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EXPERIMENTAL INVESTIGATION AND FAULT DIAGNOSIS OF A BOILER FEED PUMP UNIT

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ABSTRACT

Condition monitoring and diagnostic engineering is a novel concept which enables us to detect in advance any incipient failures with ease and confidence in any part of a dynamic system before such failure trigger off various types of failure mechanisms, which in turn may render the whole system uneconomical, unreliable, unhealthy, unsafe and even lethal. The present work highlights an experimental investigation to monitor the vibration condition of Boiler Feed Pump unit, which is a part of Boiler Feed Pump train of a large utility Thermal Power Plant. The Boiler Feed Pump is supported by 2 Bearings. Tri-axial measurements are made at the bearing supports for 12 months. Displacement and Velocity are measured along Horizontal, Vertical and Axial directions. The experimental data is plotted on Time domain for graphical analysis to ease viewing of vibration signals. Based on the experimental data, faults are diagnosed using ISO – 2372 standards and causes are predicted. It is observed that the front and rear bearings of Boiler Feed Pump are experiencing excess vibration. The work is concluded by suggesting remedial measures to ensure vibration intensity at the said points within the safe limits.

Keywords: *Condition based Monitoring, Boiler Feed Pump, Boiler Feed Pump Train.*

1. Introduction

Machine condition monitoring is gaining importance in industry because of the need to increase reliability and to decrease the possibility of production loss due to machine breakdown. Traditional preventive maintenance not only leads to unnecessary machine downtime but also premature replacement of parts. Successful implementation of a Condition Monitoring programme allows the machine to operate to its rated capacity without stopping the machine at fixed periods for inspection. In seeking to understand how they deteriorate, Collacot [2] states that, it is desirable to make a fresh approach to a new technology and in doing so, the present trend is at the frontier of new scientific discoveries useful in both safety and economic viability. As per Stipho[3], maintenance programming of industrial system in general aims at minimizing maintenance costs and maximizing the availability of the system.

Maintenance is a combination of actions carried out to retain an item or restore it to an acceptable condition. Overhaul is an action where in, machine is disassembled, all wear is identified and repaired and the machine is returned to new equipment condition. In other words, it is the search for the detection and repair of wear failure. According to Collacot [4], maintenance besides trying to better its own efficiency, must also solve the problem of preventing failure.

Maintenance as suggested by Hensey Et al. [5] needs to and must be professionally managed by optimizing the four resources, men, machines, materials and money and thus ensuring the health of assets.

The maintenance management based on the operating condition of the equipment is termed as Condition Based Maintenance (CBM). With Condition based maintenance, overhauls are carried out only when the condition of the equipment has deteriorated to a predetermined level. Thus, overhauls or replacement of parts take place only when it has definitely been proved that a fault exists and if left unrepaired would result in unsatisfactory operating condition even leading to catastrophic failure. Condition based maintenance includes 3 steps.

i. Detection of the developing failure at an early stage.
ii. Diagnosis of its origin so as to prepare spare parts.
iii. Enabling the repair date to be planned.

For Condition based maintenance to be possible, it is essential to have knowledge of machine

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condition and its rate of change with time. A number of methods have been developed in the recent years to help the maintenance personnel for prediction of the health of equipment or system. These methods include, Vibration and Noise monitoring, Temperature monitoring, Oil monitoring, Leakage monitoring, Crack monitoring, Stress monitoring, Performance monitoring etc.,

Of all these, vibration level measurements and signature analysis are the most commonly used methods for monitoring the health of a machine. Vibration is the language of the machines which if one can listen and is enough to diagnose their complaints and ailments. Machines vibrate because of defects or inaccuracies. Each kind of defect or trouble produce vibration characteristic in a unique way.

The vibration amplitude gives an idea about the intensity of vibration. The convenient representative place of measuring its intensity is the bearing housing where the vibration is properly reflected. When the overall vibration level reaches or exceeds the limit, it may be because of one or more of the following causes:

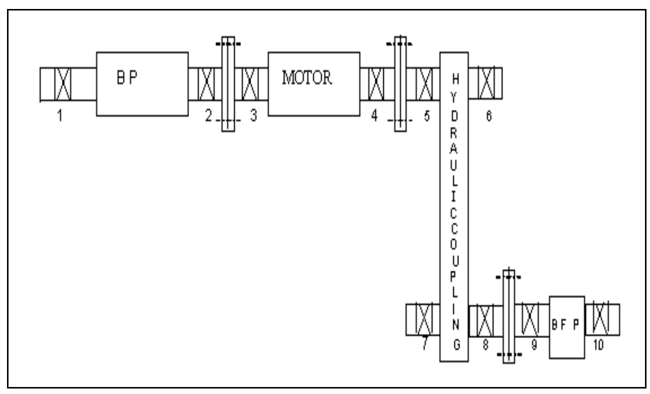
Unbalance, misalignment, looseness, damaged bearing, worn out gears, bent shaft, resonance etc. This can be identified and is to be rectified during the over haul.

Machine Condition Monitoring is gaining importance in Industry because of the need to increase reliability and to decrease the possibility of production loss due to machine break – down. The use of vibration signals is quite common in the field of condition monitoring of rotating machinery. By comparing the machine signals in normal and faulty conditions, detection of faults like unbalance, rotor rub, shaft misalignment, gear failures and bearing defects are possible. Some of the recent works in the area are done by Shiroishi Et al.[9], McFadden[10] and Nandi[11].

2. Description of the Equipment

Boiler Feed Pump (BFP) is used to pump the feed water to the boiler at high pressure. The boiler is at a high altitude compared to the level of the water. The feed pump is used to pump the water to the boiler. Pressurizing the water will be done in two stages. In the first stage, the water will be pressurized from 7.5 kg/cm^2 to 17.5 kg/cm^2 in Booster pump and from 17.5 kg/cm^2 to 180 kg/cm^2 pressurizing will be done at Boiler Feed Pump. The FK6D30 type BFP consists of FA1B56 Booster Pump (BP) directly driven from one end of the shaft of an electric Motor with rated RPM of 1440 (24 Hz). BFP is driven from the opposite end of Motor shaft through a variable speed turbo coupling. The drive is transmitted in each case through a spacer type flexible coupling.

The FK6D30 type Boiler feed Pump is a six stage horizontal centrifugal pump of the barrel casing design. The pump internals are designed as cartridge which can be easily removed for maintenance without disturbing the suction and discharge piping work or the alignment of the pump and the turbo coupling. The pump shaft sealed at the drive end and non drive end by mechanical seals and each seal is being flushed by water in a closed circuit and which is circulated by the action of the seal rotating ring. This flushing water is cooled by passing through a seal cooler, one per seal, which is circulated with clarified cooling water. The rotating assembly is supported by plain white metal lined journal bearings and axially located by a Glacier double tilting pad thrust bearing. The line diagram of the entire unit of Boiler Feed Pump Train is shown in the Fig. 1. The present work deals with the diagnostic studies of BFP shaft which is driven from the opposite end of motor shaft through a variable speed hydraulic coupling and the shaft is supported by 2 bearings.



Legend

1. Booster Pump Non – Driving end
2. Booster Pump Driving end
3. Motor Booster Pump end
4. Motor main pump end
5. Input shaft Driving end
6. Input shaft Non – Driving end
7. Output shaft Motor end
8. Output shaft Pump end
9. Boiler Feed Pump shaft Drive end
10. Boiler Feed Pump shaft Non – Drive end

Fig. 1 Line Diagram of Boiler Feed Pump Train 2

3. Experimentation

Measurements are made using “data PAC 1500”, single channel vibrometer with a frequency range of 10cpm to 4518000cpm (0.17Hz to 75.3KHz), with A/D converter, VGA resolution screen data collector from ENTEK IRD, USA, over a period of 12 months at regular monthly intervals and recorded. The instrument is mounted on 4 bearing supports along horizontal (H), vertical (V) and axial (A) directions, the axial direction is being in line with the axis of the shaft. The measurements are made for displacement and velocity. Accelerations have been computed for similar ratings. Regular logging of the data has provided the basis for performance trend monitoring of the rotating structure and prediction of faults to apply reasoning to trace the root cause.

4. Modeling of the Shaft

BFP shaft has been reduced to an equivalent shaft having length 1705 mm and diameter 91.807 mm. The shaft has been descritised with 3–D beam element by incorporating the boundary conditions at the bearing supports. First 10 natural frequencies have been computed using FEA based soft ware ANSYS8.0. In the analysis Subspace iteration and Lanczo's methods are used. The upper boundaries for the obtained frequencies are verified with conventional Rayleigh – Ritz equation given by

$$\omega_n = (\beta_n L)^2 \sqrt{\frac{EI}{\rho L^4}} = (\beta_n L)^2 \sqrt{\frac{EI}{\rho L^4}} . (\beta_n L)^2 \sqrt{\frac{EI}{\rho L^4}} \qquad (1)$$

Where β_n is a constant depends on boundary conditions. L is length of the shaft. E, I and ρ are modulus of elasticity, mass moment of inertia and density of the shaft material respectively. A source code is generated in 'C' language to validate the computation. Table 1 shows the natural frequencies of the BFP shaft Where β_n is a constant depends on boundary conditions. L is length of the shaft. E, I and ρ are modulus of elasticity, mass moment of inertia and density of the shaft material respectively. A source code is generated in 'C' language to validate the computation. Table 1 shows the natural frequencies of the BFP shaft.

Table 1: Natural Frequencies (Hz) of BFP Shaft

Subspace Iteration	Lanczos	Rayleigh Ritz	Source Code
27.710	27.71	31.46	27.71
110.57	110.57	125.84	110.57
247.79	247.79	283.14	247.79
438.32	438.32	503.36	438.32
681.12	681.12	786.50	681.12
975.55	975.55	1132.56	975.55
1322.00	1322.00	1541.54	1322.00
1720.00	1720.00	2013.44	1720.00
2170.00	2170.00	2548.26	2170.00
2833.00	2833.00	3146.00	2833.00

The table clearly indicates that there is no variation in the frequencies obtained using the two different approaches and also the frequencies obtained using Rayleigh – Ritz equation are well above the calculated values. Reference values of displacement and acceleration are calculated at each natural frequency corresponding to a velocity of 8.13 mm/sec (as per ISO 2372-Appendix I), which ensures trouble free operation of the shaft. The values are furnished in Table 2.

Table 2: Reference values of Displacement and Acceleration

Freq (Hz)	Disp (µm)	Accel (m/sec^2)
27.71	43.78	1.33
110.57	10.97	5.29
247.79	4.9	11.86
438.32	2.77	20.98
681.12	1.78	32.59
975.55	1.24	46.68
1322.00	0.92	63.26
1720.00	0.71	82.31
2170.00	0.56	103.84
2833.00	0.43	135.57

5. Results and Discussions

Triaxial measurement of displacement, velocity and acceleration are recorded on time domain at regular monthly intervals near the two bearing supports. The data is presented in Tables 3 to 4. This facilitated the computation of average values of displacement, velocity and acceleration corresponding to the rated speed of BFP.

Table 3: Tri Axial Measurements at Bearing – 9

Month	Displacement.			Velocity			Acceleration		
	H	V	A	H	V	A	H	V	A
JAN	24	11	7.6	7.3	4.2	7.1	2.2	1.5	6.69
FEB	14	17	13	6.3	7.7	5.1	2.8	3.4	2.09
MAR	26	14	9.3	7.3	5.6	7.3	2.1	2.2	5.72
APR	23	13	9	7.2	5.8	6.6	2.2	2.5	4.79
MAY	21	12	13	6.5	4.8	11	2.1	2.0	9.09
JUN	22	12	14	6.3	5.1	13	1.8	2.1	12.6
JULY	20	12	17	6.4	4.3	9.0	2.0	1.5	4.72
AUG	18	16	16	5.5	5.1	6.6	1.7	1.6	2.70
SEPT	19	13	11	5.9	4.6	8.0	1.8	1.6	5.92
OCT	9.9	16	15	5.0	4.8	6.4	2.5	1.4	2.82
NOV	13	17	17	6.0	5.9	7.6	2.7	2.0	3.49
DEC	13	19	16	6.0	6.5	8.0	2.7	2.2	4.05

Table 4: Tri Axial Measurements at Bearing – 10

Mon	Displacement			Velocity			Acceleration		
	H	V	A	H	V	A	H	V	A
JAN	26	20	7.4	9.1	6	14	3.2	1.8	24.9
FEB	22	25	9.1	8.7	11	17	3.4	4.5	32.4
MAR	29	19	8.4	8.7	7	14	2.6	2.6	24.3
APR	28	25	7.9	10	6.9	13	3.6	1.9	21.3
MAY	26	22	8.1	9.5	8.6	14	3.5	3.4	23.8
JUN	39	19	8.6	10	7.7	12	2.6	3.1	17.0
JULY	41	47	14	13	11	12	3.8	2.7	10.7
AUG	30	26	13	8.9	7.4	14	2.7	2.1	15.1
SEPT	29	25	9.1	8.4	7.4	17	2.5	2.2	30.1
OCT	22	37	15	8.6	9.6	15	3.3	2.5	13.8
NOV	26	35	13	8.2	11	18	2.6	3.2	25.8
DEC	36	42	12	9.8	11	14	2.7	2.9	16.8

Table 5 gives the velocity measurement at the bearing 9. The velocity trend on time domain is shown graphically in the Fig. 2.

Table 5: Velocity Measurements at Bearing – 9

Month	H	V	A	Conclusion
JAN	7.3	4.2	7.1	H > A > V
FEB	6.3	7.7	5.1	V > H > A
MAR	7.3	5.6	7.3	H > A > V
APR	7.2	5.8	6.6	H > A > V
MAY	6.5	4.8	11	A > H > V
JUN	6.3	5.1	13	A > H > V
JULY	6.4	4.3	9	A > H > V
AUG	5.5	5.1	6.6	A > H > V
SEPT	5.9	4.6	8	A > H > V
OCT	5	4.8	6.4	A > H > V
NOV	6	5.9	7.6	A > H > V
DEC	6	6.5	8	A > V > H

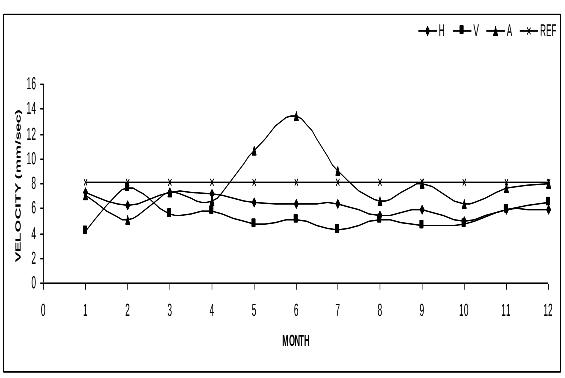


Fig. 2 Velocity Trend at Bearing – 9

It is observed that the vibrations are well within the limit till the month of May. During the month of May, June and July it is noticed that the velocity exceeding the limit along axial direction. Horizontal and vertical direction measurements are well with in the limits. The fault has been identified as Soft footing. It is suggested to check the base frame and found that the frame is distorted. Re – alignment of the frame reduced the velocity intensities during the remaining period of investigation. Table 6 gives the velocity measurement at the bearing 10. The velocity trend on time domain is shown graphically in the Fig. 3.

Table 6: Velocity Measurements at Bearing – 10

Month	H	V	A	Conclusion
JAN	9.07	5.95	13.6	H > A > V
FEB	8.66	10.6	17.2	V > H > A
MAR	8.66	6.98	14.3	H > A > V
APR	9.98	6.91	13.0	H > A > V
MAY	9.52	8.56	13.9	A > H > V
JUN	10.00	7.71	12.1	A > H > V
JULY	12.50	11.30	12.2	A > H > V
AUG	8.91	7.40	13.8	A > H > V
SEPT	8.36	7.41	16.5	A > H > V
OCT	8.60	9.61	14.5	A > H > V
NOV	8.22	10.50	18.4	A > H > V
DEC	9.84	10.90	14.3	A > V > H

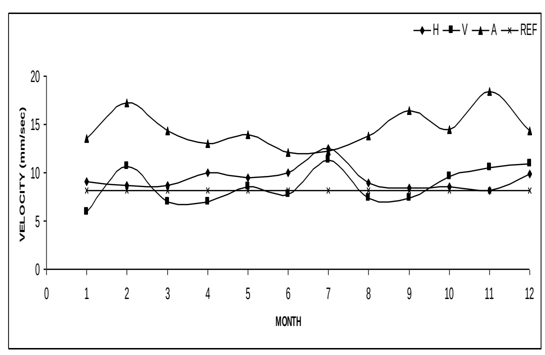


Fig. 3 Velocity Trend at Bearing – 10

It is observed that over the period of investigation, during all the months excess vibrations are recorded. The velocity intensity is above the limiting value during the month of January along Axial and horizontal directions successively where as along vertical direction its magnitude is well with in the limits. The fault causing the trend is due to Soft footing. During the month of February the velocities are exceeding the permissible values along all the three directions, with a descending nature in Axial, vertical and horizontal directions. This abnormal trend is diagnosed due to Bent Rotor. Recorded data during the months of Match to June indicates that the fault is again due to Soft footing. The table reflects that in the month of July there are excess vibrations in all the three directions, the predominant being along horizontal direction. The fault is due to Bent Rotor. During the month of August and September a similar trend as that of March and April has been observed. Again the problem is observed to be Soft footing. The velocity measurements during the months of October, November and December indicate that their values are exceeding the limiting values in all the three directions. The magnitude is high along axial directions compared to the vertical and horizontal directions. The fault is due to Bent Rotor.

It is observed that at the bearing 10, the footing is improper. It is suggested to replace the existing footing structure by a new one, since the same problem is recurring. The overall trend of vibration parameters namely displacement, velocity and acceleration are presented in Table 7. Summary of analysis based on time domain is given in Table 8.

Table 7: Overall Trend of Vibration

Bearing no.	Direc tion	Vibration Parameter		
		Disp. (µm)	Velocity (mm/sec)	Accel. (m/sec^2)
9	H	18.58	6.30	2.22
	V	14.46	5.35	2.01
	A	13.02	7.98	5.39
10	H	29.46	9.36	3.02
	V	28.38	8.65	2.74
	A	10.48	14.48	21.33

Table 8: Summary of Vibration Trend

Bearing No	Displ.	Velocity	Acceleration
9	H > V > A	A > H > V	A > H > V
10	H > V > A	A > H > V	A > H > V

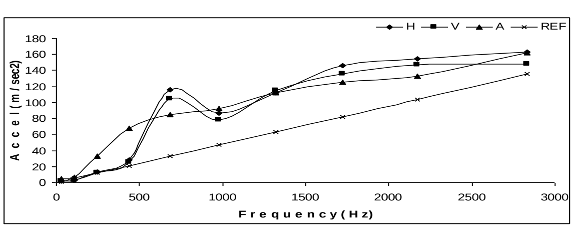


Fig. 4 Displacement Trend on Frequency Domain at Bearing – 9

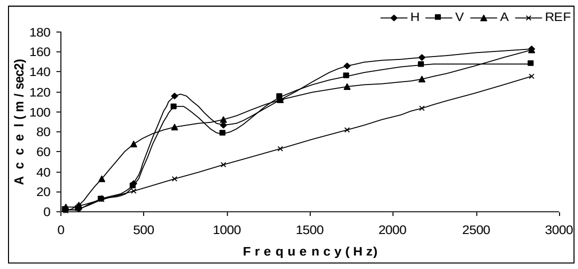


Fig. 5 Acceleration Trend on Frequency Domain at Bearing – 9

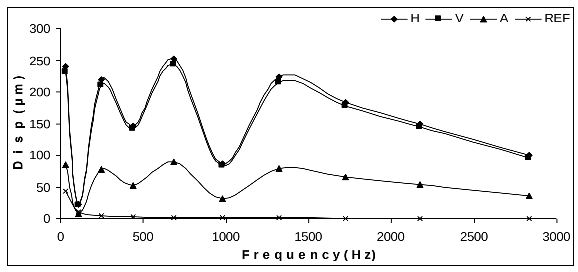


Fig. 6 Displacement Trend on Frequency Domain at Bearing – 10

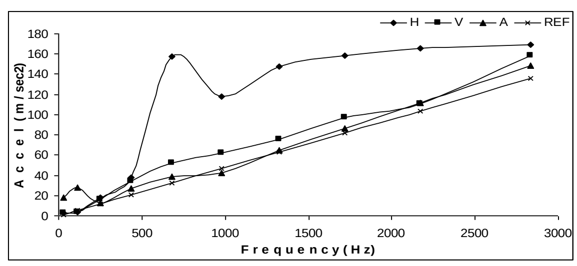


Fig. 7 Acceleration Trend on Frequency Domain at Bearing – 10

Reference values of displacement and acceleration are computed for trouble free velocity. Figures 4 to 7 show the reference signatures plotted on frequency domain and give an indication of over all trend.

6. Conclusions

In this paper an attempt has been made to highlight the application of condition based maintenance for diagnostic analysis of Boiler Feed Pump unit of Boiler Feed Pump train of a large utility Thermal power plant. The process has brought out a systematic investigation of the behavior of the rotor system. The measurement indicated that Bearings 9 and 10 are subjected to excess vibration on a few occasions. Suggestions are given based on the overall trending on time domain as well as frequency to overcome the possible faults in future there by ensuring smooth running of the unit.

Table 9: Remedial Measures at Various Key Points

Key Point	Month	Fault diagnosed	Remedial measure suggested
9	May	Soft footing	RMS1
	June	Soft footing	RMS1
	July	Soft footing	RMS1
10	January	Soft footing	RMS1
	February	Misalignment	RMS3
	March	Soft footing	RMS1
	April	Soft footing	RMS1
	May	Soft footing	RMS1
	June	Soft footing	RMS1
	July	Bent Rotor	RMS2
	August	Soft footing	RMS1
	September	Soft footing	RMS1
	October	Bent Rotor	RMS2
	November	Bent Rotor	RMS2
	December	Bent Rotor	RMS2

RMS1 - Check the base frame; RMS2 - Check the shaft in position; RMS3 - Check for non uniform mass of shaft about its center of rotation

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Appendix – I

ISO – 2372 Codes	
Speed in RPM	**Limiting Velocity mm/s**
< 500	6.35
501 – 1000	6.86
1001 – 3000	7.62
3001 – 4500	7.87
4501 – 6000	8.13
6001 – 8000	8.38
8001 – 12000	8.38
12001 – 18000	8.13
18001 – 25000	7.87
> 25000	7.62