Vibration Analysis of a Stripper Cooler Fan through SKF Micro Log Analyzer CMXA 70 – A Case Study at Bhubaneswar Power Plant Limited

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Abstract – In-situ balancing is a common balance service for different industries. In-situ balance is a technique for balancing multiple rotary systems as generators, engines, sails, fans and so on in-site. For the vibration observer it really is a challenge to identify the root cause in operating phase or operating mode behind this imbalance.

In the work currently underway, an inquiry has been carried out into in situ balancing work of a fan of strippers mounted on the power plant Bhubaneswar limited BPPL. There is still a difference between theoretical and functional conditions. There was an effort to bridge the divide in this initiative. A number of literatures were analysed in conjunction with their experimental measurements on the protocol for the abovementioned balance technique. The void thus detected is to minimise industrial fan vibration. Based on this, the imbalances in industrial fans are diagnosed, accompanied by a flow diagram, to explain them in a simple phase.

Different facets of imbalance have been analysed in this study. In fact, on the side of fan drive on the horizontal direction of the plummer block (21,55 mm/sec) the fan impeller has considerable displacement and acceleration at the fan and engine locking before any vibration measurements. It indicates imbalance of the fan impeller. SKF CMXA70 Microlog analyzer with an acelerometer on the bearings has been used to collect the vibration results.

The project work is validated through suitable procedure to reduce the vibration of industrial fan.

Key Words – In-situ balancing, Unbalance, Fan Drive end, SKF Micro log analyzerCMXA70, Flowchart.

- X -

1. INTRODUCTION

The spinning bodies may be prevented from vibrating in machines. The dynamic balancing of the matter applies to the dynamic equilibrium mechanism of moving pieces and weights, under which there is no centrifugal force and no resulting pair. The measure of imbalance by using electronic instruments is part of the initial dynamic balance method. Once the desequilibrium has been determined, so the trial weight from moving components or masses has been added or subtracted to the vibration. The dynamic nature of the mechanism is carefully predicted in the design of revolving systems.

It generally analyses the conduct of rotating systems ranging from fans, transmission trains to generators and jet motors. Rotating structures normally create instabilities that are excited and resolved by the imbalance and the internal composition of the rotor structure. This is the prime field of interest for the revolving machine designers**.**

1.1 Unbalance is the Most Common Rotor System Malfunction

"The main sign of unbalance is 1x, which may cause disastrous effects and tiredness of the components of the machine if it is excessive." It can trigger wear and tear in rollers in serious situations, which can break seals, and which can show poor results. In general, the immediate suspect is unbalanced if 1X vibrations are observed. However, as certain other malfunctions emit 1x vibration, many computers were balanced only in order to re-emerge the root cause. Thus, the equilibrium and other malfunctions causing 1X vibration may be correctly diagnosed.

We will focus on the diagnosis of imbalance as an impairment. We will explore the impact of the imbalance on the actions of the rotor system; how it manifests itself in rotor vibrations, stresses and secondary disorders, such as rub. We'll mention the other defects that result in 1X vibration and imitate imbalance. Finally, a particular case of imbalance would be discussed, the loose spinning component.

Rotor System Vibration Due To Unbalance

The 1X unbalance force acts through the Dynamic Stiffness of the systemto cause 1X vibration:

$$\text{Vibration} = \frac{\text{Force}}{\text{Dynamic Stiffness}}$$

It rotates at the same rotor speed and is used in the rotor. The unbalancing power is also synchronous (1X). Only 1X vibration is generated by a linear device with a 1X force. Rotor devices however have nonlinearities such as high-excentration rigidity in fluid film bearings. The abrupt shift in rotor device rigidity as a result of rotor-to-state rubber or looseness in the holder will also be a cause of non-linearity. These nonlinearities will produce 1 X-vibration harmonics, which can often be shown while the rotor is in resonance on a cascade map. With the higher vibration intensity of 1X, the rotor can travel across a more excentric area with a fluid-film bearing, and the sharp rise in rigidity can yield harmonics.

The complex load in the rotor coils occurs as a 1X rotor vibration. This load is transmitted by the rigidity and dämpfung of the bearing and motor housings that belong to the expanded rotor system. The vibration modes of this mechanism involve the rotor and the box of the unit. These modes can be in motion, as rotor and box travel about or out of phase, when rotor and box move around opposite one another. The amount of vibration in the case would rely on a number of variables, including the relative masses of the rotor and the case, the rigidity of rollers, the rigidity of the case and the rigidity of the case mounting and base.

The mass, humidification, and rigidity of the wear and the mounting are the same as the dynamic stiffness of the total support of the rotor, and the proportional quantities of the friction of the shaft and the cabinet are determined by the dynamic stiffness. A high ratio of casing weight to rotor weight generates typically low vibration in the case. In this group, high-pressure compressors and the HP steam turbine device are better vibrated using shaft relative transducers due to the high casing mass.

A lower ratio of the casing mass to the rotor mass, or a soft support, is likely to result in significant amounts of both the shaft relative vible and vibratory casing, which requires casing input transducers, in addition to the shaft relative transducers, moving with or without the case. The relative vibration of the shaft may be strong, even by the shaft actual vibrations may be poor if the rotor and the box vibrate out of phase. Therefore, case measures are needed to provide a full image of device vibration with shaft relative measurements.

Low rotor mass casing and comparatively compact case and supports are provided by aero derivative gas Turbines. Owing to its stability, boxes on such devices can be subject to different modes of vibration and box transducers should be placed carefully to prevent nodal points. In a liquid film rock (maybe because of malalignment), an unbalanced rotor performs with a very high excentricities ratio. It works very easily. The rotor vibration can be transmitted very quickly to the unit housing due to its stiffening of the bears. The high bearing rigidity will at the same time reduce rotor vibration. In this case, relative rotor shaft orbits are small and 1X case vibration is high. Similarly, due to their quite high stiffness, rolling element covers firmly suppress relative vibration of shaft rotor near the coiling (although the relative vibration of the midspan of the shaft may be very large) and directly vibrate to the coil. A bearer rolling part with a rigid rotor is composed of a huge lumped weight and an imbalance, as it is via the case and rotor. The subsequent shaking of the cabinet relies greatly on the rigidity of the cabinet framework and the rigidity. Machines which are quite rigidly installed have low observable 1X vibration levels, except though they have inner dynamic forces and stresses. [15]

2. LITERATURE REVIEW

2.1. Introduction

It was still a challenging task to research dynamically the rotatable high-speed machinery on rolling part layers. A special fascination is the nature of revolving machinery. The vibratory signatures in the majority of the spinning structures are needed for routine analyses. Rotating equipment has a broad spectrum of engineering applications in a variety of industries. The key objective in the architecture of revolving systems is to carry out precise study and complex properties predictive tracking.

A. The equation regulating the response to a single un-damped mass rotor device was formulated and solved by Föppl (1895)[4]. His research showed that the rotor will spin at a pace far greater than the critical speed across its mass centre; its undamped analysis predicted an eternal critical speed reaction and transitory reaction at the critical speed frequency.

The fundamental nature of a single versatile mass rotor reaction to desequilibrium was explored by H.H. Jeffcott[8] in 1919. For years, the testing of these results will not be adequate. In the 1930s, Rieger initially identified electronic and strobo-scopic measurements[12]. Even before Jeffcott showed a fundamental response to rotor systems, balance devices were used as described by Rieger[12]; in 1870 Martinson developed an equilibrium computer. Hand established the 'heavy' location in this balance machine. Jeffcott reports that the "heavy" location of the whirl coincides with the "high" spot at low speeds (sub-critical). In 1928, weights and imbalances of balances were found to act as mechanical powers by balancers like S.H. Weaver [14]. Weaver was mindful of the effects of forces in the position of a rigid rotor due to its scale and stage of the imbalance weights. The principle of coefficients of effect would be established later on, though not initially.

2.2. Balancing using amplitude only

The vibration process in the early days of vibration measurements was challenging to establish accurately. Although the vibration should be filtered to a rotation 1, it has also been calculated by means of a mechanical indicator or vibromètre that average vibration amplitude is used (see Rathbone [10] for an example). Techniques have been established to balance just the intensity of the sound, and this practise continues today. In order to balance turbine generators, G.B. Karelitz [9], of the research division of the German manufacturing company, used a trial weight. This graphics method used an immobiliser to determine the mass desequilibrium, consisting of four translucent strips which were fixed at the same end by a rotor. The technique should be used for unparalleled trial weights. F. Ribary [11] suggested a visual design that matched with the initial amplitude and the three weight test runs only. I.J. Somervaille [13] substantially simplified Ribary's graphic construction. Often known as Somervaille's building is the four-circle approach for balance without stage. The four-circle scheme that has now been commonly adopted has been established by C. Jackson[7]. Balancing with amplitude alone has led to many problems. In order to achieve a two-plan exact-point equilibrium utilising just the magnitude, K.R. Hopkirk [6] suggested an analytical solution; it took seven laps. The method was improved by the two different ways L.E. Barrett, D.F.Li, E.J. Gunter [1] and R.R. Humphris [5] used to stabilise a rotor using modal balancing weights and three ways. The previous publications appeared unfamiliar with L.J. Everett[2], who balanced two planes using only his amplitude alone. As a result of the number of lengths required for the equilibrium the amplitude technique is usually less efficient. Moreover, after such a balance is completed, one has little experience that can be utilised in the future to reduce the balance or to balance one shot. Four more runs are required for a trim balance.

2.3. Balancing using phase only

Phase details on certain early-equilibrium devices from the 1800s can be collected by the explicite labelling of the shaft by N. F. Rieger [12] and F. Ribary [11]. C. Jackson [7] demonstrated how the process could be reached with a pencil and orbit analysis (Lissajous), then the mecanics of the rotators, whether above, below or in close proximity to a crucial speed, were integrated into Jackson. This approach will take many iterations to find a solution depending on the practitioner's expertise and practise. The two-plan balancer technology, focused only on step data, was created by K.R. Hopkirks[6]. The Hopkirk approach includes a two plane accurate balance, which requires five practise runs, including the first. The graphical approach for resolving imbalance on a disc (single plane), with step data only, was introduced by I.J. Somervaille[13]. W.C. Foiles and D.E. Bently[3] found both analytical and graphical methods with just step details for the single and multi-plan balancing; their approach used a kind of effect coefficient that balanced with this partial information. Whereas single plane weight balancing requires three trial weights, only two are required. Methods have been established for single and multi-plane balance. In the Foiles and Bently document, test weights of different magnitudes were permitted[3]. The authors used a cooling tower fan and both analytical and graphical options were proposed with one flat balance (or the influence coefficients for a multi-plane balance).

2.4. Theoretical Methods to Define Frequency Response Function (FRF)

Several approaches have been used to define modal characteristics based on measures of input excitation and reaction. Juang et al. [15] suggested an own machine realisation algorithm (ERA) based on two assumptions: A time invariant (LTI) mechanism is the linearity of the structure. The excitement and response differ. There's a difference. In order to develop its implementation, the ERA is subject to continuing study. [16] Describes how rotor balance phases can decrease noise. Juang et al. Pappa et al.[17] reviewed methodologies for assessing the exactness of modal parameters defined by ERA. The efficacy of four types of modal algorithms was also examined and compared [18,19]. Tom Irvine [20,21] used the Laplace transformation function for solving the absolute response displacement steady-state transfer function, which was used to extract quantities of the transfer function as described in Table2.1.

Table 2.4.1. FRF display quantities:

Parameters	Response/ Impact force	Impact force /Response
Acceleration (g's)	Accelerance (m/s2.N)	Effective mass (s2.N /m)
Velocity (mm/sec)	Mobility (m/ N.s)	Mechanical-impedance (N.s/m)
Displacement (µm)	Dynamic-compliance (µm/N)	Dynamic stiffness (N/ µm)

In vibrant study and modal testing, all FRFs listed in Table 2.4.1 are used to identify the natural frequency of the device, damping ratio, representation in modes and the recent suggestion of use of FRF scale and step in determining the unbalance weight and location on the rotor. The induced FRF is consistent with the unit at any frequency.

3. PROBLEM STATEMENT

Systems like engine and engine rotors, machining equipment, turbocharging manufacturing, etc. When the spinning masses are unbalanced along a rotational axis, the rotor imbalance occurs. This is a real technical challenge, since it is one of the main causes of excessive vibration, especially at higher speeds. The emergence of massive unbalanced centrifuge forces can result in rolling stocks damaged and machines destroyed. This is why the resolution of the imbalance is a fundamental problem of machinery architecture and service.

3.1. Imbalance of rotor-bearing systems

Many experiments on rotor dynamics were carried out during the last few decades. Various maths for the analysis of versatile rotor-bearing systems have been employed. Using finite element techniques, the functional study of structures for rotor bearings performs higher. During the last four decades, some scholars have undertaken detailed study on rotors. Some books have been investigated for their rotor dynamic inputs, which involve crucial speed estimations, weight imbalances, normal whirl frequencies and stability thresholds.

Stress and loss: The rotor can rotate with a constant deflection form if the centerline of a rotor is moving in a rotating 1X orbit centred on the axis of the rotor structure. The rotor seems to have been deformed statically by an observer spinning the shaft. In this case, all sections of the rotor surface can be free of tension.

The static radial load in a horizontal rotor will deflect the rotor and trigger a stress cycle by 1X (either from process load or from gravity). The alternative tension of the static radial load deflection is applied to the constant stress of the 1X loop. Another 2X portion of stress could arise if the orbit is elliptical. As the orbit increases, the alternating tension part increases. It is obvious that a traditional rotor operates in a highly dynamic stress environment as it factor in non-synchronous vibration.

The nominal stresses of the rotor bending are increased by certain tension concentration variables, such as diameter changes, keyways, (drilled) trousers, shrinking shaping suits, surface finishing deficiencies, slag inclusions and corrosion. This leads to rotors that are more susceptible to crashes initiation and fatigue loss due to heavy radial loads and high 1X vibration.

Unbalanced vibration will damage connections. Unbalance bends the rotor and creates tension on stiff, diaphragm and disc pack connections, and increased wear on gear connections.

If standard casing and piping frequencies equate to running rpm, 1X vibrations may also stimulate resonance. The high resonance stresses will result in failure of the cassette or tube fatigue. For eg, during start-up a poorly supported steam injection pipe linked to a gas turbine suffered a big 1X resonance vibration excitation. The vibration caused by the imbalance will affect the connections. Due to unbalance, the bending of the rotor raises the tension of rigid, diaphragm and disc pack connections and increases wear in gear joints.

1X vibrations may also excite the resonances in case they are conveyed by a rolling speed of their normal frequencies. The high stresses induced by the resonance reaction will trigger the cabinet or the pipe fatigue failure. In one case, a poorly assisted steam injection pipe connected to the gas turbine encountered a strong 1X resonance vibration excitation in beginning operations and to a lesser extent a widespread resonance excitation at running speed. The mixture led the pipe to collapse catastrophically.

"The rotor will brush against stationary sections of the unit, known as rub, due to the high 1X vibration. The rub is most probably during a resonance process and may damage screens and decrease efficiency. An excessively cold start of the compressor gave a 1X Bode plot. As it reached the first resonance, the rotor struck a midspan seal, which was a simple bending mode, triggering the rub. The effect of the rub can be seen in the truncated resonance of the amplitude plot and the strong phase change intensity. Since the measuring sensor was next to one roller in this two-bearer unit the midspan amplitude, where the rub occurred, was far higher than the one on the storey." [15]

3.2 Electrical Noise in the Transducer System

Turbo generator sets run either at a line frequency or at some line frequency submultiples, depending on the number of poles in the generator. At 3600 rpm (60 Hz) or 3 000 rpm, a 2-pole generator would work (50 Hz). In the vibration signal, a line frequency component will occur as power line noise couples enter the signal transducer line. This kind of noise was wrong with 1X rotor vibration. This can be examined with a spectral cascade plot while starting or shutting down. As the direction of the rotor varies, the spectrum line of the electric noise is constant at the level of the rotor 1X.

3.3 Coupling Problems

Coupling issues may lead to 1X vibration of some kinds. If the axes of the two rotor machines rigidly coupled are offset (parallel error), the result of a cranking effect in one system or both produces 1x vibration as the machines rotate. This type of 1x cranking motion is also produced by an off-center or off-center connection bolt loop. The pulse is slowly continuing at unbalance-induced 1X.

The joints rely on their action on the lubricated slip of the connecting components. Should a gear coupling be locked, a sudden shift in 1X vibration and an average shaft centerline change can take place.

3.4 Shaft Crack

The rotor bending rigidity reduces in the vicinity of the crank as a split shaft is propagated around the rotor. The rotor will then be less resistant to the dynamic forces that attempt to bend the rotor and normally bow as the crank evolves. The actual heavy spot of the rotor varies, as the arc pushes the rotor mass away from the rotor axis. The cracked arc changes the machine's 1X vibration reaction.

Cracks in the shaft generally result in a transition in amplitude or phase over time. The 1X answer would normally shift steadily in the first weeks to months of crack propagation. The modification in 1X vibration could be confused by simply unbalancing. Though balance can decrease the vibration caused by crack induced arc, there remains the root cause issue and the 1X reaction will shift.

The high vibration amplitudes associated with the reaction will plastically distort the rotor near the crack, abruptly change the arcs, 1X slow rolls vectors and the successful heavy spot if the crack has been well formed. The measurement of balance based on the shutdown data may not be accurate after a large amplitude reaction occurs during initialization. The rotor would have an erratic response in this situation to multiple attempts at balance.

3.5 Loose Part or Debris

When a component moves to a rotor location, or the debris moves into a rotor position, the unbalance distribution and the 1X vibration reaction arising from the rotor can alter. Such a transition may happen periodically, intermittently, or constantly (for example, during start-up or shutdown). A loosely rotated rotor disc or thrust collar may rotate, and a system part can turn axially in some circumstances. Very certainly, if the torque reaches the pressure on the rotor interface, the component falls intermittently. Such a location shift would lead to a 1X vibration phase change, which may be found on an AHT or acceptance area plot. The vibration detected is distinct as compared to prior data whether a component falls at initialization or shutdown.

3.6 Other specified reasons:

a. Causes of an Unbalanced Impeller

"The impellers are out of alignment for two key reasons: a weight change on one hand or warping induced by unequal housing temperatures. Both cases may be caused by production handling, building material or the environment under which the impellers are worked.

b. Motorized impeller unbalance:

Uniform weight: The fluid build-up or dropping from a region of the impeller causes an uneven shift of weight. The impeller spin produces central fugitives, which, under some cases, cause the impeller to gather dust or particles from the air in the outer regions. This forces often may result in loosening of part of the rotor cover (such as paint) to the outside side of the rotor. If the collection of content fragments is constant over the impeller, the weight distribution would be unequal and thus not evenly balanced.

c. Paint spray booth gun

The concept of rear-curved centrifugal fan blades has shown in certain instances to manage the accumulation. In spray booths in particular, the accumulation of air comprising colour particles will pose a constant issue. If the blade is too curved, an accumulation on the back of the impeller in the hollow pocket will form. Back-curved ventilator systems with sloped blades are available to avoid this kind of accumulation. This impeller configuration must also be applied in spray stands and other applications that demand that the ventilator transport air containing materials like paint.

The problems of the forming of lacquer residue on impeller blades also do not involve a lack of equilibrium, since the residue needs to be uniformly formed. It is advised that the rollers be checked periodically and that excess paint is washed as a prevention measure in order to mitigate problems.

Droplet degradation and corrosion are both sources of unequal weight. The first one happens where a tiny region is eroded, mostly water dripping on a portion of the pulse. A difference in the weight of that side of the impeller results in the flow of this product. Corrosion often comes because a chemical has a certain impact and interacts with the impeller. The fan should be held away, if there is a need to treat air or smokes containing toxins, from places that contribute to droplet erosion. In this scenario, it is proposed to construct polypropylene.

d. Uneven Temperature

Non-uniform temperature is another common source of imbalance. A difference between top and bottom can arise when a ventilator rotor remains in rest during shutdown. A connected, but less pronounced, thermal expansion or warping may lead to a temperature difference inside the shaft. This will result from the difference of as little as 1 degree between top and bottom of the shaft. The fan is close a heat source. The majority of times it occurs.

Moreover, if the impeller is running at a rate which is too fast for its nature in terms of revolutions per minute (rpm), it can become hot and induce warping in serious instances. The fan should be disabled for long periods in the event that a heat source influences one region to prevent warping, and the fan should be powered at the pace prescribed by the manufacturers.

e. Causes of an Unbalanced Rotor

Unbalance is also loosely described as the uneven distribution of a rotor's weight on its rotating centerline. The above are causes of imbalance:

f. Blow Holes in Castings

Often cast rotors like pump impellers, big sheavings or sand traps have been created via the casting method. Blow holes can be found within the object which may produce a major source of imbalance whilst being detectable by regular visual examination.

g. Eccentricity

Eccentricities occur where a part does not match the geometric central axis. The rotor might be completely circular, but the centre of rotation is oriented off for whatever purpose.

Addition of keys and keys: a company should match its commodity with a complete key, a half key or no key whatsoever. Thus, if both a slider and an engine maker had to align their parts without keys, the weight of a key would be imbalanced.

h. Distortion

The weight distribution and balance of a rotor can be altered after manufacturing, distortion or shift of form. Stress relaxation or thermal distortion is also the source of distortion. Stress relief, unless done during production, is often an issue with soldering-manufactured rotors. Naturally, every portion formed by pulling, painting, bending, extruding, etc. is subject to strong inner stresses. They will distort this tension over time. Temperature changes are caused by thermal distortion. Once hot, most metals stretch. Rotors usually have small defects and irregular heating experience, which leads to unprecedented distortion. Thermal distortion is typical for devices with high temperatures, including electrical engines, fans, blowers, compressors, expanders, turbines and so on. Often thermal distortion may involve a balanced rotor at its usual operating temperature.

i. Clearance Tolerances

The most prominent cause of imbalance is the build-up of tolerances in a machine's assembly. The bore in a squirrel is bigger than the shaft diameter, for example. To cover the void, the shaft will have to be pressed on one side of the middle of the shaft.

j. Corrosion or Wear

Many rotors, particularly those engaged in material handling, are subject to corrosion, abrasion or wear, especially fan, blower, compressor, pump rotary. Unbalancing results if the corrosion or wear isn't consistent

k. Deposit Build-Up

The unfair build-up of deposits (dirt, lime, etc.) on the rotor will lead to rotors used in the material handling unbalanced. When sections of reserves begin to split up, the resulting progressive increase of imbalances will easily become a significant concern. The vibration rises as tiny deposits disrupt, which in turn breaks down even more deposits and thus causes a severe imbalance. Routine inspection and cleaning will mitigate the effect, but the rotor should normally be eliminated and finally balanced.

l. Manufactured Unsymmetrical Configurations

There are also rotors producing dissymmetry. These include: rough textures on forgings, core cast changes, irregular bolt hole or location and unsymmetrical sections, such as crankshafts, and others. [16]

4. DIAGNOSIS OF UNBALANCE IN INDUSTRIAL FANS

4.1 Diagnosis of unbalance in industrial fans

"The most popular predictive technology in an industrial fan is vibration analysis, the easiest way to diagnose imbalance".

Unbalance is characterised by a radial rotational frequency governed by the rotor (1X). If an aircraft is out of balance, the rotor phase is 00 at both ends. You should search for imbalance in the plane or static imbalance if the fan is idle, remove it from the motor and rotate it to see if it has a fixed idle spot. The fan is unbalanced if it conducts aounter-rotation to return to its idle position.

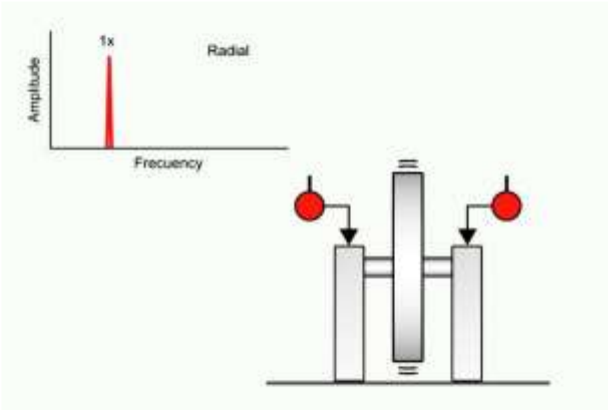


Fig.4.1.1 Unbalance in one plane or static unbalance

Be sure that there are evidence of high vibration and imbalance there is no problem at all. Seek loose bolts or holes. Often the analyst in his feet may sense a high vibration of the base, signaling an issue. If the bench or foundation is an issue, do not diagnose unbalance. The imbalance can occur, but the structural issue must first be fixed.

Fig .4.1.2 Crack in bench near foundation bolt.

Check the temperature, the cleanliness (or wear), the fixation of the fan to the rotor, any axial clearance while the fan is rotated and the appearance of corrective weights from recent or manufacturer balancings whether the fan may be switched off and tested visually. It is is a smart question to ask if the service has ever equalised the fan, adjusted it lately or made updates.

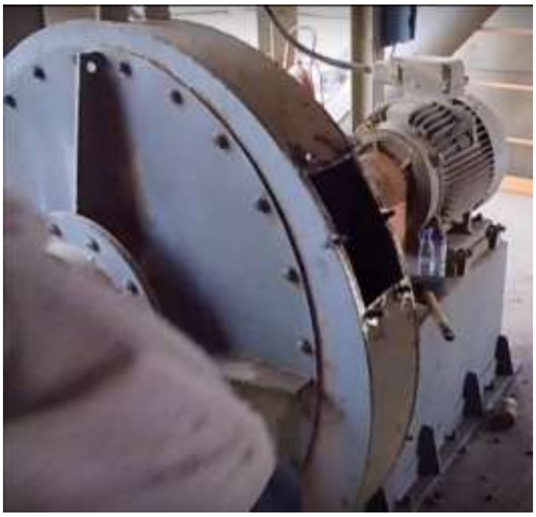


Fig. 4.1.3 Fan and Motor assembly

If there is no signs of a material buildup, blade wear or basic problems that a fan has to be balanced on a daily basis, you may have a problem with resonance. A one-time adjustment operation is a machine balance. Take range measurements from fan start-up at different operation temperatures in case you expect a thermal impact or wish to avoid temperature unbalance.

4.2 Methodology on vibration analysis for a fan

In the vibration analysis of a fan, the most critical thing after an accurate diagnosis is to notify the repair service of the state of the fan unbalance.

A vibrations report must contain:

- Identification (of the asset)
- Introduction
- Operational conditions
- Evolution
- Diagnosis
- Reading points
- Readings
- Recommendations
- Author

4.3. Balancing Plan

The first approach to stabilise the uneven rotor will be discussed in this portion. The balance plan takes the user through the proper sequence: first sprint, the inclusion of the weight of the test, second run and, ultimately, the corrective weight size and location. The plan to balance one plane as seen in fig.4.3.1 in the flux diagram.

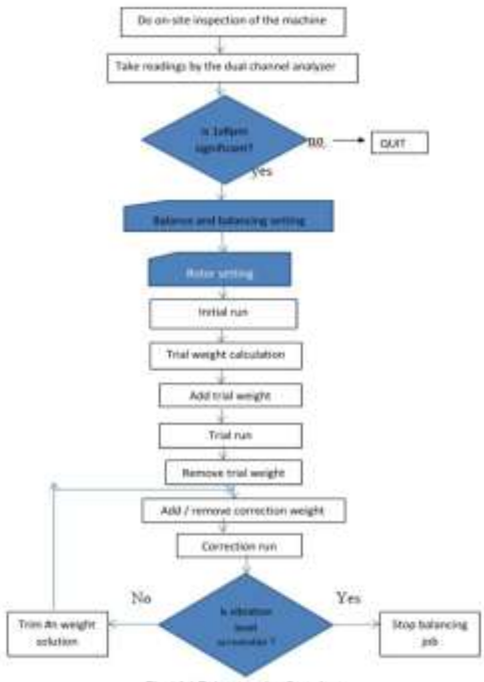


Fig 4.3.1 Balancing plan flow chart

4.4. Calculation of Trial Weight

"ISO 1940/1 defines the permissible residual unbalance as being:

$$U_{per}\ [gmm] = 9549 \times G \times W/N \ \ldots\ldots\ldots\ldots\ldots \quad (1)$$

Where:

G = Balance quality grade

W = Rotor weight [Kg]

N = Maximum service speed [RPM]

Most of the industrial rotors can be balanced, considering a Balance quality grade of 6.3.

In formula (1), by replacing the maximum service speed N with the balancing speed N_b you can calculate the trial weight (T_W):

$$T_W\ [g] = 9549 \times 6.3 \times W/N_b/R \ \ldots\ldots\ldots\ldots\ldots \quad (2)$$

Where, R = Trial weight radius

W = Rotor weight [Kg]

N_b = Balancing speed [RPM]

For a two plane balancing, the value of the trial weight will be half of T_w in each balancing plane ($T_W/2$)." [24]

In the complex balance of rotors, the correct size of the test weight is a major step. The weight of the test must be big enough to shift the unbalance vector significantly, but at the same time must be minimal enough that the rotor cannot be destroyed. As seen in Fig.4.4.1, the unbalance can be described by the original unbalance of the rotor and the vector V2 by adding the weight of the test.

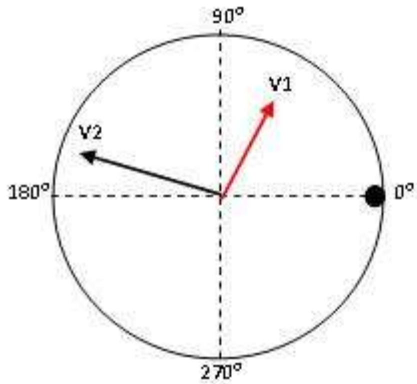
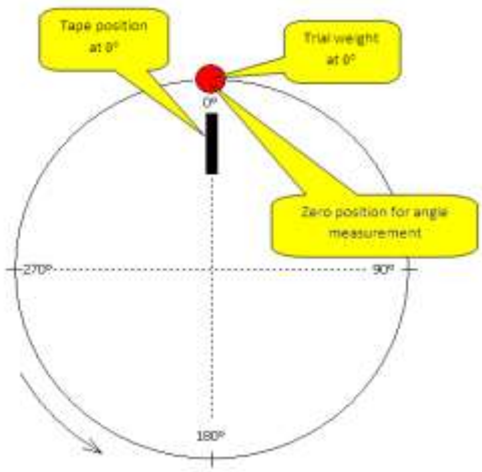


Fig.4.4.1

4.5 Conclusion

To avoid confusions, it is a good practice to define the angle for the trial weight at zero (default setting).

In this condition, the *zero position for the angle measurement* will always be on the trial weight position. In other words, the position of the trial weight determines the *zero position for the angle measurement*.



The reflective tape is much simpler to bring into the position of the trial weight if you are inexperienced. As such, the tape location is a clear null-spot.

Alternatively, please remember where each trial weight would be attached. This is the angles of the balancing plane calculation reference. The right and left balance aircraft cannot have the same zero location. It is not necessary.

If the corner indicator is mistakenly positioned empty, the adjustment weights will be positioned incorrectly and the balance results will be wrong.

4.5.1 Best practice

- Do not adjust the default angle value when editing the test weights (must be always 0).
- Simply choose the spot while attaching the test weight. You can fit the weight of the test everywhere, just label the place. This would be the reference point of this equilibrium plane for angle calculation.
- The comparison plane to the left and right balancing may be any. When you mark all locations, the reflecting tape location or the reference points in the conjugate equilibrium plane should not be taken into account." [17]

5. RESULT AND DISCUSSIONS

5.1 Reading Procedure and data collection:

1. Vibration data was collected from the machine using SKF CMXA 70 Machine condition Analyzer.
2. Software used-SKF@ptitude Asset Management system.
3. Instrument Specifications:

Dual Channel Analyzer-SKF CMXA 70

a) ***Input sources***-Acceleration, Velocity and Displacement

Universal Tachometer Input-Accepts pulse inputs in the range +-25 volts

b) ***Preprocessing***-Enveloper (Demodulator): With 4 selectable input filters for enhanced bearing and gear mesh fault detection.

c) *Filter Selection*:

5 Hz-100 Hz

50 Hz-1000 Hz

500 Hz-10 KHz

KHz-40 KHz

d) ***Input parameters***

Tachometer:

RPM range 1 to 99,999

Tachometer power supply output + 5 volts at 100 mA

e) *Amplitude accuracy*: 5 %

f) Data Processing and Storage:

Micro-processor: Intel Xscale PXA255 at 400 MHz

DSP processor: Motorola DSP56307

Memory: OS Storage Application,

Internal RAM-64 MB

g) Fmax-you can set upto 2,400,000 CPM

h) Sensitivity: 100 mV/EU

5.1.1 Vibration Limits as Per ISO 10816 Standards

(Velocity in mm/sec-RMS)

Below mentioned are standard vibration levels for class I machines. (0-15KW)

Standard Vibration Level	Machine Condition
UP TO 1.8 MM/SEC.	NORMAL
1.8 TO 4.5 MM/SEC.	MARGINAL
ABOVE 4.5 MM/SEC.	CRITICAL

Below mentioned are standard vibration levels for class II machines. (15-75KW)

Standard Vibration Level	Machine Condition
UP TO 2.8 MM/SEC.	NORMAL
2.8 TO 7.1 MM/SEC.	MARGINAL
ABOVE 7.1 MM/SEC.	CRITICAL

Below mentioned are standard vibration levels for class III machines. (>75 KW)

Standard Vibration Level	Machine Condition
UP TO 4.5 MM/SEC.	NORMAL
4.5 TO 11.2 MM/SEC.	MARGINAL
ABOVE 11.2 MM/SEC.	CRITICAL

Below mentioned are standard vibration levels for class IV machines. (Generator Set)

Standard Vibration Level	Machine Condition
UP TO 7.1 MM/SEC.	NORMAL
7.1 TO 18.0 MM/SEC.	MARGINAL
ABOVE 18.0 MM/SEC.	CRITICAL

Below mentioned are standard vibration levels for class V machines. (Reciprocating M/c)

Standard Vibration Level	Machine Condition
UP TO 11.1 MM/SEC.	NORMAL
11.1 TO 28.0MM/SEC.	MARGINAL
Above 28.0 mm/sec.	CRITICAL

NORMAL:-Equipment can be run as usual and no specific maintenance action is required envisaged now.

MARGINAL: - Symptoms indicates onset of abnormality. Trend of symptoms is suggested for determining next course of action.

CRITICAL: - Specific abnormality / Problems identified corrective action is required immediately.

5.1.2 Machine Index

1. EQUIPMENT:STRIPPER COOLER FAN-1B			
MonitoredItem:	**KW** : 230	**Acceptable: (4.5 mm/sec)**	**Still Accep**
Bearing Measuring Points: 04		**MachineMounting:**	
OperatingCondition:		*GroundLevel:*	×
Coupled,withload		*AboveGroundLevel:*	√
Engineeringparameters:		*OnConcreteColumn*	√
Impeller weight -635 kg.		*OnConcreteFloor:*	×
Diameter of Impeller – 1340 mm		*OnsteelFrame:*	√
		OnVibroDamper:	×

Equipment Block Diagram:

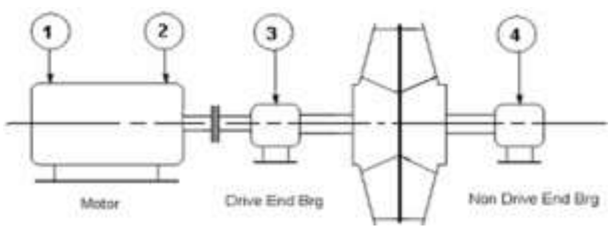


Figure 5.1 Block diagram of the SC Fan

Figure 5.1.1(a) Side view of the SC Fan

Fig 5.1.2 (b) Side view of fan vanes

5.1.3 Overall Vibration Values:

Monitoring Point	*Direction*	*Before Balancing Velocity(Rms) (mm/sec)*	*After Balancing Velocity(Rms) (mm/sec)*	*Before Balancing Acceleration (Peak)g*	*After Balancing Acceleration (Peak)g*
MNDE (1)	H	2.95	0.82	0.20	0.19
	V	0.67	0.68		
	A	1.04	0.63		
MDE (2)	H	3.28	1.38	0.29	0.21
	V	0.80	0.84		
	A	2.02	0.80		
FDE (3)	H	**21.55**	5.39	**4.96**	2.25
	V	3.61	2.51		
	A	**8.08**	3.85		
FNDE (4)	H	**8.69**	3.98	**3.00**	2.83
	V	**7.18**	3.88		
	A	4.50	4.85		

5.2 Machine & Problem Description:

- SC FAN-1B with simply supported structure
- Normal operating speed was from 3000 RPM.
- Job began with a call to balance this fan.

5.3 Onsite Inspection & Initial Data

- Prior to any vibration tests, fan inspections were performed at site in the horizontal direction (21,55 mm / second) showing noticeable movement / vibration at the fan

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and motor room on the fan driving end side, in particular on the fan drive end side.

- SKF Microlog CMXA 70 analyzer with accelerometer attached to the coils has collected vibration info.

- Then, attempts were made to equalise the fan to further decrease its vibration speed.

5.4 Initial Balance Plan

- Initial balancing was performed at 2987 RPM.

- Trial mass-11 gm at blade no.1 as marked in the impeller (Trial mass removed)

- Final balancing was performed at 2987 RPM with corrected mass of 31 gm at 177 0 in anticlockwise direction from blade no.1.

5.5 Results & Conclusions

- Balancing of fan reduced 1 x rpm vibration levels from 21.55 mm/sec to 5.39 mm/sec.

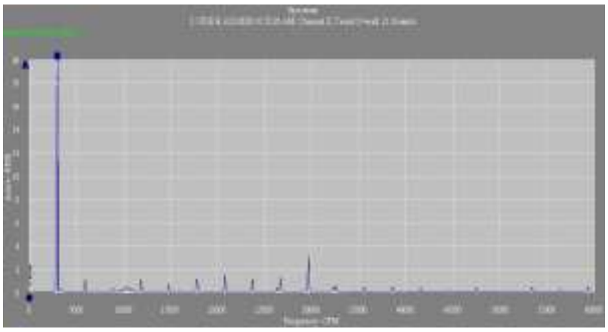
- However shows peak at 8.9X indicating for bearing loose on the shaft at FDE side. Same to be checked on opportunity.

- Fan was run though its normal speed range @ 2987 RPM.

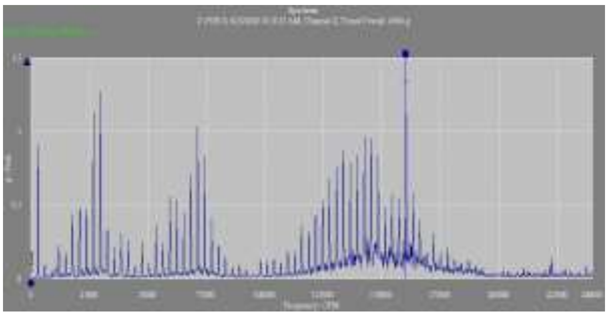
- Normal operation can be performed.

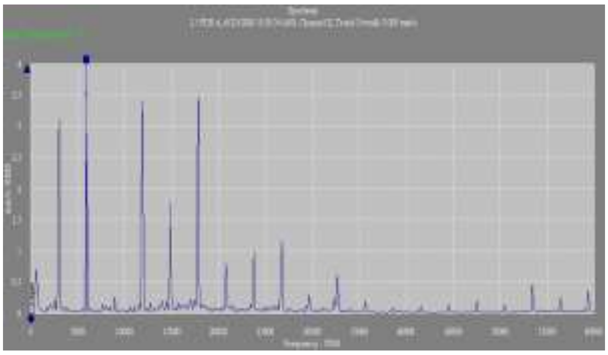
5.5.1 Spectrums:

FDE H-BEFORE BALANCING

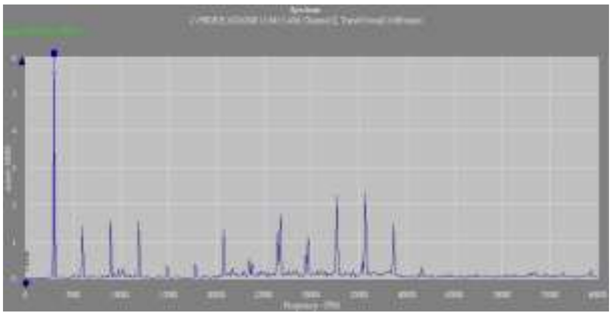


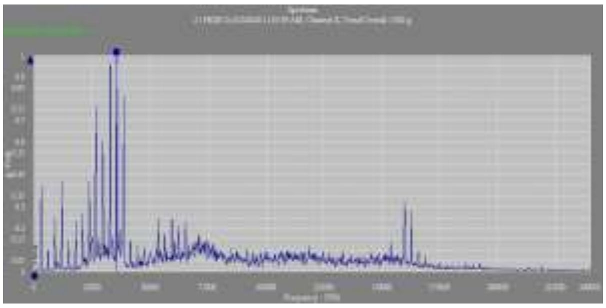
FDE G-BEFORE BALANCING



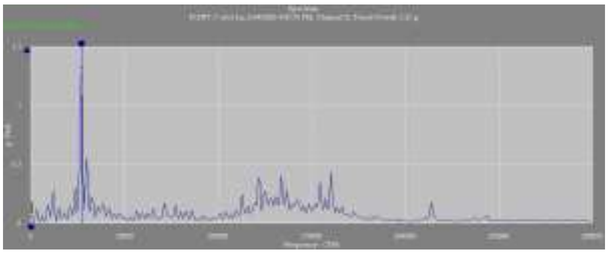


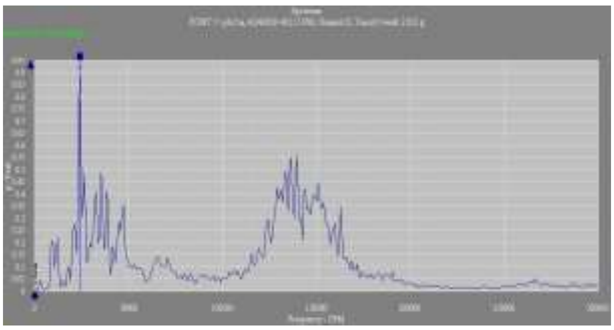
FNDE H-BEFORE BALANCING

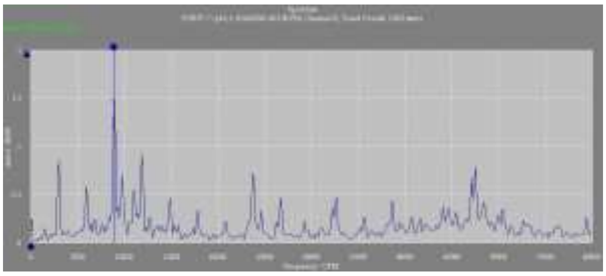




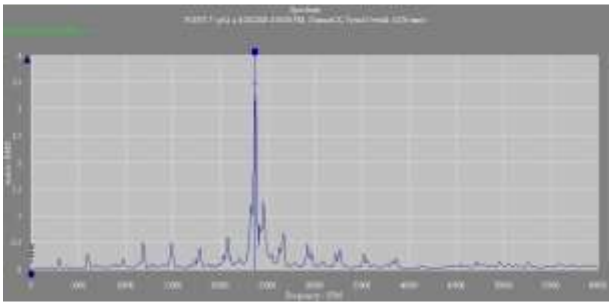
FDE G-AFTER BALANCING







FNDE A-AFTER BALANCING



REFERENCES

[1] L.E. Barrett, D.F. Li and E.J. Gunter (1978). Second mode balancing without phase measurement using the three point method, in: Selected Papers on Field Balancing of Rotating Machinery Advanced Theory and Techniques, E.J. Gunter, ed., No. UVA/643092/MAE81/137, ROMAC Report 80, Charlottesville, VA, 1978. York, 1992.

[2] L.J. Everett, Two plane balancing of a rotor system without vibration phase measurements, J. Vibration, Acoustics, Stress.

[3] W.C. Foiles and D.E. Bently (1988). Balancing with phase only (single-plane and multiplane), Trans. American Society of Mechanical Engineers, J. Vibration, Acoustics, Stress, and Reliability in Design 110(2), pp. 151–157.

[4] A. Föppl (1895). Das problem der lavalschen turbinenwelle, Der Civilingenieur 4, pp. 335–342.

[5] E.J. Gunter, H. Springer and R.R. Humphris (1982). Balancing of multipass flexible rotor bearing system without phase measurements, in: 11th Biennial Conf. on Mechanical Vibration and Noise, Boston, MA, September.

[6] K.R. Hopkirk (1940). Notes on methods of balancing, The Engineer 170, pp. 38–39.

[7] C. Jackson (1979). The Practical Vibration Primer, Gulf Publ., Houston, Texas, 1979.

[8] H.H. Jeffcott (1919). The lateral vibration of loaded shafts in the neighbourhood of a whirling speed – the effect of want of balance, Philosophical Magazine 37, pp. 304–314.

[9] G.B. Karelitz (1928). Field balancing rotors at operating speed,Power 67(7), pp. 286–289. 334 W.C. Foiles et al. / Review: Rotor balancing

[10] T.C. Rathbone (1929). Turbine vibration and balancing, Trans. American Society of Mechanical Engineers, Part I, 51, pp. 267–284.

[11] F. Ribary (1936). The balancing of masses in rotating bodies, Brown Boveri Review 23, pp. 186–192.

[12] N.F. Rieger (1986). Balancing of rigid and flexible rotors, The Shock and Vibration Information Center, Naval Research Laboratory, Washington, DC.

[13] I.J. Somervaille (1954). Balancing a rotating disc, simple graphical construction, Engineering, pp. 241–242.

[14] S.H. Weaver (1928). Balancing of rotors in factory and at installation, General Electric Review 31(10), pp. 542–545

[15] http://vlabs.iitkgp.ernet.in/rmfs/exp3/index.html#:~:text=Because%20the%20unbalance%20is%20part,unbalance%20is%20synchronous%20(1X).&text=The%20bearing%20stiffness%20and%20damping,of%20the%20extended%20rotor%20system.

[16] https://www.axair-fans.co.uk/all-technical-information/causes-unbalanced-impeller/

[17] https://www.mainttech.se/wpcontent/uploads/manuals

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