Improving Life Time for Output Shaft (Gear Box) in Dryer Unit-by Mathematically

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Abstract–This paper reports the results of failure analysis of dryer gear box output shaft, the shaft operates continuously for a year but the failure occurs periodically for the paper industry. Which fail permanently after every three months of continuous working, this is one of the major problems in industry. The major causes of failure of the shaft are to the impulse force and radial force. Beach marks on the fracture were clearly visible. Fatigue cracks were initiated at the corners of the bearing. Relatively small final fracture surface indicated that the shaft was under a low stress at the time of failure. The premature failure occurred due to high stress concentration at the corner of the shaft bearing, which led eventually to fatigue crack initiation, crack growth and final fracture. Improved design and manufacturing practice suggested that this would prolong service life of the shaft.

Keywords—Dryer unit O/P shaft, EN24 Material, shear stress, Bending moment, Mathematical Analysis.

INTRODUCTION

Shaft is a mechanical component for transmitting torque and rotation, usually used to connect other components of a drive train that can't be connected directly because of distance or the need to allow for relative movement between them. Drive shafts are carriers of torque, they are subjected to torsion and shear stress, equivalent to the difference between the input torque and the load. Drive shafts frequently incorporate one or more universal joints or jaw couplings. Sometime a Splinted joint or prismatic joint to allow for variations in the alignment and distance between the driving and driven components.

Dryer Gearbox Output Shaft

It is the rotating machine element which is used to transfer power (Tangential force &Resultant torque) from the main gearbox drive to the dryer unit. In order to transfer the power, the various members along with the forces exerted upon them causes the shaft to bending.

CAUSES FOR SHAFT FAILURE

1. Stress and strain
2. Effect of deformation
3. Cracks in shaft
4. Overload
5. Smearing of the metal
6. Vibration

a).Vibration due to balancing
b).Vibration due to lubrication

EN24 MATERIAL

EN24 is a very popular guide of through hardening alloy steel, which is readily machinable in the "T" condition. EN24T is the most suitable for the manufacture of parts such as heavy –duty axles and shafts, gears, bolts and studs.

Table 1.EN24 Equivalents

Indian IS	40Ni2CrMo28
BS 970:1955	EN24T
BS 970:1991	817M40T
German DIN	36CrNiMo6
American AISI/SAE	4340
French AFNOR	35NCD6
German Worst-off No	1.6582
European standard	EN10277-5

Table 2.EN24 Chemical composition

Carbon	0.36-0.44%
Silicon	0.10-0.44%
Manganese	0.45-0.70%
Sulphur	0.040 max
Phosphorous	0.035 max

Chromium	1.00-1.40%
Molybdenum	0.20-0.35%
Nickel	1.30-1.70%

SHAFT SPECIFICATION

Application of shaft : Paper Industry (Dryer)
Input power : 50 kW
Output speed : 300 rpm
Gear ratio : 1.96/1
Gearbox input speed : 588 rpm
Shaft length : 624 mm
Dia of shaft at failure : 69.85 mm
Max shear strength : 35 N/mm2
Shaft Material : EN24

LITERATURE REVIEW

[1] Anup A Bijagare,P.G. Mehar& V.N Mujbaile This paper reported about the replacement of conventional two-piece-steel drive shaft with a single-piece high strength carbon/epoxy composite drive shaft for an automotive application. The design parameters were optimized with the help of Genetic Algorithm (GA) with the objective of minimizing the weight of composite drive shaft. The design optimization also showed significant potential improvement in the performance of drive shaft. The results of GA are used for modeling of carbon/epoxy composite drive shaft & steel drive shaft using Pro-E software, to perform static, buckling & modal analysis of both the drive shafts using ANSYS software. Finally compare both analysis results.

[2]Sachin G Mahakalkar and Nilesh D Khutafal This study deals with a review of optimization of drive shaft using the genetic algorithm and ANSYS. Substitution of composite material over the conventional steel material for drive shaft has increasing the advantages of design due to its high specific stiffness and strength. Use of conventional steel for manufacturing of drive shaft has many disadvantages such has low specific stiffness and strength. Conventional drive shaft is made up into two parts to increase the weight of drive shaft which is not desirable in today’s market. Many methods are available at present for the design optimization of structural systems and these methods based on mathematical programming techniques involving gradient search and direct search. In practical structural engineering optimization, almost all the design variables are discrete. This is due to the availability of components in standard sizes and constraints due to construction and manufacturing practices. This paper discusses the past work done on composite drive shafts using ANSYS and Genetic Algorithm.

[3]A.H.V Pavan and K.S.N Vikrant reported about the pinion shaft made of 18CrNiMo7-6 material are used for transmitting torque from motor to gearbox used in bowl mills of fossil fuel fired power plants. In 2013 this work elucidates the metallurgical investigation that was carried out on a failed pinion shaft for analysing the cause for failure. Fractography revealed the initiation of a crack from the keyway corner. Mechanical testing indicated that the yield strength of the material was lower than the specified value. The bowl mill at site after failure indicated that hard lumps were present in the bull ring segment, which clearly made it evident that was sudden jamming of it which in turn led to overloading of the pinion shaft leading to the initiation of crack. A small overload fracture zone was also observed in the interior of the shaft suggesting low stress but high stress concentration torsion failure. Hence, this investigation concluded that this was a consequential failure.

[4] Mohammad Reza Khoshravan and Amin Paykani reported about the vibration analysis of composite propeller shafts. A propeller shaft is not limited to vehicles, but in many transmission applications can be used, but in this paper the aim is to replace a metallic drive shaft by two piece composite drive shaft. Designing of a composite drive shaft is divided in two main sections, design of the composite shaft and design of couplings. In composite shaft design some parameters such as critical speed, static torque and adhesive joint are studied, the behavior of materials is considered nonlinear isotropic for adhesive, linear isotropic for metal and orthotropic for composite shaft. Along with the design all the analyses are performed using finite element software (ANSYS). The result show significant points about optimum design of composite drive shafts.

[5] Belawagi Gireesh, Sollapur Shrishail B and V.N. Satwik reported about composite material such as E-Glass/Epoxy and HM-Carbon/Epoxy for the purpose of automotive transmission applications. A one-piece composite drive shaft for automobile is designed and analysed using ANSYS software respectively for E-Glass/ Epoxy and HM-Carbon/Epoxy composite with the objective of minimization of weight of the shaft which is subjected to the constraints such as torque transmission, tensional buckling strength capabilities and natural bending frequency. Finite element model of the drive shaft will be generated and analysed using ANSYS version 13 commercial software. Cylindrical local coordinate dataset has been defined to align the material direction of the composite lay-up and to apply ends supports and loading. Comparison of drive shaft with steel material and composite shaft shows that composite shaft gives advantages in terms of strength, weight reduction and ultimately power consumption in automobile.

[6] S.N. Vijayan and M.Makesh kumar reported about the Rotary kiln support roller and shaft are located under the kiln, which is in contact with the kiln ring. In recent years, it is estimated that many roller shaft failure occurs due to continuous loading condition that prevails in the industries due to inevitable facts. The shaft is affected by high stresses with respect to kiln load on the roller. A through insight over literature reports that it can be avoided by proper design and supportive remedial measures of the roller and shaft have been done by FEM.

[7] S.Vijayaranga and R.A.Chandrasekar reported about overall objective of this paper is to design and analyze a composite drive shaft for power transmission applications. A one piece drive shaft for rear wheel drive automobile was designed optimally using E-Glass/Epoxy and High Modulus (HM) Carbon/Epoxy composites. In this paper a Genetic Algorithm (GA) has been successfully applied to minimize the weight of shaft which is subjected to the constraints such as torque transmission, torsion buckling capacities and fundamental natural frequency. The results of GA are used to perform static and buckling analysis using ANSYS software. The results show the stacking sequence of shaft affects buckling torque.

[8] D.K. Padhal and D.B. Meshram reported about the overall subject of this paper is the frequent failure analysis of output shaft of gear motor used for cold rolling mill to drive the Pay-off Four-HI (Horizontally inserted). One end of that shaft is connected to the claw coupling to the drive of the Pay-off Four-HI. Torque acting on the output shaft are determining using which the stresses occurring at the failure section are calculated, stresses is also carried out with analytical method and comparing these results with the software results that is done using the ANSYS software after getting all the different parameter redesign of shaft is done and again analyzing the shaft using ANSYS and hope that shaft may torsion rigid.

[9]R.A. Gujar L and S.V. Bhaskar reported about the shaft employed in an inertia dynamometer at 1000rpm is studied. Considering the system, forces, torque acting on a shaft is used to calculate the stresses induced. Stresses analysis also carried out by using FEA and the results are compared with the calculated values. Shaft is having varying cross sections due to this stress concentration is occurred at the stepped, keyways, shoulders, sharp corners etc. caused fatigue failure of shaft. So, calculated stress concentration factor from which fatigue stress concentration factor is calculated. Endurance limits using Modified Goodman Method, fatigue factor of safety and compared results with FEA.

[10]Deepan Marudachalam M.G, K.Kanthavel, R.Krishnaraj reported about the frequent failure of shaft employed in a spinning machine is studied. Considering the drive system, forces and the torques acting on the shaft are determined using which the stresses occurring at the failure section are calculate. Stress analysis is also carried out by using finite element method (FEM) and the results are compared with the calculated values. Stress concentration factors at the failure cross section are calculated using least square method from which fatigue stress concentration factors are calculated. Endurance limit using Goodman's method, Fatigue of safety and the theoretical number of cycles sustained by the shaft before failure is estimated. In conclusions it is seen that the increase in fillet radii of the shaft and change in the position of support decrease the stress concentration factor, increase the endurance limit and fatigue factor of safety of the shaft.

MATHEMATICAL CALCULATION

Twisting moment

$$M_t = \frac{P \times 60}{2\pi N}$$

$$= \frac{50 \times 10^3 \times 60}{2\pi \times 300}$$

$$= 1591.55 Nm$$

Where,

P - Input Power in Kw

N – Speed in rpm

Tangential force on gear

$$F_t = \frac{2M_t}{D}$$

$$= \frac{2 \times 1591.55}{0.425}$$

$$= 7849.65 N$$

Where,

M_t - Twisting moment on gear

D -Pitch circle diameter

Normal load acting on the tooth of gear

$$W = \frac{F_t}{Cos\alpha}$$

$$= \frac{7489.65}{Cos 20^\circ}$$

$$= 7970.32 N$$

Where,

α - Pressure angle

Tangential force on bevel gear

$$F_t = \frac{2T}{D}$$

$$= \frac{2 \times 1591.55}{0.4225}$$

$$= 7533.96N$$

Where,

D – Pitch Circle diameter of Bevel gear

Load Acting on the Tooth of Bevel gear

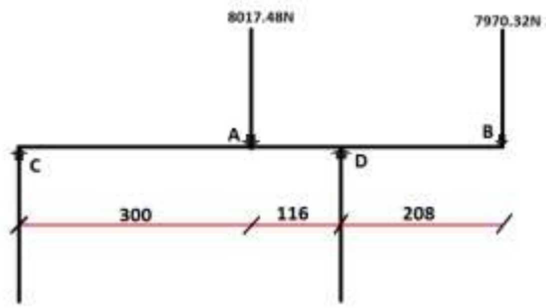
$$W = \frac{F_t}{Cos\alpha}$$

$$= \frac{7533.96}{Cos20°}$$

$$= 8017.48N$$

Bending Moment

The shown figure was drawn by using the paper mill dryer shaft dimensions.



Taking bearing supports as the fixed support,

$$R_A + R_B = 15987.8N$$

$$R_C + R_D = 15987.8N$$

Taking bending moment about B

$$(R_D \times 0.208) + (R_C \times 0.624) = 8017.48 \times 0.324$$

$$0.208R_D + 0.624R_C = 2597.66$$

$$R_D + R_C = 15987.8$$

By solving the equations,

$$R_D = 10989.17N$$

$$R_C = 4998.63N$$

BM at C=0,

$$D = 7970.32 \times 0.208 = 1657.83Nm$$

$$A = 4998.63 \times 0.3 = 1499.59Nm$$

$$B = 0$$

Taking maximum bending moment,

$$M_B = 1657.83Nm$$

Equivalent twisting moment

$$M_{t\ mean} = 1.3M_t$$

$$= 1.3 \times 1591.55$$

$$= 2069.02Nm$$

$$M_{tE} = \sqrt{(M_{tmean}^2 + M_b^2)}$$

$$= \sqrt{(2069.02^2 + 1567.83^2)}$$

$$= 2651.27Nm$$

Checking for shear stress

$$\tau = \frac{16M_{tE}}{\pi \times d^3}$$

Where,

d – Diameter of the shaft

The diameter of shaft failure is 69.85mm. The next standard diameter of the shaft are 70, 71, 75 etc..,

i. If the diameter of shaft is 70mm,

$$\tau = \frac{16 \times 2651.27 \times 10^3}{\pi \times 70^3}$$

$$= 39.37 N/mm^2$$

ii. If the diameter of shaft is 71mm,

$$\tau = \frac{16 \times 2651.27 \times 10^3}{\pi \times 71^3}$$

$$= 37.73 N/mm^2$$

iii. If the diameter of shaft is 75mm,

$$\tau = \frac{16 \times 2651.27 \times 10^3}{\pi \times 75^3}$$

$$= 32.01 N/mm^2$$

The allowable shear stress is 35 N/mm^2.so the diameter of the shaft 75mm is acceptable for design. Shaft doesn't failure at the stress limit.

CONCLUSION

The major causes of failure of the shaft are to the impulse force, radial force. These loads are acting over the shaft during its operation and lead to the shaft failure. The failure of the shaft can be avoided by redesigning the shaft. The redesigning of the shaft can be obtained by either increasing the diameter of the shaft or by changing the material properties of shaft.

By changing the material property of the shaft cost will be increased, instead by changing the shaft design the cost is not much increased and also we can avoid the periodical failure of the shaft. we suggest that from the mathematical result for redesigning the shaft. From theoretical calculation the shaft doesn't fail if its diameter is 75mm or above. Hence the diameter of the shaft 75mm is suggested for avoid the failure.

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