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Fault Diagnosis of Turbo Machinery Rotor Using Artificial Neural Network

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***Abstract-* This paper details faults diagnosis by means of vibration analysis with the help of experimental simulator of turbo machinery rotor. Literature indicates the fault diagnosis of rotating machines with the help of online monitoring system, as faults in the real system encountered randomly, hence identification of these faults is also complex in nature. The vibration analysis incurred in the online diagnosis programme needs an experience or the man who will diagnose the system, should have the feel of vibration problem. It is thus realized that there may be need of an experimentally simulated model with the help of which one can improve the art of diagnosis of the system. Even fault diagnosis is subject too wide ranging to allow comprehensive coverage of all areas associated. Comprehensive coverage of various faults in real system has been artificially created on simulated model. Detection of these faults with the help of signature analysis is specifically detailed using ARTIFICIAL NEURAL NETWORK. Further several aspects of proposed simulated model are discussed.**

I. Introduction

The machine itself is talking with us through a translator i.e. FFT analyzer with a special language is called vibration signature, matter is only to read these signature to identify corresponding faults. This would requires an field experience. Proposed model will help to him or her to become familiar with signature analysis & would enhance their knowledge to become expertise.

The simulated model is established by the rotor placed on the two journal bearings. One perforated disk has been placed centrally on the rotor.

Artificial faults can be created by imbalance condition can have created by placing mass on perforated disk. Misalignment condition can be, similarly created by disturbing the bearing position. Shaft cocked on bearing looseness in structures restraining rotor being influenced by imbalance in the rotor. Every fault indicates its special vibration signature so one can come up to a stage to identify the faults with particular regards to simulated model. The complete analysis is carried out using artificial neural network

II. Design of Proposed Simulated Model

A. Nomenclature

Total weight of rotor, W_2= 10 kgs
Total weight of Slip Rings, $W_1 = W_3$=5 kgs
Coefficient of friction, $\mu = 0.3$
Angle of wrap, $\theta = 180$ degree
Speed of Rotor, N = 1440 rpm
Critical Speed of Rotor, Nc
Stiffness of the Rotor, St
Absolute viscosity of the lubricating oil, z kg /ms
Speed of the journal, n rps
Bearing pressure on the projected bearing area, P N/mm^2
Factor to correct for end leakage, k
Projected area of the bearing, A m^2
Temperature of bearing surface, tb ^{o}cTemperature of surrounding air, ta ^{o}c
Difference between outlet and inlet temperature of oil, t ^{o}c
Modulus of Elasticity, E = 2 * 10^6 ^{kg}/cm^2

B Design of Rotor

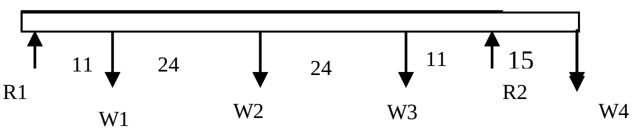
Rotor consists of three basic elements disc, shaft and bearings. Rotor is made of plain carbon steel a carbon steel content of 0.15 to 0.4 percent. Rotor is subjected to radial loads and axial load. The radial load produces torque and bending moment[2] while the axial load produces tension and bending moment acting at various sections of the rotor.

B.1 Weight calculation:

1. Total Weight of Rotor
   A. $W_2 = 10$ kg
2. Total Weight of Slip Ring
   $W_1 = W_3 = 5$ kg
3. Total Weight of Pulley
   Tension calculation over a pulley:
   Torque, $T = 60 * P / 2*\Pi*N$
   $T = 5$ Nm
   $T_1 - T_2 = 3.33$ kg
   $T1/T2 = 3$
   $T1 = 1.7$ kg
   $T2 = 5$ kg

Total Weight of the Pulley is W4 = 10 kg

B.2 Reactions:



Σ Fy = 0
R1 + R2 = W1 + W2 + W3 + W4
R1 + R2 = 30 kg

Taking moment at R1

W1 * 11 + W2 * 35 + W3 * 59 – R2 * 70 + W4 * 85 = 0

5 * 11 + 10 * 35 + 5 * 59 – R2 * 70 + 10 * 85 = 0

R2 = 22.14 kg and R1 = 7.86 kg

Moment of Inertia of Rotor:

$I = \prod/64 [(do)^4 – (di)^4]$

$= \prod/64 [(3.86)^4 – (3.56)^4]$

$= 3\ cm^4$

B.3 Deflections:

Bending Moment at any section distant x from R1 is given by

Using macaulay's method

$EId^2 y /dx^2 = - 10x \big| -22.14 (x-15) \big| -5(x-26) \big| -10(x-50) \big| - 5(x-74)$

Integrating, we get

$EIdy/dx=-5x^2+c_1 \big| 11.07(x-15)^2 \big| 2.5(x-26)^2 \big| 5(x-50)^2 \big| 2.5(x-74)^2$

Integrating again, we get

$EIy=-1.666 x^3+ c_1x \big| + c_2\ 3.69 (x-15)^3 \big| - 0.833 (x-26)^3 \big| -1.666 (x-50)^3 \big| - 0.833 (x-74)^3$

Applying Boundary Conditions

At $x = 0$, $y = 0$ …. $\mathbf{c_2 = 0}$

At $x = 85$, $y =0$ …..c_1 comes out to be

$c_1 = 13.44$

Deflection at W1 where x_1 =11 cm

$\delta_1 = 0.51282$ mm

Deflection at W_2 where x_2 =35 cm

$\delta_2 = 0.02164$ mm

Deflection at W_3 where x_3 =59 cm

$\delta_3 = - 0.0923$ mm

Deflection at W_4 where x_3 =85 cm

$\delta_4 = - 0.1737 * 10^{-5}$ mm

B.4 Critical Speed of the Rotor:

$Nc = 945 / (\sqrt{\ } \delta_1 + \delta_3 + \delta_4 + (\delta_2/1.27))$

$= 1198$ rpm

B.5 Stiffness of the Rotor:

$St = W / \delta_2$

$= 10 / 0.02164$

$= 4.621 * 10^5$ N / m

C. Design of Journal Bearing

Journal bearings are used to furnish lateral support to rotating rotor. In a [3] journal bearing, journal is the part of the rotor that runs in the bushing or sleeve. It is usually stationary and supports the journal. In hydrodynamic lubrication system, load supporting fluid film is created by the shape and relative motion of the sliding surface.

Load on the journal = 30 kg

Speed of the journal = 1440 rpm

Material of journal Bearing: GUNMETAL

Diameter of the bearing:

D = 38.85 mm

Diameter of the journal: d = 38.60 mm

Diametric clearance:

c = D – d = 0.25 mm

Radial clearance:

$c_1 = R – r$

c/2 = 0.125 mm

Diametric clearance ratio:

c/d = 0.0064

Minimum Oil Film Thickness:

Minimum distance between the bearing and the journal, under complete lubrication condition, ho = c/4 = 0.0625 mm

Type of Bearing:

Long bearing

Bearing Pressure:

P = W / (l * d)

$= 30 * 9.81 / 5 * 3.86 = 15.25\ N/cm^2$

Bearing characteristic Number:

Type of oil is SAE 10, for which viscosity at 65^o c = 7.5 mpa

Zn/p = 7.5 * 24 / 0.1525 = 1180

Velocity of Journal:

v = Π*d*N / 60

v = 2.91 m /sec

Coefficient of Friction:

The coefficient of friction in design of bearing is of great importance, because it affords a mean for determining the loss of power due to bearing friction. Coefficient of friction for a full lubricated journal bearing is a function of the three variables i.e. zn/p , d/c , l/d

$\mu = c_1c_2 / 306 \times \sqrt{\ } p/v$ Constant $c_1 = 2$ and $c_2 = 2$

$\mu = 0.0015$

Sommerfield Number:

$S = (Zn/p)(r/c)^2 \times 10^{-9}$

S = 0.02813

Heat Generated:

$Hg = \mu * W * V$

$= (\mu * W / J)* (\prod * d * n)$

= 1.2846 watt

Heat Dissipated :

Hd = C * A * (tb – ta)

Comparing heat generated with heat dissipated

Hd = Hg

1.2846 = 33.5 * 0.05 * 0.0386 * (tb - 25)

$tb = 45^0$

Rise in Temperature Oil:

$t_{oil} = 2(tb - ta) + ta$

$= 65^0$

Power Loss in Bearing:

$P_L = \mu * W * V$

= 0.00172 HP

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III ARTIFICIALLY GENERATED FAULTS

Testing of the proposed simulated model needs sophisticated instruments including FFT analyzer, computer, accelerometer pickups, phototach etc. However the testing is done properly with high care.

- For measurement of rpm of the shaft and phase angle phototach is used.
- Phototach can be mounted at stationary part of machine by means of magnetic holder.
- Accelerometer pickups are used to take vibration signals from machines. They are placed at bearings in horizontal vertical and axial directions.

TABLE 1:

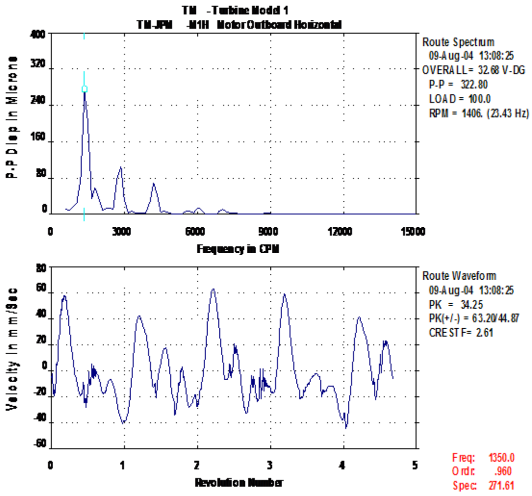
INDICATION OF FREQUENCY FOR VARIOUS CAUSES

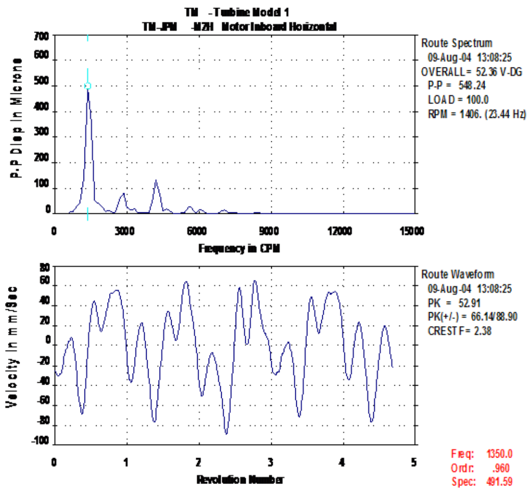
Frequency in terms RPM	**Most Likely Causes**	**Remarks**
1X RPM	Unbalance	Most common cause of Vibration.
2X RPM	Mechanical Looseness	Usually accompanied by unbalance and /or Misalignment
3X RPM	Misalignment	Usually a combination of Misalignment and excessive axial clearances (looseness)

A. Artificially created fault analyzed by Artificial Neural Network :

A.1Unbalance

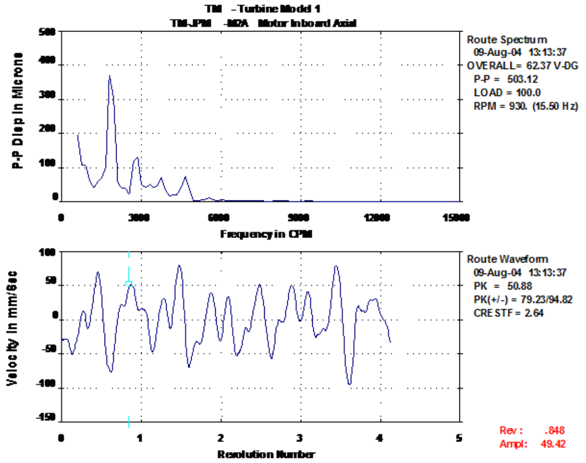
Unbalance condition is created in simulator by putting extra mass on the perforated disk. The extra mass is attached to the perforated disk by means of nut and bolt. So as to cause known unbalance to the total system. To confirm created unbalance, signature in horizontal and vertical directions, it is seen that the amplitude level at 1X rpm is indication of unbalance, which can also be obtained in tracking mode. By addition or removal of mass, the change in 1X rpm amplitude level can be observed from correction. The signature at M1H , M2H, M1V, M2V, shows unbalance condition of shaft.





A.2 Misalignment

Misalignment in the centerline of the shaft can be created with disturbing the position of shaft supporting bearing.
It will be in both radial and axial directions. This is the reason that axial vibration readings are taken. Normally at the vibration frequency 1X rpm; however, when the misalignment is severe, second order (2X rpm) some times third order (3X rpm) vibration frequencies may appear. If highest redial (Horizontal or vertical) reading is observed, then misalignment or a bent shaft should be suspected. Signature at M1A and M2A shows misalignment of the shaft center line where as signature at M1H, M1V, M2H, M2V shows possible bent shaft.



CONCLUSION

It has been seen that, the proposed experimental simulator is helping to identify the different types of faults. It is anticipated that the future extension would be to scale the simulated model with the parameters such as thermal distortion bearing oil temperature and number of stages.

REFERENCES

1. G.Stepean, "*Vibration Problem on Steam Turbo Generator at Low Power*",pp1344,1995
2. R S Khurmi. J K Gupta "*A Text Book of Machine Design"* Eurasia Publishing House (pvt)Ltd, New Delhi, 1997
3. B D Shiwalkar, "*Design of Machine Element*",Central Techno Publication, Nagpur, 130-139.
4. B.Vytatitas et. All "*Rotor Dynamics of steam Tubodynamics*