Hot-run-down technique for condition-based maintenance on turbo-generators

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The author discusses the significance of hot-run-down technique adopted by the National Thermal Power Corporation at its power stations and how it helps to identify the reasons for chronic problems. Inferences drawn through field studies and experience are brought out, along with many remedial measures for improvement.

1 Introduction

NTPC has commissioned several turbo-generators of different makes and are in operation for varying periods from one year to a decade. Some of these machines had experienced high axial/radial bearing vibrations during commissioning and just after recommissioning, even after maintaining the maintenance norms. There had been instances wherein high axial and radial vibration problems were experienced in the operating frequency range, namely, 48.5 to 51.5 Hz.

In order to identify the probable causes of high axial / radial vibrations, extensive studies on different machines have been conducted by taking hot run-down signatures by the author. Efforts have been made to assimilate the probable causes and their associated reasons based on field experiences. To substantiate the various actions suggested, the experience gained at NTPC stations are included in this paper, which will serve as a guideline in resolving similar types of problem elsewhere.

A detailed interpretation of hot run-down signatures, run-down strategies, probable causes, remedial measures and its applications are presented in this paper along with acceptable criteria for vibration limits being followed internationally.

2 Significance of hot run-down

Hot run-down technique is used for tracking and plotting of vibration amplitude versus phase as a function of speed, sometime called Bode Plot, and used for identification of machine malfunctions caused by system design and maintenance deficiencies such as resonance and unbalance.

Plots of vibration amplitude and phase versus rpm are extremely useful for identifying critical speeds and other resonant condition. From the plot (**Fig.1**), it can be noted that significant amplitude peak vibration was detected at two operating speeds. Obviously, if the machine in question were designed to operate at or near either of these speeds, some modification would be necessary to permit smooth operation. Simultaneously phase plots are obtained along with amplitude plots to confirm the existence of resonances which will characteristically reveal a 180° phase shift as shown. Of course, machinery will consist of many spring mass systems and it is not usual to find out two, three or more speeds where peak amplitudes occur. In addition, it is very likely that the mass-spring-damping characteristics of the machine and its components will vary considerably in horizontal, vertical and axial directions. Consequently, the resonance frequencies will most likely differ in each direction and it will be necessary in many cases to obtain plots for each direction to ensure that no significant conditions exist for the normal operating speeds of the machines. While plots of vibration amplitude and phase versus rpm are primarily used to identify resonance problems in machine structure, piping, foundation, pedestal etc., this analytical technique can also be valuable in selecting the most suitable or desirable operating speed and for evaluating the effects of increasing speed of older equipment.

2.1 Monitoring strategies and applications

A possible strategy to adopt for the run-down frequency could be to record all run-downs as and when they occur, because it contains much more information than can be obtained from the on-load fingerprint. This is because the run-down signature contains the response to a wide range of driving / exciting frequencies, rather than just 50 Hz. The measurements should be taken under established conditions and will need to cover the full range of speeds from synchronized down to 300 rpm, or to less than half the first generator critical speed. Furthermore, it is not considered that routine monitoring is of sufficient importance to justify the removal of a machine from service specifically to record a run-down. It is therefore proposed that run-downs should be recorded as and when they occur, provided that 1000 hours of operation have taken place on the machine since the last recording.

The method can be used during development by the manufacturer and after commissioning of equipment to ensure that the system will not be operated at or on resonance frequencies which could result in a catastrophic failure.

2.2 Detection of rotor crack by monitoring run-down / run-ups

Run-down / run-up monitoring is used to detect shaft cracks. When a crack appears in a shaft, it affects stiffness of that shaft. The stiffness of the shaft in turn depends on the depth of the crack. When the crack is on the top of the shaft, the weight of rotor forces the crack to close. Conversely, when it is at the bottom, the weight opens it. As a result, stiffness varies over each revolution.

The stiffness is not proportional to the forces causing it and so, as the shaft rotates, the variation causes harmonics of the running speed to be generated. When one of these harmonics coincides with a rotor critical frequency, as in the case of run-up or a run-down, the vibration response to that frequency changes.

A run-down spectrum of a cracked shaft will therefore show a change in response at a frequency corresponding to one of the known rotor critical speeds.

Symptoms of cracked shaft

There are two fundamental symptoms of a cracked shaft.

The vital and primary symptoms of a shaft crack are changes in the synchronous 1X RPM amplitude & phase. The changes in synchronous 1X RPM amplitude and phase are caused by the shaft bowing due to an asymmetric transverse crack.

The secondary and classical symptoms of the occurrence of the 2X component is caused by asymmetry of the shaft. The 2X component is due to a combination of a transverse crack which causes shaft asymmetry and a steady state radial load.

Once a crack is suspected, its size and position can be pinpointed by means of another vibrational technique. After removal from the machine, the shaft is freely suspended on slings from a crane gantry and is connected to a 'shaker'. Its natural resonances and the corresponding deformation patterns (or, mode shapes) are measured for different shaft orientations and will vary slightly depending on whether crack is open or closed. The changes can be related theoretically to both the location of the crack and its depth. Ultrasonic examination or other non-destructive testing methods can then be applied locally for final confirmation.

3 Interpretation of hot run-down spectrum

When a machine passes through a resonance frequency during coast down or start-up, the usual indications will be a peak in amplitude at the resonance speed, accompanied by the characteristic phase shift of approximately 180° (**Fig.1**). Occasionally, however

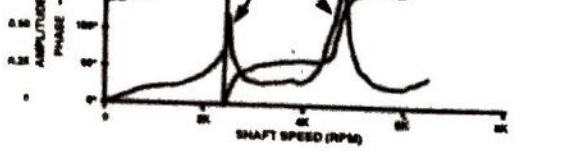


Fig.1

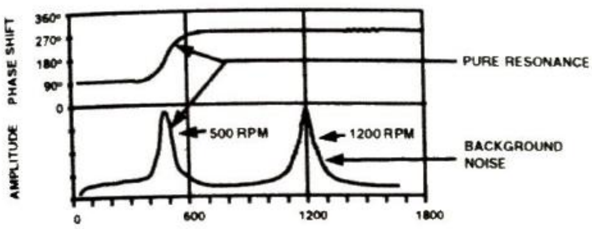


Fig.2

the recorded data may reveal some unusual conditions of system response as illustrated by the following examples:

Example 1 In (**Fig.2**), two distinct amplitude peaks are noted at 500 rpm and 1200 rpm suggesting two resonant speeds. However, the recorded phase data shows the characteristic 180° phase shift for only the lower speed (500 rpm) amplitude peak. There is virtually no phase shift associated with the higher 1200 rpm peak. Based on the data it can be concluded that the lower 500 rpm peak is, in fact, a true resonance. Then, what would cause a peak in 1X RPM vibration which is not due to resonance? One possible cause of the 1200 rpm amplitude peak might be the presence of significant background vibration at the indicated frequency.

Example 2 Refer to the amplitude and phase plots in (**Fig. 3.**) An amplitude peak is noted at a speed of approximately 600 rpm. A 180° phase shift corresponding to the amplitude peak confirms this as being a resonance. In addition, it is also noted that a 180° phase shift occurred at approximately 1400 rpm,

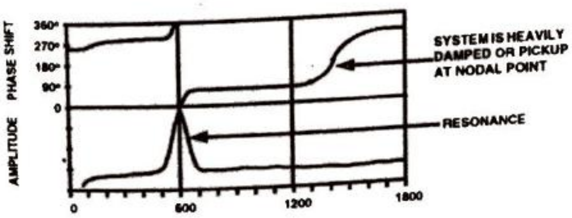


Fig.3

but there is no amplitude peak at this speed to suggest resonance. The 180° phase shift, however verifies that there is, in fact, a resonance at 1400 rpm. Then, how can there be a resonance without a peak in amplitude?

The absence of an amplitude peak at resonance might be the result of one of the following:

1. If the exciting force corresponding to the resonant frequency is very low, or if the system is heavily damped, little amplification in amplitude may occur at resonance. (See **Fig.4** for the effect of damping on system response).

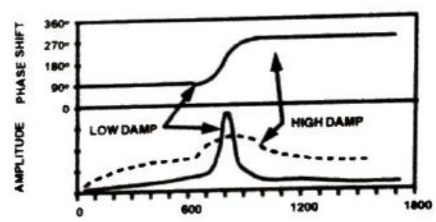


Fig.4

2. If the pick-up is located at a nodal point of the resonant system, little or no noticeable increase in amplitude may be noted at the resonance frequency. When a rotor shaft or structure is excited to vibrate at resonance, it will likely assume one of the vibratory modes (**Fig.5**). Each mode has one or more nodal points which are points of minimal amplitude. Of course, if the vibration pick-ups were located at a nodal point,

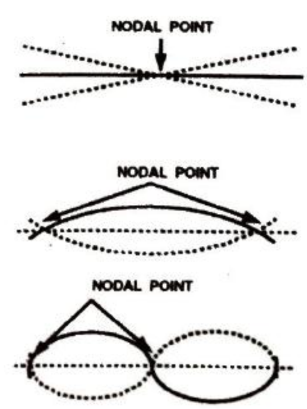


Fig.5

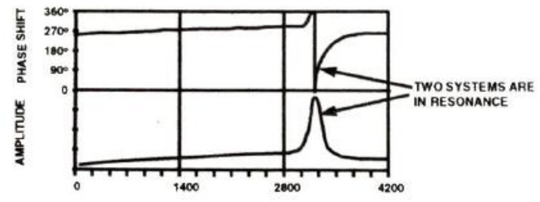


Fig.6

there may be little noticeable amplification of the vibration amplitude when passing through this resonance. However, the phase plot will reveal the characteristic 180° shift. When this occurs, the vibration pick-up should be repositioned to a new location and a second plot obtained to confirm the resonance.

Example 3 Referring to the amplitude and phase-versus-rpm data in (**Fig.6**), it can be noted that an amplitude peak occurred at approximately 900 rpm. However, checking the phase information, it is noted that the amplitude peak is accompanied by a phase shift of approximately 360° instead of the usual 180° . In this case, the phase data suggests that there are actually two systems in resonance at or near the same frequency, and each of the two resonances is contributing a 180° phase shift for a total phase shift of 360°. In this case identifying and correcting only one of the resonant systems may not totally solve the problem.

Example 4 Referring to the sample data in (**Fig.7**), the somewhat unusual portion of the data which deserves explanation is the dip in vibration amplitude which occurs at approximately 1000 rpm, along with the accompanying 360° shift in phase. As in example 3, the 360° phase shift at 1000 rpm in (**Fig.7**) suggests two spring-mass systems in resonance at or very near the same frequency. The dip or reduction in vibration amplitude at this frequency sometimes referred to as an anti-node can be explained by referring to the spring mass systems in (**Fig.8**).

First, assume that the initial exciting force is the unbalance, U. If spring-mass system *A* is resonant with the unbalance frequency, then the actual vibration of mass *A* lags the unbalance force by 90° Now, if spring-mass system *B* is resonant at the same frequency, then its exciting force will be derived from the motion of mass *A*. Therefore, the actual vibration of mass *B* must lag the motion of *A* by 90° . As a result, the actual vibration of mass *B* will lag the initial unbalance force by a total of $90° + 90° = 180°$. This produces two opposite forces acting on a mass *A*, one being the initial unbalance force and the second being the force of vibration of mass *B*. Since the two forces acting on mass *A* are opposite, mass *A* will exhibit a minimal vibration amplitude as illustrated by the dip in **Fig.7**. Mass ***B***, on the other hand, may reveal an extremely high amplitude of vibration. For this reason, when the data suggests a possible anti-node, it would be advisable to identify the second system in resonance and plot its response as well, to ensure satisfactory performance.

The principals of the anti-node are sometimes put to beneficial use to control a resonance condition. For example, if a machine or structure was found to be vibrating at resonance, the addition of a second resonant spring mass system may serve to control or minimize the vibration of the machine or structure. For example, if the bearing pedestal in (**Fig.8**) was vibrating at resonance, the addition might be used to minimize bearing vibration in a situation where

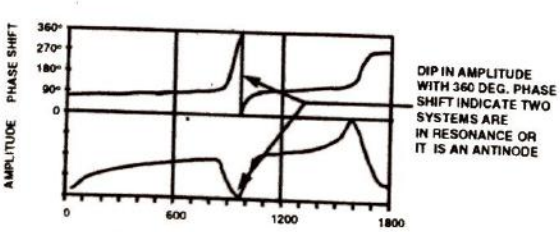


Fig.7

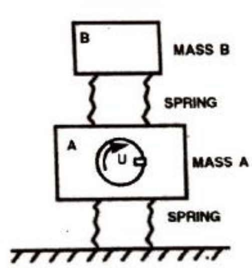


Fig.8

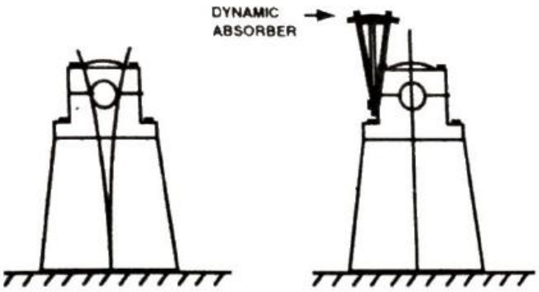


Fig.9

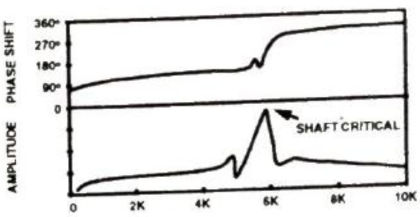


Fig.11

changing the exciting frequency (rpm) or changing the bearing natural frequency was not possible or practical. The resonant system added to the machine or structure is often called a dynamic absorber.

The position of the weight on the bar in Fig.9 can be adjusted to accurately fine-tune the resonant frequency of the bar to the known exciting frequency.

Example 5 Plots similar to that in Fig.10 are sometimes obtained when non-contact pick-ups are used to identify rotor critical speeds. The plot does display a peak amplitude and the normal 180° phase shift. However, it should be further noted that the amplitude plot includes a dip at approximately 5000 rpm, and, in this case, just slightly below the shaft critical speed. The amplitude dip in Fig.10 does not have a substantial corresponding phase shift and, thus, is not likely the result of an anti-node, plotted in **Fig.7**. The cause of the indicated amplitude reduction (dip) in the plot, **Fig.10** can unusually be traced to excessive electrical and/or mechanical run-out of shaft at the non-contact pick-up target area. The non-contact or proximity pick-up cannot distinguish between actual shaft vibration and any run-out or eccentricity of the shaft journal. As a result, the non-contact pick-up provides a signal proportional to the vector sum of run-out and actual shaft vibration.

Operating well below its critical speed, a shaft is considered to be rigid and, thus shaft vibration will essentially be in phase with unbalance. However, as the rpm of the shaft approaches its critical speed, shaft deflection will progressively increase. In addition, the deflection in the shaft (actual shaft vibration) begins to lag the heavy spot of unbalance. In fact, operating at critical speed, shaft deflection lags the unbalance heavy spot by 90°, and at operating speeds above shaft critical, deflection will lag the unbalance by 180°. The important point to note here is that the amplitude and phase of shaft vibration is undergoing a change as the shaft passes through critical speed. As a result, the pick-ups, the presence of excessive run-out may give a very distorted picture of rotor response. The recorded data may suggest a condition which is far better or far worse than the true response of the rotor. Therefore, it is advisable to carefully measure run-out amplitude before response measurements are taken. Of course, excessive run-out should be physically or electronically eliminated so that plots of true rotor response can be obtained.

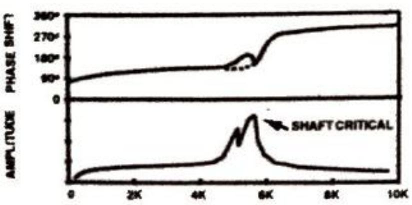


Fig.10

Example 6 The recorded plot in **Fig.12** illustrates a situation where the recorded data can be somewhat misleading. The suspicious data in **Fig.12** are the somewhat unusual phase shifts that occur at 6000 and 7000 rpm. While these apparent phase shifts may suggest resonant frequencies which were not indicated on the amplitude plot, it is quite possible that these are not phase shifts at all. It is quite possible it could be due to the absence of a sufficient vibration

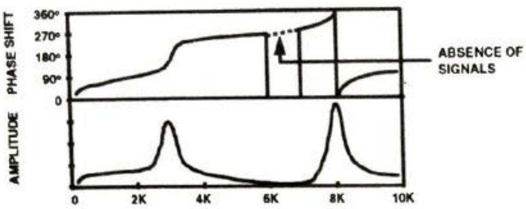


Fig.12

Example 7 The phase plot in Fig.13 illustrates another situation where the data can be somewhat misleading. From the phase plot, it appears as though there may have been three abrupt phase shifts of 360° each within the speed range of 4500 to 6000 rpm. However, this type of phase plot is not unusual where the phase indication tends to vary around 0 to 360° and does not suggest anything unusual or abnormal as far as system response is concerned. It is more likely that the unusual phase shifts between 4500 and 6000 rpm in Fig.13 result from a phase indication which may have been varying only slightly, say between 1° and 359°, which is only a 2° variation. Of course, if the phase changes from 1° and 359°, the plotter pen must travel all the way from the bottom to the top of the phase chart. And, if the phase should shift slightly from 359° back to 1°, then the pen must, again, travel the full distance from the top of the phase chart to the bottom. This tends to produce a cluttered and somewhat confusing phase picture. However, as long as the cause is understood, the data should not be misinterpreted.

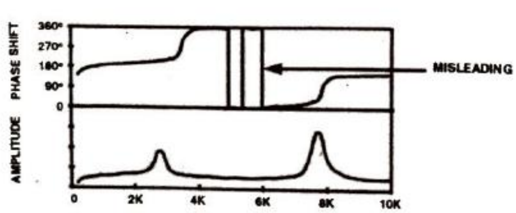


Fig.13

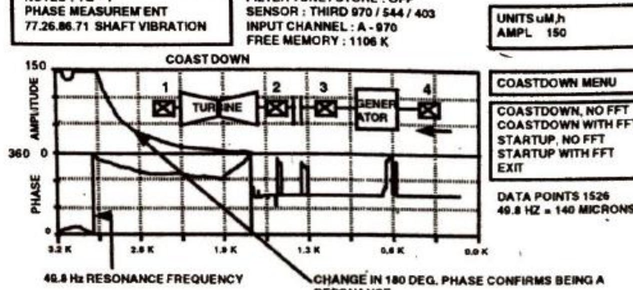
4 Discussion on hot run-down signature experienced at NTPC

The objective of the recording of hot run-down is to see how the rotor dynamic response changes with varying speed. High axial vibration problems were experienced at BCPP TG1, KhSTPS TG2 projects and high radial vibration problem at KSTPS BFP2A.

The RhSTPS Generators had experienced wrapper resonance problem which was identified by taking hot run-down with and without excitation, and detuned the resonance frequency away from operating frequencies range by adding mass of about 10 **t** on the wrapper of Generator Unit-1 & 2. In order to find out the cause of high axial / radial vibrations problem, hot run-down of problematic bearings and pedestals was recorded and the probable reasons such as resonance, unbalance, inadequate stiffness and damping identified. The case studies experienced by the author are presented here along with their interpretation.

Case study 1: Resonance - BCPP TG 1 (run-down date 12/7/93)

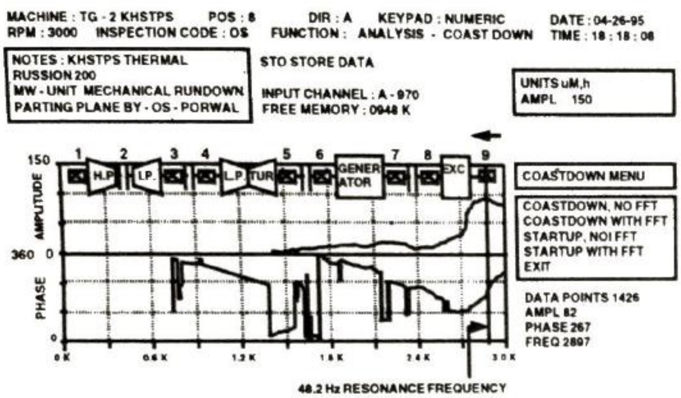
TG1 Generator rear bearing had high axial vibration since commissioning of the unit. In order to identify the reason, a hot run-down spectrum was taken in axial direction at bearing no 4 as shown in **HTRDN#1** which revealed that bearing top half is resonating at 49.8 Hz and later confirmed by putting crane hook weighing 500 kg weight on the bearing, when it reduced axial vibration to 68 microns pk-pk from 140 microns pk-pk. As a temporary measure, dynamic absorber has been made and installed which finally reduced the axial vibration to an acceptable limit of 68 micron pk-pk.



HTRDN#1

Case study 2: Low amplitude resonance - KHSTPS TG2 (run-down date 26/4/95)

Exciter front bearing had high axial vibration problem since commissioning of the unit. The hot run-down spectrum was taken as shown in **HTDRN#2** and it

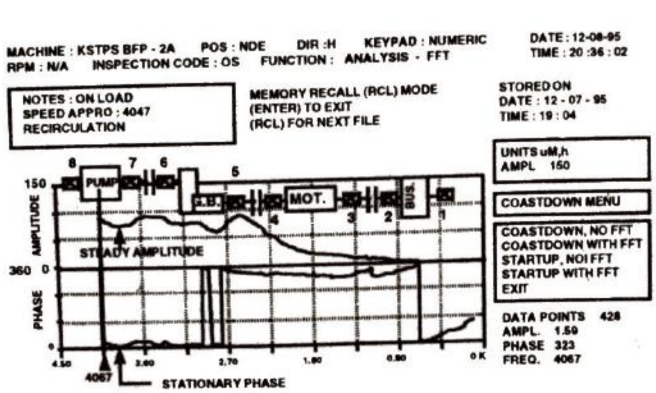


HTRDN#2

was found that bearing top half is resonating at 48.2 Hz. The maximum vibration of the order of 82 micron pk-pk was observed at exciter front bearing at unexcited condition which came down to 72 micron pk-pk under excitation. It could be due to inadequate stiffness of the bearing pedestal. As the vibration level at present is below 70 micron pk-pk, the machine was allowed to operate at full load as per **ISO 3945**. For further reduction of axial vibration, dynamic absorber can be installed.

Case study 3: Unbalance- KSTPS BFP#2A (Run-down date 07/12/95)

The KSTPS BFP2A has been experiencing high horizontal vibration since re-commissioning after balancing of cartridge on balancing machine at RhSTPS. In order to find out the reasons, hot



HTRDN#3

run-down was taken and identified that this is due to unbalance as amplitude and its phase is constant throughout operating frequencies range upto 4067 rpm as in **HTRDN#3**.

Case study 4: Resonance & humming sound - RhSTPS Generator#1 & 2 (Run-down dates July & Sept. 1992)

The RhSTPS generators had experienced design deficiencies caused by inadequate core-bore rings stiffness which resulted in shift of natural frequency towards operating frequency zone, creating resonance and humming sound. In order to off-tune the resonance frequency, stiffening of core and bore ring was done by providing additional stiffeners in various zones. After carrying out this modification, generator wrapper had started vibrating at around 100 micron at second harmonic. Hot run-down was taken and identified that wrapper is resonating in operating frequency zone. In order to detune the natural frequency of wrapper towards under frequency zone, a 1.5 t of mass was applied, equally spaced at six locations, which resulted in reduction in vibration level at second harmonic i.e., 40 micron pk-pk. The **HTRDN#4 & 5** were taken before and after added mass modification. The mass added on the generator wrapper is shown in **Fig. 14**.

5 Causes, remedies, applications and recommendations

5.1 Probable causes

(a) Initiation of crack or inhomogeneity in the rotor or asymmetry of shaft

(b) Excessive or inadequate damping and stiffness

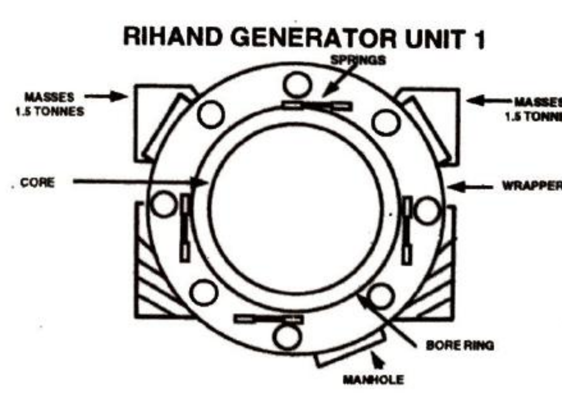
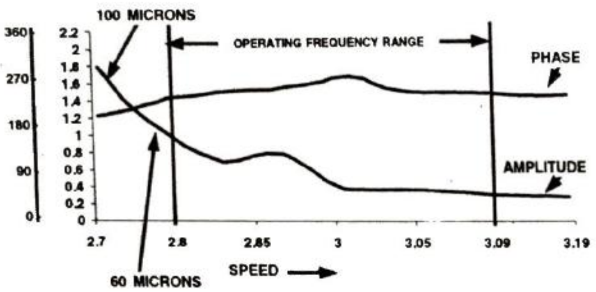
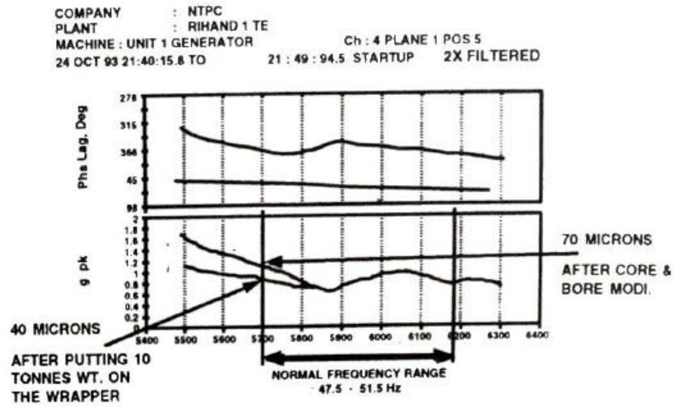


Fig.14

RIHAND UNIT 1 GENERATOR
PLOT OF 2F VIBRATION AMPLITUDE AND PHASE VS SPEED



HTRDN#4



HTRDN#5

(c) Drift in system natural frequency

(d) Change in critical speed

(e) Inherent rotor defect

(f) Deterioration of generator core end winding clamping tightness

(g) Unbalance

(h) Imperfect manufacturing and design deficiencies

5.2 Remedial measures

(a) Natural frequency can be increased or decreased by increasing or decreasing the stiffness or mass of the object

(b) Resonance could be avoided by eliminating the exciting source. Balancing to lower than normal levels will sometimes reduce effect of resonance

(c) Dynamic absorber is normally recommended to install for reducing the effect in the operating frequencies zone as a temporary measure

5.3 Applications and recommendation

(a) Hot run-down spectrum of bearing / shaft and its associated pedestal should be taken at every six month as per Operation Directive, because changes in spectrum is an indication of deterioration of machine performance.

(b) Hot run-down should be taken during PG test for acceptance of resonance-free machine or system (include foundation, bearings rotor, pedestal).

(c) The hot run-down of bearings and shaft should be taken to identify rotor design deficiency and same can be used for modifications.

(d) The hot run-down technique can be used to distinguish clearly whether machine is operating under resonance or having unbalance may be caused by blade breakage etc.

(e) In order to carry out such analysis, it is proposed to have a multi-channel off-line FFT analyser for taking run-down at different bearings in different directions for detailed analysis.

(f) On-line crack detection monitoring systems are available and can be installed to avoid costly breakdown and catastrophic failure.

6 Vibration limits for steam & gas turbine generators & exciters

In general, bearing vibration of steam & gas turbo generator sets is evaluated in terms of limits specified in ISO 3945 and ISO 2372, which are defined for transverse (horizontal/vertical) vibration. It is not normal practice to specify the limits of axial vibration because these can vary significantly depending on the geometry of the bearing supports, and also because for the machine supported by sleeve bearings there is no correlation between axial vibration of the pedestal and rotor. The only exception to this is for thrust bearings where axial forces can be transmitted into the bearing.

"The evaluation of axial bearing vibration depends on the bearing function and bearing construction.

In the case of thrust bearings, axial vibrations correlate with thrust pulsations which could cause damage to the metal liners of sliding bearings, or to part of antifriction bearings. The axial vibrations of these bearings should be judged in the same manner as transverse vibrations.

Where bearings have no axial restraints, a less exacting requirement may be permissible"

.ISO 3945. ISO 2372 Axial vibrations

It is clear therefore from the above that other than for thrust bearings, axial vibration measurements are not normally regarded as significant and are not used for machine evaluation except in case of resonance. There is no common relationship between transverse and axial vibrations but, in general, significantly higher axial vibrations upto 1.25X the allowable transverse vibration are regarded as acceptable for continuous operation considering machine is free from resonance and impending trouble.

Required actions during normal operation

The vibration levels given in **Table 1** are measured at the bearing housings on the shaft centerline of the turbinegenerator or exciter bearings when the machine is at normal operating speed and under steady state conditions. The levels expressed in units of rms velocity apply to whichever is the larger of measurements made in the vertical direction and measurements made in the horizontal direction at right angles to the shaft axis.

All turbo generator rotors to a greater or lesser degree are susceptible as regards vibration levels to changes on rotor excitation current and as such any vibration evaluation should refer to the given operating conditions. In certain cases, it will be necessary to optimize the vibration levels in order to achieve

Table 1

Quality ZONE	Operation Guidelines	Vibration Range RMS (VELOCITY)
A.	Target range for newly commissioned machines. Machines with vibration magnitudes within this Zone are regarded as good.	3.8 mm/sec 34 microns pk-pk at 50 Hz
B.	Machines whose vibration magnitudes are below the upper limit of this Zone are considered acceptable for unrestricted long term operation. Action to reduce the magnitude of vibrations occurring within this Zone is not normally justified from technical/ economic considerations.	3.8 - 7.5 mm/sec 34 - 68 microns pk-pk at 50 Hz
C.	Machines with vibration magnitude within this Zone are normally considered unsatisfactory for long term continuous operation. Generally, the machine can be operated for a limited period in this condition until a suitable opportunity arises for remedial action.	7.5 - 11.8 mm/sec 68 - 106 microns pk-pk at 50 Hz
D.	Sustained operation at vibration levels within this range is likely to cause damage. Investigations should be initiated and a decision made on the safety of continued operation. If the vibration levels are unstable, or if they are steadily trending upwards and cannot be contained by changes in operating conditions, e.g. load rotor current etc., then action should be taken to shut the machine down. When the levels of vibration approach the upper limit of this range it is common practice to initiate a trip.	11.8 - 18 mm/sec 106 - 162 microns pk-pk at 50 Hz

satisfactory limits over the range of operational conditions.

Generator vibration behavior can be influenced by electrical winding irregularities such as inter-turn shorts and earth faults. Methods are available for monitoring both and a generator condition monitor can give additional information for correlation with vibration behavior.

Actions required during run-up and run-down

The vibration levels for a machine in good condition will normally be within the upper limit of Zone C defined in **Table - 1**. When levels during run up/run down exceed this then consideration should be given as to the cause and the possible need to implement corrective action.

Changes in vibration levels

When there is either a significant step or progressive change in vibration levels (although still within acceptable limits) under steady state load conditions, then investigation should be initiated to establish the cause. A significant change for guidance purpose can be classified as when on increase or decrease in vibration magnitude exceeds 25% of the upper limits of zone B in **Table - 1.**

7 Advantages and disadvantages of hot-run-down signatures

The advantages of this method are that the frequencies generated by the equipment can be produced under normal operating conditions, and that resonance frequencies can be quickly differentiated from speed dependent frequencies. This method can be used during development of equipment to ensure that the system will not be operated at or near a resonance frequency which could result in a catastrophic failure. Its disadvantage is that it can be carried out only at cost of machine availability for generation, unless withdrawal from service happens to be necessary for other reasons.

8 Effect of resonance on the machine

(a) Incorrect operation
(b) Loss of product or quality
(c) Difficulty in balancing
(d) Accelerated fatigue and wear
(e) Machine failure
(f) Cracks in the structure and frequent bearings failure

9 Conclusions

Findings of various case studies leads the author to conclude that rundown of the system/machine should be taken atleast every six months, or before and after recommissioning to ensure that all the machines and associated components are free from resonance in operating frequencies zone.

References

a) A review of the vibration monitoring and diagnostic management techniques applied in a modern 2000 mw power station by **D.H. Brown** (CEGB Fawley Power Station)

b) Operation Services Study Reports of **NTPC.**

c) ISO and VDI standards

d) Didcot upgrades vibrational analysis by **Andrew Whittingham,** National Power, Didcot, Jeremy Kingston, Sensonics Ltd, Chesham, UK.

e) Predictive maintenance programme for rotating machinery using computerised vibration monitoring system by **S. ZIMMER** VDI, Debentle, USA.

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