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Condition Monitoring of Low Pressure Steam Turbine Blades

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Abstract**: *Vibration monitoring is widely recognized as an effective tool for the detection and diagnosis of incipient failures of turbines. Out of all the bladed disks in operation, the one on the last stage of the low-pressure turbine is going to be affected by lot of operating parameters. In this study, the vibration amplitudes, blade lean and blade twist are measured from Blade Tip Timing or Time of Arrival data obtained from Blade Vibration Monitoring System that was installed on a working Steam turbine. In this study, the blade lean and blade twist of the last stage of low pressure turbine are analyzed and trended to notice any changes that occur over time. The design validation is performed using synchronous vibrations for performance assessment. Analysis and trending of synchronous and asynchronous vibrations is done for detection of any deterioration of blades. The blade vibration data is correlated with the operating parameters for severity criteria assessment.***

Keywords:* *NSMS, Blade Vibration Monitoring System, Blade Tip Timing, Campbell diagram, Steam Turbines

I. INTRODUCTION

As the increasing capacity in turbine machinery, the dimensions of the blades assembled on the rotor are becoming larger and larger. In transmitting the power of the turbine, the torsional and centrifugal loads are exerted on the blades, which are complicated and probably unstable, especially in the off-designed conditions or the varying loads conditions. In the view of recently occurring failures in last stage of low pressure turbine, Blade Vibration Monitoring System (BVMS) is installed and the health of each blade is monitored continuously.

Vibrations of Turbine blades in service could reduce their fatigue life by increasing the risk of crack formation. In this frame, the blade health monitoring is an important challenge in order to prevent unexpected blade failures. Traditionally, the rotor blade vibrations have been detected using strain gauges which still represent nowadays the most reliable measurement system. The strain gauges have the disadvantage that, since they need to be stuck on the blade airfoil, they cannot be used in the engine in service. For this reason, in the last years for the blades monitoring, the non-contact measurement technique Blade Tip-Timing (BTT) has gained ground.

In traditional approach strain gauges are applied on the turbine blades monitor the blade vibration. The deformation is measured based on the variations of the resistance. To output the signal, a slip ring is used and installed. As the noise and fluctuating resistance are unavoidable in the slip ring between the rotating component and the stationary component, the application of the strain gauge methods is greatly restrained in rotating machinery. Also, the service life of slip ring decreases rapidly with the increasing rotating speed. The online Blade vibration monitoring system (BVMS) is used by mostly all manufacturers as a complement of strain gauges.

Online blade vibration monitoring system uses Non-contact sensors to detect the time of arrival blade by blade. And the deformation of the blade is acquired based on the comparison of the expected time of arrival and the actual time of arrival. As there's no need to install the sensors on the blades, it avoids the usage of the slip ring and enhances the reliability. The blade vibration monitoring system uses non-contact sensors arranged around the rotor precisely determine the time of arrival (TOA) of each blade tip at each sensor. This data is used in conjunction with a once per revolution sensor located on the shaft of turbine to compute the blade tip Time of arrival and lag which is a measure of tip deflection.

BVMS uses blade tip timing method to measure the blade vibration amplitude, blade lean, tip clearance and blade twist. The main purpose of BVMS is to continuously monitor, acquire, analyze, archive and trend the blade parameters like blade lean, blade twist, asynchronous vibration, torsional vibration etc. w.r.t. time and operating parameters. This helps in detecting any deterioration taking place in the condition of blades viz., cracking, rubbing, foreign object damage etc.

The main purpose of BVMS is to continuously monitor, acquire, analyze, archive and trend the blade parameters like blade lean, blade twist, asynchronous vibration, etc w.r.t. time and operating parameters. This helps in detecting any deterioration taking place in the condition of blades viz., cracking, rubbing, foreign object damage etc.

This paper deals with the monitoring of changes in the vibrations of the last stage of low pressure steam turbine, blade by blade. The last stage blades of the low-pressure steam turbine are of the most concern because they not only experience the pressure from steam but also the vacuum pressure from the condenser. In addition to this the steam at this point is containing a high moisture content. The software used in monitoring the blades are Acquire, Analyze and Monitor. The Acquire collects all the data from the blades and stores it on hard drive in a raw binary format. The Monitor uses that data to plot the amplitudes of asynchronous vibrations and blade twist. The Analyze is used for analyzing each time stamp file. There is a utility in the software to stitch multiple time stamp files for analyzing an event that occurs in two or more time stamps. The frequencies at which these amplitudes are occurring can also be extracted from the time stamp files using Analyze software.

II. LITERATURE SURVEY

In this rapidly expanding field, the state of the technology continues to advance. New solutions to troubling problems are emerging. In turbo machinery as blade failures can be the result of application of mechanical stresses there is a need for blade health monitoring systems. Although some significant advances have been made during past 15years in the field of blade health condition and monitoring, blade failures still continue to take place, and hence efforts are still on to understand totally the blade health monitoring which has resulted in large number of technical papers. A few selected papers that describe some interesting advancements in this field are discussed here.

Ahd. M. Abdelrhman et al have listed diagnostic tools for turbine blade faults and have concluded that vibration could detect most types of blade faults on the basis that dynamic signals are correctly extracted using the most appropriate signal processing method.

R. S. Mohan et al have formulated the equations that govern the vibrations of turbine blades and validated them with a finite element model. They have stated that Flexural natural frequency increases as the rotational speed increases due to centrifugal stiffening effect. The torsion mode frequency is almost constant over rotational speed as it is unaffected by the centrifugal load acting in axial direction.

Jingsong Xie and Michael Pecht has reviewed the applications of in-situ health monitoring and prognostic sensors. They also introduced a physics-of-failure based life-consumption monitoring methodology for the health monitoring and life prediction of electronic systems.

S Madhavan, Rajeev Jain also reported on condition monitoring of turbine rotor blade on a gas turbine engine. The authors investigated the vibration behaviour of LP turbine rotor blade of a gas turbine during forced response excitation numerically and experimentally. The experimental results correlate well with FE predictions for resonance frequency. This paper had focussed specially on the possible cause of fatigue failure which may occur as a direct consequence of anomalies in the experimental vibration characteristics of the blades. The vibration characterization of the blades revealed an unusual behaviour of blades in terms of magnitude response as well as static blade position. From the study, he explained that likely the cause of blade failure is considered to be HCF as revealed by experimental analysis and confirmed from visual examination.

Ray Beebe has published a paper on Condition monitoring of steam turbines by performance analysis and concluded that condition monitoring by performance analysis is a most valuable input in assuring continued operation, and when deciding and justifying major maintenance actions.

Pavel Prochazka and Frantisek Vanek have suggested several diagnostic methods for damage assessment of steam turbine blades under operation. New magneto-resistive sensors have induced further development of contactless diagnostic methods featuring a wide operational velocity range of blade tips in the range from 0 m/sec to 700m/sec with constant amplitude of output voltage.

Todd M. Pickering reported on Methods for Validation of a Turbo Machinery Rotor Blade Tip Timing System. He developed two innovative test methods that were used to experimentally evaluate the performance of a novel blade tip timing (BTT) system from Prime Photonics, LC. The research focused on creating known blade tip offsets and tip vibrations so that the results from a BTT system can be validated. The topic of validation is important to the BTT field as the results between many commercial systems still are not consistent. While the system that was tested is still in development and final validation is not complete, the blade tip offset and vibration frequency validation results show that this BTT system will be a valuable addition to turbo machinery research and development programs once completed.

I B Carrington, J R Wright, J E Cooper and G Dimitriadis have conducted experiments and formulated three vibrational analysis methods and applied to blade tip timing problem for the first time, using data obtained from a simple mathematical blade tip timing simulation.

A.Rama Rao, B.K.Dutta explained that LP blades, especially in nuclear plant where the last stage blades interact with wet steam, the chances of blade vibration and failure is even higher. Intrusive health monitoring system provided by some suppliers has not proved reliable for providing early indications of high vibrations or failure. Besides, they are very expensive and intensive in data processing and interpretation. Blade health monitoring through LP casing vibrations has been shown to be a reliable method for early indication of blade vibration/failure. The method is non-intrusive and easy to implement even in operating plants. As blade vibration is vital for safety and economy of the plant, it is advisable to invest in the technique which is less expensive than the bearing and shaft monitoring system for turbine generator set.

III. THEORY OF BLADE VIBRATION MONITORING SYSTEMS

The fundamental blade vibration involves natural frequency and mode shapes. Mass and stiffness properties determine the natural frequencies and mode shapes of the structure. Excitation levels and damping determine the actual amplitude of the vibration response. Vibration Properties are determined by the structural features alone.

A. Vibratory Forces

Some alternating forces must exist to excite a structure to vibrate. These forces have inherent frequencies and shapes just as bladed disk do. In a steam turbine, the most common sources of excitation are nozzle passing frequencies and running speed harmonics.

Running speed harmonics occur due to interruptions in the fluid flow path. Frequencies of running speed are multiples of rotor operating speed. For example, the tenth harmonic of running speed would be a force that occurs ten times for every revolution of the wheel. For example, a turbine rotor running at a speed of 3000 RPM (50 cycles/sec or Hz) would have running speed harmonics occurring at 100HZ, 150Hz, 200HZ…

Nozzle passing frequency excitation is caused by steam flowing through a nozzle or diaphragm which is used to turn and direct steam flow onto the blades. Because of their design, nozzles have flow interruptions at regular or cyclic intervals which cause the force imparted on the blades to be cyclic. An analogous situation would be running a stick along a picket fence. This cyclic or periodic force can excite bladed disk into their characteristic mode shapes.

B. Resonance

Each blade on a rotating turbine disk experiences a dynamic force when it rotates through a non-uniform flow from stationary vanes. The dynamic response (e.g. stress, displacements, etc.) levels experienced by the bladed depend on:

1) The natural frequencies of the bladed disk and their associated mode shapes
2) The frequency, the shape and the magnitude of the dynamic force which are ion of the turbine speed, number of stationary vanes and their location around the annulus and/or the number of interruptions in the flow passage e.g. struts and their location around the annulus an
3) The energy dissipating properties called damping – provided by blade material, frictional slip between joints, aerodynamic damping from steam, etc. A turbine bladed disk may get into a state of vibration where the energy build up is a maximum in its response (stress, displacement, etc.) and minima in its resistance to the exciting force. This condition is called a state of 'RESONANCE'. There are two simultaneous conditions for the energy built up per cycle of vibration to be a maximum. These conditions are:
4) The frequency of the exciting force equals the natural frequency of vibration.
5) The exciting force profile has the same shape as the associated mode shape of vibration. hus, for a resonance to occur, both of the above conditions must be met.

C. Campbell Diagram

Using modal analyses at several operating conditions (with different temperatures and rotation speeds) the engineer can produce a Campbell Diagram, Figure. The Campbell diagram simply plots natural frequencies versus engine operating speed. The horizontal axis shows the engine speed (in Rev/min). The vertical axis shows the modal frequencies (in Hertz). The Campbell Diagram shown uses fabricated data for illustrative purposes. The horizontal, drooping lines are the natural frequencies of the blade. The straight lines of various integer slopes are the excitation frequencies caused at certain engine speeds. The slope of the line is called the Engine Order. This is an integer factor equal to a rotational symmetry or repetition found somewhere in the path of the turbine

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blades such as the number of nozzles before the blade or the number fuel nozzles in the combustion chamber. Any repeated feature that the blade will see in its travel around the engine represents a potential for resonant excitation. If an engine order line crosses a natural frequency line at an operating speed it is said to be a "resonant crossing." This is a visualization tool that allows the engineer to easily see which modes of vibration will be excited at which engine speeds. The following diagram shows an example of a Campbell Diagram with 7 natural frequencies and 6 engine orders of concern. A crossing between the 22E driver and 4th mode has been identified with a red circle at the second engine operating condition.

Again, the horizontal axis is the engine rotational speed (in RPM) and the vertical axis is the vibration frequency in Hertz. The conversion from RPM to Hertz is 1/60 (revolutions per minute to cycles per second).

The natural frequency lines are not perfectly horizontal. Usually the lines are not perfectly horizontal because natural frequency will vary depending on engine speed and temperature. Although engine speed and temperature are not directly related, they are often highly correlated. For the purposes of the Campbell Diagram, we assume a constant known temperature for each operating condition. For turbine blades, the natural frequency lines usually droop with higher engine speed because thermal softening effects overtake stress-stiffening effects. For fan blades, the lines of natural frequency typically increase slightly with respect to engine operating speeds because temperature increase is small and stress stiffening effects.

D. Formulation

Turbine blades have their axial dimension much greater than the cross-sectional dimensions. Further, the turbine blade is rigidly attached to a stiff rotor at the root. As such to first order, they can be approximated as cantilever beams rotating about an axis perpendicular to the beam. Two modes of vibration arise in this case: - (1) vibration in the plane of rotation (lapping), (2) Vibration out of the plane of rotation (flapping). In order to gain physical insight into the flexural dynamics of such turbine blades with the inclusion of the rotor dynamic effect, the blade geometry is simplified to that of uniform and straight cantilever beams. The equation of motion for this simple model is available in the literature. The solution of these equations is also accomplished in earlier works through semi-analytical techniques. The formulations clearly highlight the rotor dynamic effect regarding that of the well-known case of vibration of beam in a stationary frame.

Consider a beam attached radially to a rotating disc as shown in Figure 3.1. Let m(x) be mass per unit length, E be Young modulus, I be area moment of inertia and f (x; t) be the loading per unit length of the beam. Consider a differential element of length dx at the axial position x. The forces induced on this different element are: - 1. Shear force (V) 2. Axial force (T) at the cross-section caused due to centrifugal effect. Also, the differential element experiences a bending moment (M) as shown in Figure 3.2. Axial force due to centrifugal effect is given by [2]

$$T - \int_x^l m(x)\Omega^2(r+x)dx.$$

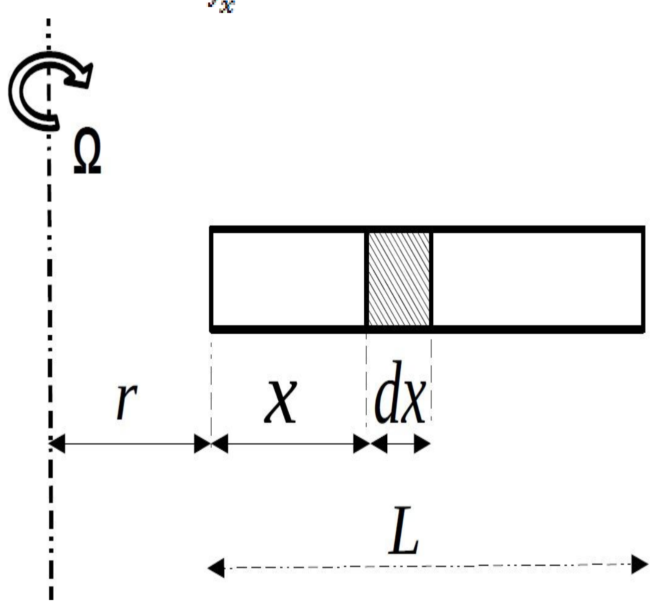

Fig. 1 Beam rotating at speed Ω

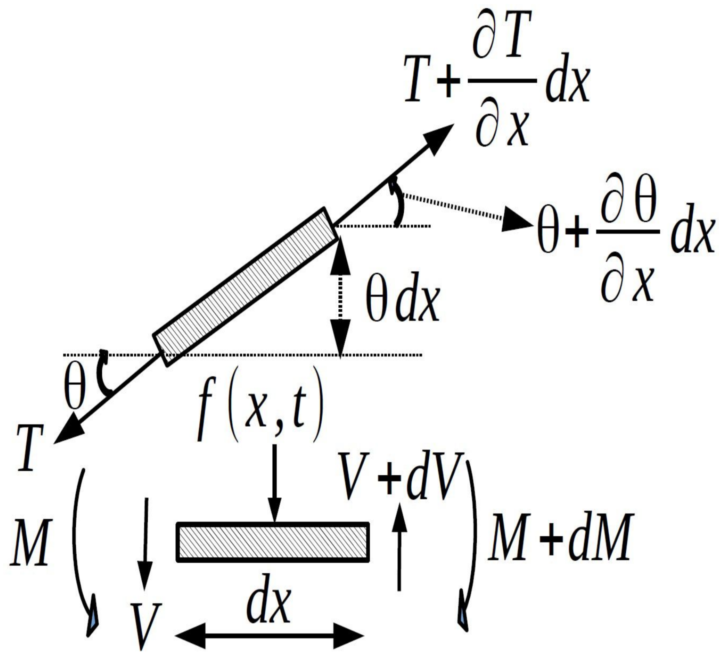

Fig. 2 Axial force acting on 'dx'

As shown in Figure 2, the deformation of the beam enforces non-collinearity of the axial forces acting on either side of the differential beam element. This in turn leads to an additional moment acting in the opposite sense of the bending moment (refer Figure 2). Thus, the effect of rotation is to resist the deflection which in other words implies stiffening of the structure. Alternatively, if observed on a rotating frame moving at the same angular velocity as the blade, the centrifugal action creates a pre-stress (equivalently pre-strain). Thus, the rotor dynamic analysis in such cases is equivalent to the dynamic analysis of a pre-stressed structure in an inertial frame. The governing equation of a rotating Euler-Bernoulli cantilever beam is given by [2]

$$\frac{\partial^2}{\partial x^2}\left(EI\frac{\partial^2 W}{\partial x^2}\right)+m\frac{\partial^2 W}{\partial t^2}-\frac{\partial}{\partial x}\left(T\frac{\partial W}{\partial x}\right)=f(x,t)$$

E. Tip Timing Theory

Tip timing system uses a special analog circuit to detect the exact moment of zero crossing of the eddy current sensor signal. Although zero crossing times can be derived from the DAQ data, it is not as accurate. The DAQ samples, necessitating an interpolation to derive the actual crossing time. The analog circuit triggers precisely on the zero crossing and generates a time stamp. While tip timing information can be derived from the data recorded by the DAQ. Data from the system is stored in a series of files i.e *.inf files. These files can be located and data can be analyzed by using Analyze blade vibration software.

There are some geometrical parameters of the blade that are indicative of damage in blade health monitoring.

1) Time of arrival of blade
2) Blade deformation
3) Inter blade angle
4) Blade position
5) Frequency shifts in Campbell diagram

The structure of the signal returned from the sensor will depend on whether it is a capacitive, eddy current or optical sensor. Blade passage can be determined by BVSI on the repeatable feature of analog pulses generated by sensors. There are two patterns in the analog signal to generate a blade passage pulse. Triggering at a fixed voltage on the raising edge of the pulse. This is recommended for signals already in TTL form or sharp, saturated signal such as optical sensor signal. Adaptive triggering on the falling edge of the pulse. The triggering level is adjustable between 0 to 90% of the pulse's peak value. As shown in Figure 3.4 the pulse width depends little on the pulse amplitude, making the triggering stable. The triggering threshold should be selected to correspond to the largest skew rate.

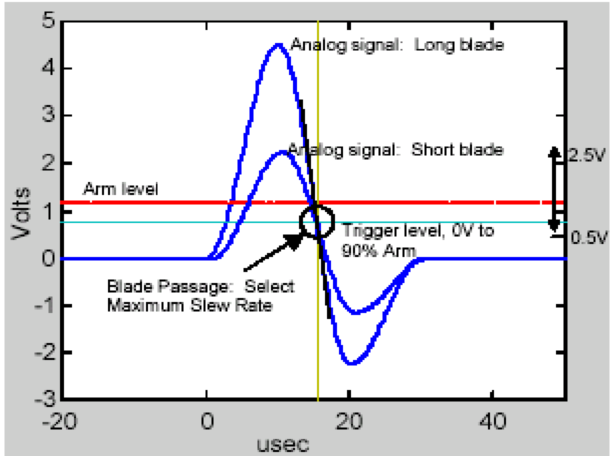

Fig. 3 Blade Passage Logic and BVSI settings

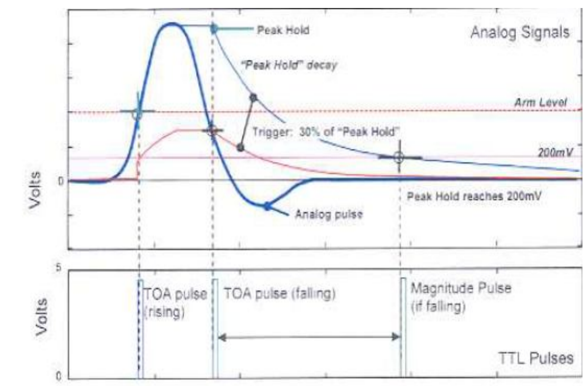

Fig. 4 Raising/falling triggering

Figure 3 shows arm and trigger setup as described below

6) *Rising edge:* The time of arrival (TOA) pulse is generated when the pulse rises above the arm leve
7) *Falling edge:* The peak hold circuit is activated as soon as the signal rises above the arm level. The trigger is adaptive, equal to a fraction of the: peak-Hold" voltage. The TOA pulse is generated when signal fails below the adaptive trigger level.

The blade vibration can be divided into synchronous and asynchronous vibrations. By test, we can obtain the tip-timing time sequence and the synchronous speed time sequence, but there are differences between these two time-sequences in the amplitude frequency and phase of the extraction.

F. Synchronous Vibration Analysis

The synchronous time averaging technique is used to detect the source of vibration at the 1X frequency. 1X frequency is the running speed of the machine. The vibration probe collects the data & the speed probe give the speed information mounted on the reference shaft. This running speed is tracked & the time averaged 1X data is monitored & analyzed. This is very useful in the machine trains with variable speeds or if many other machines are running at close proximity to machine under measurement. Rotor related problems like imbalance, misalignment; rotor related looseness & rotor rub can be identified by using this technology. Other peaks are discarded. Synchronous response analysis can be done in two ways

G. Circumferential Fourier fit

It is direct approach to reconstruct the sinusoid from several sensors. It requires 3 sensors (& 1/rev) to measure 2 times of flight, position, amplitude and phase. Additional sensor allow one to estimate goodness fit.

H. Single degree of freedom Fourier Fit

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Single degree of freedom Fourier Fit requires only one sensor. It calculates magnitude, phase, resonance factor, resonance frequency. To calculate resonance frequency must find excitation order. Magnitude and phase are calculated by assuming Engine order.

I. Asynchronous Vibration Analysis

Asynchronous time averaging technique is used to detect the vibration components that are not related to rotating speed & are above 1X speed i.e. above the running speed. These faults are bearing fault, electrical noise, cavitation, etc. Resonance also produces asynchronous frequencies.

J. Converting time of arrival to blade deflection or blade lean

As there are no direct measurements of strain or stress with the BVMS, the deflection or lean of the blade tip must be determined from the blade TOA relative to the TOA of a rotating reference point. This reference could be a flat or tooth on the rotor that a sensor can trigger from at once per revolution (OPR) or an average position relative to the TOA of the other blades on the rotor.

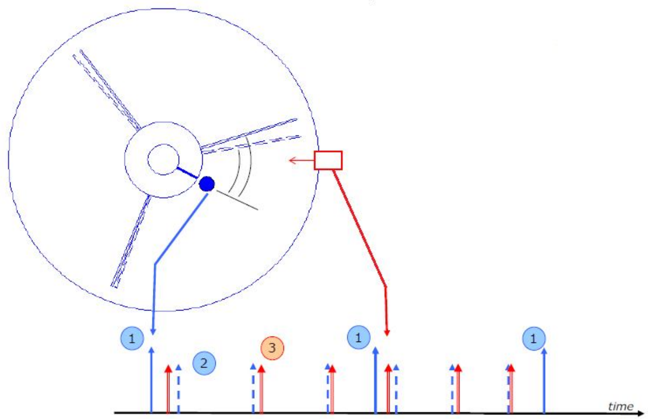

Fig. 5 Measuring Deformation from shaft reference

Blade lean or deformation is defined as the average position (average taken over 60 seconds of blade arrival data) of a blade relative to its perfect position. In a perfect rotor, the blades would be equally spaced, the location of the probes precisely known, the speed of the engine constant (or acceleration known), and the reference for the OPR directly in line with blade 1. The BVM system tracks the arrival position of the blade during every revolution and the maximum value is maximum occurred during 24hour period.

The blades can deflect in either direction as shown in Fig 6

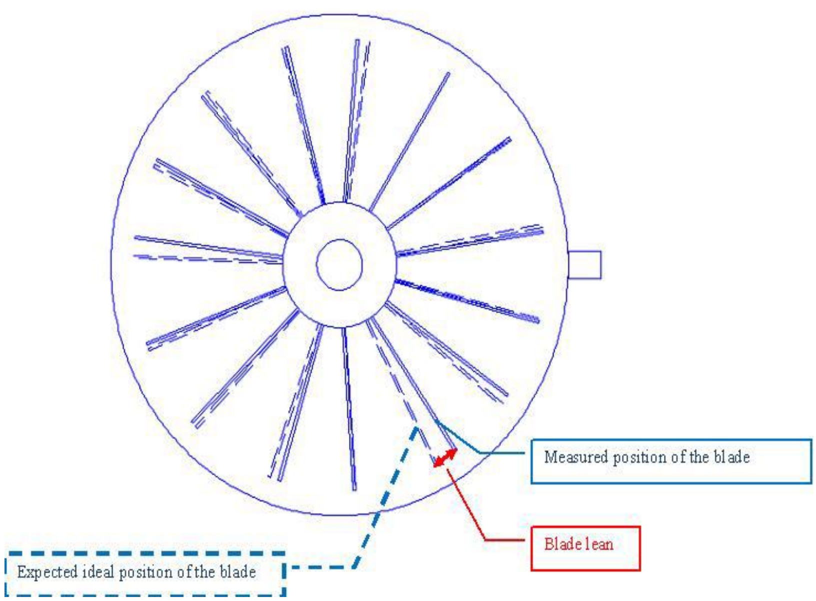

Fig. 6 Deflection of blades

Measuring Deformation by taking shaft Reference

Distance = Velocity × Time

To measure the velocity and acceleration of the rigid motor shaft reference is taken. Arrival time of rigid blades can be calculated using rotor diameter, velocity and acceleration. After that actual time of arrival is measured. To get deformation the difference between times of arrival is calculated and multiplied with velocity i.e.

Blade lean = Deformation = (Calculated TOA – Measured TOA) × Velocity

Measured TOA = $TOA_{B1}^{R1} - TOA_{Shaft}^{R1}$

Calculated TOA = C/ω

Angular velocity, ω = 2πN/60

K. Conversion of Time of arrival to Inter blade angle

In a turbine if we consider an ideal rotor then the Inter Blade Angle (IBA) between all the blades remain same (i.e. all the blades are equally spaced)

IBA (ideal rotor) = C / NB

Where NB = No of Blades

C = Circumference = πd

In a real rotor Inter Blade Angle (IBA) varies from blade to blade

L. Conversion of time of arrival to blade position

To convert the Time of arrival into Blade position the formula can be used is as follows.

Blade position = $\dfrac{TOA_{B1}^{R1} - TOA_{Shaft}^{R1}}{TOA_{shaft}^{1} - TOA_{shaft}^{2}}$

TOA_{B1}^{R1} = 0Time of arrival of blade 1 for 1st rev

TOA_{Shaft}^{R1} = Time taken for a shaft to make 1st rev

TOA_{shaft}^{2} = Time taken for a shaft to make 2nd rev

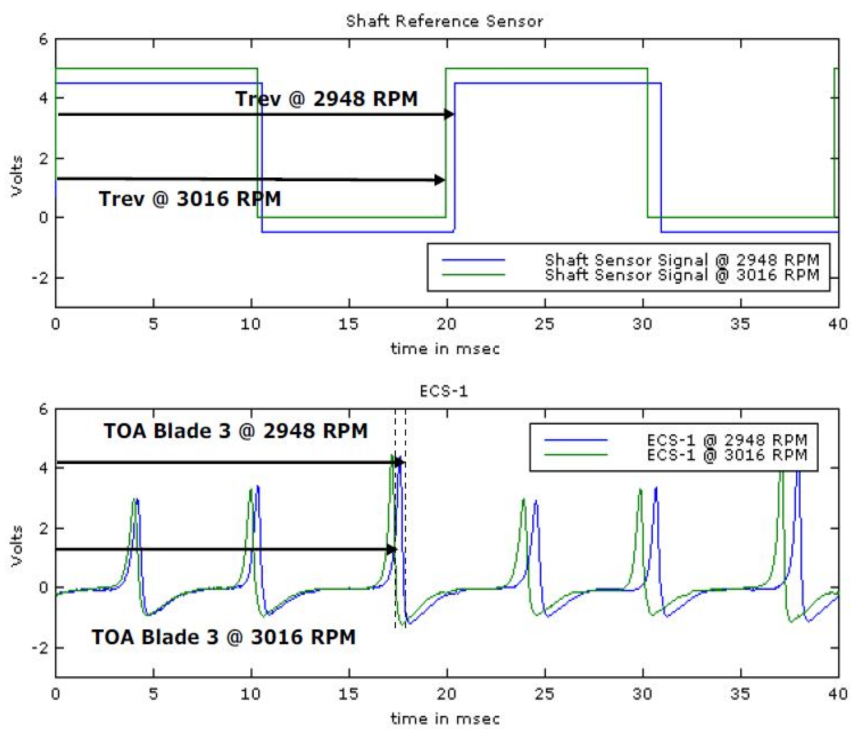

Fig. 7 TOA to Blade position

M. Conversion of time of arrival to Time of flight

Time of Flight is the time taken for a blade to travel from one sensor to another sensor as shown in below Figure 8

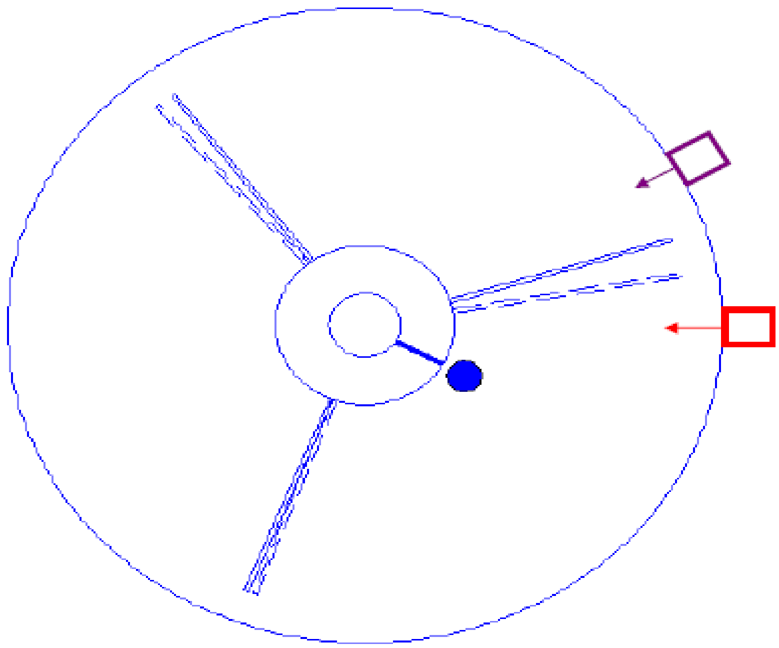

Fig. 8 Time of flight

Time of flight can be calculated form the below formula

Time of flight = $TOA_{B1}^{sensor1} - TOA_{B1}^{sensor2}$

$TOA_{B1}^{sensor1}$ = Time of arrival of blade 1 with respect to sensor 1

$TOA_{B1}^{sensor2}$ = Time of arrival of blade 1 with respect to sensor 2

IV. RESULT AND DISCUSSIONS

The following plots give the trend in the blade vibrations against the operating parameters. The eight Operating parameters used for the analysis can be listed as follows

A. Load
B. Condenser vacuum pressure
C. Shaft 3X vibrations
D. Shaft 4X vibrations
E. Low pressure Heater level
F. LPBP Water Injection Percent
G. LPBP Water Injection flow
H. LP Exhaust Temperature

These operating parameters are not the only conditions that control the vibrations of the last stage of low pressure turbine. There are so many of them which are causing the blade vibrations. But these were chosen because they were considered as crucial after extensive studies done on the subject and also by the previous experiences faced by the company. The other parameters include electrical disturbances and Grid operations.

Figure 9 shows the trend in Asynchronous vibrations for more than two years of data. It also correlates the operational parameters with the vibrations. The first subplot showing the vibrations of each blade at that particular time as seen by the Turbine side mid-chord sensor (Turbine_1091_MC). The following subplots showing the operational parameters values at that particular time, thus enabling us to correlate any changes in parameters that lead to high amplitudes in blade vibrations.

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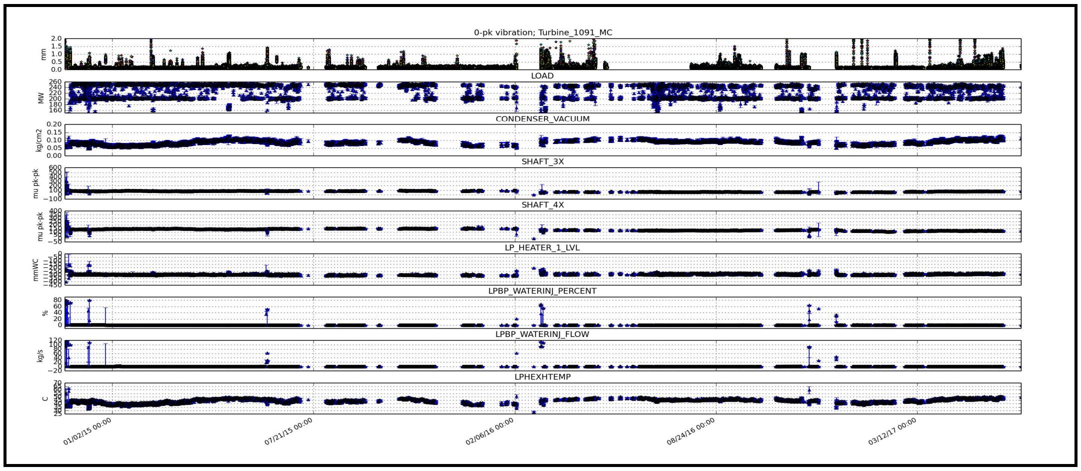

Fig. 9 Trend of Turbine side Asynchronous Vibrations vs. Operating Parameters

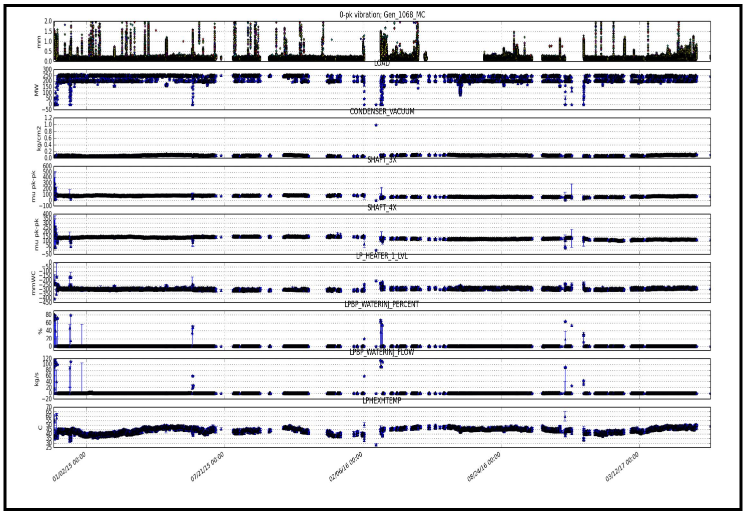

Fig. 10 Trend of Generator side Asynchronous Vibrations vs. Operating Parameters

Figure 10 shows the trend in the vibrations of Generator side wheel as seen by Generator side mid-chord sensor (Gen_1068_MC). Figures 11 and 12 gives the trend in blade lean plot for the same timespan as in above plots. The former showing data about turbine side rotor and the latter about the generator side rotor.

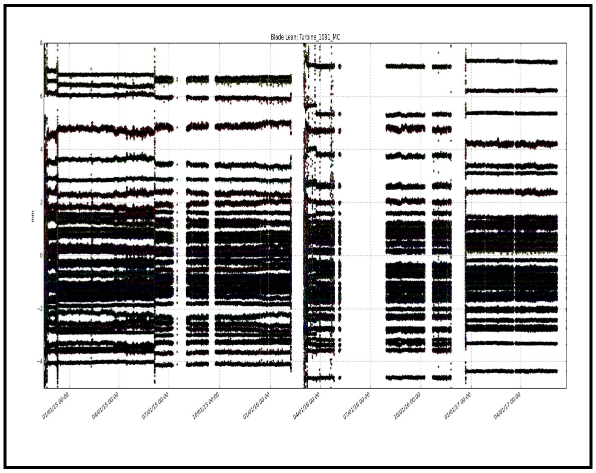

Fig. 11 Trend of Blade lean on Turbine side wheel

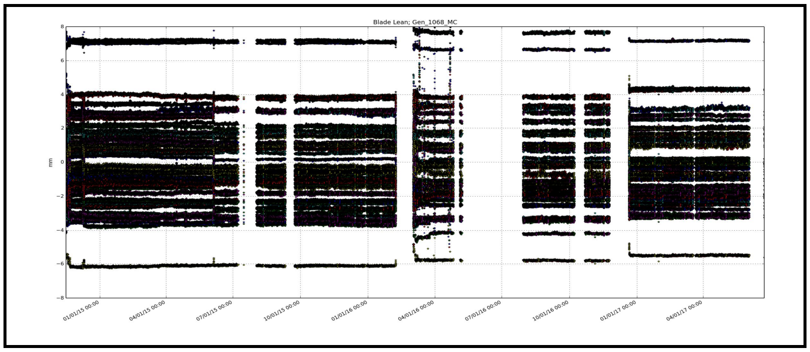

Fig. 12 Trend of Blade Lean on Generator side Wheel

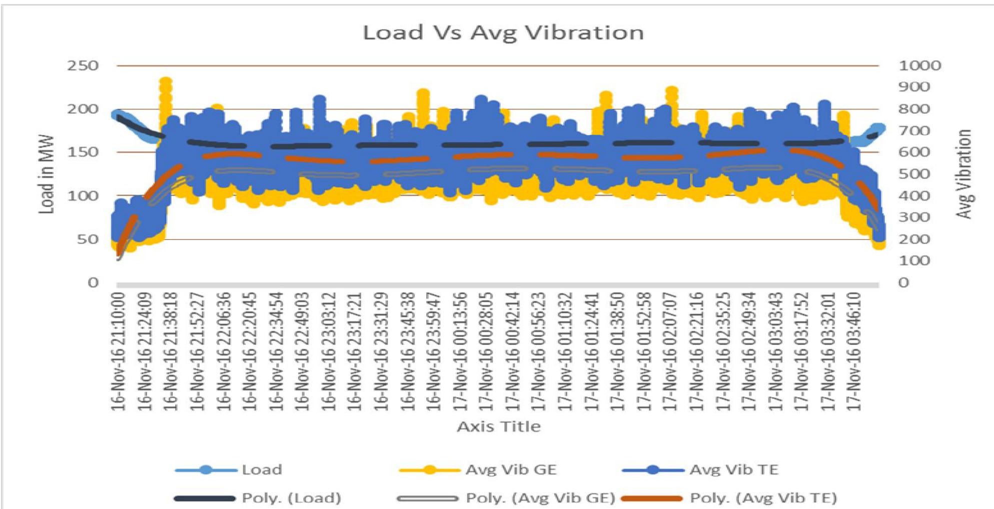

Fig. 13 Load vs. Average Vibrations

Figure 13 shows that the decrease in load results in increasing the amplitude of vibrations. The detailed variation of amplitudes with changes in load from 80MW to 250MW is shown in Figure 14.

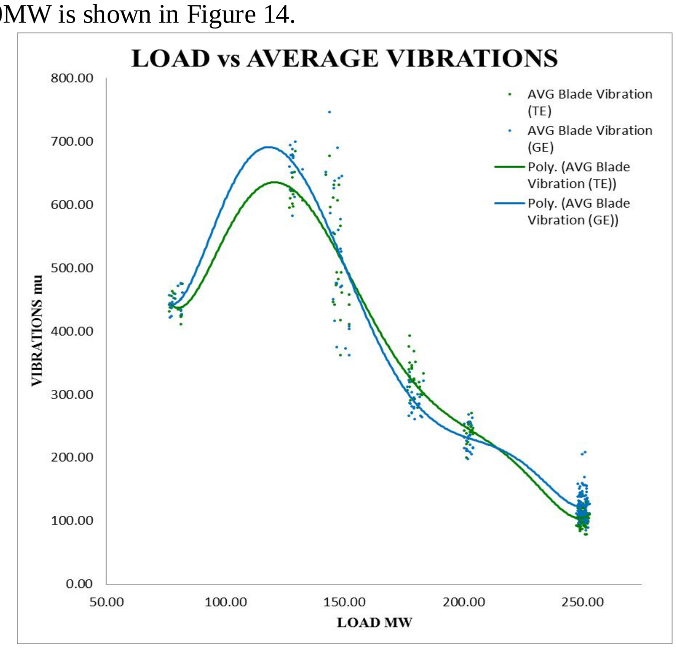

Fig. 14 Load vs. Average vibrations

It is clearly observed from the figure 5.6 that as the load decreases from 250MW to 110MW, the blade amplitudes increase and from 110MW to 80MW the amplitudes decrease. At 80MW usually the circuit trips and load falls to zero, after which the steam is cutoff and turbine comes to zero speed gradually.

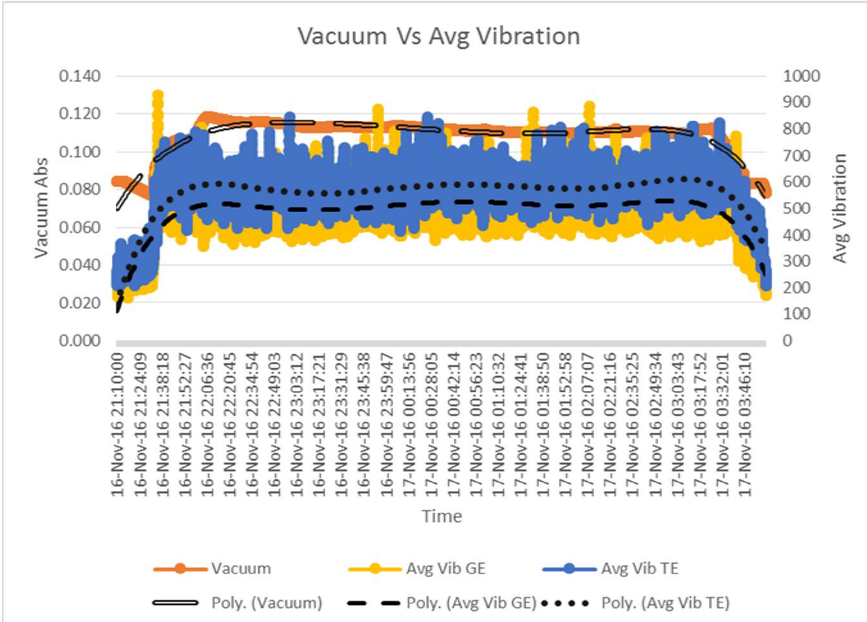

Fig. 15 Condenser Vacuum pressure vs. Average vibrations

Figure 15 shows variation of blade amplitudes with the change in condenser vacuum pressure. It can be seen that as the condenser vacuum increases blade vibration amplitudes increases.

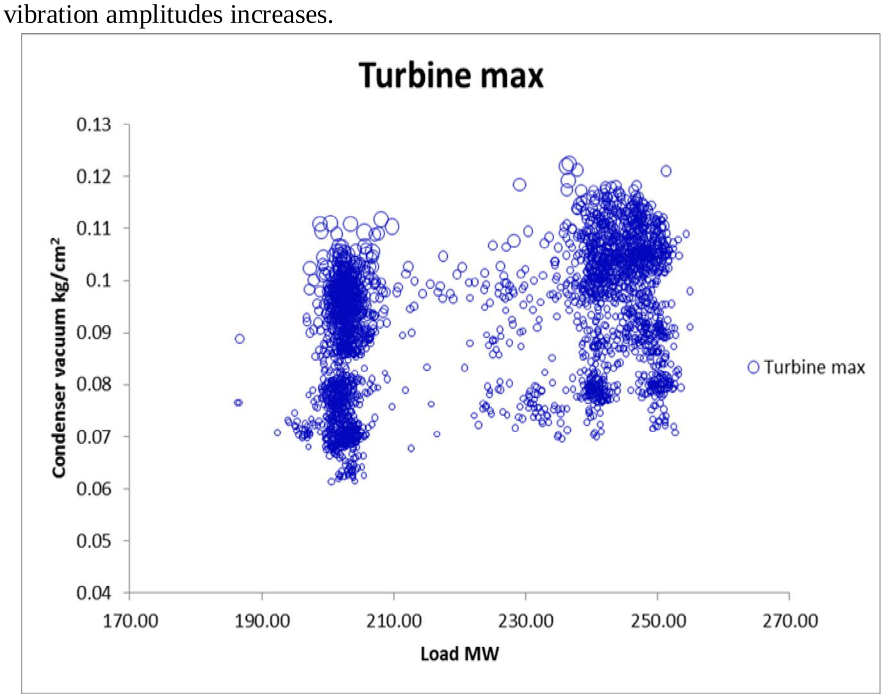

Fig. 16 Load vs. condenser vacuum vs. vibrations

In Figure 16 the blade vibrations are plotted against Load on X-axis and condenser vacuum pressure on Y-axis. The size of the bubble indicates the amplitude of the blade vibration. It is observed from the graph that at any point of load if the condenser vacuum pressure is high, the blade vibrations are observed to be increasing.

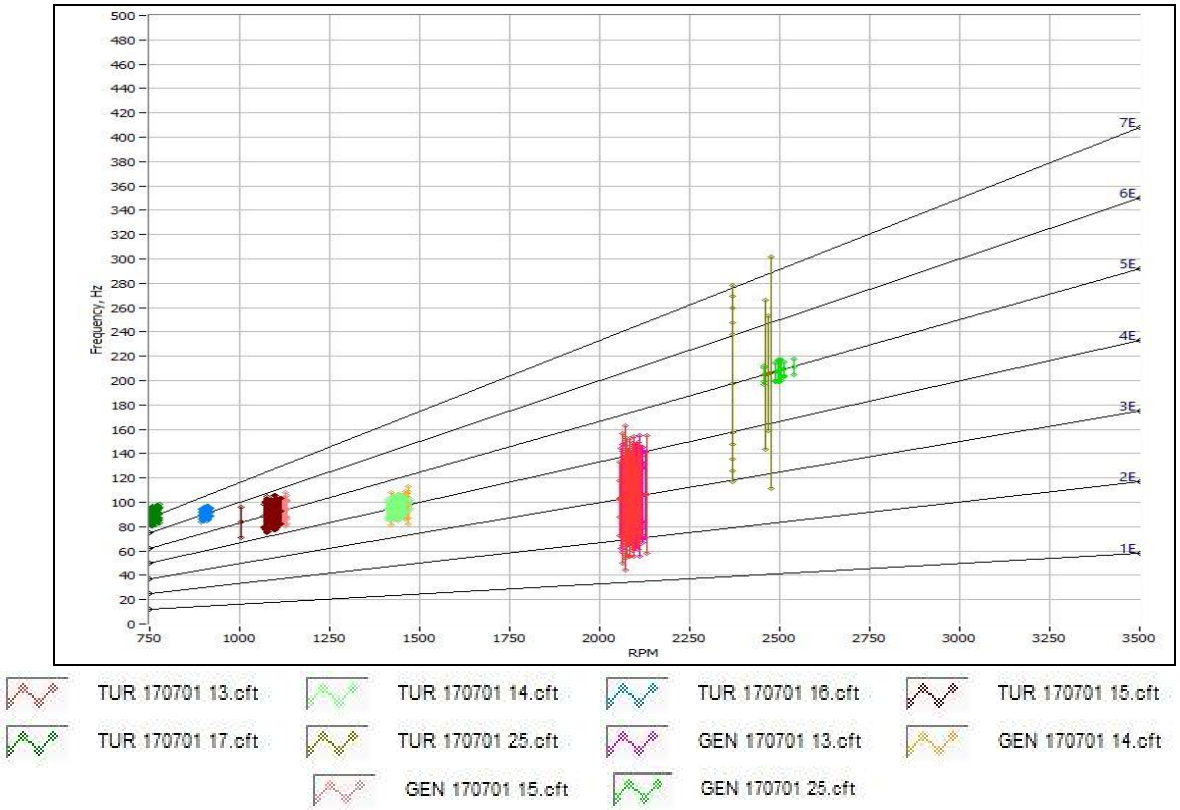

Fig. 17 Campbell diagram

The Campbell Diagram (Figure 17) results obtained from BVMS has been validated with FE analysis using BladePro Software as given in Figure 18 and both were found to be closely matching.

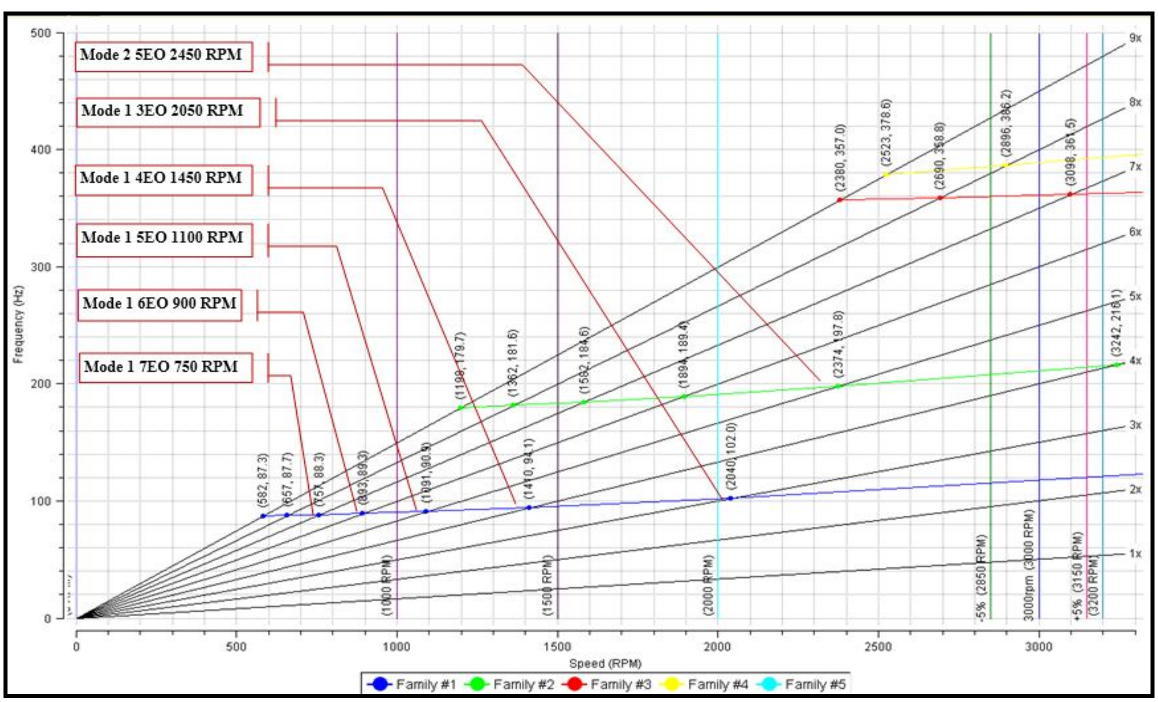

Fig. 18 Campbell Diagram of Blade generated using Blade-Pro Software

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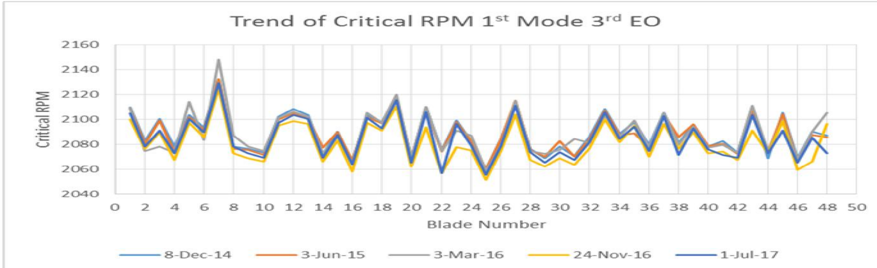

Fig. 19 Trend of Critical RPM

Figure 19 shows the trend in critical RPM for each blade. It can be observed that there is not much change recorded since 8^{th} Dec 2014 implying that the rotor blades are healthy. It also shows how no two adjacent blades are having same natural frequency. The blades are mistuned intentionally while manufacturing to avoid resonance in case of foreign object hitting the blade and making it vibrate at its natural frequency. If a blade is vibrating at its natural frequency and all the blades having same vibrational characteristics then the amplitude of blades gets larger and larger leading to catastrophic failure. By mistuning the adjacent blades this can be avoided.

V. CONCLUSION

A. From the plots shown, it can be concluded that as follows

1) As the load decreases from 250MW to 130MW the average amplitude of non-integral vibrations (Asynchronous vibrations) is observed to be increased to 700µm and as it further decreases the amplitude decreases from then.

2) At 250MW, the condenser vacuum pressure above 0.11 kg/cm^2 results in higher amplitudes and at 200MW, even 0.1 kg/cm^2 of condenser vacuum results in increase of amplitude.

3) Also, it is observed that as load decreases, condenser vacuum pressure should also be decreased in order to control the vibrations. As the load decreases, the amount of steam hitting the blades is brought down by the governor. At this situation if the condenser vacuum pressure is kept at same level, this negative pressure will impart more load on to the blades which increases the amplitude of vibrations.

4) The validation of synchronous vibrations is done by comparing the Campbell diagram obtained from blade vibration monitoring system with that obtained from FEA model. The Campbell diagram obtained from the actual vibrations using Blade Vibration Monitoring System is in good correlation with that obtained from FEA model.

5) The trend of critical RPM at five scheduled shutdowns in 3 years shows that there is no damage on any blade so far.

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