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Using Vibration Analysis to Identify & Correct an Induced Draft Fan's Foundation Problem of a Pollution Control Device

- A Case Study

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Abstract

Condition monitoring engineers regularly report of high vibration of induced draft(ID) fans that run in tandem with the pollution control devices. These fan's mean time between failures (MTBF) is much higher than the common ID fans and the reason is that due to the high abrasive nature of the dust that the ID fan blades are frequently subjected due to the filter bag failures of the pollution control devices.

In this case study, high vibrations were observed from a recently commissioned ID fan connected to a Blast Furnace stock house pollution control device and analysis showed that cause of vibration was principally due to resonance. The fan's prime mover was of constant rpm hence speed could not be altered to avoid resonance. Second option was to increase the concrete base mass but due to site restrictions that was not feasible either.

Balancing the unit reduced the vibration to certain extent as masses were being added to the fan blades but within a very short period high vibrations re-occurred due to wearing out of the fan blades as they were subjected to abrasive dust particles and the mean time between failures (MTBF) was as low as 30 days. In this case study the whole sequence of the fault detection and its analysis with corrective measure taken has been shown and the conclusion drawn is that whenever a high vibration of a pollution device ID fan is observed with resonance as a key factor and if the prime mover is of constant speed then instead of increasing the base mass which is quite expensive and at times not permissible due to space restriction and if the capacity of the prime mover permits it is economical to increase the weight of the fan preferably by hard facing the impeller blades with deposition of abrasion resistant materials which can offset resonance due to extra added mass on the impeller blades and increase MTBF (mean time between failures)substantially and at the same time prevent impeller blade wear.

Keywords: ID Fan, Blade Wear Out, Fan Vibration, Resonance. MTBF, Pollution Control Device

INTRODUCTION

At the stock house of a Blast Furnace a pollution control unit(PCU) has been installed to arrest the dust generated by the conveyor belts transporting the raw materials and the induced draught fan (Figure:1) draws in the dust fines generated from the conveyor belts which after passing through the bag filters passes through the ID fan as a comparatively cleaner air and escapes through the chimney into the atmosphere. (Fig:1)

Figure 1. Picture of the ID Fan for the Blast Furnace Stock House

The specification of the blower and the motor is given in Table 1.

Table 1. Motor & Blower Specification:

Motor Type	Motor Rating	Motor Voltage	Motor rpm	Full Load Current	Motor Bearings	Fan Bearings (Water Cooled)	Cooling Water Pressure	Number of Impeller Blades
Induction Motor	1120 kW	6.6kV	992	117 A	Anti-friction Bearings	Anti-friction Bearings	0.0429 MPa	12

The blower is coupled to the motor with a flexible coupling as shown in the Figure:2 . The measuring points for vibration monitoring has been numbered.

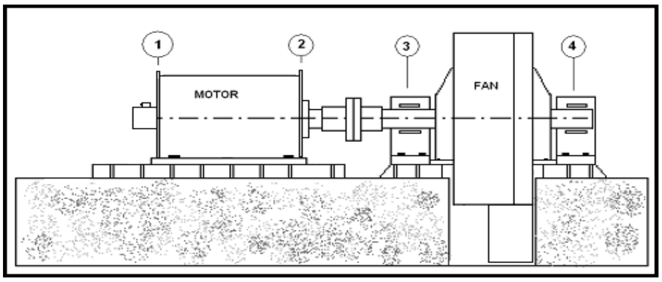


Figure 2. Schematic layout of the blower & motor unit showing the Vibration Measuring Points

After commissioning the blower unit the vibrations, both of the fan and motor, were found to be quite high and the plant's condition monitoring team were asked to monitor and diagnose the reason for the high vibrations .The initial vibration readings of the motor & blower drive end & non-drive end bearings at the measuring points are shown in Table:2

Table 2. Vibration of Motor & Blower after commissioning. (30th August 2014) (Abnormal vibrations marked *).

Measuring Points	Measuring Axis		
	Horizontal (H) (Unit: RMS value mm/sec)	Vertical(V) (Unit: RMS value mm/sec)	Axial(A) (Unit: RMS value mm/sec)
Point No:1 (Motor NDE)	11.3*	3.1	2.4
Point No:2 (Motor DE)	9.8*	0.8	2.7
Point no:3 (Blower NDE)	5.8*	1.8	3.5
Point No:4 (Blower DE)	4.9*	3.0	4.0

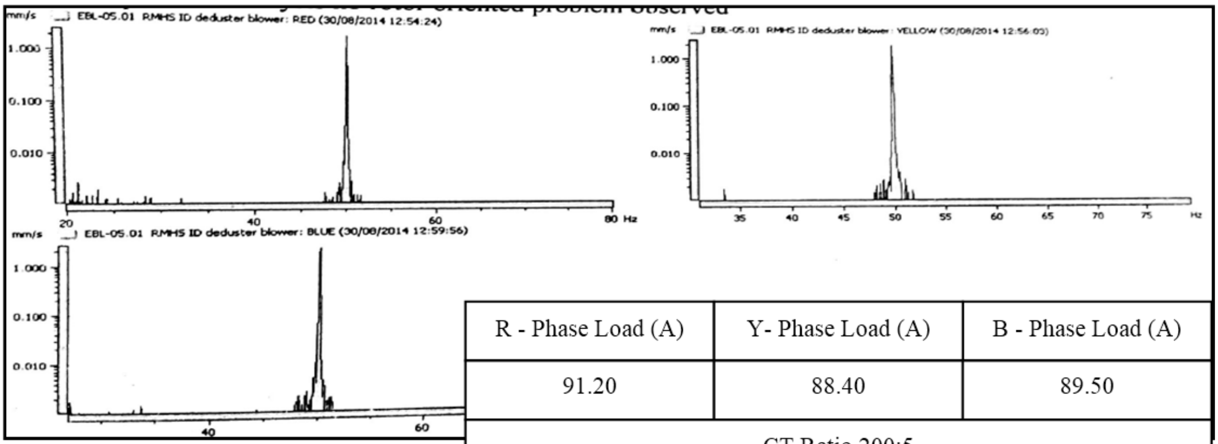
It was observed that the motor drive end & non-drive end vibrations in the horizontal axes were higher than the accepted vibration level as per ISO -10816-3.[1] as compared to the blower unit horizontal vibrations. The other axes vibration of the fan and the motor were within the accepted standard[1]. As the horizontal vibration readings were comparatively much higher than the vertical & axial readings in this case study we have considered only the horizontal readings in all our analysis.

The first step was to analyze the reason for high vibrations in the motor and as William R Finley et al. [2] had observed that to solve a vibration problem in an induction motor one must differentiate between cause and effect and in order to get this happen one has to first find the root cause of each abnormal vibration. Glenn H Bate [3] had observed that the vibration problems of the induction motor can be the combination of two groups which can be called '*magnetic'* and '*mechanical'.* Vibrations due to *magnetic* phenomenon can also be due to two reasons (a) air gap variation (b) current variation. In (a) air gap variations typical magnetic vibration has been listed as (i) static eccentricity (ii) weakness of stator support (iii) dynamic eccentricity and (iv) loose rotor bars. In (b) current variation typical magnetic vibration has been listed as (i) stator winding faults (ii) broken and cracked rotor bars (iii) shorted rotor laminations. Glenn H Bate[3] also described a simple test called 'power trip test' wherein in the test the *magnetic* components of vibration will disappear immediately once the power is removed. This test was carried out and it was observed after power switch off the vibrations remained but gradually minimized with the drop in the motor speed. It was confirmed that the components of the vibration were *mechanical* in nature so it became evident that the specific components of *mechanical* vibrations have to be sorted through vibration signature analysis and countermeasures to take on specific causes to reduce the overall vibration amplitude to the acceptable limit. Mikhail Tsypkin [4] had observed that major vibration sources of mechanical origin in induction motors are: shaft bow, rotor imbalance, misalignment, discrepancies in bearing operations as well as in the couplings ,sheaves, and other mechanical rotating elements of the assembly. Mechanical looseness, foundation problems and/or structural resonances also may significantly changes vibration signal amplitudes and configurations.

Though the 'power trip test' indicated the problem of high vibration in the motor was *mechanical* in nature still current signature analysis was also carried out to negate any problems with the rotor bars.(Fig:3)

CURRENT SIGNATURE ANALYSIS-1

From Figure 3 it is observed that there is no phase unbalance in the 3 phases and from the spectrum analysis we find there is no rotor initiated problem. With the current signature analysis it was confirmed that though the motor vibration is high but there is no broken rotor bar related issue of the motor.



R - Phase Load (A)	Y- Phase Load (A)	B - Phase Load (A)
91.20	88.40	89.50
CT Ratio 200:5		

Figure 3. Current Signature Analysis Report and the Load in the RYB Phases.

PHASE ANALYSIS

In the next step a phase analysis was carried out of the motor's both end bearings (Figure: 4) and results are shown in Table:3

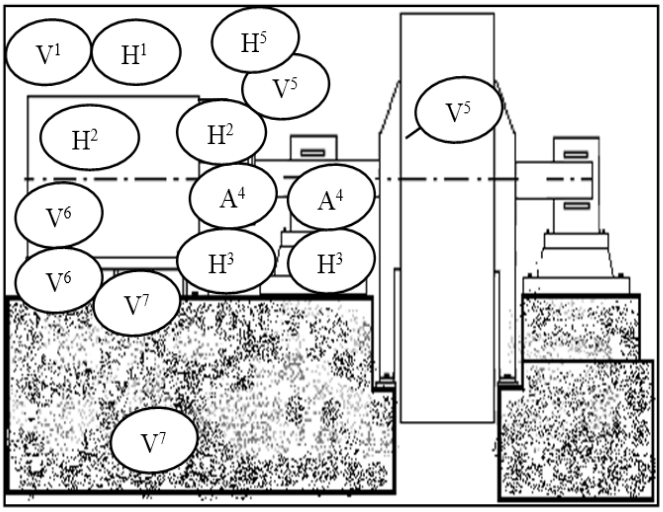


Figure 4. Phase analysis axis & location are being shown

Table 3. Phase Analysis Measuring Results

Location	Velocity RMS (mm/s)			Acceleration RMS		
	Horizontal	Vertical	Axial	Horizontal	Vertical	Axial
Motor NDE	10.9	3.5	2.1	1.2	0.6	0.4
Motor DE	10.1	1.7	2.1	1.1	0.4	0.6
Fan DE	6.7	1.4	3.2	2.2	2.3	2.4
Fan NDE	5.1	2.8	3.6	1.3	1.5	2.1

The maximum overall vibration amplitude recorded was 10.9 mm/sec in horizontal direction at motor NDE & 10.1 mm/sec in motor horizontal direction at DE bearing.

Phase Analysis at various locations of Machine base:

1. Motor NDE horizontal to vertical :12^0
2. Motor NDE horizontal to Motor DE horizontal :1^0
3. Motor DE horizontal to Fan DE horizontal :14^0
4. Motor DE axial to Fan DE axial:17^0
5. Fan DE horizontal to vertical :110^0
6. Motor base vertical to base frame vertical:1^0
7. Motor base vertical to foundation vertical :4^0

From the above result we find the phase shift (delta) value is 0^0 or 180^0 (approximately) vibration analysis concludes weak base rigidity/mechanical looseness at structure and unbalance and to confirm the same the next first step was to do the bump test to check for resonance..

BUMP TEST

Jack D Peters [5] in his work has described Bump Test as :

- Measured response of an impact to an object
- The force of the impact is not controlled or measured.
- The response of the object is not controlled but measured.
- A single channel response measurement.

In the same work Jack D Peters described the necessity of the Bump Test:

- To excite and measure the natural frequency of an object .
- To identify a resonance.
- To understand a change in mass.
- To understand a change in stiffness.
- To understand a change in damping.

Before doing the Bump test the vibration of the equipment and the base was mapped.(Fig:5)

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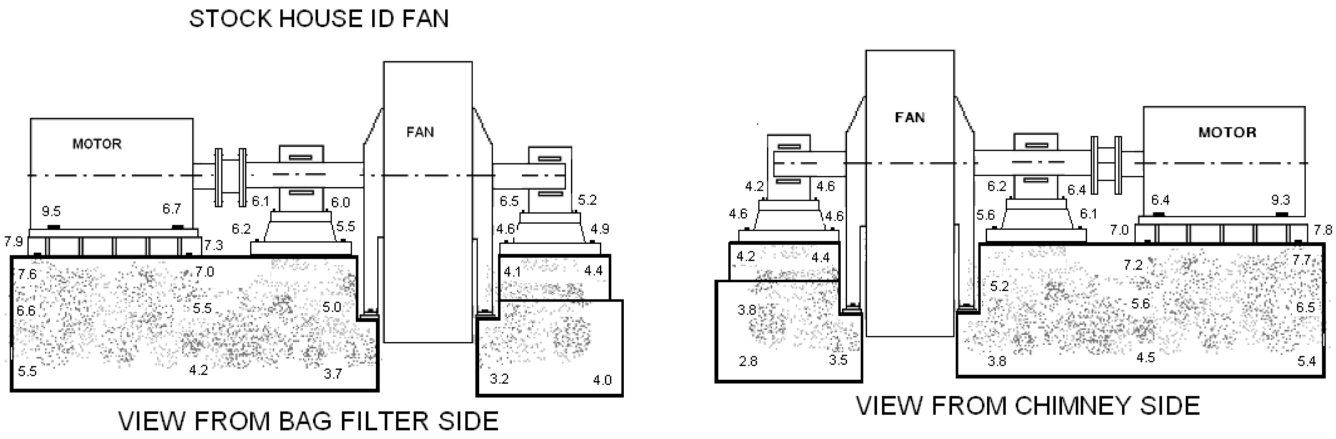

Figure 5. Vibration level mapping of the equipment and the base(view from both side).Dimension mm/sec

It is evident from the above vibration mapping that the higher vibration was at the motor side and the same was being transmitted to the base.

After the mapping of the equipment and the base the bump test was conducted and the result is given in Figure:6

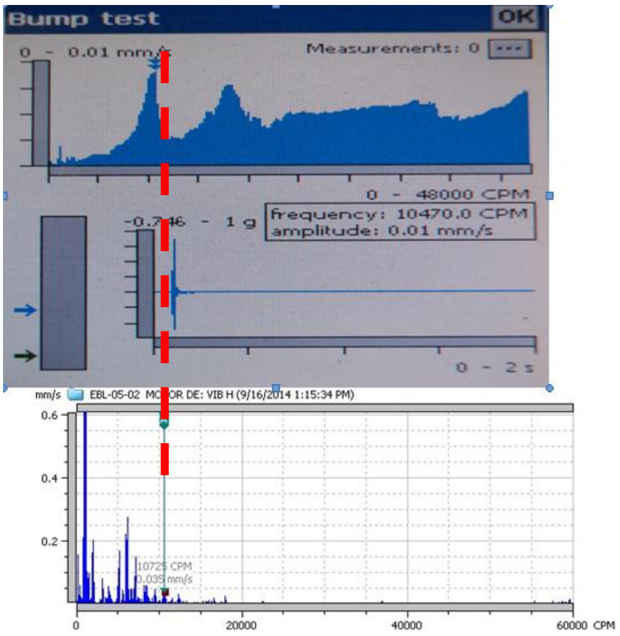

Figure 6. The result of the bump test.

From the above result of bump test it is evident that the blade pass frequency(the impeller has 12 numbers of blades) and the natural frequency of the concrete base is coinciding which is causing resonance.

Normally to avoid the resonance the usual practice is to either reduce the fan speed or to increase the stiffness of the base further. In this case study it was not possible to reduce the motor speed as it was a fixed speed induction motor and due to site restriction to increase the base size was not possible. So the next step is to find an alternative method to off-set the resonance.

RESONANCE CHARACTERISTICS:

Eugene Vogel[6] has expressed a method for solving vibration problems due structural resonance .Structural resonance refers to excessive vibrations of non-rotating components usually machine parts or supporting structures. Even slight vibratory forces from residual unbalances and misalignment effects of the machine can excite the resonant base structure resulting in severe vibration. The Bode Plot (fig:7) shows the operating speed versus the amplitude.

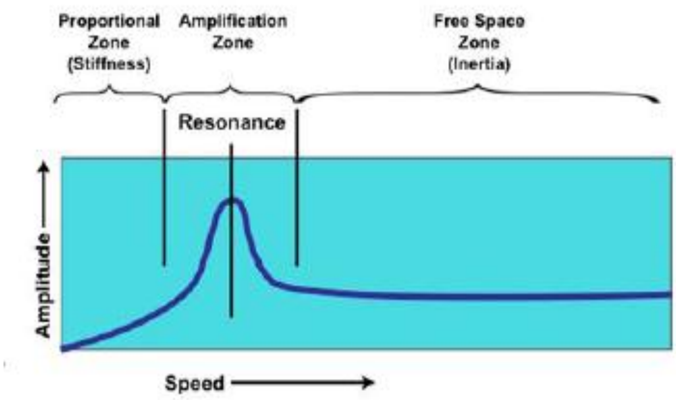

Figure 7. Bode Plot Amplitude versus Speed

From the above plot it is observed that as the operating speed is increased beyond the resonant frequency the vibration amplitude will decrease somewhat but in this case study it was not possible to vary the motor speed so a next alternative had to be implemented.

The formula for calculating the natural frequency is:

$$F_{(hz)} = \tfrac{1}{2}\pi \sqrt{k/m} \qquad \textit{(equation-1)}$$

Where k is the stiffness of the structure & m is the mass

Core formula is :

$$k/m \approx \textit{stiffness/mass}$$

Barry T Cease[7] of Cease Industrial Consulting also in a similar case study of resonance in foundation had carried out a modification work on the following basic principles of vibration.

- From Newton's 2nd Law of Motion :F=m*a where 'F'is Force ,'m' is mass & 'a' is acceleration. And when we rearrange the equation we get the following :a=F/m
- Thus, when we increase the mass(m) , in general , we will lower the acceleration levels(a)
- Also from Hooke's Law: F=k*x where 'k' is stiffness & 'x' is displacement.
- Thus, when we increase the stiffness(k), in general ,we will lower the displacement levels(x).
- By upgrading the foundation we may very well have moved a potential fan natural frequency higher(> 900 rpm) thus reducing or eliminating its amplification of our vibration levels.
- The upgrading of the foundation was not feasible in our case study because of space restriction so if we increase the mass of the rotating impeller the acceleration level will be lower

From equation (1) and also considering and above basic vibration principles it is evident that increased stiffness will raise the natural frequency and increased mass will lower it. That's logical since stiffness creates a force that is always directed against motion , while mass has inertia, which is force always directed with motion. Resonance is what happens when these two opposing forces are equal and they cancel each other out increasing vibration. The option was to either increase the concrete base mass or increase the mass of the impeller but as because of area constraint it was not feasible to increase the concrete base mass so the sole alternative was to increase the mass of the rotating impeller.

A third factor which can reduce resonance is damping but as the whole fan structure was a heavy mass weighing about 6 metric tonne it was directly mounted on the concrete base and no shock absorber were used.

INCREASING THE MASS OF THE IMPELLER:

The rotating speed of the impeller is constant so the exciting force will be the residual unbalance of the impeller at the rotating speed. In reference to the readings noted in Table :3 it was noted that both in the fan & motor bearings the vibration in the horizontal direction is much higher than the vibrations in the other two axes i.e. vertical & axial. The vibration signature analysis of the motor non-drive end(MNDE) & drive end(MDE) bearing's horizontal axis and also the fan's drive end(FDE) & non-drive(FNDE) end was done for further analysis(Fig:8)

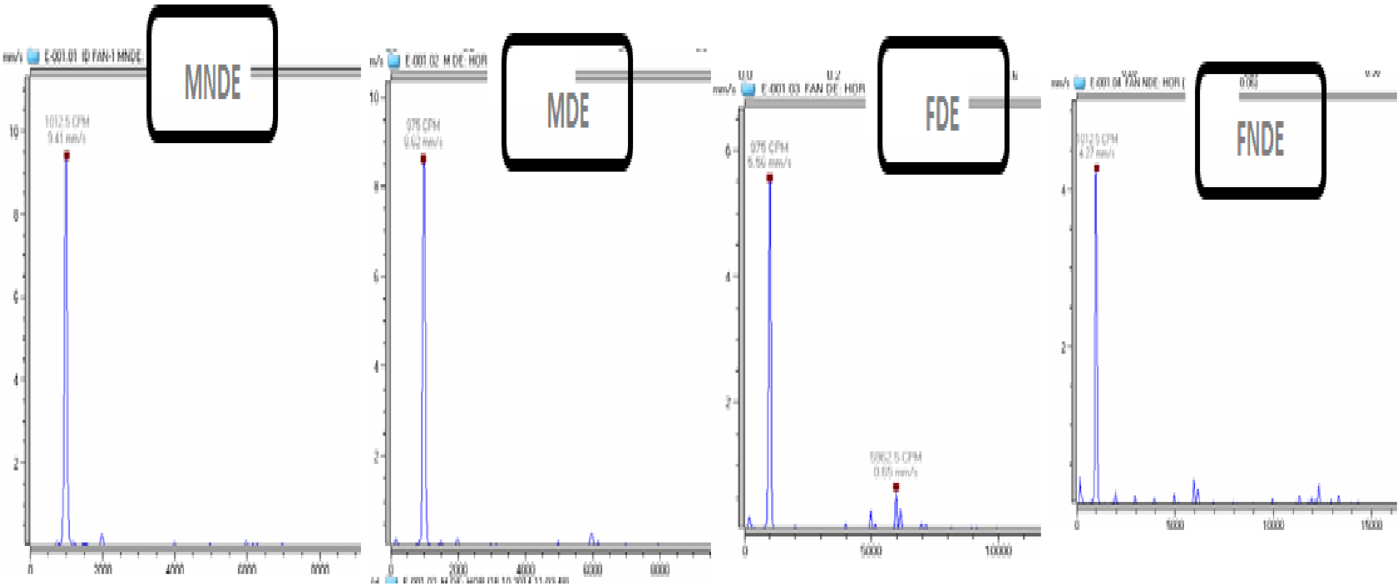

Figure 8. Vibration Signature Analysis of the Motor & the Fan Bearings in Horizontal axis

In the vibration signature analysis it was observed that the highest amplitude was at 1* running speed(X).

Troy D Feese & Phillip E Grazier [5] had observed that generally when machinery experiences high vibration at 1 x running speed the first cause can be of unbalance of the impeller and the same to be checked first. As the signature analysis indicates there is some unbalance in the impeller it was decided to check the same first and to correct the unbalance as there will be some addition of weights which may off-set the resonance forces.

BALANCING OF THE IMPELLER-1ST TIME (25TH MAY 2014)

The first balancing of the impeller was done and the correction mass that was added is given below:

Total 780 grams weight added at 315 ^{0} on the impeller.

After the balancing there was an drastic improvement in the fan and the motor vibration.(Table:4)

Table 4. Vibration Reading Before & After Balancing -1st time

Vibration Reduction Record Before & After Balancing (all units are RMS mm/sec)			
Measuring Points	Before Balancing	After Balancing	% Reduction
Motor Non-Drive End Horizontal Axis	16.17	4.73	70.7
Motor Drive End Horizontal Axis	16.06	4.25	73.5
Fan Drive End Horizontal Axis	11.09	3.06	72.4
Fan Non-Drive End Horizontal Axis	8.09	2.49	69.2

Since horizontal axis vibration were the highest in both the fan and the motor we have shown its readings only but similarly there reduction in vibration in all the axis.

It is evident that after addition of mass there was a noticeable shift in the resonance which resulted in the reduction of vibration in all the axes.

However after some weeks later again the vibration in all the axes increased considerably and signature analysis indicated increase in vibration in the 1X in the horizontal axis for both the motor and the fan.

Balancing of the Impeller-2nd Time(20th July 2015)

The second balancing of the impeller was done and the total correction mass added was1500 gms. And after addition of extra weights for balancing the vibration dropped drastically.(Table:5)

Table: 5. Vibration Reading Before & After Balancing -2nd Time

Vibration Reduction Record Before & After Balancing (All Units are RMS mm/sec)			
Measuring Points	Before Balancing	After Balancing	%Reduction
Motor Non-Drive End Horizontal Axis	23	4.9	78.89
Motor Drive End Horizontal Axis	21	4.3	79.52
Fan Drive End Horizontal Axis	14	3.2	77.14
Fan Non-Drive End Horizontal Axis	8.9	2.7	69.66

It was observed that after adding mass to the impeller during balancing resonance has drastically reduced for which overall vibration had drastically dropped but after few weeks vibration increased again . On close observation of the impeller blades it was found that though the weights added in the previous balancing was intact but there was a considerable wearing out of the impeller blades (Fig:9)

Figure 9. The tip and the edges of the impeller blades were wearing out very fast

The reason of fast wearing out was due to abrasion of sand materials which were leaking from the partially damaged filter bags of the pollution control device. In the pollution control device there are 12 chambers and each chamber has 192 bags and to locate and replace the damaged from the total 2304 total bags is a time consuming task which needs a long shut down which the operation cannot afford because of production schedule.

This abrasion was reducing the overall mass of the impeller and resonance was occurring for which the overall vibration was increasing again. As it was not possible then to overhaul the pollution control device unit the fan was again put into operation after addition of weights as required for balancing.

On expected lines again after few weeks the overall vibration increased and fan was stopped again for inspection.

Balancing of the Impeller-3rd time(24th Aug 2015)

On inspection it was found that the wear of the impeller blades were increasing now at a faster rate. (Fig:10)

Figure 10. The impeller blades were wearing out at a faster rate

For the third time again the impeller was balanced by adding weight 1310 gms. and after adding the mass overall vibration reduced.(Table:6)

Table 6. Vibration Reading Before & After Balancing -3rd Time

Vibration Reduction Record Before & After Balancing (All Units are RMS mm/sec)			
Measuring Points	Before Balancing	After Balancing	%Reduction
Motor Non-Drive End Horizontal Axis	11.41	2.58	77.39
Motor Drive End Horizontal Axis	10.05	2.35	76.62
Fan Drive End Horizontal Axis	7.22	1.78	75.35
Fan Non-Drive End Horizontal Axis	5.4	1.74	67.78

Balancing of the Impeller

It was evident from the above analysis and vibration records that high vibration of the fan was due to resonance and as the same could not be avoided by varying the motor speed or increasing the base stiffness it was achieved by increasing the impeller mass as extra weights were added to balance the impeller but the corrective measure was not very effective as the total effect of the additional weight was getting counterbalanced as the fan blades were wearing out at an extremely faster rate due to abrasion as because it was subjected to constant flow of sand and other particles due to the damage of some filter bags of the pollution control device unit. The average mean time between failures (MTBF) was as low as 45 days on the average.

After the third balancing , the impeller had to be balanced further two times for the same reason but during the 5th balancing it was observed that due to frequent high vibration the fan base was getting damaged and if the same is not contained the damage of the cast base will be irreversible. (Fig:11)

Figure 11. Initiation of Crack Propagation at the Base

Final Diagnosis:

Till now after each balancing the fan would be operating within the vibration limits [1] for certain number of days

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only. The graph(Fig:12) shows that the MTBF of the motor & the fan drive & non-drive end bearings..

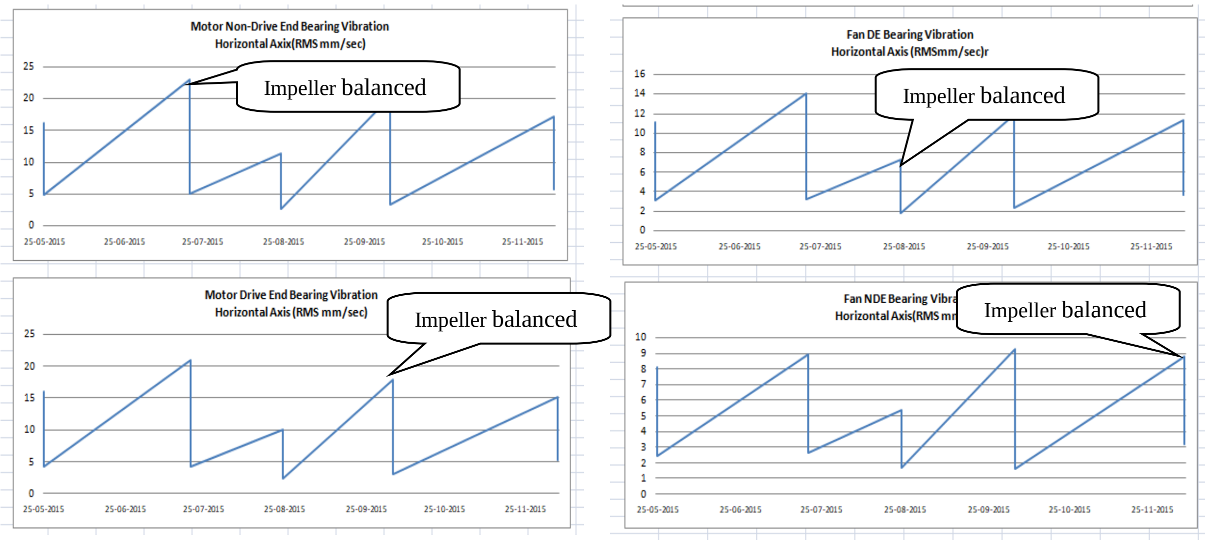

Figure 12. Horizontal vibration of the motor & the fan DE & NDE bearings after each balancing.

From the above graphs it is evident that the mean time between failures was as low as 45 days on the average and this was due to the fast wear rate of the impeller blades due to abrasion of the dust effusing from the partially damaged pollution control unit filter bags.

The reason for fast rate of wear of the impeller blades was found after checking the impeller blade material grade as given in the general drawing supplied by the overseas original equipment manufacturer . The fan blade material specification was Q235B whose IS equivalent is IS: 2062 which is a common structural steel material but is not suitable for abrasion resistance.(Table:7)

Table 7. IS :2062(IS equivalent of Q235B)

Carbon %	Silicon%	Manganese%	Phosphorous %	Sulphur %	Hardness
0.20	0.35	1.40	0.045	0.045	22 HRC(approx.)

From the Manganese % it is evident it is not a abrasive resistant material and not suitable for an impeller blade which is subjected to high velocity abrasive particles. This was the reason for high wear out of the impeller blades.

With even best maintenance practices filter bags getting damage is a routine aspect and most of the time operation need to be continued with partial damaged filter bags till a major shut down is available to replace the bags. In this case study the filter unit has 2304 bags distributed in 12 chambers. (Fig:12)

Figure 12. Picture of the 12 Chamber Filter Bag Unit

Frequent balancing of the fan was a hindrance to production and there were two counter measures to solve the issue.:

1: Change the damage filter bags to arrest abrasion on the impeller blades .

2: Coat the impeller blade surface with abrasion resistant material to reduce erosion due to abrasion.

The 2nd. option was certainly better as to locate the damaged bags from a bundle of 2304 bags is by itself a mammoth task and even after replacing the damaged bags there is no guarantee that other bags in due course of time will not get damaged in phases and start the process of corrosion. Routine damages of filter bags occur due to aging of the filter bags and natural deterioration due to common wear and tear.. So it was decided to coat the impeller blades with abrasion resistant material.

Abrasion Resistant Coating of the Impeller Blades:

Based on the base metal of the blades an abrasion resistant electrode [8]was selected with the following description:

‘Newly formulated heavy coated high recovery electrode designed specially for extreme abrasion resistance at higher temperature to about 600 ^{0} centigrade. Deposits consists of high percentage of primary chromium carbide and secondary vanadium carbides and Borides for optimum results . Hardness 61-66 HRC [8]

After coating the impeller blades the total weight of the electrodes added was 35 kg. After checking the final balance the fan was commissioned on 16th June 2015. And reading taken before and after the final balancing is given in Table :7

Table 7. Vibration Reading Before & After Welding the Impeller Blades with Abrasion Resistant Metal & Balancing.

Vibration Reduction Record Before & After Balancing (All Units are RMS mm/sec)			
Measuring Points	Before Balancing	After Balancing	%Reduction
Motor Non-Drive End Horizontal Axis	9.12	4.32	52.6
Motor Drive End Horizontal Axis	8.23	3.69	55.1
Fan Drive End Horizontal Axis	6.28	2.96	52.8
Fan Non-Drive End Horizontal Axis	4.68	2.72	41.8

After the coating of the impeller blades fan has run un-interrupted for more than two years and monitoring done during schedule maintenance showed that all the vibrations were well within the acceptable range[1].(Fig:13)

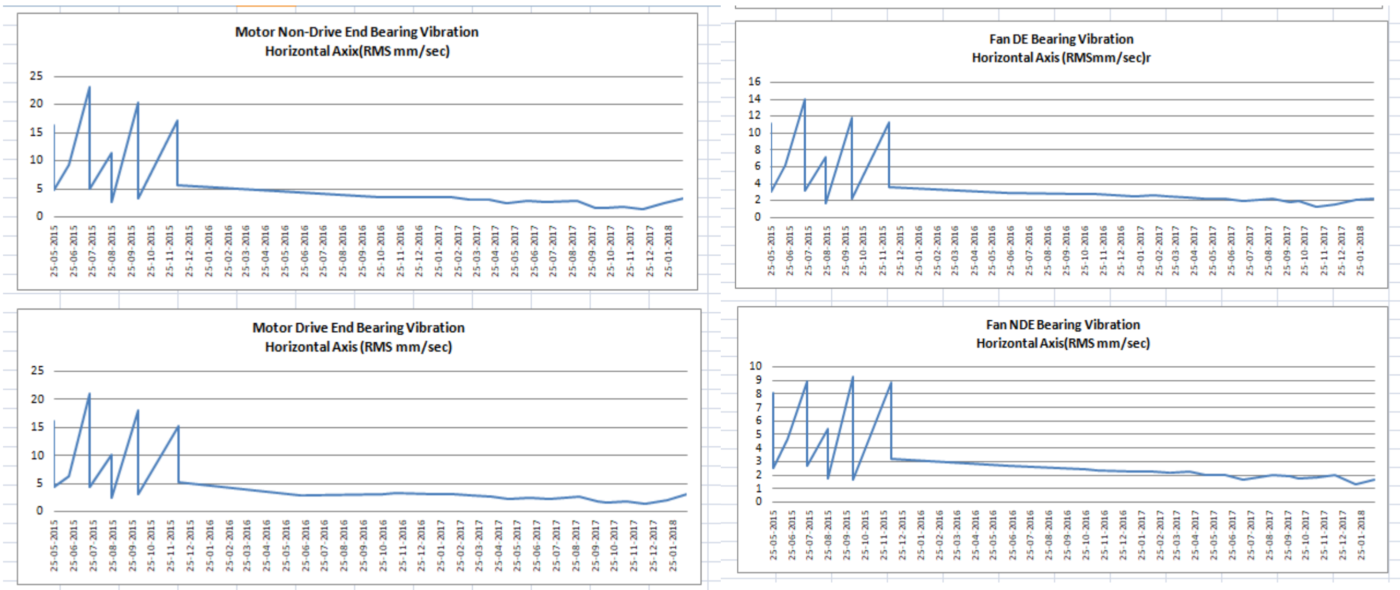

Figure 16. Motor & Fan’s DE & NDE Bearing Horizontal Vibrations after Abrasion Coating Till 2018

DISCUSSION

In this case study the initial problem was that an abnormal vibration was observed after commissioning of an ID fan's motor and fan unit and by vibration analysis the reason of high vibration was found out to be because of resonance. As the motor was of fixed rpm neither its speed could be varied nor due to space restriction the stiffness of the concrete base could be increased by adding extra mass. to avoid the resonance . As the motor was not running on full load current even with 81% damper opening the impeller mass was increased and balanced to avoid resonance. When the problem was analyzed and solved with vibration analysis ,a new dimension was added and that was due to the depletion of the impeller blades as they were subjected to constant abrasion particles that leaked from some of the damaged filter bags of the pollution control unit. This metal erosion problem was then solved by overlaying of the impeller blades with an additional abrasion resistant material which not only arrested the fast wear out of the impeller blades but also added substantial mass to the impeller blades as desired to avoid resonance.

The experiment was successful as before welding the abrasive resistant material mean time between failure (MTBF) was a low as 30-45 days but after the welding the fan operated with out a single failure for the next two years.

CONCLUSION

From the above discussion we can conclude that in a ID fan with a fixed speed prime mover if there is an vibration increase due to resonance and due to economical or space restriction increasing the concrete base stiffness is not feasible and if there is unbalance of the fan impeller then the best option is to balance the impeller .The added mass on the impeller will lower the fan acceleration level and off-set the resonance.

In case the blower unit is connected to a pollution control device handling abrasive material like sand dust or any other raw material dust then its feasible to weld a suitable abrasive resistant material on the impeller blades provided the prime mover is operating below its rated capacity(full load current) and there is some provision to load the prime mover further . Apart from increasing the impeller mass it will also prevent the wearing out of the impeller blades which not only weakens the blades but also reduce the weight of the impeller and re-create resonance and increase vibration. In the above case study it has been demonstrated that with the corrective action MTBF could be increased from barely 45days to two years and more.

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