FAILURE ANALYSIS OF MTC 12/1 SLEWING GEARBOX AND INPUT COUPLING MODIFICATION

[1]R. Venkatakrishnan, [2]R. Rakesh Sharma, [3]R. Rajavel and [4]J.Sridhar bala
[1] Department of Mechanical Engineering, Assistant professor, IFET College of engineering, villupuram, India
[2,3,4] Department of Mechanical Engineering, Student, IFET College of engineering, villupuram, India

ABSTRACT

Neyveli lignite corporation (NLC) limited is a leading public sector undertaking owned by a government of an India. In NLC mines, open cast mining technology is adopted in which numbers of specialized mining equipment’s (SME) are used. Mobile transfer conveyor (MTC) plays a vital role in mining of lignite/burden from the ground level of the earth. In MTC there are many components present and many mechanisms among analyzing major problem have raised from MTC gear box, then the common failure are produced by the gearbox is planetary carrier was sheared and the input teeth was broken. When the all failures of the gearbox are studied and analyzed. In coupling the claw will be damaged due to an action of shaft twisting moment and bending moment. The coupling of an SME is an imported type we have to an overcome the failure of a coupling raised in it way of an increasing the diameter of a coupling in a proposed length and not varying length of the shaft and coupling.

Keywords: Mobile transfer conveyor (MTC) gear box, imported type coupling.

I. INTRODUCTION

Mobile transfer conveyors are “Multiuse” equipment used to increase performance and flexibility of various mining operation. A MTC consist of single conveyor are used to transfer the material without having material transfer point at the center of a machine. The conveyor consists of two sections are receiving and discharge sections and they can be lifted to adjust among various transfer points, the slew bearing is placed between the machine undercarriage and superstructure are provide slewing movement of the superstructure. Further the second slew movement between the receiving boom and discharge boom. In order to provide machine stability along any slew on luffing position, the receiving boom thus equipped with counterweight and consequently service weight of a machine higher than other machines [1, 2].

A MTC 12/13 slewing gearbox is driven by the motor through shaft. The coupling was placed around the shaft by reducing its frequency rate and corresponding bending moment, twisting moment are increased [3]. When the rated power 300 KW motor started by electric supply and the power transmitted to the slewing gearbox then failure was occurred at the time of sudden force will be produce to control the slewing action. In slewing gearbox the force was converted in the form of loads are transformed to one stage to another stage and it will reaches final stage the planetary carrier was sheared and corresponding teeth broken occurred [4].

1.1 Gear box data

- Gear box make : PEKRUN - Germany
- Model no : PKR 400
- Rated power : 300 kw
- Input RPM : 980/86.5 min^{-1}
- Type of gear box : bevel helical
- Number of stages : 2
- Transmission ratio : 1:11:36
- Gear box weight : 2100 KG
- Oil grade : 460 oil
- Oil quantity :165 LTRS.

1.2 Motor data

Rated power of motor : 300 KW.

1.3 Coupling data

The coupling used is of imported type and getting the spares for the coupling and the rubber. Elastomer proved to be a difficult, time consuming and more costly.

We analyzed the gearbox and the coupling was found and the detail manner of the failure were analyzed and the drawback of the coupling which that they are not much hard to withstand a speed of the motor given through the shaft. It have cause failure to the coupling by an overcome this problem we have increase the coupling diameter which gives an more advantageous than older coupling type which used in performance wise and the life of coupling have an increased [7, 8].

1.4 Shaft subjected to twisting moment and bending moment

When the shaft is subjected to combined twisting moment and bending moment, then the shaft must be designed on the basis of the two moments simultaneously. Various theories have been suggested to account for the elastic failure of the materials when they are subjected to various types of combined stresses. The following two theories are important from the subject point of view:

1. Maximum shear stress theory or Guest's theory. It is used for ductile materials such as mild steel.
2. Maximum normal stress theory or Rankin's theory. It is used for brittle materials such as cast iron.

Let

τ = Shear stress induced due to twisting moment, and

σ_b = Bending stress (tensile or compressive) induced due to bending moment.

According to maximum shear stress theory, the maximum shear stress in the shaft,

$$\tau_{max} = \frac{1}{2} \times \sigma_b^2 \times 4\,\tau^2$$

Substituting the values of τ and σb, we have

$$\tau_{max} = \frac{1}{2}\sqrt{\left(\frac{32}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2} = \frac{16}{\pi d^3}\left[\sqrt{M^2 + T^2}\right]$$

$$\frac{\pi}{16} \times \tau_{max} \times d^3 = \sqrt{M^2 + T^2}$$

The expression $\sqrt{M^2 + T^2}$ is known as equivalent twisting moment and bending moment and is denoted by T_e. the equivalent twisting moment may be defined as that twisting moment, which when acting alone, produces the same shear stress (τ) as the actual twisting moment. By limiting the maximum shear stress (τ_{max}) equal to the allowable shear stress (τ) for the material, the equation (i) may be written as

$$T_e = \sqrt{(M^2 + T^2)} = \frac{\pi}{16} \times \tau \times d^3 \quad \ldots \text{(ii)}$$

From this expression, diameter of the shaft (d) may be evaluated. Now according to maximum stress theory, the maximum normal stress on the shaft,

$$\sigma_{b\,(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{}\,(\sigma_b)^2 + 4\tau^2 \ldots \text{(iii)}$$

$$= \frac{1}{2} \times \frac{32M}{\pi d\text{^}3} + \frac{1}{2}\sqrt{\left(\frac{32}{\pi d^3}\right)^2 + 4\left(\frac{16T}{\pi d^3}\right)^2}$$

$$= \frac{32}{\pi d^3}\left[\frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)\right]$$

Or $$\frac{\pi}{32} \times \sigma_{b\,(max)} \times d^3 = \left[\frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)\right] \quad \ldots \text{(iv)}$$

The expression $\left[\frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right)\right]$ is known as equivalent bending moment and is denoted by M_e. The equivalent bending moment may be defined as that moment which when acting alone produces the same tensile or compressive stress as the actual bending moment. By limiting the maximum normal stress [$\sigma_{b\,(max)}$] equal to the allowable bending stress (σ_b), then the equation (iv) may be written as

$$M_e = \frac{1}{2}\left[\left(M + \sqrt{M^2 + T^2}\right)\right] = \frac{\pi}{32} \times \sigma_b \times d^3 \quad \ldots \text{(v)}$$

From this expression, diameter of the shaft (d) may be evaluated [10].

Fig:1 Mobile transfer conveyor 12/13 slewing gearbox

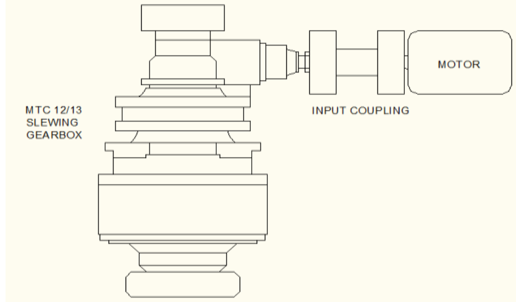

Fig: 2 MTC Slew drive planetary gearbox setup

II. MOBILE TRANSFER CONVEYOR (MTC) FAILURE

Table:1. Failure of MTC 12

SL NO	UNIT NO	NATURE OF PLANT	READY & ISSUED DATE
1	U 480	Final planetary carrier sheared.	04-09-2015
2	A519	Input coupling teeth broken – 3nos.	03-09-2015
3	A 535	1. Final planetary carrier sheared. 2. Final stage bearing broken.	28-08-2016
4	U 480	Input coupling teeth broken – 5nos.	28-0-2018
5	U 480	Final planetary carrier sheared.	23-11-2018

Table: 2 Failure of MTC 13

SL.NO	UNIT NO	NATURE OF COMPLAINT	REASDY & ISSUED DATE
1	U 480	Final stage planetary carrier sheared	06-04-2013
2	A 139	Input coupling claw damaged	27-11-2016
3	A 150	Final stage planetary carrier sheared	06-03-2017
4	A 150	Input coupling broken	12-09-2017
5	A 139	Final stage planetary carrier sheared	13-09-2017
6	A 535	Final stage planetary carrier sheared	27-09-2008
7	A 139	3^{rd} stage planetary top broken	28-09-2018

III. COUPLING FABRICATION

In this imported type coupling the claw will be damaged due to frequency rate of the shaft at low bending moment and twisting moment. The original input coupling as shown in figure 3. According to reducing its frequency rate by increasing the shaft diameter and a coupling were modified [15, 16].

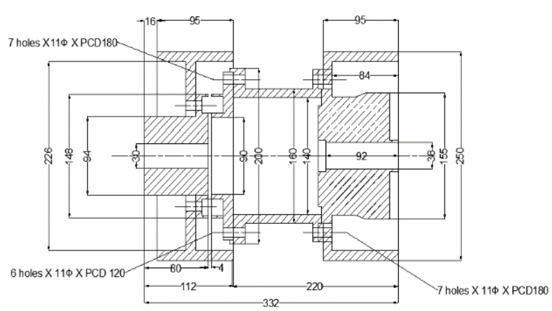

Fig:3 Dimensions of Original Input coupling assembly

IV. Design calculation

The diameter of the shaft was increased from 170mm to 180mm for reducing its frequency. Then the calculation are followed in detail [10,18].

1. To find the equivalent twisting moment of shaft diameter=170mm

For cast steel

$\tau = 950\ N/mm^2$

$\sigma_b = 1600 N/mm^2$

Equivalent twisting moment (T_e)

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$
$$= \frac{\pi}{16} \times 950 \times (170)^3$$
$$= 4.67 \times 10^9 \times (0.916)$$
$$T_e = 9.14 \times 10^8\ N/mm^2$$

Equivalent bending moment (M_e)

$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3$$
$$= \frac{\pi}{32} \times 1600 \times (170)^3$$
$$= 7.86 \times 10^9 \times 0.098$$
$$M_e = 7.76 \times 10^8\ N/mm^2$$

2. To find the equivalent twisting moment of shaft diameter=180mm

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$
$$= \frac{\pi}{16} \times 950 \times (180)^3$$
$$= 5.54 \times 10^9 \times (0.916)$$

$$T_e = 1.088 \times 10^9\ N/mm^2$$
$$M_e = \frac{\pi}{32} \times \sigma_b \times d^3$$
$$= \frac{\pi}{32} \times 1600 \times (180)^3$$
$$= 9.33 \times 10^9 \times 0.098$$
$$M_e = 9.16 \times 10^8\ N/mm^2$$

The twisting moment and bending moment of a shaft diameter 180mm is greater than the shaft diameter 170mm, so choose shaft diameter 180mm and features are followed

- Failure rate and mode of failure of the MTC 12 slewing gear box was analyzed.
- Final planet carrier shaft got sheared more often in out finding
- It is proposed to analyze the shaft diameter as increase in shaft diameter benefit to reducing the failure rate.
- By subjecting the shaft diameter of present to increase in diameter suggested to equivalent bending moment and twisting moment theory.
- Material of the planet carrier is kept same.
- By increasing the shaft diameter, it is obvious that the bending moment and twisting moment got increased.
- Increase in the both bending and twisting moment may reduce the failure rate considerably.
- It is suggested to increase the shaft diameter at the shearing area.
- Accordingly suitable bearings may be used.
- Cost wise margined increase may occur to the suggested version.
- But considering the reduction in failure rate the cost involved may become advantageous.

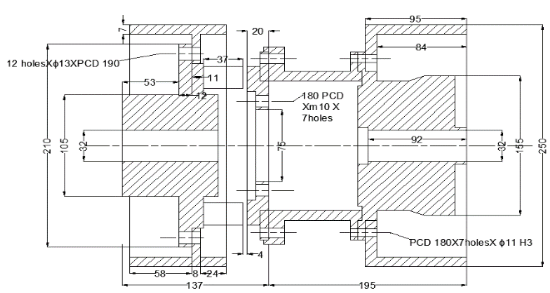

Fig: 4 Dimensions of modified input coupling.

V. CONCLUSION

- Failure analysis of final planetary carrier pertaining to MTC 12\13 slewing gearbox carried out.
- Increasing in diameter of shearing area shaft suggested keeping the material constant as length as follow in present period.
- Equivalent bending and shearing stress worked out found to have an increase an value when compare to present value of bending and shearing stress.
- By increasing the diameter failure rate will come down.
- Input coupling modification work taken up using the coupling suggested by dept.
- Since the present coupling use is imported one,by using the failure notes coupling cost will get reduced.
- While modify the coupling existing width of coupling maintained and ensured that no changes in fixing holes.
- By implementing the both, failure rate will come down and cost will got reduced.

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