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Design and Development of Experimental Setup for Vibration Analysis of Coupling

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Abstract-In this paper the design and development of experimental setup is explained. The experimental setup is used to observe the non linear behavior of coupling response under misalignment condition. For solving the purpose the three jaw elastomeric coupling is incorporated as a means to transfer torque from driver to driven shaft. The setup can be characterized as a three rotor system where two rotors are mounted intermediately between the two bearing and third rotor act as a load drum thereby providing a resisting torque (load) to the motor. It makes use of ball bearings with arrangement for creating different misalignment conditions. A variable frequency drive is used for varying the speed of a.c motor while allowing constant torque transmission. Hence, the experimental setup facilitates us for observing the behavior of coupling under different misalignment conditions, different loading and at different speed.

Index Terms- Three jaw elastomeric coupling, variable frequency drive, ball bearing, A.C motor.

1. PRESENT STATE OF ART

The experimental setup designed so far is used for studying the vibration spectrum obtained either by varying speed or by varying load torque at specified misalignment condition for different couplings. The theory formulated on the basis of these vibration spectrums explains the non linear behavior of amplitude with increase in misalignment.

2. PROPOSED CONCEPT

The concept used for developing the model for analysis is based on the fact that the model could be rearranged to observe the vibration response produced under different parameters. The parameters that can be varied are:

(1) Misalignment: for creating different misaligned positions the bearing pedestals can be adjusted vertically or horizontally.

(2) Speed: the speed can be varied with the help of variable frequency drive which also ensures a constant torque transmission.

(3) Load: the provision for varying load can be made at non drive end by suspending different weight.

3. CONSTRUCTIONAL DETAILS

The schematic diagram of the system is shown in Figure 1. An electric motor of rated capacity .5 hp is used in the experimental setup. The two rotors R1, R2, and load drum BRD are placed on shaft S2. The intention for placing these rotors on the shaft is to provide the load on bearings. A motor shaft S1 is coupled with the shaft S2 with help of elastomeric two jaw coupling JC. The shaft S2 is mounted on two self aligned ball bearings B1 and B2. At the end of shaft S2 a provision of load drum BRD is provided which is shown Figure 4.2. The assembly of shaft S2 is being fitted on the two pedestals of bearing B1 and B2 with the help of nuts and bolts. These two pedestals are already fixed on a 800 mm X 200 mm X 20 mm sized cast iron plates. However, the arrangement is also made in the experimental setup for providing misalignment in radial and lateral direction.

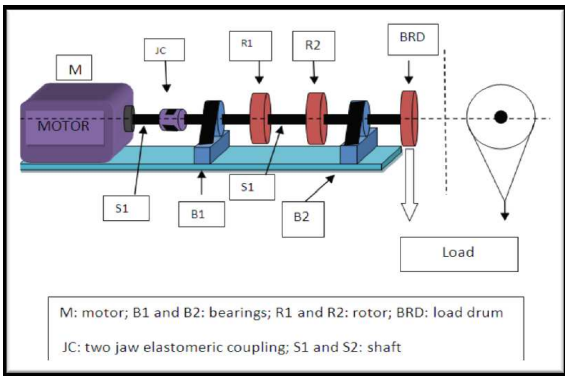

Fig. 1. Schematic diagram of experimental setup.

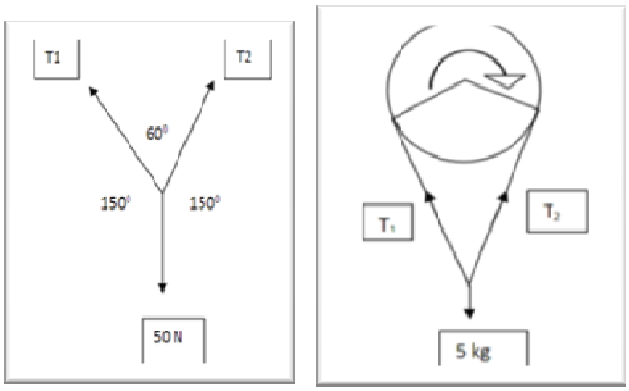

Fig. 2. Schematic diagram of load drum

4. LOAD TORQUE CALCULATION

The load torque is obtained by following formula,
Load Torque,

$T_L = (F_1 - F_2)$*(Diameter of Drum/2) (1)

In the present work a load of 5Kg is applied at the end 'A' of load drum. Find out the tensions produced in the band due to this load, and it can be obtained as under.

T2= 8.1 N and T1=28.86 N, if diameter of drum=90 mm, where T1= Tight side tension, T2= Slack side tension, μ= Coefficient of friction between band and the drum. Then, the load torques applied at drum shaft S2 is TL=934 N-mm.

5. SHAFT DESIGN

The In Figure 3, let Ra and Rb be the reactions developed at bearings B1 and B2 due to rotors C, D and load drum. The force acting at the end E is combined effect of weight of load drum and tension induced in the belt due to applied load. So, the force at E is likely to change with the change in applied load. The detailed procedure is systematically elaborated in following article.
load torque.

5.1. *Design torque*

$$T_d = \frac{60*P*1000*K_L}{2\pi\ N},$$

K_L = 2.2, Power = .5 hp, Speed = 1440 rpm

$$T_d = \frac{60*\ .372*1000*2.2}{2\pi*\ 1440}$$

Td = 5.28 Nm

5.2. *Calculation of bending moment*

Force at E = W + T1 + T2
= 200 + 144.33 + 40.88
= 385.21 N

$\sum Fv = 0$
Ra + Rb = 785.21
$\sum M_A = 0$
Rb =(200x80+200x160+ 385.21x300)/240
Rb = 681.51 N
Ra = 103.69 N
$M_A = M_B = 0$
M_B = -385.21 * 60 = -23.11 Nm
M_D =-385.21 * 140 + 681.51* 80
= 0.591 Nm
M_C = -385.21*220+681.51*160-200*80
= 8.2 Nm

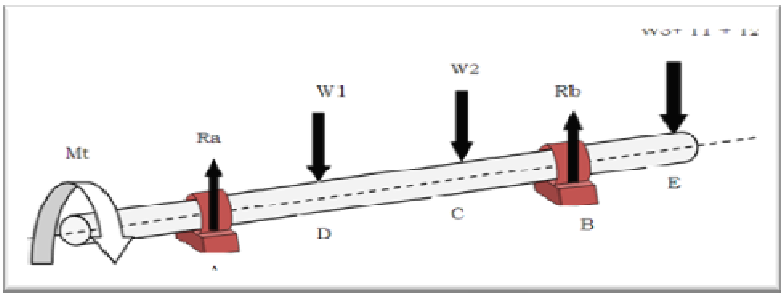

Fig. 3. Various forces acting on shaft S2.

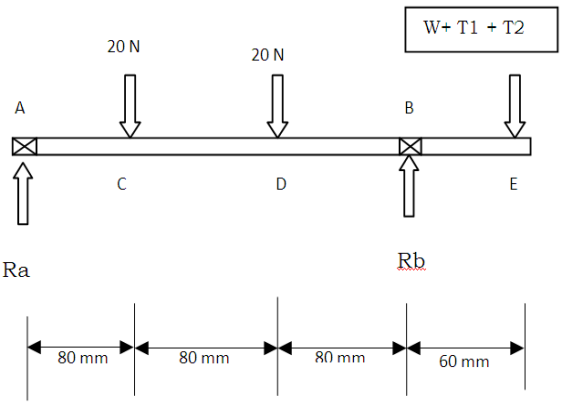

Fig. 4. FBD of shaft S2

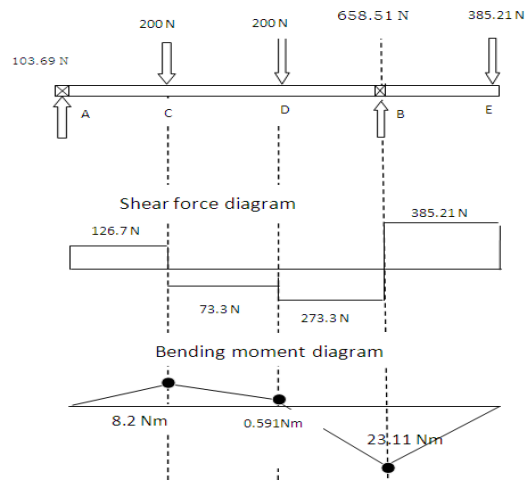

Fig. 5. Shear force and bending moment diagram

Selecting maximum bending moment at B,
$M_B = 23.11$ Nm

5.3. *Maximum stress, τ_{max}*

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(K_t M_t)^2 + (K_b M_b)^2}$$

$$\tau_{max} = \frac{0.5 * S_{yt}}{f.s}$$

S_{yt}= 296 MPa,
Factor of safety, f.s = 2
Kb = 3 and Kt= 3
Torsional moment, Mt = 5.28 Nm
Bending moment, M_B = 5.214 Nm

$$74 = \frac{16 \times 1000}{\pi d^3}\sqrt{(3 \times 5.28)^2 + (3 \times 23.11)^2}$$

d = 17.8 mm
Selecting standard diameter, d = 25 mm

6. DESIGN OF ANTIFRICTION BEARING

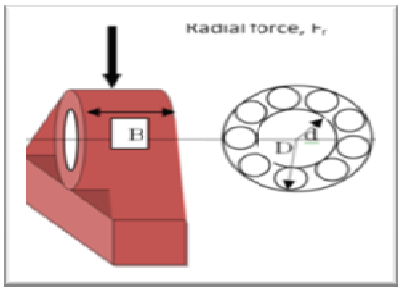

Fig. 6. Antifriction bearing

There are two antifriction bearings B1 and B2 used in the experimental setup. The maximum reaction developed at bearing B2 i.e. Rb = 681.63 N is considered for designing the bearing.

6.1. *Equivalent load coming on bearing, Fe, N*

$Fe = (XF_r + YF_a)\, K_s K_o K_p K_r$
F_r= 681.5 N
F_a= 0 N
$e = F_a / F_r$
e = 0
Selecting self aligning ball bearing
X = 1, Y = 2.3
Ko = 1 (constant rotational speed)
Kp = 1 (no preloaded bearing)
Kr = 1(outer race fixed inner race rotating).
Ks = 2 (moderate shock load)
$Fe = (XF_r + YF_a)\, K_s K_o K_p K_r$
= (1x 681.5 + 0) x 1 x 1 x 1 x 2
= 1363 N

6.2. *Life of bearing, L (million revolution)*

$L = (C/Fe)^n$ Kret.
L = 500 (demonstration model)
n = 3 for ball bearing
Kret = 1 (reliability = 90%)
$C = (500)^{(1/3)} \times Fe$
C = 10818.138 N
Selecting series 02xx (C = 11000)
Dimension d = 25 mm,
D = 52 mm,
B = 15 mm.

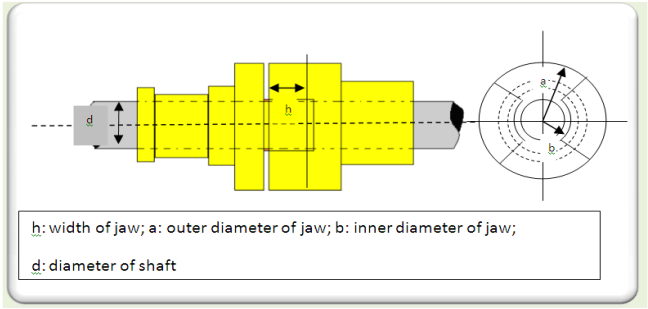

Fig. 7. Three jaw elastomeric coupling

7. DESIGN OF COUPLING

Number of jaws, n = 2
Shaft diameter, d = 25 mm
a = (1.8 – 2.5) x d
= 2.1 x 25
= 52.5 mm
b = (1.5 – 2.1) x d
= 1.8 X 25
= 45 mm
h = (0.3d + 12.5) = 20 mm

8. OPERATION OF EXPERIMENTAL SETUP

A load is being applied at the point A of a load drum thus it creates difference in the tensions of the band. This difference indeed provides the load torque on the drum shaft. On the other hand, an electrical energy is fed to motor which converts it into the mechanical energy. This mechanical energy is nothing but the driving power which is the product of driving torque and angular velocity of shaft. Thus, the driving power at 1440 rpm is getting transferred to the load drum shaft with the help of jaw coupling JC. This driving torque actually overcomes the load torque offered at the load drum.

9. CONCLUSION

The experimental setup has been developed for observing the behavior of coupling under misalignment condition. The vibration behavior two jaw coupling is noted under different loading condition at different speeds. In future the model could be used for testing different types of couplings by making some modifications.

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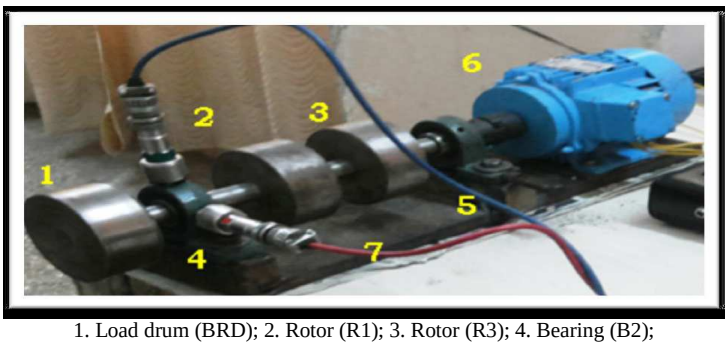

1. Load drum (BRD); 2. Rotor (R1); 3. Rotor (R3); 4. Bearing (B2);
5. Bearing (B1); 6. Motor (M); 7. Shaft (S2)

Figure 8: Schematic diagram of experimental setup

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