International Journal of Research
Available at https://edupediapublications.org/journals
e-ISSN: 2348-6848
p-ISSN: 2348-795X
Volume 04 Issue-17
December 2017

Analysis of Hot Air Circulation Blower by Condition Monitoring Tool

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Abstract: *Over the years machinery health management has become vital part of the plant operation. Earlier day's machinery maintenance is only focused on reactive maintenance. In later stages, Vibration Monitoring is a critical component of any Predictive Maintenance (PdM) Practices. Vibration Monitoring and subsequent analysis has helped in identifying an earlier consequence of breakdowns. [1]The idea of performing Predictive Maintenance to perform maintenance on the machines they exhibit signs of mechanical failure has become known as Condition Based Maintenance (CBM). Condition Based Monitoring System (CBMS) is proven technology to be less costly than the failure. A simple Consequence of Failure Analysis (CFA) is made to justify preventive maintenance activities. This evolutionary process of machinery maintenance has allowed the maintenance operation to more "Proactive" than reactive in their maintenance tasking. This paper pertains to one such work made at M/s MRF Ltd, who are the leading tyres manufacturer in India, on a Main Circulation Blower which extremely critical for their production. We have observed the machinery health condition based on vibration measurements and vibration analysis which really helped us in identifying a failure sequence. In-situ dynamic balancing of main circulation blower ,which focuses the importance vibration analysis to reduce the induced vibrations from unbalance forces and Significant reduction in the vibration levels and which increased the machinery availability for production.*

Keywords:
Predictive Maintenance (PdM) Practices, Condition Based Maintenance (CBM), Consequence of Failure Analysis (CFA), In-situ dynamic balancing and Main Circulation Blower

1.**Introduction:**

This particular paper represents real time case which has become so critical for a plant operation and implementation of CBM practices has helped in identifying the failure sequence well in advance. M/s MRF Ltd is a leading manufacturer of tyres in India. They incorporated the CBM methodologies for regular maintenance practices to achieve their Total Productive Maintenance (TPM) standards. As a part of this, vibration data was collected regularly on particular equipment and we have observed phenomenal change in vibration trend. By using Vibration Analysis, machinery fault frequencies were identified and appropriate action was taken in order to reduce the vibration which in turn increased the machine availability with uninterrupted production.

Condition based monitoring is the most important goal of any maintenance program is the elimination of machine breakdowns. Very often a catastrophic breakdown will cause significant peripheral damage to the machine, greatly increasing the cost of the repair. Complete elimination of breakdowns is not at present possible in practice, but it can be approached by a systematic approach to maintenance.

Condition Based Monitoring System (CBMS) is proven technology to be less costly than the failure. A simple Consequence of Failure Analysis (CFA) is made to justify preventive maintenance activities Define the need for corrective maintenance (Condition Based Monitoring) as a part of preventive maintenance. Correcting the machine problem is defined as planned and scheduled maintenance. Preventive maintenance activities are primarily condition-based. Practicing this maintenance philosophy the level of planned and scheduled corrective maintenance will increase production to over 80% and total maintenance volume and costs will go down 20% to 30%. Over the years of our experience in the field of condition monitoring and Laser Shaft Alignment technology helped to serve the need of the machinery availability for uninterrupted continuous production. Condition Monitoring through Predictive maintenance would be helpful tool in achieving the above goal.

2.Brief description about vibration:

Need of Condition Monitoring:

- The need to predict equipment failures
- The need for a holistic view of equipment condition.
- The need for greater accuracy in failure prediction.
- The need to reduce the Effect of Maintenance cost.
- The need to improve equipment and component reliability.
- The need to optimize equipment performance
- Benefits of Condition Monitoring:
- Increase in Overall Equipment efficiency (OEE).

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p-ISSN: 2348-795X
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- Increase in Life Cycle Profit (LCP) of the equipment.
- Reduced maintenance costs.
- Extended the machine availability.
- Total Productive Maintenance (TPM) can be achieved.
- Uninterrupted production is possible

3.Frequency:

The cyclic movement in a given unit of time. The units of frequency are:

RPM = revolutions or cycles per minute.

Hertz (Hz) = revolutions or cycle per second.

These are related by the formula:

F = frequency in hertz = RPM/60.

Amplitude:

The magnitude of dynamic motion of vibration. Amplitude is typically expressed in terms of either Peak to Peak: 0 to Peak: RMS (Root Mean Square).

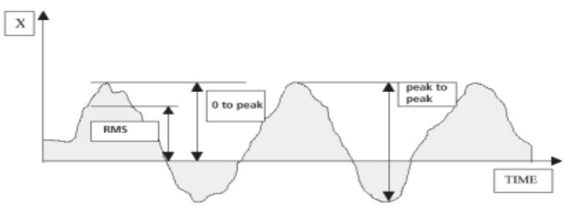


Figure: 1. On a sine wave identifying the P-P,0-P & RMS

The sketch below illustrates the relationship of these three units of measurement associated with amplitude.

Harmonics

These are the vibration signals having frequencies that are exact multiples of the fundamental frequency (i.e. 1 x F, 2 x F, 3 x F etc.).

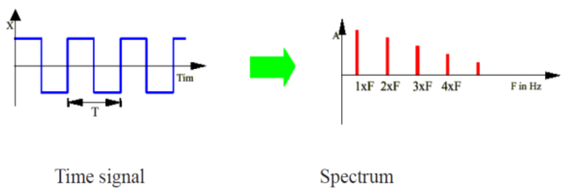
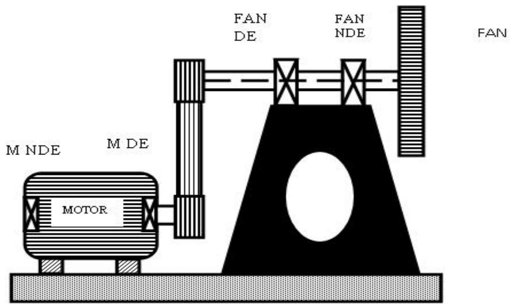


Figure: 2. Converting time signal to spectrum

Vibration frequency spectrum:

Machinery vibration consists of various frequency components as illustrated below. The amplitude of each frequency components provides an indication of the condition of a particular rotating element within the machine. Simply stated, a vibration frequency spectrum converts a vibration signal into a true amplitude representation of the individual frequency components. Since most machinery faults are displayed at or near a frequency component associated with running speed the ability to display and analyse the spectrum as components of frequency is extremely important.



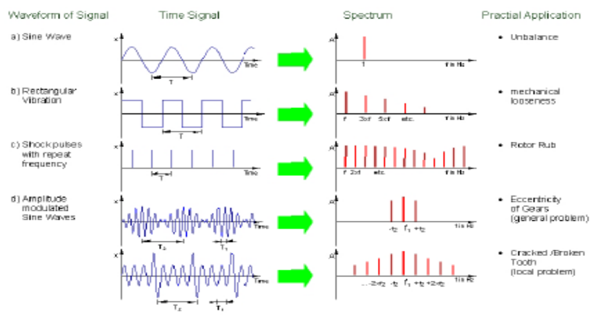


Figure: 3. how we are converting the time wave form to FFT

FFT (Fast Fourier Transform):

FFT is predominantly the most used tool in analysis of spectral data with respect to vibration analysis of machine components. Fourier transform is a mathematical operation which decomposes a time domain function into its frequency domain components.

Fault Detection:

As we have discussed the main objective of a vibration monitoring program is the detection of incipient machine failures. The methodologies associated with fault prediction usually involve comparing current vibration information with a vibration description of that machine or a similar machine in satisfactory operating condition.

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This comparison is made by two methods:

1. Comparison to industrial standards - ISO 10816 - 3

2. Comparison to a previously measured reading.

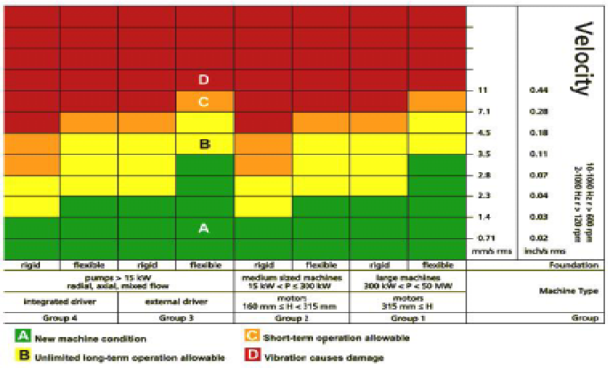


Figure:4. ISO 1086-3 vibration standards

Instrument used for Analysis:

The following equipments were used to carry out the

Vibration analysis: ON-LINE SHAFT VIBRATION

Hardware: French Make OROS Model OR-36 - MOBIPACK 16 Channel Analyzer for measurement of "Shaft Vibration" from online vibration Monitor.

Software: OrbiGate BEARING HOUSING (FIELD) VIBRATION

Hardware: PRUFTECHNIK Make VibXpert – 2 Channel FFT Analyzer for "Bearing Housing" Measurement from the Field.

ISO 10816- VIBRATION STANDARDS

Velocity measurements can be categorized as follows:

Class I machines may be separated driver and driven, or coupled units comprising operating machinery up to approximately 15kW (approx 20hp).
Class II machinery (electrical motors 15kW(20hp) to 75kW(100hp), *without special foundations*, or *Rigidly mounted* engines or machines up to 300kW (400hp) *mounted on special foundations*
Class III machines are large prime movers and other large machinery with large rotating assemblies *mounted on rigid and heavy foundations* which are reasonably stiff in the direction of vibration.
Class IV includes large prime movers and other large machinery with large rotating assemblies *mounted on foundations which are relatively soft in the direction of the measured vibration* (i.e., turbine generators and gas turbines greater than 10MW (approx. 13500hp) output.

Related typical zone boundary limits are outlined as follows:

VIBRATION SEVERITY PER ISO 10816

Vibration Velocity Vrms in/s	mm/s	Class I small machines	Class II medium machines	Class III large rigid foundation	Class IV large soft foundation
0.01	0.28				
0.02	0.45				
0.03	0.71		good		
0.04	1.12				
0.07	1.80				
0.11	2.80		satisfactory		
0.18	4.50				
0.28	7.10		unsatisfactory		
0.44	11.2				
0.70	18.0				
0.71	28.0		unacceptable		
1.10	45.0				

Figure:5. ISO 10816 Machine classes & vibration severity.

Vibration data:

Vibration amplitudes were measured in three different directions at each of the bearing location as shown in the below figure 6.

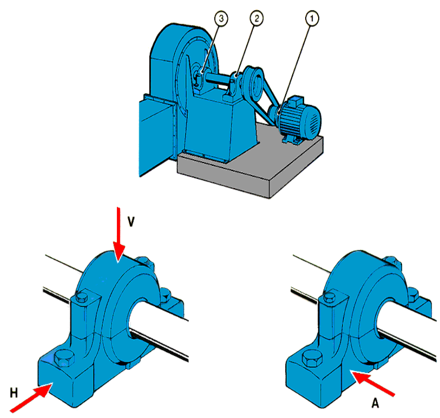


Figure: 6. Different locations on Bearing

Horizontal:

A directional force which is parallel to the ground and perpendicular to shaft axis.

Vertical: A directional force which is perpendicular to the ground and perpendicular to shaft axis.

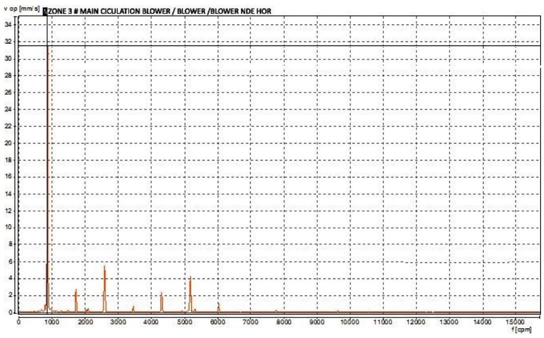
Axial:A directional force which is parallel to the ground and parallel to shaft axis.

Interpretation:

From the above table, it is observed that the highest vibration amplitudes were observed at Blower NDE bearing location and they are not in the allowable limits with respect to ISO-10816-3 standards [3]. As the Highest vibration was in Horizontal direction, it indicates presence of radial forces influencing the vibration. Hence, the highest vibration amplitude indicates presence of unbalance in the blower impeller.

Spectrum analysis:

Vibration Spectrum Analysis is non-destructive technique which helps in identifying early indication of machine problems. Spectrum Analysis is also helpful detection of machine fault frequencies which gives an early indication of machine deterioration. Hence, the maintenance action can be taken without any interruption to production.



Graph:1. Vibration Spectrum of Blower NDE Bearing Location before Balancing Vibration amplitude levels in RMS Velocity (mm/s) were measured at 1500 RPM of the drive motor and 890 RPM of the blower. The above spectrum indicates dominant vibration amplitude at 880 RPM, which is clearly 1X RPM. Vibration measurements & frequency analysis showed significant amplitude at 1x rpm frequency component attributed to unbalance induced vibration levels in Main circulation blower. As a corrective maintenance plan, it is suggested to carryout In-situ dynamic balancing for minimizing the effect of unbalance forces induced and minimizing vibration amplitudes.

Balance quality grades for various groups of representative rigid rotors in accordance with

ISO 1940 [6] and ANSI S2-19-1975 [7] standards were considered for carrying out the balancing

Unbalance:

International Standards Organization (ISO) defines unbalance as “The condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces [4].”

The technical way to describe unbalance is as a condition where “A shaft’s geometric centerline and mass centerline do not coincide” or else “the center of mass does not lie on the axis of rotation”. In other words, “uneven distribution of mass over the rotor”.

The rotating centreline being defined as the axis about which the rotor would rotate; also called as Principle Internal Axis. The geometric centreline (shaft axis) is the physical centreline of the rotor, when the centreline are coincident the rotor will be in state of balance and when they are apart, the rotor will be out of balance.

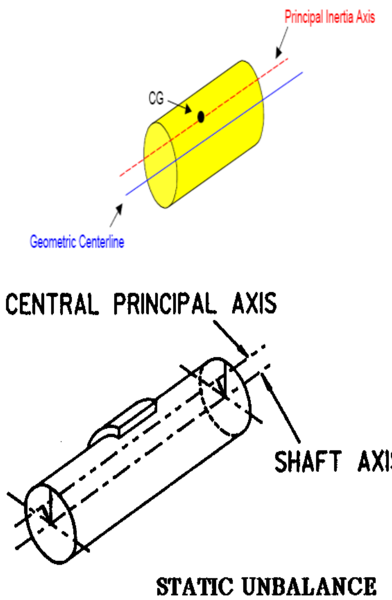


Figure: 7. Geometric centerline & principal inertia axis of shaft.

e-ISSN: 2348-6848
p-ISSN: 2348-795X
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Different types of unbalance can be defined by the relation between these two centrelines. These include:

Static Unbalance: The condition where the Principle Internal Axis (PIA) is displaced parallel geometric centreline (as shown above)

Couple Unbalance: The condition where the Principle Axis intersects the geometric centreline at the centre of gravity

Dynamic Unbalance: The condition where the Principle Axis and the geometric centreline do not coincide. The most common type is dynamic unbalance [5].

Dynamic balancing:

Dynamic Balancing is the process of the doing balancing of impellers/rotors at rated speed of the machine. [2] Unbalance forces in machines will always produce vibration at the bearings where the radial and tangential IX components are 900 out of phase. This is a sure test of unbalance and can be used to help distinguish unbalance from misalignment. Unbalance is a condition where the center of rotation of a rotor does not correspond to its center of mass, or in other words, its center of gravity does not lie on its axis of rotation. This result in a centrifugal force applied to the bearing at the IX frequency.

$$F = I_m r \omega^2$$

Where F = Unbalance Force/Centrifugal Force,

Im = Mass of the Rotor, r = Radius form centre,

ω (omega) =The Angular Velocity, equal to 2p times the frequency in Hz.

Action Taken:

In-situ dynamic balancing at Blower side bearing was performed at machine running speed and unbalance correction was effected by mass addition on the blower side and the Main circulation blower component at appropriate locations as identified by influence-coefficient method. By using phase method for carrying out In-situ dynamic balancing, the correction weight to be added at required correction angle is calculated.

Balancing with Phase Method:

Step1: Initially we need to measure the influence of unbalance by measuring the overall vibration amplitude (V_o) and phase angle (φ_0).

Step2: We can create some unbalance in the rotor by adding Trail Mass (T_m), then measure the vibration amplitude (V_1) and phase angle (φ_1).

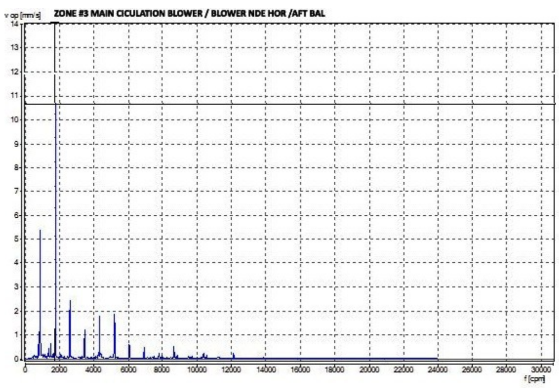
Step3: With the help of vector method calculation done by the VIBXPERT instrument, it will give us the correction angle (φ_c) at which we need add the mass for minimizing the influence of unbalance force. And the correction mass would be calculated by using the below formula.

Correction Mass (Cm) = (V_o / V_1)*T_m

With the help of this method, In-situ dynamic balancing can be performed on all types rotors for reducing the effect of centrifugal force in order to minimize the influence unbalance forces.

Table: 1. Overall Vibration amplitude levels measured at main circulation blower after balancing. The measured amplitudes are absolute amplitudes levels.

Equipment Name :Main Circulation Blower			
Measuring Location	Horizontal reading in mm/sec	Vertical reading in mm/sec	Axial reading in mm/sec
Motor NDE	8.55	6.50	6.72
Motor DE	8.02	5.75	6.12
Blower DE	12.96	10.52	9.06
Blower NDE	31.52	14.50	10.20



Graph: 2. Vibration Spectrum of Blower NDE Bearing Location after Balancing

Spectrum Interpretation:

With reference to the above spectrum, it clearly evident that the vibration amplitudes have decreased from 31 mm/sec to 10 mm/sec, however, with respect to the ISO 10816 standards, the amplitudes still not within the acceptable limits. After observing the spectrum closely, the vibration amplitudes related with structural looseness symptoms were observed. Even if the vibration amplitudes have shown significant reduction, the there are still structural looseness related symptoms were identified with sub harmonics in the spectrum. See Fig 1.5

Corrective action Recommended:

e-ISSN: 2348-6848
p-ISSN: 2348-795X
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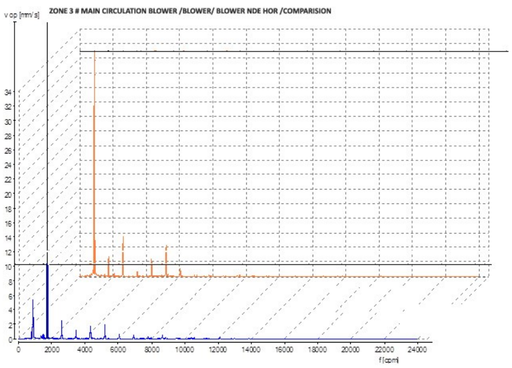
We have recommended an action for strengthening the foundation, as shown in figure 1.0, the blower bearings were mounted on the pedestal structure.

The foundation was strengthened by adding channels plates across the base frame and also we found some cracks in the base plate which were also rectified by welding.



Graph: 3. Spectrum Comparison between before & after balancing blower

Results:

Equipment Name :Main circulation blower			
Measuring Location	Horizontal reading in mm/sec	Vertical reading in mm/sec	Axial reading in mm/sec
Motor NDE	3.10	2.32	2.90
Motor DE	2.02	1.96	2.65
Blower DE	10.85	6.10	5.99
Blower NDE	10.56	5.01	6.02

Table: 2. Overall Vibration amplitude levels measured at main circulation blower before balancing.

Conclusion:

With reference above waterfall diagram we would be able toidentify the change in vibration measurements with respect to the frequency.

Spectrum analysis would be great tool identify various vibration related problems and help the maintenance personnel to reduce the failuresSignificant reduction in the vibration levels was observed and improved balance quality of the Main circulation blower assembly was achieved by in-situ dynamic balancing. Noise levels perceived in the structure level is also considerably reduced after balancing the air handling unit.

Mass added for balancing should not be removed / altered in any manner; doing so would disturb the present achieved balancing condition of the machine assembly.It is recommended to run the Main Circulation blower up to a drive motor speed of 1450 rpm and fan speed of 867 rpm for lower noise levels.Due to insufficient structural rigidity at the base locations the forced frequency excites the machine natural frequency which tends to move in the resonance mode increase the high vibration in the given equipment.

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