2.

Total moment of 756000 lb-in (as suggested in structural analysis) was applied on one side of the shaft keeping the other side fixed. Maximum stress of 39763 psi (20 tons/in^2) was achieved at the step (see Fig. 5) which was same as calculated in structural analysis (i.e., 20-25 tons/in^2). Density of dots is showing the variation of stresses at different points along the length of the shaft.

When the sharp edge was replaced by a smooth round curve, the stress was reduced to 22 tons/in^2 which is still very high. Previously the factor of safety (FOS) was kept equal to 1.5 which

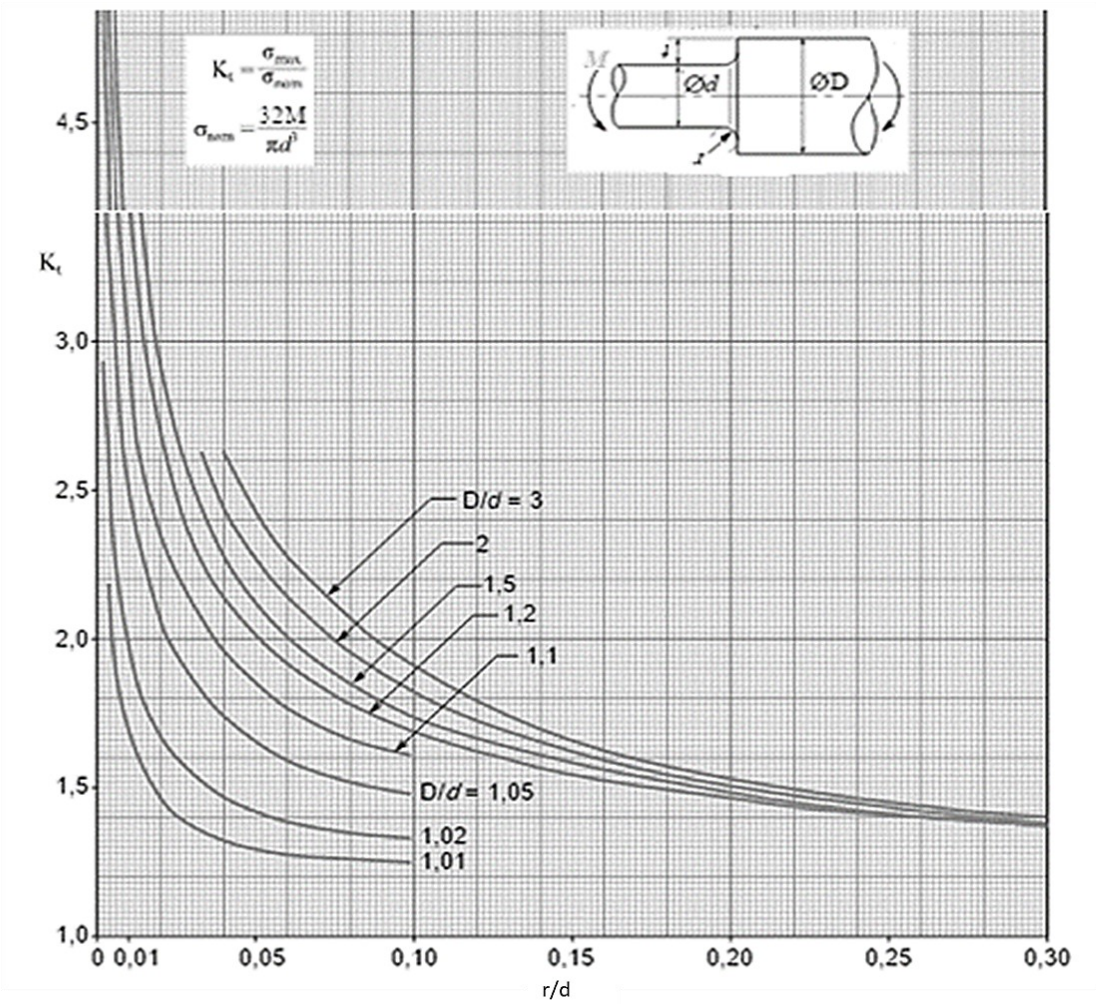

Fig. 4: Stress concentration factor versus diameter ratio [12]

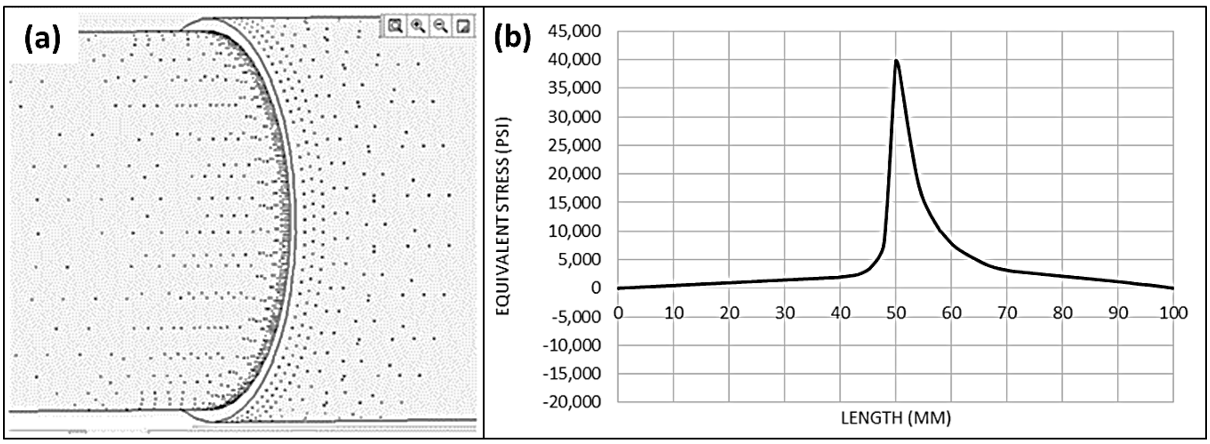

Fig. 5: Variation of stresses along the length of 8 in. diameter shaft as calculated using ANSYS workbench

was extremely low. As a result of the present analysis, it was suggested to increase the FOS to approximately 10 and the new diameter of the shaft for the safe operation of ride was proposed to be 16 in. Analysis of 16 in. diameter shaft showed that the stresses were reduced to a safe level of just 5036 psi (03 ton/in^2) as shown in the graph of Fig. 6.

6. Conclusions and Recommendations

The detailed failure investigation shows that it was a fatigue failure due to which the rotating wheel fell down in an amusement park in Lahore. Fatigue was aided by stress concentration formed due to sudden reduction in the external diameter of the

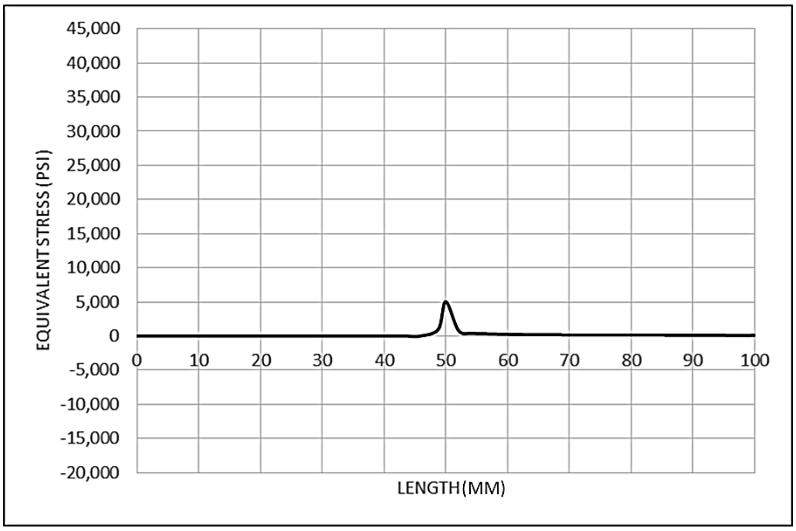

Fig. 6: Reduced stresses due to thicker shaft of 16 in. diameter as calculated using ANSYS workbench

middle shaft. The structural and computational analysis validated the reasoning of failure.

In order to redesign the giant wheel ride and to avoid failures following recommendations are made.

- The most critical part of the Giant Wheel was the central shaft, so the Factor of Safety (FOS) should be kept much more than the current value.
- Since proper heat-treatment improves the fatigue strength of the steel, such shafts which are heavily loaded, as in this case, should be made up of heat-treated steel.
- Such machines whose failure can risk human lives must be inspected on regular basis.

7. References

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