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Failure Analysis and Redesign of Shaft of Overhead Crane

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Abstract
This paper deals with the failure analysis and redesign of shaft of overhead crane having capacity 25 tonne. There is problem of failure of the shaft in gear box which is mounted on the crane. The shaft breakage occurred due to dynamic, alternating low tensile– compressive stresses and simultaneous torsional load.
An overhead crane is mechanical equipment for lifting and lowering a load and moving it horizontally, with the hoisting mechanism which is an integral part of the machine.
To carry out the failure analysis of shaft it is necessary to model the shaft in any modelling software like PRO-E, CATIA etc. Then analysis of existing shaft is done analytically and with ANSYS11. The shaft is then redesigned with suitable material available in data book of machine design and then analysis of new design of shaft is done with the help of ANSYS-11 software.
Keywords— Shaft failure, Overhead crane, failure analysis, stresses.

I. INTRODUCTION

A shaft is a rotating member usually of circular cross-section (solid or hollow), which is used to transmit power and rotational motion in machinery and mechanical equipment in various applications. Most shafts are subjected to fluctuating loads of combined bending and torsion with various degrees of stress concentration. For such shafts the problem is fundamentally fatigue loading. Failures of such components and structures have engaged scientists and engineers extensively in an attempt to find their main causes and thereby offer methods to prevent such failures.

Eccentric Shaft is widely appreciated for its features like corrosion resistant, long service, effective performance and reliability [1].

A crane is [2] mechanical equipment for lifting and lowering a load and moving it horizontally with the hoisting mechanism an integral part of the machine. A crane with a single or multiple girder movable bridge, carrying a movable trolley or fixed hoisting mechanism and travelling on an overhead fixed runway structure is known as overhead crane. Material handling is a vital component of any manufacturing and distribution system and the material handling industry is consequently active, dynamic and competitive. Overhead crane is used for material handling purpose and hence it is very useful for any industry.

Figure 1: Shaft in the gear box of overhead crane

II. Background of failure analysis

Failure analysis is [3] the process of collecting and analyzing data to determine the cause of a failure and how to prevent it from recurring. It is an important discipline in many branches of manufacturing industry. Such as the electronics industry where it is a vital tool used in the development of new products and for the improvement of existing products. However, it also applied to other fields such as business management and military strategy. Failure analysis and prevention are important functions to all of the engineering disciplines. The materials engineer often plays a lead role in the analysis of failures, whether a component or product fails in service or if failure occurs in manufacturing or during production processing. In any case, one must determine the cause of failure to

prevent future occurrence or to improve the performance of the device, component or structure.
Failure analysis can have three broad objectives.

1. Determining modes of failure.
2. Failure Cause
3. Root causes.

Failure mode can be determined on-site or in the laboratory, using methods such as fractography, metallographic and mechanical testing. Failure cause is determined from laboratory studies and knowledge of the component and its loading and its environment. Comparative sampling or duplication of the failure mode in the laboratory may be necessary to determine the cause. Root failure cause is determined using knowledge of the mode, the cause and the particular process or system. Determining the root failure cause require complete information about the equipment's design, operation, maintenance, history and environment. A typical failure analysis might include fractography, metallographic and chemical analysis. Failure analysis of a rear axle of an automobile was discussed in [4].

Figure 2: Failure of shaft

III. Causes and Analysis of Shaft failure

1. Causes of failure

Austin H. Bonnett [5] discussed the causes of shaft failures. He has focused on failures associated with fatigue. XU Yanhui [6] says that shaft damaged can be induced by sub synchronous resonance (SSR).

According to J. feller [7] fatigue loading on wind turbine drive trains due to the fluctuating nature of wind is major cause of premature failure of gearboxes.

Table 1: Causes of shaft failure

Cause of shaft failure	Percentage
Corrosion	2
Fatigue	25
Brittle fracture	16
Overload	11
High temperature Corrosion	7
Stress concentration fatigue	6
Creep	3
Wear, abrasion and erosion	3

The shaft failed due to fatigue, which arises due to following reasons [8].

a. Presence of cyclic over-loads.
b. Stress concentration: They may be due to production or operation causes e.g. under cuts, machining, traces, notches etc.
c. Wrong adjustment of bearing, insufficient clearances.

In corrosion failures, the stress is the environment and there action it has on the shaft material. At the core of this problem is an electrochemical reaction that weakens the shaft.

Corrosion is a process that occurs when oxygen, water, acids and salts mix together. The temperature must be above 0˚C, when the relative humidity is below 40% almost no corrosion from 40-60% (relative humidity) significant corrosion is to be expected [9].

IV. Design of shaft

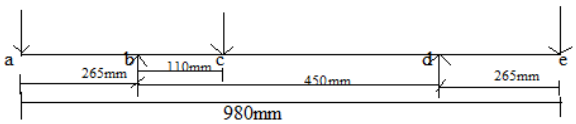

Figure 3: Free body diagram of shaft

The shaft is connected to motor from one side.
Power of motor (P) = 60 HP = 44.13 KW
N (RPM of motor) = 980 RPM
α (Pressure angle) = 20˚
Force on brake drum = 1081.75N
Torque acting on shaft

$$T = \frac{P * 60}{2\pi N}$$

T = $430*10^3$ N-mm

We know that
Diameter of gear Dp = No. of teeth on gear*module
Dp = 15*4= 60mm
Radius of gear
R=30 mm
Assuming that torque at C, therefore tangential force on the gear

$F_{tc} = \frac{430*10^3}{30} = 14333.33N$

Wa = F_{tc}* tan α
=32065.22 N
Bending moment at point B and C is 286663.7 Nmm and 1559053 Nmm respectively.

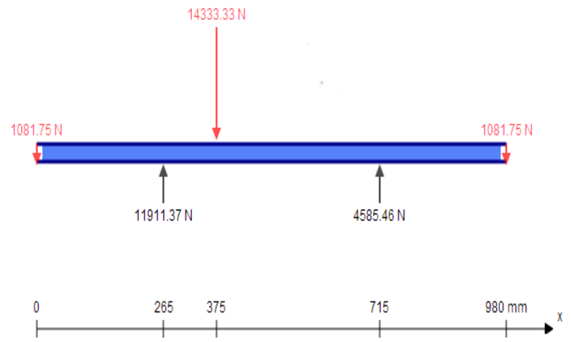

Figure 4: Reaction forces on the shaft

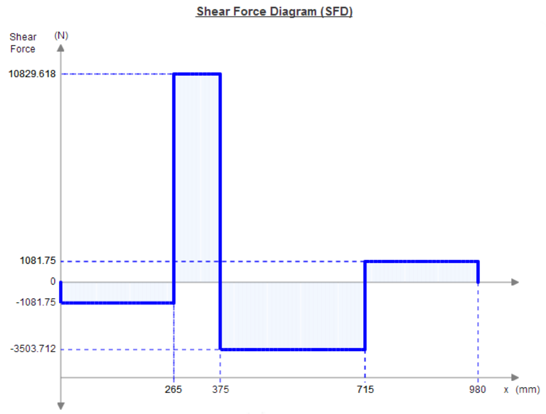

Figure 5: Shear force diagram

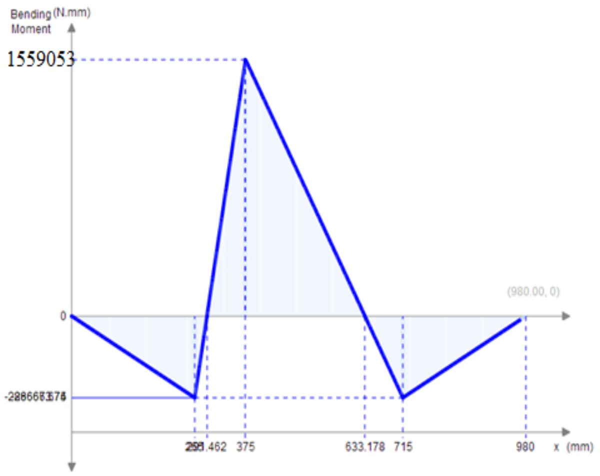

Figure 6: Bending moment diagram

Bending moment due to axial load

$M_2 = Wa*R$

$= 9601950$ Nmm

Resultant bending moment

$M = \sqrt{{M_c}^2 + {M_2}^2}$

$M = 9727697.058$N-mm

$T_e = \sqrt{(Km * M)^2 + (Kt * T)^2}$

$T_e = 14597880.07$N-mm

$$\tau = \frac{16Te}{\pi Dp^3}$$

$\tau = 344.19MPa$

Equivalent bending moment

$M_e = \frac{1}{2}\,[km * M + Te]$

$M_e = 14594712.79$ N-mm

Bending stress acting on shaft is given by

$\sigma_b = \frac{32Me}{\pi d^3}$

$\sigma_b = 650.243$MPa

The equivalent torque is maximum at location near the gear because the induced shear stress will be maximum at that section. Now the induced shear stress is equated to allowable shear stress and the minimum diameter calculated. The A.S.M.E. code defines the permissible or design shear stress S_{ds} [10].

$$s_{ds} \leq 0.18Sut \text{ or } 0.30Syt$$

Hence, 0.18*1000 or 0.30*680

=180 MPa or 204 MPa

Taking Design shear stress= 180 MPa

$S_{ds} = 0.75 * 180$

$S_{ds} = 135\,MPa$

$S_{ds} = \frac{16Te}{\pi Dp^3}$

The calculated diameter from the above equation,

D = 81.96 mm

Table 2: Comparison of Analytical result with allowable stresses

Parameter	Analytical Result	Allowable Stress
Bending stress	650.243MPa	680MPa
Shear stress	344.19MPa	340MPa

From above table it can be conclude that, the bending stress is within the allowable range but shear stress is greater than allowable stress.

Shaft can be redesign by keeping diameter more than 81.96 mm instead of 60 mm. The analytically redesign shaft shows stress acting are within allowable stress. But the client will not be interested because he has to redesign the complete system including the gear train and other devices such as brake drum dynamometer attached to it. So an alternate way to redesign the shaft is to change the material of the shaft.

V. Finite Element Analysis

The finite element method (FEM), sometimes referred to as finite element analysis (FEA), is a computational technique used to obtain approximate solutions of boundary value problems in engineering. Simply stated, a boundary value problem is a mathematical problem in which one or more dependent variables must satisfy a differential equation everywhere within a known domain of independent variables and satisfy specific conditions on the boundary of the domain. Boundary value problems are also sometimes called field problems. The field is the domain of interest and most often represents a physical structure.

V.1 Modelling of shaft

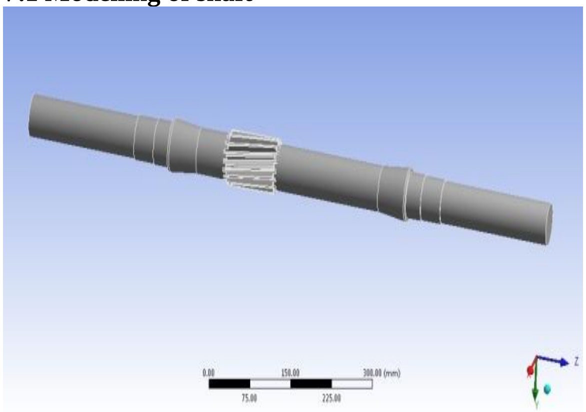

Figure 7: CAD model of shaft in Pro-E

Pro-E wildfire-5 is used for modelling of shaft. CAD software like PRO/E is higher end software which is feature based solid modelling systems. It is the only menu driven higher end software. It provides mechanical engineers with an approach to mechanical design automation based on solid modelling technology.

V.2 Forces applied on the shaft

Figure 8 shows the forces which are applied on the shaft.

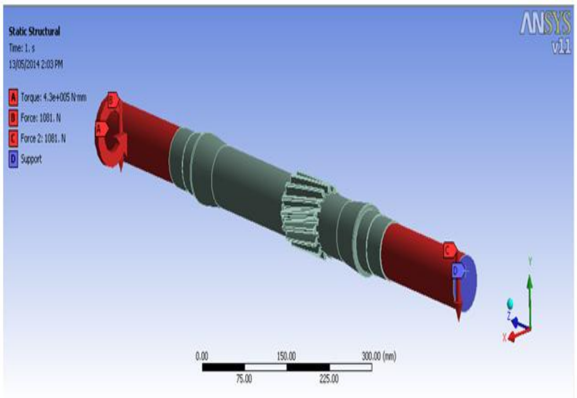

Figure 8: Forces applied on the shaft

The torque T= 430Nm is applied to one end of the shaft. 1081.75N is applied at both end of the shaft by brake drum dynamometer.

V.3 Shear stresses on the shaft

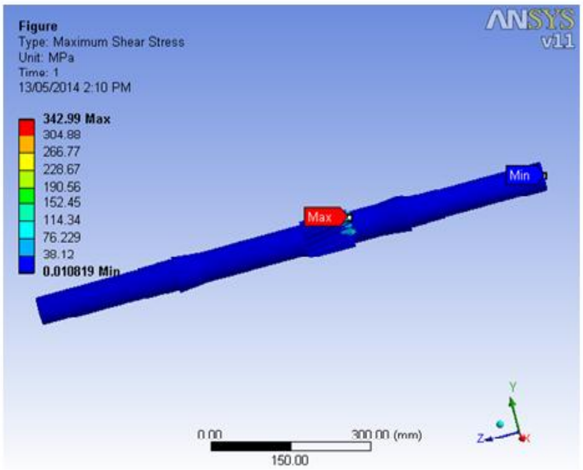

Figure 9: Maximum shear stress on the shaft

Figure shows the maximum shear stress on shaft and it is found 342.90MPa which is greater than the allowable shear stress at the location near the gear of the shaft.

V.4 Equivalent (von-Mises) stresses:

Figure 10 shows the maximum equivalent stress is 617MPa and it is well below the allowable stress.

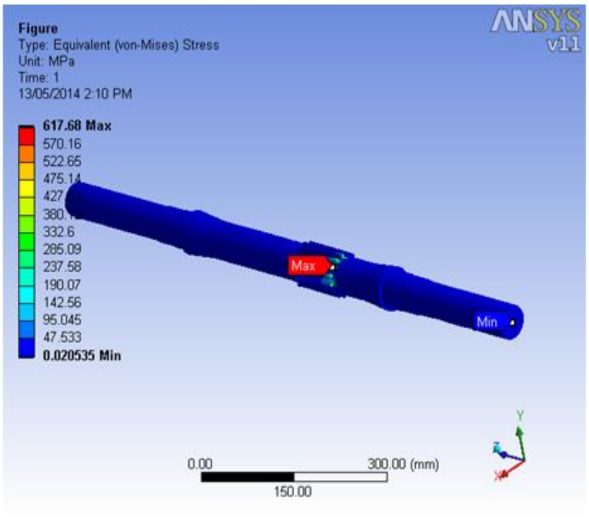

Figure 10: Equivalent (von-Mises) stress on the shaft

The maximum Equivalent (von-Mises) stress is acting on the shaft near the gear.

Table No.3 Comparison of allowable stress and ANSYS result

Stress	ANSYS Result	Allowable stress
Shear stress MPa	343	340
Bending stress MPa	617	680

From the above table and the figures we can conclude that maximum shear stress acting on shaft is greater than the allowable shear stress value.

VI. Redesign of shaft by using various material

VI.1 SAE 6145 (Chromium Vanadium steel)

SAE 6145 is a fine grained, highly abrasion resistant carbon-chromium alloy steel. Very good shock resistance and toughness are also key properties of this alloy in the heat treated condition. It is used for torsion springs and spring for truck, engine, vehicle parts and shaft.

Ultimate Strength (S_{ut}) = 1570 MPa
Yield Strength (S_{yt}) = 1430 MPa

Now the same forces are applied on the shaft with SAE 6145 and it is observed that maximum shear stress is 391.76 MPa which is less than allowable shear stress 715MPa.

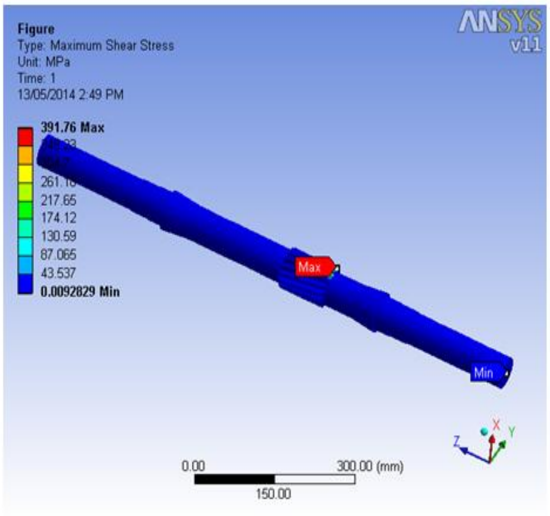

Figure 11: Maximum shear stress value of SAE 6145

VI.2 SAE 4140 (Chromium Molybdenum Steel)

SAE 4140 alloy Steel is chromium, molybdenum alloy steel. It has high fatigue strength, abrasion and impact resistance, toughness and torsion strength etc. It is used extensively in most industry sectors for a wide range of application such as axle shaft, bolts, crankshaft and part lathe, spindle, motor shaft, nut, pinions, pump shaft, worm, etc.

Ultimate Strength (S_{ut}) = 1300 MPa
Yield Strength (S_{yt}) = 1130 MPa

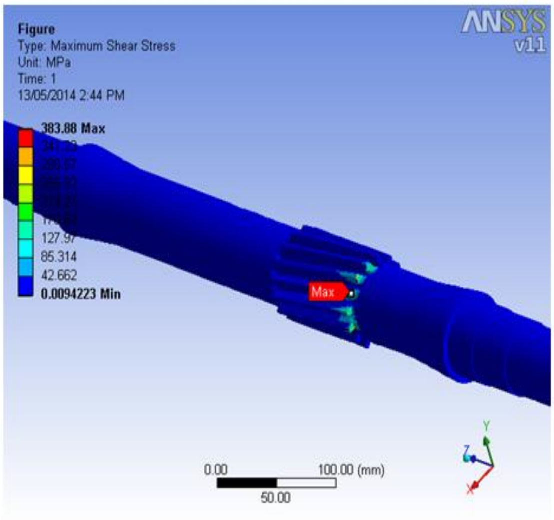

Figure 12: Maximum shear stress of SAE 6140

It is observed that maximum shear stress value of SAE 4140 is 383.88 MPa which is less than allowable shear stress limit of SAE 4140.

VI.3 SAE 6150 (Chromium Vanadium Steel)

SAE 6150 is a fine grained, highly abrasion resistant carbon chromium alloy steel and very good shock resistance and toughness also key properties of this alloy in heat treated condition. It is commonly employed in stressed machinery part including shaft, gears, and pinions and also in hand tools components, etc.

Ultimate Strength (S_{ut}) = 1690 MPa
Yield Strength (S_{yt}) = 1200 MPa

It is observed that the maximum shear stress value of SAE 6150 is 338.79 MPa from ANSYS result and allowable shear stress limit of SAE 6150 is 600MPa.

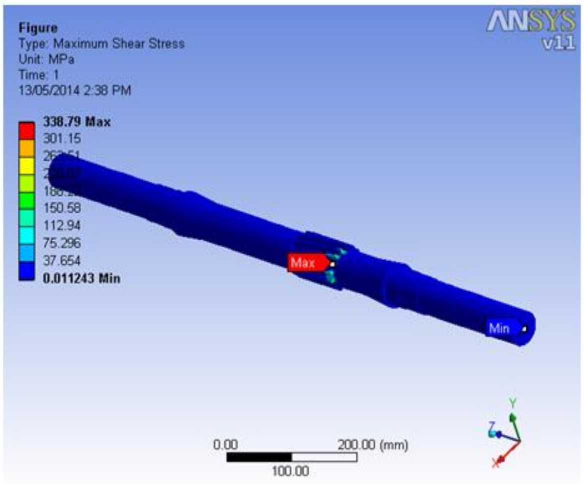

Figure 13: Maximum shear stress on SAE 6150

From above analysis we can conclude that SAE 6150 has minimum shear stress value than SAE6145 and SAE4140. Hence SAE 6150 material is safer and it can be used for the manufacturing of the shaft.

VII. Results

Failure of shaft is mainly due to the corrosion, fatigue, overload, creep, wear, abrasion, erosion. The diameter of shaft is very less as compared to stress developed on the tooth load of the shaft. So we conclude that shaft failure occurred due to minimum diameter of shaft as compared to stress developed on the shaft.

Table 3: Comparison of material

Specifications	EN24	SAE6150	SAE6145	SAE4140
Youngs Modulus MPa	206	190	209	210
Poissons Ratio	0.291	0.27 - 0.30	0.27 - 0.3	0.27 - 0.3
Density Kg/m^3	9490	7700	7850	7750
Results from Ansys				
Maximum Shear Stress MPa	343	339	392	384
Allowable stress MPa	340	600	715	565

The SAE 6150 (Chromium Vanadium Steel) has minimum shear stress value than SAE6145, SAE4140 and existing material. So SAE 6150 best material suggested for manufacturing of shaft because its low shear stress value than allowable shear stress value.

VIII. Conclusion

Spectro analysis test is carried out at P. R. Khedkar calibration and testing center, Nagpur and it is concluded that material of shaft is EN24.

Analysis of shaft is carried out by using analytical method and using ANSYS-11 software. Both these methods showed that maximum stresses are generated near the portion of gear.

Static loading for equivalent stress is safe and develops bending stress up to 650.243MPa. But maximum shear stress is 344.19MPa which is exceeding allowable shear stress of 340MPa. The diameter of shaft is less as compared to stress acting on tooth. So it can be conclude that shaft is failed due to less diameter.

Materials of shaft are selected from data book and shear stresses acting on the shaft calculated. SAE6150 (Chromium Vanadium Steel) is best material suggests for manufacturing of shaft because its shear stress value is very less than allowable shear stress.

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