Transfer coal conveyor shaft Failure analysis

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ABSTRACT

Now day’s major breakdown in continuous running plant is to costly. The motor and gearbox was directly coupled through chain and sprocket to the shaft. The overall subject of Project is the frequent failure analysis of drive shaft used in transfer coal conveyor of thermal power plant. An investigation was performed determine the failure root cause and factors affecting on transfer coal conveyor shaft. ansys analysis was also performed to quantify the stress distribution in the shaft. It was concluded that the shaft failed due to stress concentration at the end of sprocket where the diameter of shaft was small to carry load of coal.

Keywords— Coal Conveyor shaft, Failure analysis, Ansys analysis.

I. INTRODUCTION

The coal feeding arrangement of transfer coal conveyor, the coal from coal plant transfer into the coal bunker and coal bunker to transfer coal conveyor through into combustor of boiler. The 700 ton/day coal burned into combustor of boiler. Technical description of conveyor unit-Make-Demech, speed (m/s)-0.1, Head (mmwc)-10000pa, Flow Rate (Ton/Hr)-46. The combined mass of the motor and gearbox is 300 kg–this load is shared between the conveyor pulley shaft and the mounting hinge pin. The mounting arrangement was motor, gear box, drive sprocket with duplex chain, drive shaft, chain assembly, both sides Plummer block unit driven shaft of a coal conveyor was replaced with an over-hauled unit during scheduled maintenance. After approximately two days of operation, the shaft failed at the end of sprocket.

II. LITERATURE SURVEY

This describes the failure analysis of a drive shaft used in a transfer conveyor which had been involved in a load carried out by weight coal. The shaft was found to break into two pieces. The investigation was carried out in order to establish whether the failure was the cause design Or the material failure. An evaluation of the failed shaft was undertaken to assess its integrity that included a visual examination, photo documentation. Results indicate that the shaft shear as a result of improper diametric cross section change areas. So the Redesign of shaft was change with high diameter shaft having double sprocket. We therefore conclude that the failure was the maximum shear stresses acting on at the neck of sprocket.

Sandeep Gujarani and Shivaji Gholap describe the failure analysis of Fatigue Analysis of Drive Shaft” Geometry of shaft is modified by introducing a step at both side of the shaft before bearing junction and giving max. Radius, so that stress concentration could be minimized at the bearing junction. (1)

Deepan Marudachalam M.G, K.Kanthavel, R.Krishnaraj describes the failure analysis of Optimization of shaft design under fatigue loading using Goodman method. Conclude that the failure occurred due to tensional-bending fatigue.(2)
Nakul R. Deshmukh, Vaibhav G. Pardeshi, Kartik R. Pardeshi, Shashank S. Bhalekar, Prof. A. J. Patil, describe the failure analysis of Design and Analysis of Stepped Shaft in Inclined Position, As shaft is revolving with low speed the centrifugal component of force acting on shaft is negligible. Stepped shaft used will be stable in inclined position from above results. (3)

R. A. Gujar, S. V. Bhaskar, describe the failure analysis of Shaft Design under Fatigue Loading By Using Modified Goodman Method, Conclude that the The fatigue life prediction is performed based on finite element analysis. The manufacturer to improve the fatigue life of the dynamometer shaft using FEA tools. It can help to reduce cost, critical speed and times in research and development of new product. (4)

D. K. Padhal, D. B. Meshram, describe the failure analysis of Analysis and Failure Improvement of Shaft of Gear Motor in CRM Shop, Conclude that the By changing the material of good shear strength, the shaft may sustain in the maximum loading condition. Re-design the shaft with modified diameter and use in Scraped motor. (5)

F. Bernta, A. van Bennekomb, describe the failure analysis of Pump shaft failures a compendium of case studies Conclude that the, many factors can and do contribute towards pump failures, but if we are aware of these factors and how they affect the performance of pump components we will be in a better position to try and avoid these failures in the future. (6)

The E. Rusiński, J. Czmochowski, P. Moczo, discusses the investigation of dumping conveyor breakdown. It consist of two half shaft joining the track grid to carriage Due to slippage of one half shaft causes damage to slip-out protection. It also shows crack initiition from groove edge. FEM analysis shows that main cause is due to poor protection against slipping out & the fracture on half shaft lock is due to fatigue failure. (7)

The Sandip Bhattacharyya, A. Banerjee, I. Chakrabarti, S. K. Bhaimuk, deals with the failure analysis of an input shaft of skip drive gearbox, which had failed after 13 months of service against an expected life of 15 years. (a)

Use material as per the specification. (c) Choose appropriate heat treatment to achieve hardness profile as per the specification. (8)

The J. D. Coagula & R. G. Todkari deals with the failure analysis of Blister machine cam shaft, stress concentration at the cam step region failure of shaft. (9)

The Dejan Momcilovic, Zoran Odanovic, Radivoje Mitovic, Ivana Atanassovska, Tomaz Vuherer deals with the failure analysis of Failure analysis of hydraulic turbine shaft material & material quality are not fulfill the requirement, shaft corrosion and high stresses at the time of star and stop. (10)

III. METHODOLOGY

The damaged 75 mm shaft as shown in fig considering that the failure occurred during operation and that the shaft pieces continued to rub against each other. In order to determine the real cause of this damage, we have performed a detailed Analysis of the problem. By integrating experimental methods and ansys analysis.

A. Shear stress in shaft

\[ P = 2 \times \pi \times N \times T / 60, \quad 18.5 \text{ kw} = 2 \times 3.14 \times 1450 / 60; \quad T = 0.1218 \text{ NM.} \]

Shear stresses \( S = 16 \times T / \pi \times D^3, \text{ Where, } D = 80 \text{ mm,} \)

\[ T = 0.1218 \text{ nm, } s = 16 \times 0.1218 / \pi \times (0.8)^3 \]

\[ S = 1.2122. \]

B. Tensile shear stress in shaft \( S_s = T^*C/J. \)

Where, \( T \)-Torque-Radius of shaft-polar moment of Inertia.

\[ J = \pi / 2 \times D^4 / 32, \quad J = 3.14 \times (0.8)^4 / 32, \quad J = 0.04021. \]

\[ S = 0.01218 \times 0.4 / 0.04021, \quad S = 1.2116. \]

Find the load at sprocket pitch circles=\( DP = Nt / Pd, \)

\[ Dp = 11 / 119.8, \quad Dp = 9.18. \]

Find the stress=\( M, \quad Z = \pi \times D^3 / 32, \quad Z = 3.14 \times (0.8)^3 / 32, \)

\[ Z = 0.05026 \]

Find the maximum moment \( M = Ft / 4 = 0.061 \times 1.414 / 4, \)

\[ M = 0.02156. \]

\[ S = M / Z = 0.02156 / 0.05026, \quad S = 0.4290. \]

Find the combined maximum shear stresses-\( ((5.1 / D^3)^\times((T^2 + M^2)^2) / 2)^0.5, \)

\[ = 5.1 / (0.8)^3 \times ((0.01218)^2 + 0.02156 / 2)^0.5 \]

\[ = 5.1 / (0.8)^3 \times ((0.02476)^2) \quad \xi = 2.4664 \text{ N/m}^2 \]

If the coal conveyor carry the weight coal then Normal operating capacity of coal conveyor is 40 tph & Design capacity is 48 tph. Then calculate maximum limit of drive so 50TPH=136.20N force acting on shaft.

Resultant force acting on shaft is=\( Fr = Ft / Cos \theta, \quad Fr = 0.061 \times 0.4 / 0.02156, \quad Fr = 0.026N. \)

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\[ = 5.1 / (0.8)^3 \times ((0.02476)^2) \quad \xi = 2.4664 \text{ N/m}^2 \]
The allowable shear stresses $1135.04$ is more than the design shear stresses $850n/mm^2$. So the shaft was break.

**Redesign of Shaft** - Due to single sprocket & cross section change the under the heavy load transfer conveyor shaft was broken. Redesign of shaft having two drive sprockets for shearing maximum load.

![New Arrangement](image1) ![Old Arrangement](image2)

**C. Design calculation for shaft**

Motor power $= 7.5kw = 7.5*10^3$ W, Motor RPM $= 1440$, Motor RPM $= 24$, Gearbox Ratio $= 60:1$, Sprocket Dia Between shaft $= 347.3mm$. Sprocket weight $= 100N$. Chain Tension $= 2:1$ (Fc1/Fc2) Yield strength of EN 8 material $= 465$ N/mm2. Tensile strength of material $= 603/N/mm2$. Combined shock and fatigue factor for EN 8 material $= K_b=1$, $K_t=1$, Shaft Key dimension $W=32,L=145,H=25mm$.

**D. Bending moment on shaft**

According to ASME code the maximum stress on the shaft $=0.79*0.18*683 = 85.74N/mm2$. Or, $0.3*0.79 = e*Sut$, $=0.79*0.3*465 = 110.20N/mm2$. The which value is smaller, $110.20 < 85.74$, So the shaft was break.

**E. Equivalent Torque**

$Te=\sqrt{(K_b*M)^2+(K_t*T)^2}$, $Te=\sqrt{(1.5*16.84*10^6)^2 }+ (2.94*10^6)*2$ $Te=25.43*10^6$. Direct stress induced in key $=d2 = (2*16.84*10^6)/100*32*145$, $12.86 = N/mm2$. The allowable shear stresses $12.86 N/mm2$. is less than the design shear stresses $850n/mm^2$, So the shaft was not break.

**F. Bearing selection**

shaft dia $= 115mm$, shaft rpm $= 24$ rpm, select the bering for $1000$ hrs for operation, radial load-fr $= 51.65*10^3$ n, axial load-fa $= 12*10^3$, Million revolution of bearing $= 1.44$, calculate the equivalent dynamic load on from the equitation- $p=x * fr + y* fa$, $(fa/fr)=(12/51.65)=0.23$. $(fa/fr) > e$ the values of “ $y$" obtaining by linear interpolation-$y=1.2$, n load life relation $=l=(c/p)^p$, $p=10/3$ for roller bearing, $c=pl(10/3)$, $=42960*(1.44)^3/10$, $=47926.20$, by chart-shaft dia $=115mm$, dynamic load $c=758$
kn,(maximum),static load-(co)=930,ref.speed=2600rpm,limiting speed=3600.bearing is selected-22226ek/c3,and adopter sleeve-h3126.

IV. CONCLUSION
In order to determine the real cause of this damage, we have performed a detailed analysis of the problem. By integrating analytical methods and ansys analysis we were able to define the actual forces acting on the drive shaft. The major, direct cause of quarry conveyor drive shaft fractures is the improper shape at the sprocket area, which leads to overload in view of the immediate. The shaft failed as a result of shear. The cyclic load leading to shearing load was caused by the weight of weight coal on the gearbox and motor being carried (partially) by the conveyor sprocket shaft. And stresses caused the crack Initiation the mechanical design and material selection of the shaft is appropriate for its intended service. Failure can be avoided by redesign of new shaft with higher diameter & with two sprocket arrangement on shaft for shearing a maximum load on the shaft.

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REFERENCES