**REF: NIMDC/RIHAND/19/03**
**DATE 24.08.2019**

**TO,**
**SHRI A. K. MUKHERJEE**
**HOD (PROJECT) /ED**
**RIHAND SUPER THERMAL POWER STATION,**
**P.O BIJPUR, RIHANDNAGAR, NTPC LIMITED.**
**DISTT -SONEBHADRA**
**UTTAR PRADESH- 231223**

**SUB:- STUDY REPORT ON HIGH SHAFT VIBRATION PROBLEM OF TURBINE GENERATOR OF RIHAND UNIT#4 (500 MW)**

Dear Sir,

Thanks for giving an opportunity to "NIMDC Private Limited" to carryout study & analysis of high shaft vibration problem of TG#4 at Rihand STPS.

TG Unit #4 was shut down for major overhauling on 16th June 2019, during which HP, IP and LP rotors were replaced with new rotors. Especially, HP rotor was replaced with refurbished rotor and coupling bolts of all the rotors have been replaced with new ones. Subsequently, unit was rolled to 3000 rpm on 28th July 2019 and since then; it had been running with high shaft vibrations of the order of maximum 205/253 microns pk-pk at HPT & LP-Generator bearings respectively. HPT shaft vibrations had a rising tendency with rise in load and MS temperature. In order to reduce shaft vibration, balancing was attempted many times using LP-Generator, HPT and even IPT bearings as planes, but it could not be successful.

NIMDC was contacted on 14th August 2019 and requested to visit immediately to study the problem in detail in running condition. Accordingly, executive from NIMDC visited Rihand STPS on 14th to 17th August 2019. During the stay at Rihand, a detailed study was conducted using Bentley Nevada System, installed in CCR for demonstration purpose and erection protocols alongwith operation parameters. The orbit analysis revealed that bearing No#3 is heavily preloaded due to misalignment and the same is confirmed by the orbit's shape to resemble character "8" and causing the shaft to go into reverse precession due to appearance of a 2X component in the response.

Preloaded is defined as unidirectional axial or radial (side) load due to external or internal mechanism. It can act to stabilize or destabilize the dynamic condition of the machine.

There are two types of preloads, soft and hard preloads. Soft preloads include the effects of gravity and thermal misalignment that can act on the rotor system. But, hard preloads are due to high eccentricity ratios resulting in increased direct stiffness in the bearing. Probable causes include gross shaft to shaft misalignment and piping forces acting on the machinery casing. In either case, preloads can be damaging if the forces involved are strong enough to create fatigue. The results of this can be realized by excessive bearing wear, shaft fatigue, etc.

Based on the study of operation parameters, protocols & analysis of steady state & transient data, it is recommended to inspect all the bearings of HP, IP, LP and Generators for any abnormality. The alignment of all couplings HP/IP/LP/GEN should be checked after removal of high points on the couplings faces. Catenary of the machine is to be maintained along with the alignment of all rotors. HP turbine has to be inspected/ replaced with the available module and carryout swing and roll checks etc.

A detailed report on preliminary study conducted at Rihand STPS is enclosed alongwith recommendations /action plan for your kind information please.

With kind Regards,

Yours Sincerely

24/08/2019

**(C.G.Porwal)**
**Chairman cum Managing Director**

**Copy for kind information pl**
**SHRI PRAKASH TIWARI**
**DIRECTOR (OPERATIONS),**
**NTPC BHAWAN, SCOPE COMPLEX,**
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**EXECUTIVE DIRECTOR (OS)**
**NTPC LIMITED**
**7th FLOOR**
**ENGINEERING OFFICE COMPLEX**
**NTPC EOC NOIDA 201301**

# STUDY REPORT ON HIGH SHAFT VIBRATION PROBEM OF TG#4 (500 MW) AT RIHAND STPS [6x500 MW]

**Prepared By NIMDC**

**"NITISH INDUSTRIAL MACHINERY DIAGNOSTICS CENTRE PRIVATE LIMITED"**
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**(14th to 17th August 2019)**

**RIHAND SUPER THERMAL POWER STATION**
**NTPC LIMITED, RIHAND, (UP)**

# SUB: STUDY REPORT ON HIGH SHAFT VIBRATION PROBLEM OF TURBINE GENERATOR UNIT#4 (500 MW)

## 1.0 INTRODUCTION

Rihand Super Thermal Power Station is one of the oldest power plants of NTPC, having an installed capacity of 3000 MW (6X500 MW) units, located at Rihand, Sonebhadra district, Uttar Pradesh. The Stage-I (2X500 MW) units was supplied by NEI & commissioned in March 1988 and July 1989. The Stage-II (2X500 MW units) was commissioned in January and September 2005 and Stage III (2X500 MW units) commissioned on 10th May & 31st August 2012 respectively. These units were designed by KWU and supplied by BHEL.

TG Unit #4 was shut down for major overhauling on 16th June 2019, during which HP, IP & LP rotors were replaced with new rotors and particularly HP rotor was replaced with refurbished rotor. The coupling bolts of all the couplings of HP-IP, IP-LP and LP -Generator have been replaced with new ones. Subsequently, Unit was rolled to 3000 rpm on 28th July 2019 and since then; it had been running with high shaft vibrations of the order of maximum 205/253 microns pk-pk at HPT & LP-Generator bearings respectively. In order to reduce shaft vibration, balancing was attempted many times using LP-Generator, HPT and even IPT bearings as planes, but it could not be successful.

NIMDC was contacted on 14th August 2019 and requested to visit immediately to study the problem in detail in running condition. Accordingly, executive from NIMDC visited Rihand STPS on 14th August 2019 to 17th August 2019. During the stay, a detailed study was conducted using Bentley Nevada System, installed in CCR for demonstration purpose and erection protocols alongwith operation parameters. The orbit analysis indicated that Brg No#3 is experiencing heavy preloading due to misalignment and causing the shaft to go into reverse precession.

## 2.0 VIBRATION LIMITS:

The alarm was set by KWU for absolute shaft vibration is 200 microns pk-pk. The trip value is set 300 microns pk-pk any of the two bearings out of 14 pickups, as mounted in the system. The bearing pedestal vibration should be less than 70 microns pk-pk for normal operation as per VDI 2056. In this unit all the pedestal vibrations were within the permissible limits.

## 3.0 THE BEARINGS & CATENARY

### A) BEARING & COUPLING

**Coupled & uncoupled run outs Of Coupling**: - Excessive coupled and uncoupled run out values can give rise to high vibration levels in the radial directions and may show symptoms similar to misalignment existing in the system. It is to be ensured that the coupled and uncoupled runs out values are not more than 0.03mm.

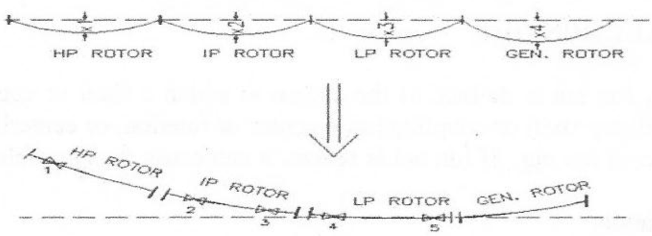
**Axial Face Run Out Of LP-Generator Half Coupling :-** High radial vibrations on the generator front bearing can also be attributed to excessive axial face run out of LP-Generator half coupling which is therefore to be kept below 0.02mm.

### B) CATENARY ADJUSTMENTS

The catenaries of the machine are very important for a turbine and Generator assembly to achieve proper alignment of various rotors and loading on their bearings. Any deviation may lead to various operational problems in the machine like high shaft vibration, high bearing vibration, high Babbitt metal temperature of the bearings etc. To avoid these problems it is necessary to maintain the catenaries of the machine during erection and subsequent realignment/overhaul of the unit. Many times it is observed that though the alignment of rotors are within limit but the catenaries as a whole get deviated from the prescribed design value of the machine.

- ✓ When rotors are resting in free condition on their bearings, there will be individual sags in them due to its dead weight and dimensions.
- ✓ Due to above the mating coupling faces of the rotors shall be at an angle. In this condition it is very difficult to couple the rotors.
- ✓ To align the coupling faces the rotors are given vertical adjustment through bearing or pedestal adjustment so that they become parallel and coupling becomes easier. One rotor normally LP rotor is kept as MASTER.
- ✓ From above adjustment the deflected lines of rotors also approaches the Center line which reduces the Centrifugal forces on the rotor.

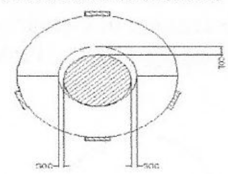
**TURBINE CATENARY**




## C) MEASUREMENT OF BEARING CLEARANCES

Bearing clearance is one of the most important parameters in the operation of a bearing. Therefore, it is important to determine the installed clearances along with the bore contour and concentricity to the outside fit diameter.

TOP OIL CLEARANCE (T.O.C.)
& SIDE OIL CLEARANCE (S.O.C.)



## D) ALIGNMENTS

Improper alignment settings give rise to high radial and axial vibrations on the bearings adjacent to coupling. Actually misalignment is the number one cause of turbine and rotating equipment failure. When shafts are not aligned properly, they cause excessive vibration, rotor bending, and cascading impacts that lead to premature bearing failure and reduced seal life. Because misalignment is a condition that compounds itself, if left unchecked, can cause total equipment failure and replacement.

The main objective in setting and maintaining rotor alignment is to achieve satisfactory dynamic behavior of the running shaft line.

**Parallel misalignment** occurs when the shaft centerlines are parallel but displaced from one another. Possible causes of misalignment are: -

- Thermal expansion due to a process working with heat (as with a turbine).
- Most machines are aligned cold, then as they operate and heat up, thermal growth causes them to grow misaligned.
- Machine directly coupled not properly aligned.
- Forces transmitted to the machine by piping and support members.
- Foundation uneven, shifting, or settling.

Phase measurements are a very useful tool for diagnosing misalignment.

**NOTE:** All phase values are ± 30° because of mechanical variance.

- Angular Misalignment – In the axial position, a phase shift of 180° will exist across the coupling or machine.
- Parallel Misalignment – In the radial direction, a phase shift of 180° will exist across the coupling or machine. A 0° or 180° phase shift will occur as the sensor is moved from the horizontal to the vertical position on the same bearing.
- Combination Angular and Parallel Misalignment – In the radial and axial positions, a phase shift of 180° will exist across the coupling or machine.

## E) IMPORTANCE OF SEALS & CLEARANCES

Sealing fins reduces the leakage's of steam through inter stage and turbine ends there by leading to high efficiency and better Heat Rate .They also result in balancing of the Thrust. The seal clearances are to be kept within limits which otherwise may result in high temperatures of the bearing pedestals especially the LP pedestal which may lead to warping. This may give rise to high vibration levels.

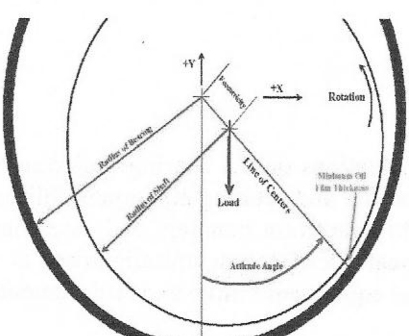
## F) COUPLING OR SHAFT RUN OUT

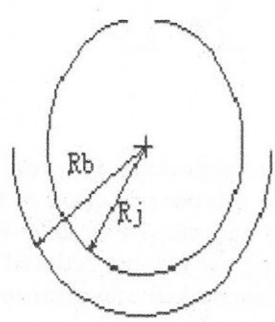
On rotating machinery, run out is defined as the degree to which a shaft or coupling deviates from true circular rotation. Every shaft or coupling has a center or rotation, or centerline. Any stray from concentricity is considered run out. If run out is severe, it can cause many problems with equipment, such as:-

- ✓ Excessive vibration.
- ✓ Seal wear.
- ✓ Bearing damage.

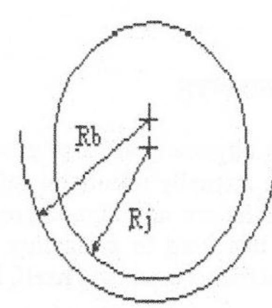
## G) PRELOAD

Besides clearance the next most important factor of a bearing is preload. A preloaded bearing simply means that the centre of curvature of the pad is not coincident with the geometric center of the bearing. Preload can be thought of as the percentage of clearance reduction. A preload is defined as a directional load or force applied to the rotating shaft. Preloads can be defined into two basic categories, external and internal.




Zero Preload




Positive Preload

**External Preloads**: Only one type of external preload exists. This is one whereby the directional force is applied to the rotating shaft through the coupling element. This is caused by angular or parallel misalignment, plus a stuck or partially stuck coupling. The magnitude of this preload is a function of the amount of misalignment as well as the type and condition of the coupling, the highest existing for a rigid shaft, and the lowest with diaphragm or other spring type couplings.

**Internal Preloads**: Internal preloads are those forces generated internal to the rotating machine. Common examples of internal preloads are:-

- Elliptical or lemon type bearings where the clearance is reduced in one plane to provide a preloading action on the shaft,
- A cocked bearing due to thermal distortion, or installation misalignment.
- Internal misalignment on machines will also produce a preload.
- Distortion of a machine casing may cause the internal seals to act as partial bearings preloading the shaft.

## H) SWING CHECK

The swing check is the measurement of radial throw caused due to coupling face geometric form of the two rotors coupled together. This is measured on opposite end of the coupling and at the free end of the rotor. ***The higher swing check value may cause higher shaft vibration, higher bearing shell temperature etc***. The value of swing check depends on axial run out of coupling faces, the diameter of the coupling and the length of the rotor. During the machining of rotors in the works some tolerances are permitted by Designer's on the coupling faces of the rotor resulting to some swing check values.

The maximum swing check values permitted caused due to the above tolerance for different diameter and length of the rotor can be worked out. However it is recommended to keep the minimum swing check values for better results during operation of the machine. In a multi rotor Turbine-Generator system it is essential to measure the swing check value in both extreme end and where the weight of the rotors are light.

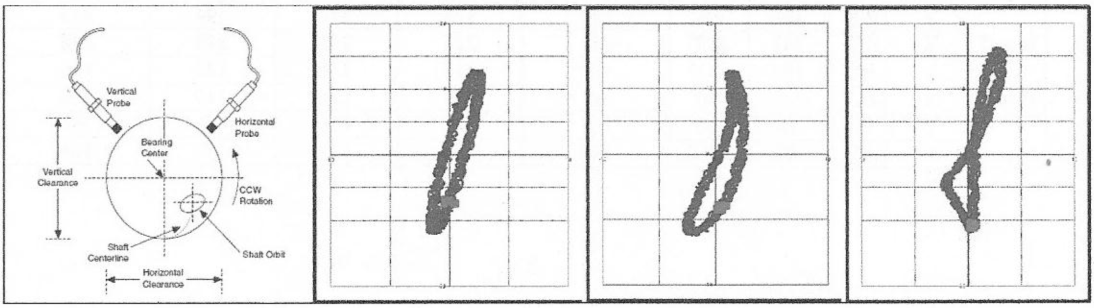
## I) ROLL CHECK

In KWU machine the minimum radial clearances HP, IP and LP casings are measured by actual by moving the casing radial direction at site while rotor is rotated by hand. This is a very accurate and fast method of measuring the minimum radial clearance of any of the casing. The major variation in these reading may cause vibration in the machine and obstruction in barring gear operation etc. After completing the alignment of rotors the casing alignments are carried out by roll check method but equal importance should also be given to the centering of HP and IP casing. The centering of casing may not be fully sacrificed in comparison of rolling test readings and a compromise between these two readings should be made.

## J) ORBITS

An orbit measurement is taken with a pair of proximity probes (non-contact probes), mounted radially, 90° apart (or as near as is practical). Typically, the probes are mounted at ±45° as shown below. The orbit display shows a single trace which illustrates how the shaft moves within the bearing as it rotates, in the style of a Lissajous figure. Both filtered and unfiltered orbits can be viewed.

**Misalignment**

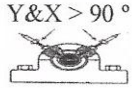
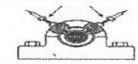



----Increasing Severity of Misalignment--------->

**4.0 OBSERVATIONS** Our observations recorded during the study are as follows:-

i) The proximity pickups mounted on Brg No 1, 3, 4 & Bearing No 5 are having phase difference of around 180° degrees between Y & X, instead of 90° (Table –I) .

**TABLE-I**
**PHASE READING BETWEEN Y/X PICKUPS**

| DATE | RECORDED ON 30th July 19 AS INITIAL REFERENCE | | | | | 13th August 2019 BENTLY DEMO | |
|---|---|---|---|---|---|---|---|
| LOCATION | POSITION | Overall / | 1x / | Phase | PHASE BETWEEN Y&X > 90 °  | Overall/ 1x/ phase **3001** | PHASE BETWEEN Y&X > 90 °  |
| BRG No 1 | Y | 159 | 138 | 97° | 196° | 184/ 158 /113° | **205°** |
| | X | 202 | 189 | 293° | | 304/ 269/ 318° | |
| BRG No 2 | Y | 88 | 76 | 221° | 129° | 142/ 104 /203° | **110°** |
| | X | 184 | 167 | 92° | | 274 /224 /113° | |
| BRG No 3 | Y | 52 | 48 | 110° | 122° | 69/ 43 /88° | **186°** |
| | X | 88 | 80 | 232° | | 117 /107 /274° | |
| BRG No 4 | Y | 131 | 114 | 107° | 226° | 87/ 81/ 126° | **207°** |
| | X | 135 | 140 | 333° | | 86 /73 /333° | |
| BRG No 5 | Y | 273 | 260 | 308° | 153° | 237/ 222/ 318° | **166°** |
| | X | 242 | 243 | 155° | | 195/ 187/ 152° | |
| BRG No 6 | Y | 103 | 100 | 291° | 161° | 108/ 102/ 305° | **164°** |
| | X | 113 | 96 | 130° | | 122/ 102 /141° | |
| BRG No 7 | Y | 90 | 78 | 108° | 222° | 102/ 87/ 123° | **214°** |
| | X | 85 | 80 | 330° | | 103/ 100 /337° | |

ii) The machine was rolled to 3000 rpm on 28th July 2019 and many balancing attempts were made considering Generator bearing, LP rear bearing, HPT Bearings, IPT bearing as planes, but it could not be successful to reduce the shaft vibrations at HPT and even at Generator bearings.

**TABLE-II**
**FINAL VIBRATION READINGS RECORDED AFTER THE BLANCING**

| Date | POSITIN | REFERENCE 30.07.2019 | | | GENERATOR FRONT PLANE 03/08/2019 | LP FRONT & REAR 05/08/2019 | IPT FRONT & REAR 08/08/2019 | HP FRONT & REAR BRG 10.08.2019 | 10.17 AM 400 MW 17/08/2019 |
|---|---|---|---|---|---|---|---|---|---|
| Location | | Overall | 1X | PHASE | | | | | 403 MW |
| Brg 1 | Y | 159 | 138 | 97° | 102 | 184 | 274 | 206 | 338 |
| | X | 202 | 189 | 293° | 84 | 144 | 201 | 154 | 239 |
| Brg 2 | Y | 88 | 76 | 221° | 134 | 198 | 265 | 221 | 285 |
| | X | 184 | 167 | 92° | 83 | 111 | 118 | 127 | 135 |
| Brg 3 | Y | 52 | 48 | 110° | 142 | 137 | 121 | 105 | 135 |
| | X | 88 | 80 | 232° | 62 | 36 | 69 | 57 | 81 |
| Brg 4 | Y | 131 | 114 | 107° | 179 | 103 | 82 | 83 | 81 |
| | X | 135 | 140 | 333° | 139 | 97 | 96 | 63 | 107 |
| Brg 5 | Y | 273 | 260 | 308° | 203 | 198 | 212 | 194 | 213 |
| | X | 242 | 243 | 155° | 242 | 251 | 242 | 213 | 248 |
| Brg 6 | Y | 103 | 100 | 291° | 107 | 118 | 138 | 122 | 136 |
| | X | 113 | 96 | 130° | 65 | 93 | 126 | 119 | 128 |
| Brg 7 | Y | 90 | 78 | 108° | 141 | 89 | 116 | 97 | 96 |
| | X | 85 | 80 | 330° | 149 | 98 | 122 | 95 | 104 |

iii) BRG No 1, 2, 3 & 4 had SOC clearances less than the normal values (Table-III). The presence of an oil wedge is indispensable to the normal operation of a hydrodynamic bearing.

**TABLE –III**

**BEARING CLEARANCES DRAWING AND ACTUAL CLEARANCES**

| BRG No | SHAFT Diameter "d" | TOC DRAWING "b" | TOC ACTUAL | SOP DRAWING "c" | SOP ACTUAL "C" FRONT L | FRONT R | REAR L | REAR R | Remarks |
|---|---|---|---|---|---|---|---|---|---|
| B #1 | 250 | 0.30 TO 0.36 | 0.35 | 0.37 TO 0.40 | **0.30** | **0.30** | 0.40 | **0.30** | **Very low** |
| B #2 | 380 | 0.46 TO 0.53 | 0.49/0.50 | 0.57 TO 0.60 | **0.40** | **0.45** | **0.40** | **0.50** | **Very low** |
| B #3 | 450 | 0.51 TO 0.62 | 0.52/0.53 | 0.36 TO 0.40 | **0.35** | **0.30** | **0.35** | **0.30** | **Very low** |
| B #4 | 500 | 0.60 TO 0.70 | 0.57 | 0.40 TO 0.44 | **0.35** | **0.35** | **0.35** | **0.35** | **Very low** |

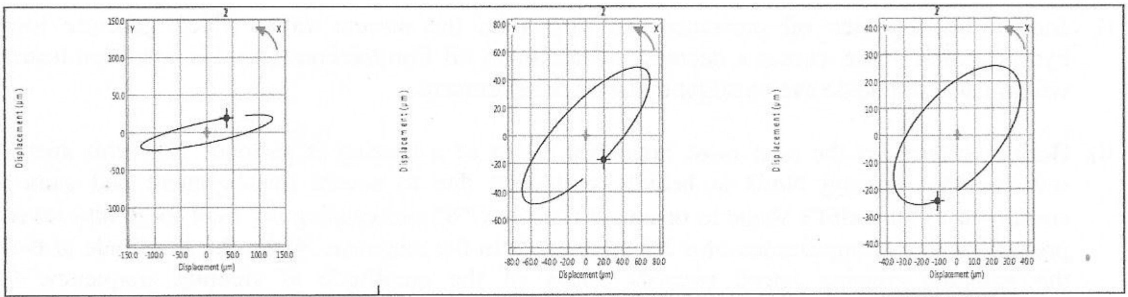
iv) The swing check value recorded after final assembly was very high in the 1st set of readings more than 150 microns whereas in the 2nd set readings around 90 microns. If we take average of these two readings, it will come around 120 microns, indicates built in eccentricity in the system, therefore HPT front shaft vibration are showing high in the beginning and maintaining high at various loads. This needs to be looked into seriously.

**TABLE –IV**

**SWING CHECK PROTOCOLS**

| Swing check | | | | |
|---|---|---|---|---|
| HP-IP COUPLING | 1st | | 2nd set | |
| HOLE NO | Left | right | Left | Right |
| 01 | 50 | 50 | 55 | 45 |
| 16 | 50 | 50 | 53 | 47 |
| 15 | 49 | 52 | 51 | 49 |
| 14 | 48 | 52 | 53 | 48 |
| 13 | 50 | 52 | 53 | 48 |
| 12 | 51 | 50 | 55 | 45 |
| 11 | 52 | 49 | 57 | 43 |
| 10 | 55 | 46 | 58 | 42 |
| 09 | 58 | 42 | 59 | 40 |
| 08 | 61 | 39 | 59 | 41 |
| 07 | 62 | 38 | 60 | 40 |
| 06 | 62 | 37. | 59 | 41 |
| 05 | 62 | 37 | 58 | 42 |
| 04 | 61 | 39 | 58 | 43 |
| 03 | 60 | 40 | 56 | 44 |
| 02 | 58 | 42 | 56 | 44 |
| 01 | 55 | 45 | 55 | 45 |
| Swing | 0.14 | 0.15 | 0.09 | 0.09 |

v) Orbit analysis of HPT bearings indicates that vibrations are in forward precession whereas on IPT, it is under reverse precession which may be due to misalignment.



HPT FRONT BRG#1 HPT REAR BRG#2 IPT BRG#3

vi) The Orbit plot taken during the load at 410 MW on 15th August 2019 indicated that Brg No 3 is highly preloaded, confirmed by orbit shape to resemble character "8" and causing the shaft to go into reverse precession due to appearance of a 2X component in the response.

Figure No 1 indicating higher amount of preload on the rotor which causes the orbit to become more elliptical and distorted. This is where the spring constant is much higher in the opposite direction to the pre-load than in the perpendicular direction to the preload. This produces the classical twice per rev frequency associated with alignment.

BRG No #3 Direct Orbit (Banana shape)

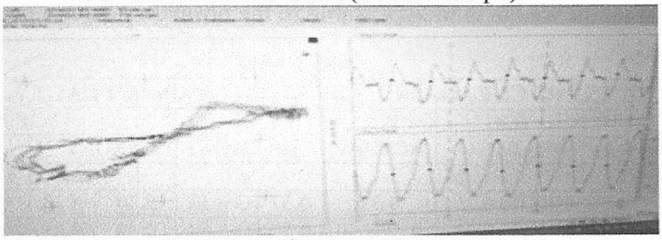


Figure No 1

vii) A bearing preload due to cocked assembly can also cause orbit to have lower amplitude in one axis that makes the ellipse look thinner as shown in Figure 2(a). But heavy preloading due to misalignment can cause the shaft to go into sometime reverse precession; can be seen in Figure No 2(b). The forward precession is normal; reverse is not can be seen in full spectrum taken during load on $15^{th}$ August 2019.

Brg No#3 1x Orbit

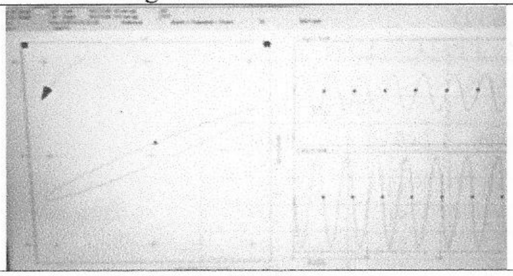


Figure No 2 (a)

Reverse Precession

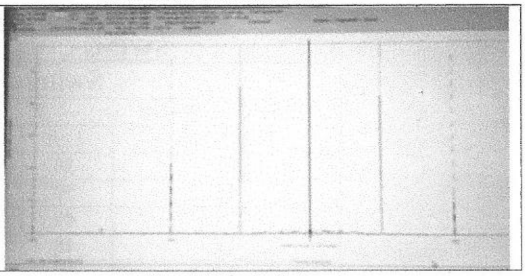


Figure No 2 (b)

viii) An Orbit analysis of HPT bearings indicating truncation in the orbit which is an effect of misalignment.

HPT shaft vibrations had a tendency to rise with rise in MS temperature & loads due to thermal disturbances/distortions recorded on $16^{th}$ August 2019 at 410 MW.

| B1 | B2 | B3 | B4 | B5 | B6 | B7 |
|---|---|---|---|---|---|---|
| 340/238 | 293/142 | 138/80 | 81/106 | 212/247 | 135/128 | 104/111 |

## 5.0 VIBRATION & OPERATION PARAMETERS ANALYSIS

Based on recorded online data, installed in control room and physical observations of protocols, it is revealed that high vibration is due to system misalignment:-

i) Journal bearing side oil clearances are less than the normal value which generate lower hydrodynamic force, causes a decrease of minimum oil film thickness. In this condition bearing will not accommodate even a slight dynamic misalignment.

ii) Besides clearances the next most important factor of a bearing is preload. The orbit analysis revealed that bearing No#3 is heavily preloaded due to severe misalignment and same is confirmed by the orbit's shape to resemble character "8" and causing the shaft to go into reverse precession due to appearance of a 2X component in the response. When the amplitude at twice the machine running speed exceeds 150% of the amplitude at running frequency, the misalignment is producing a severe action at coupling (rotor bend/asymmetry).

Similarly, a bearing preload due to cocked assembly sometimes can also cause the orbit to have lower amplitude in one axis that makes the ellipse look thinner.

iii) All are indicators of excessive rotor loading that may be the result of misalignment coupled with excessive pipe strain and/or severe bearing problems. A complete machinery train experiencing such problems should be considered a candidate for immediate physical inspection.

iv) This gross misalignment may sometime shift the natural frequencies of the rotor so much that a critical speed range may approach the normal turbine speed. Also, piping forces or foundation distortion, etc. can change the turbine casing support conditions to an extent that the casing's natural frequencies are altered. As a result, the casing may operate in a resonance. Its vibration is then transmitted to bearing pedestals, increasing bearing vibrations and thus rotor vibration. Casing is also an important factor in rotor vibration. There are many ways that casing temperature fluctuations can cause steam turbine vibrations. Casing problems can cause misalignment in many different ways, mostly related to expansion and contraction due to temperature fluctuations.

## 6.0 RECOMMENDATIONS- ACTION PLAN

i) Inspect all the Bearings of HP, IP, LP, Gen front for any abnormality. Check all the bearings for self - alignment & its seating and maintain proper clearances.

ii) HP turbine has to be inspected/ replaced with the available module. Check swing value, which is an indication of high eccentricity, reduce to as minimum as possible.

iii) Check HP & IP coupling alignment

iv) As IP rotor is highly preloaded/misalignment, check IP rotor internals for any partial rubbing or contact,

v) The coupling alignment of all rotors should be checked after removal of high points on the couplings faces.

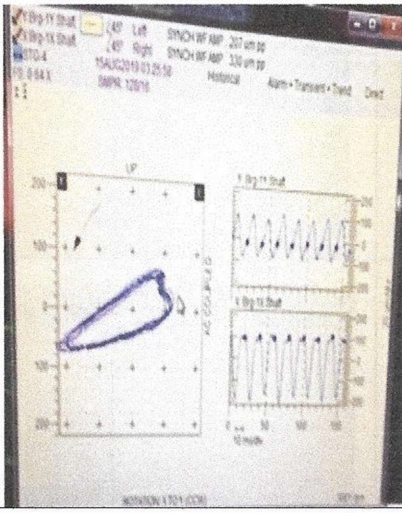
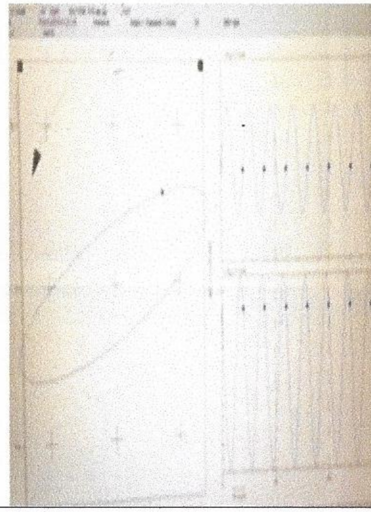
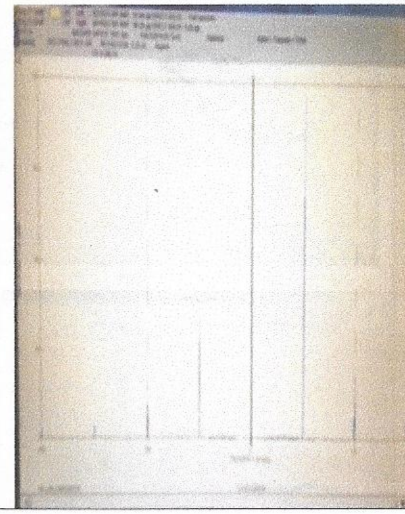
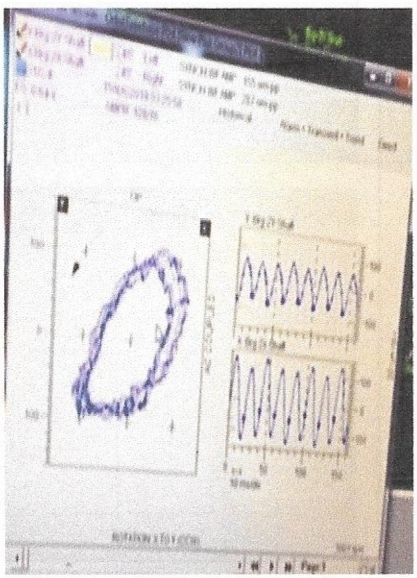
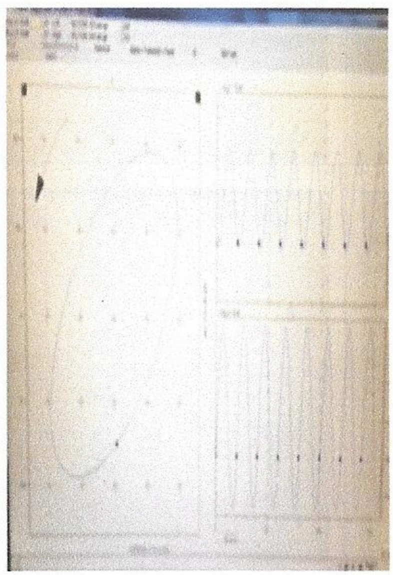
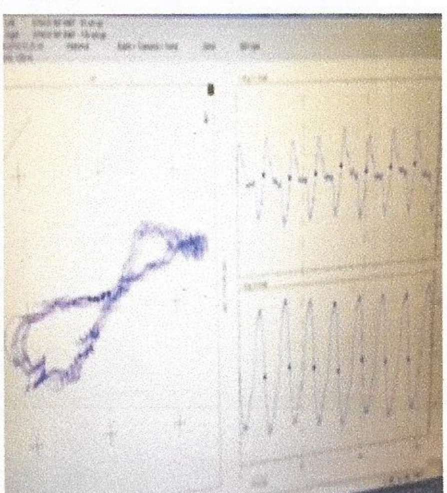
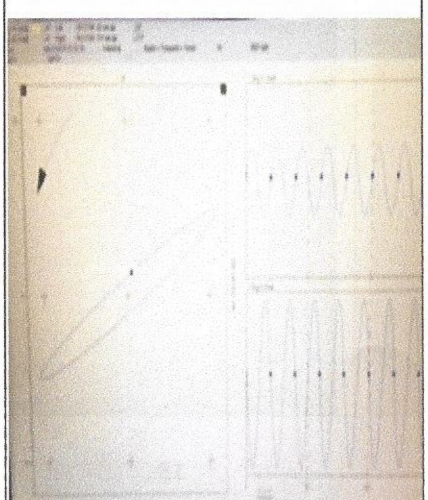
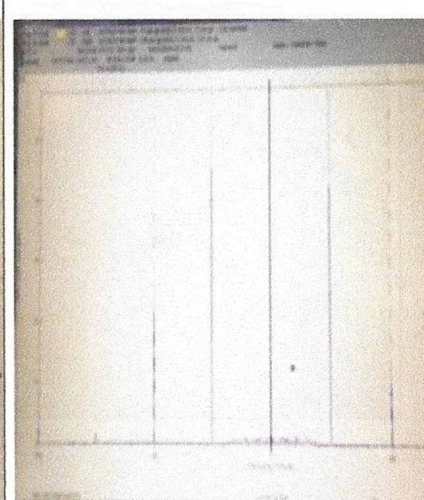
vi) Carryout roll checks of HP & IP for centering.

vii) Check the catenary as per protocols.

viii) LP-Gen Coupling faces have to be blue matched.

24/08/2019.

( C G PORWAL )

| Overall vibration orbit | Filtered orbit | Full Spectrum |
|---|---|---|
|  |  |  |
|  |  |  |
|  |  |  |