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# Diagnosis of Intermittent High Vibration Peaks in Industrial Gas Turbine Using Advanced Vibrations Analysis

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***Abstract*—** This paper provides a comprehensive study pertaining to diagnosis of intermittent high vibrations on an industrial gas turbine using detailed vibrations analysis, followed by its rectification. Engro Polymer & Chemicals Limited, a Chlor-Vinyl complex located in Pakistan has a captive combined cycle power plant having two 28 MW gas turbines (make Hitachi) & one 15 MW steam turbine. In 2018, the organization faced an issue of high vibrations on one of the gas turbines. These high vibration peaks appeared intermittently on both compressor's drive end (DE) & turbine's non-drive end (NDE) bearing. The amplitude of high vibration peaks was between 150-170% on the DE bearing & 200-300% on the NDE bearing from baseline values. In one of these episodes, the gas turbine got tripped on "High Vibrations Trip" logic actuated at 155µm. Limited instrumentation is available on the machine, which is monitored with GE Bently Nevada 3300 system having two proximity probes installed at Turbine NDE, Compressor DE &at Generator DE & NDE bearings. Machine's transient ramp-up & steady state data was collected using ADRE SXP & DSPI 408. Since only 01 key phasor is installed at Turbine high speed shaft, a derived drive key phasor was configured in ADRE to obtain low speed shaft rpm required for data analysis. By analyzing the Bode plots, Shaft center line plot, Polar plot & orbit plots; rubbing was evident on Turbine's NDE along with increased bearing clearance of Turbine's NDE radial bearing. The subject bearing was then inspected & heavy deposition of carbonized coke was found on the labyrinth seals of bearing housing with clear rubbing marks on shaft & housing covering at 20-25 degrees on the inner radius of labyrinth seals. The collected coke sample was tested in laboratory & found to be the residue of lube oil in the bearing housing. After detailed inspection & cleaning of shaft journal area & bearing housing, new radial bearing was installed. Before assembling the bearing housing, cleaning of bearing cooling & sealing air lines was also carried out as inadequate flow of cooling & sealing air can accelerate coke formation in bearing housing. The machine was then taken back online & data was collected again using ADRE SXP & DSPI 408 for health analysis. The vibrations were found in acceptable zone as per ISO standard 7919-3 while all other parameters were also within vendor defined range. As a learning from subject case, revised operating & maintenance regime has also been proposed to enhance machine's reliability.

***Keywords*—**ADRE, bearing, gas turbine, GE Bently Nevada, Hitachi, vibration.

## I. Introduction

ENGRO Polymer & Chemicals Limited, a Chlor-Vinyl complex & the only PVC producing plant in Pakistan has two 28 MW gas turbines (make Hitachi) to meet in house power requirements. Hitachi's H-25 is a variant of GE Frame V with similar power generation capacity. The generator is installed on the suction inlet air side & is coupled with compressor shaft with gearbox in between. The gas turbine has a 17 stage axial compressor and a 03 stage turbine with can-annular type combustion chamber. The turbine is a single spool with in-line Heat Recovery Steam Generator (HRSG) on the exhaust side. The turbine is supported radially by two bearings installed on each end i.e. Bearing#01(radial bearing) is installed on turbine's non-drive end while the Bearing#02(radial + thrust bearing) installed on compressor's drive end. Both the bearings are fluid film tilting pad bearings. Both bearings are equipped with two radial proximity probes 90 degrees apart & a key phasor is installed at Bearing#01 side. There's no sensor for bearing metal temperatures but temperature transducers are installed on drain oil line of each bearing. The configuration of turbine is shown in Fig. 1.

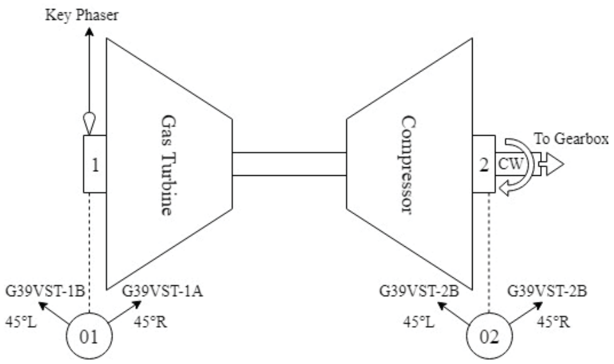


Fig. 1 Bearings configuration & instrumentations installed

### A. Intermittent Vibrations

One of the Hitachi's H-25 gas turbine was tripped on 27th October, 2018 at 0957hrs due to Turbine Bearing#01 high vibration on both X & Y probes (G-39VST1-1 & G39VST1-2). At that time to identify the root cause of this tripping, complete data of last 03days were extracted from Mark-VI & all the relevant parameters like vibrations at other bearings, lube oil supply temperatures, lube oil supply pressures, lube oil bearing drain temperatures, lube oil sump pressure, machine load,

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ambient conditions etc. were trended to find any irregularity or deviation from the normal baseline data (as shown in Fig 2). The only observation was the gradual increase in Bearing#02 vibrations to 76µm pp at the time when Bearing#01 cross the trip limit of 155µm pp. No other anomaly was found in any of the other parameter during the analysis. After necessary checks by machinery, instruments & electrical teams, GT was successfully started again on 30th October, 2018 at 1350 Hrs.

After running smoothly for around next 14 hours, high vibration peaks were started to appear at regular intervals & every event was having a peak value greater than the last one. Following was the sequence of events:

1) First event at 0415 Hrs on 31st October, 2018, maximum vibration amplitude was 77µm pp at Bearing#01 having a deviation of 208% & maximum vibration amplitude was 58µm pp at Bearing#02 having a deviation of 28%
2) Second event at 1100 Hrs on 31st October, 2018, maximum vibration amplitude was 95µm pp at Bearing#01 having a deviation of 280% & maximum vibration amplitude was 66µm pp at Bearing#02 having a deviation of 46%
3) Third event at 1500 Hrs on 31st October, 2018, maximum vibration amplitude was 127µm pp at Bearing#01 having a deviation of 400% & maximum vibration amplitude was 70µm pp at Bearing#02 having a deviation of 55%
4) Fourth event at 0615 Hrs on 1st November, 2018, maximum vibration amplitude was 157µm pp (tripping occurred) at Bearing#01 having a deviation of 528% & maximum vibration amplitude was 77µm pp at Bearing#02 having a deviation of 72%

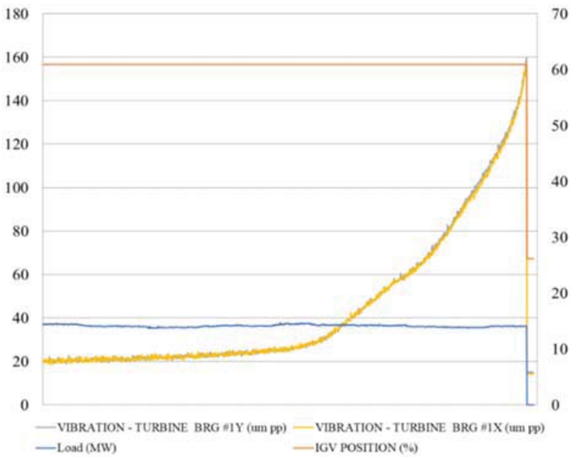


Fig. 2 (a) Bearing#01 vibrations trend against Load & IGV

## II. Advanced Vibrations Analysis

After having multiple high vibration events it was decided to go for advance vibrational analysis [1]. For this, Machine data was collected using ADRE SXP & DSPI 408 on 1st November, 2018. Machine is monitored with Bently Nevada 3300 system having two proximity probes installed at Turbine NDE, Compressor DE & at Generator DE & NDE bearings. However, only 01 key phasor was installed at Turbine high speed shaft due to which a derived drive key phasor was configured in ADRE [2] to obtain low speed shaft rpm required for important data analysis. Details of complete terrain used for route creation are shown in Fig. 3.

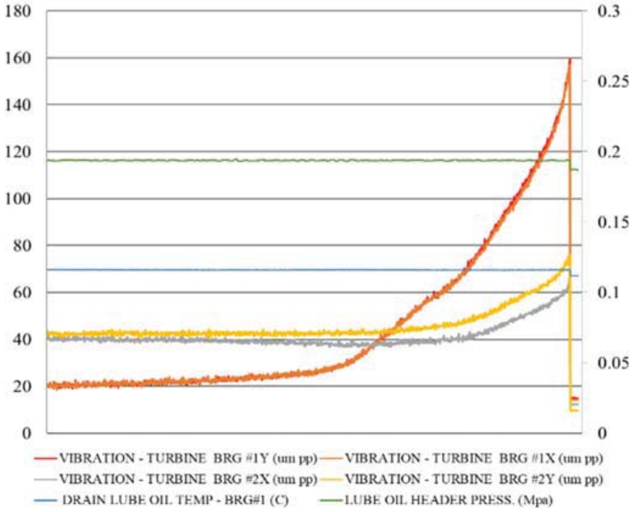


Fig. 2 (b) Turbine Bearings vibration trend against Lube oil drain temperature & header pressure

### A. Diagnosis

Vibration peaks was observed during steady state data recorded on 1st November 2018. Vibration peak was recorded at 1435hrs up to 40.09 & 35.54µm pp at Turbine Bearing#01 against normal vibrations of 22 & 24µm pp & at Turbine Bearing#02 up to 64.09 & 58.98µm pp against its normal trend of 43 & 45µm pp. As per vibration trend data, vibration peaks were observed to appear at different times & the peak lasted for duration of approximately 15 minutes (as shown in Fig. 3).

Basic rotor dynamic theory [3] states that when rotating at constant speed, rotor synchronous response remains constant, both in amplitude & phase. The occurrence of phase changes at constant speed could be caused by several different things, including rubs between stationary & rotating parts, angular shift of shrink fit components, a crack propagating through the rotor. In this case almost all vibration change was observed at frequency of 1X Turbine rpm while significant change in 1X phase (up to full 360° rotation) was also observed at Bearing#01 which most probably corresponds to symptoms of rubbing [4] (as shown in Fig 4). Also, as per shaft centerline plot, movement of rotor in horizontal plane was evident at Bearing#01 (as shown in Fig. 5)

Spectrums plots of Turbine Bearing#01 & 02 also indicated dominant vibration at 1xTurbine rpm frequency during vibration peak [5]. However, unbalance was not suspected due to increasing / decreasing vibration which did not correspond to normal unbalance symptoms. Also, based on periodic nature of vibration, misalignment between turbine & gearbox was not suspected as the cause of vibration.

During vibration increase, turbine operating parameters such as inlet steam pressure, exhaust pressure, bearing temperatures & load had remained constant Therefore the vibration change was not related to flow / load changes.

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*B. Conclusion*

Based on the data gathered from ADRE SXP, there was a vibration increase with high 1X rpm response & 1X phase change [6], [7] is most probably suspected due to intermittent rubbing either in Turbine NDE bearing or bearing seals. Rubbing in the bearing oil (labyrinth) seal may be caused by build-up of deposits resulting in localized reduction in seal clearance. Carbon deposits may be formed if lubricating oil comes in contact with high temperature causing oil to overheat. This may cause intermittent contact of rotor with deposits resulting in high vibration which tends to become normal upon clearance of deposits; however, vibration cycle may be repeated upon formation of new deposits.

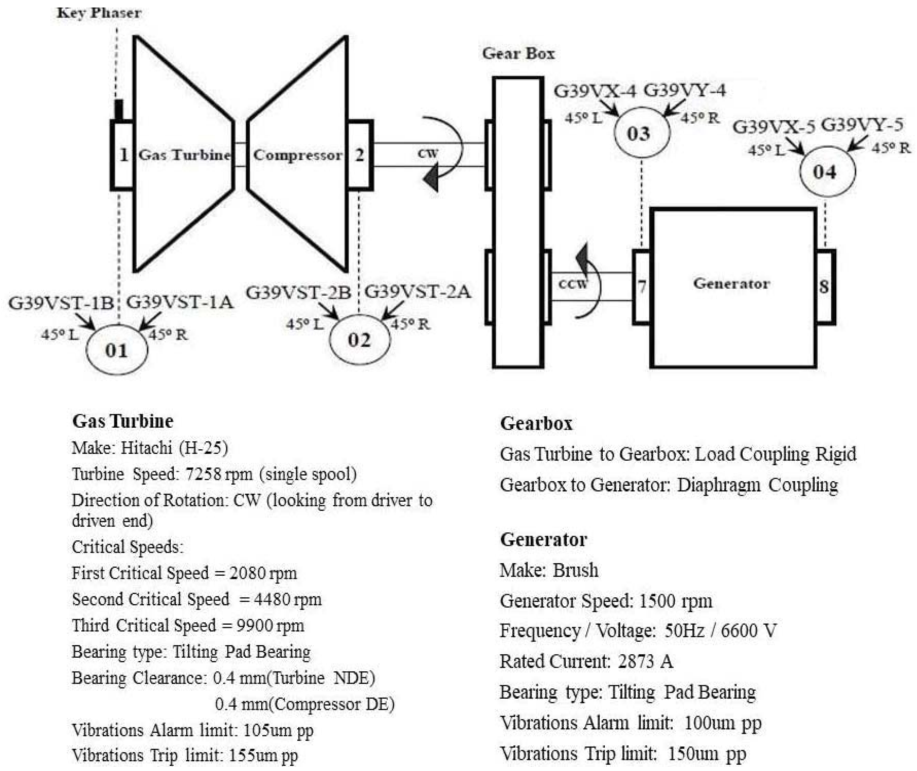


Fig. 3 Details used for route creation in ADRE DSP 408

## III. Bearings Inspection

As per diagnosis using advanced vibrational analysis, rubbing symptoms at Bearing#01 were evident [7] although its severity & actual rotor health could not be exactly predicted. Due to this a confident decision could not be made to continue operations with turbine as this could have led to a major failure & would result in larger downtime & hence both production loss & costlier maintenance. So based on recommendations of advanced vibrational analysis, it was decided to inspect & replace the Journal Bearing#01 located at turbine's non-drive end. In parallel, inspection of Bearing#02 (thrust & journal) was also planned installed at compressor's drive end. Historically Bearing#01 usually operates at lower vibrations as compared to Bearing#02 & had earlier no issues so this was the first time this bearing was going to be inspected & replaced since the commissioning of turbine back in 2009. The Bearing#02 had already been replaced in 2016. Access to Bearing#01 was more difficult than the Bearing#02 due to the opening of annular diffuser casing bolts from inside of Bearing#01 tunnel. After getting access, Bearing#01 pedestal's upper half was removed & initial as-found observations were taken before removing the bearing as shown in Fig 6.

At Bearing#01, two labyrinth seal sets installed in the bearing housing each on forward & after side. In the forward side labyrinth seals clear rubbing marks were present, covering around 25 degrees in the inner radius of bearing housing. The bearing pedestal end & labyrinth seal close to turbine side had excessive coke deposition. A layer of coke varying in thickness from 5 mm on outer radius to 2 mm on inner radius was formed on the forward side bearing pedestal face. The cavities of the first set of labyrinths were completely choked because of heavy coke deposition. This coke was brownish black & was hard but not tough. There was a small area from that layer where coke was removed showing signs of an impact load (as shown in Fig. 7). Coke deposition was also observed on the after-side labyrinth seals, but the severity of deposition was comparatively less. However, the mechanical integrity & geometric profile of labyrinth seals were intact & they were installed back after thorough cleaning followed by inspection.

Two out of three shoes in the upper half of the bearing had symptoms of damage on the Babbitt layer. One shoe had a crack on the Babbitt while the other had pitting across its ends (as shown in Fig. 8). The two shoes in lower half of the bearing had signs of rubbing, oil patches & Babbitt discoloration. The observations on the Bearing#01 pads were quite usual for a bearing with equivalent operating hours of more than eighty-two thousand hours. The shaft journal area had signs of minor rubbing. Old bearing was then removed after which thorough cleaning & inspection of shaft journal area were performed as per OEM guidelines.

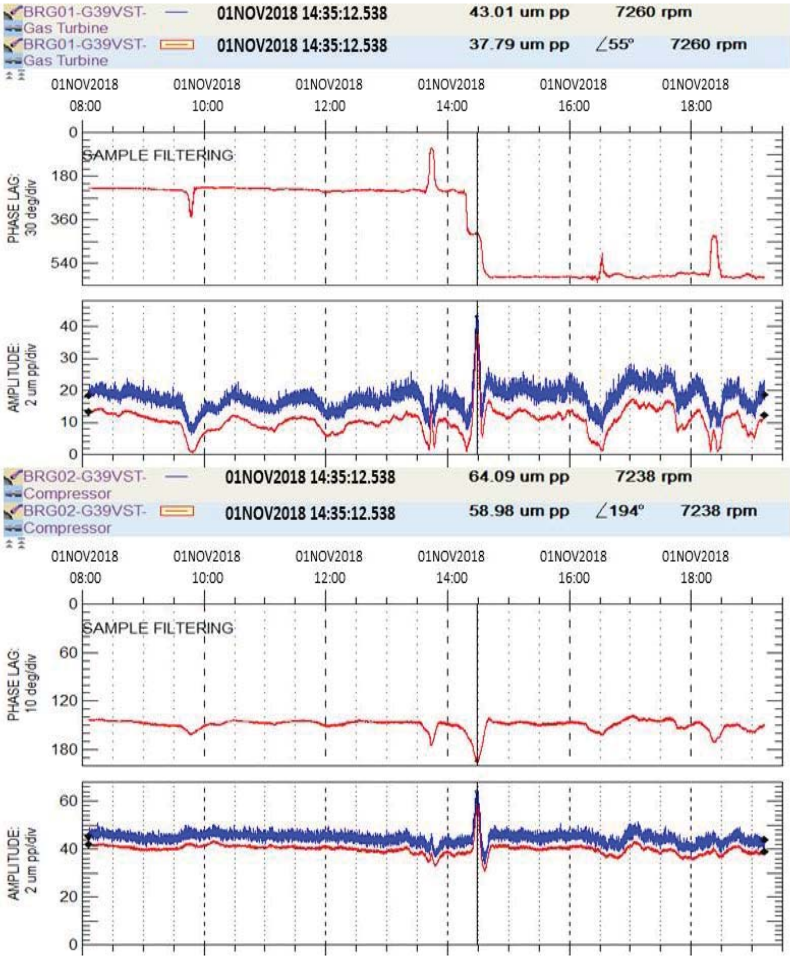


Fig. 4 (a) Time base trend of Bearing#01 & 02 with Phase angle

After thorough cleaning & following the all quality hold points, new bearing was installed at location. Quality check measurements like semi height, bearing crush & radial clearance were also taken. The radial clearance of bearing was maintained at 0.45mm (shaft journal OD: 184mm). Bearing#02 was also inspected & did not have excessive coke formation like Bearing#01. The radial Bearing#02 at the driving end had minor signs of rubbing & oil marks. Its condition was comparatively better but replaced with new one with the top clearance of 0.40 mm. The thrust bearing was healthy & installed back after quality checks & inspection. Before final box-up, cooling & sealing air lines were thoroughly blown with instrument air to remove any suspected choking & installed back.

### *A. Post Maintenance Analysis*

After that machine was handed over to operations & taken into service. Machine coast up was started at time 22:26:42 on 09th November, 2018 as recorded in ADRE & machine reached operating speed of 7272 rpm at time 22:42:13, in about 16 minutes. As per trend plots, as machine reached operating speed (7272 rpm) at 22:42:12 vibrations were initially found reduced to 10/16 & 48/41µm pp at turbine NDE & DE bearing respectively. However, as machine was synchronized & taken to 3.5MW load, vibrations were found increased to 35/50 & 42/54µm pp at on turbine NDE & DE bearing respectively at time 23:34:37. Vibrations were found more or less stable with further increase in load. Steady state vibrations recorded on Gas Turbine at 21 MW operating load were found 35/38µm pp & 38/54um pp at NDE & DE bearing respectively. Steady state vibrations recorded on Generator at 21 MW load were found 10/14um pp & 17/19um pp at DE & NDE bearing respectively. The vibrations were found in acceptable zone as per ISO standard 7919-3/10816-7 [8], [9] (as shown in Table I). As per spectrum plots, the dominant vibration was observed at frequency of 1X of turbine rpm. As vibration response was not directly related to increasing machine speed, therefore unbalance was not suspected as the cause of vibration & the 1X vibration most probably was related to shaft bow condition during the machine start up. Orbit plots were found indicating 1X orbits & remained highly flat/elliptical throughout speed range from 2000 rpm to 5000 rpm (as shown in Fig. 9) suggesting possible rub during coast up. At operating speed i.e. 7272 rpm no such observation was found in 1x orbit plots (as shown in Fig. 10). As per Shaft center line plots, maximum shaft vertical upward movement of 134 microns was observed at Turbine NE Bearing#01 when machine was loaded from FSNL to 21 MW, which was within bearing clearance boundaries (as shown in Fig 11).

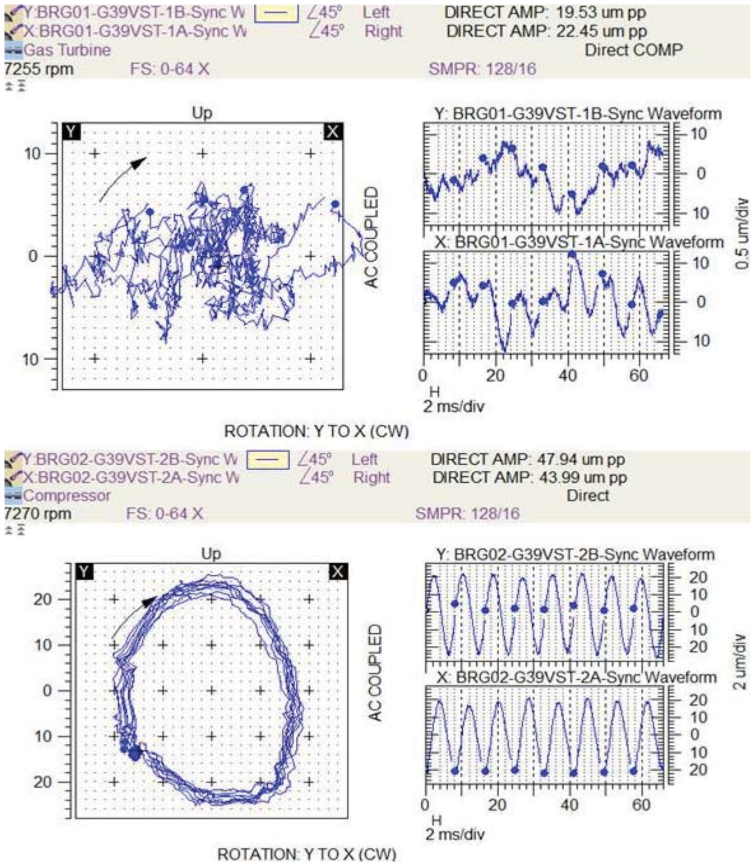


Fig. 4 (b) Orbit Plot of Turbine bearings showing rubbing symptoms at Bearing#01

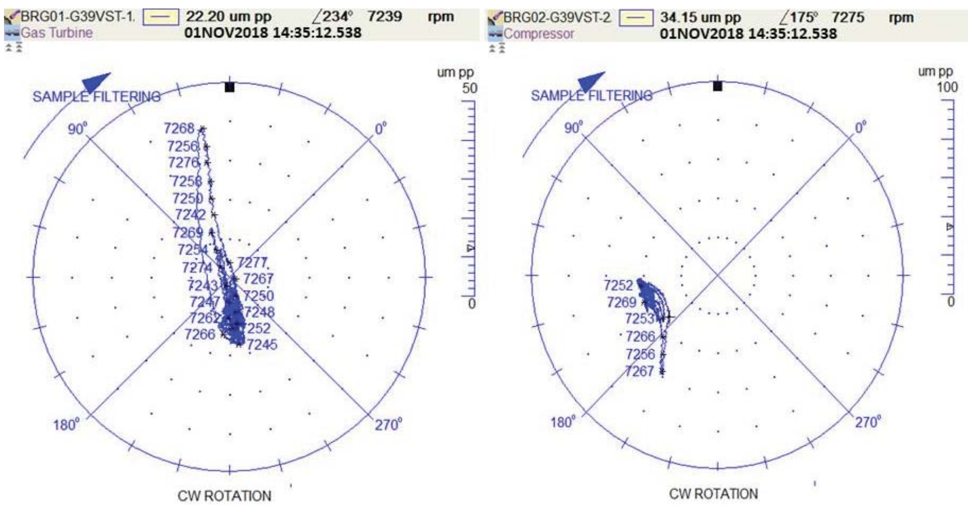


Fig. 4 (c) Polar Plot of Turbine bearings showing phase change of almost 360$^{o}$ at Bearing#01

Start-up data of machine, as explained above, suggests some rubbing may have taken place, possibly due to shaft bow condition [10] in turbine-compressor during coasting up of the machine speed. But Steady state shaft vibration levels of Gas Turbine were found within acceptable limits & found to be in zone A/B as per ISO 7919-3 / 10816-7, thus the machine was declared healthy for continuous operation.

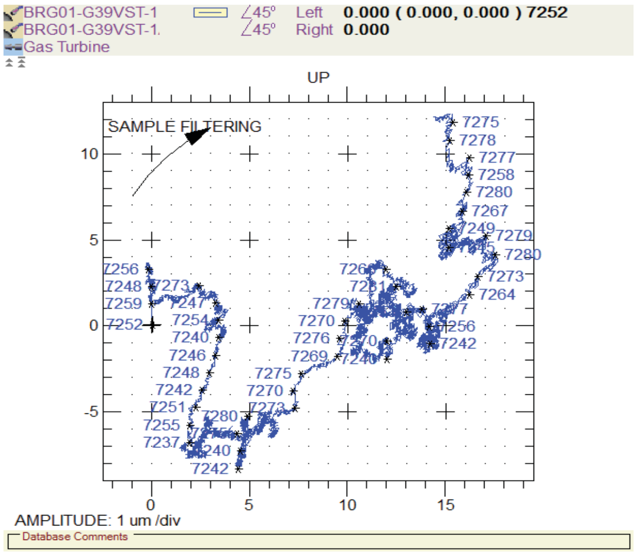


Fig. 5 Time base trend of Bearing#01 & 02 with Phase angle

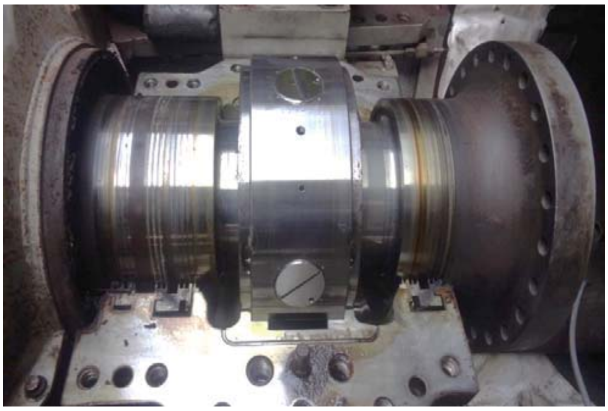

Fig. 6 Bearing#01 with housing upper half removed

Fig. 7 Heavy coke deposition, rubbing marks on Bearing#01 housing

Fig. 8 Pitting and cracks on Babbitt layer of Bearing#01

TABLE I
MECHANICAL VIBRATIONS LIMIT

| Vibration Limits as per ISO 7919-3 / 10816-7 | | | | |
|---|---|---|---|---|
| **Gas Turbine & Compressor** | | | | |
| Zone | RPM | Min. Bearing Clearance (µm) | ISO 7919-3 (µm pp) | ISO 10816-7 (µm pp) |
| A/B | 7258 | 400 | 56.3 | 132 |
| B/C | 7258 | 400 | 105.6 | 200 |
| C/D | 7258 | 400 | 154.9 | 280 |
| **Generator** | | | | |
| Zone | RPM | Min. Bearing Clearance (µm) | ISO 7919-3 (µm pp) | ISO 10816-7 (µm pp) |
| A/B | 1500 | Not Provided | 123.9 | — |
| B/C | 1500 | Not Provided | 232.4 | — |
| C/D | 1500 | Not Provided | 340.8 | — |

Do not have data of generator side bearing clearances, so vibration limits as per ISO 10816-7 could not be determined.

Zone A: Newly commissioned machines in preferred operating range; Zone B: Unrestricted long term operation in allowable operating range; Zone C: Limited operation (Alarm); Zone D: Risk of damage (Danger or Trip)

## IV. COKING

Coke is the solid residue created when oil undergoes severe oxidative and thermal breakdown at extreme temperatures. Higher the temperature, harder, blacker and more brittle the coke deposit residue form. Coking performance of oils varies based on formulation and the machine's environmental conditions. Coke formation in lube oil can be of different types like thin films, mist, puddles and dynamic lumps.

In the subject case puddle formation was observed in the labyrinth of the bearing housing (as shown in Fig. 12). Puddles form from high surface to volume ratio of lube oil layers & by long residence time of oil in a cavity. Physically it's in form of thick chunks of varying size & hardness depending on the age of coking. Its surface texture varies from shinny to matte like finish.

Coking phenomena occurs because of continuous exposure to elevated temperatures and long oil residue time of oil. Operational factors such as hot shutdowns can influence coke formation. Similarly, obstructions to flow or directional changes in lube oil path cause reduction in flow rate & increase the oil residence time.

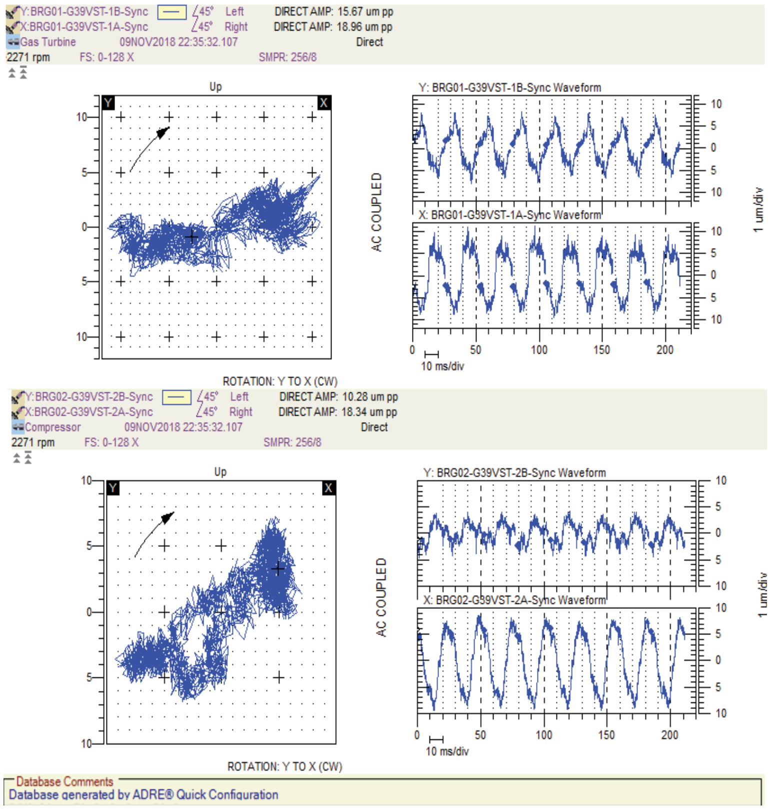


Fig. 9 (a) Orbit plot of turbine bearings at 2271 rpm

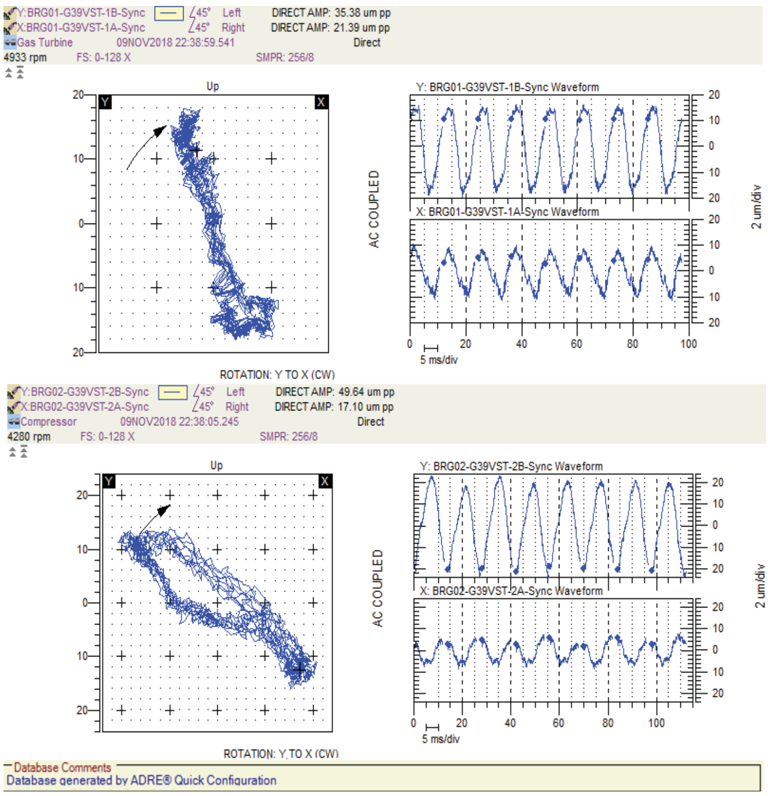


Fig. 9 (b) Orbit plot of turbine bearings at 4933 rpm

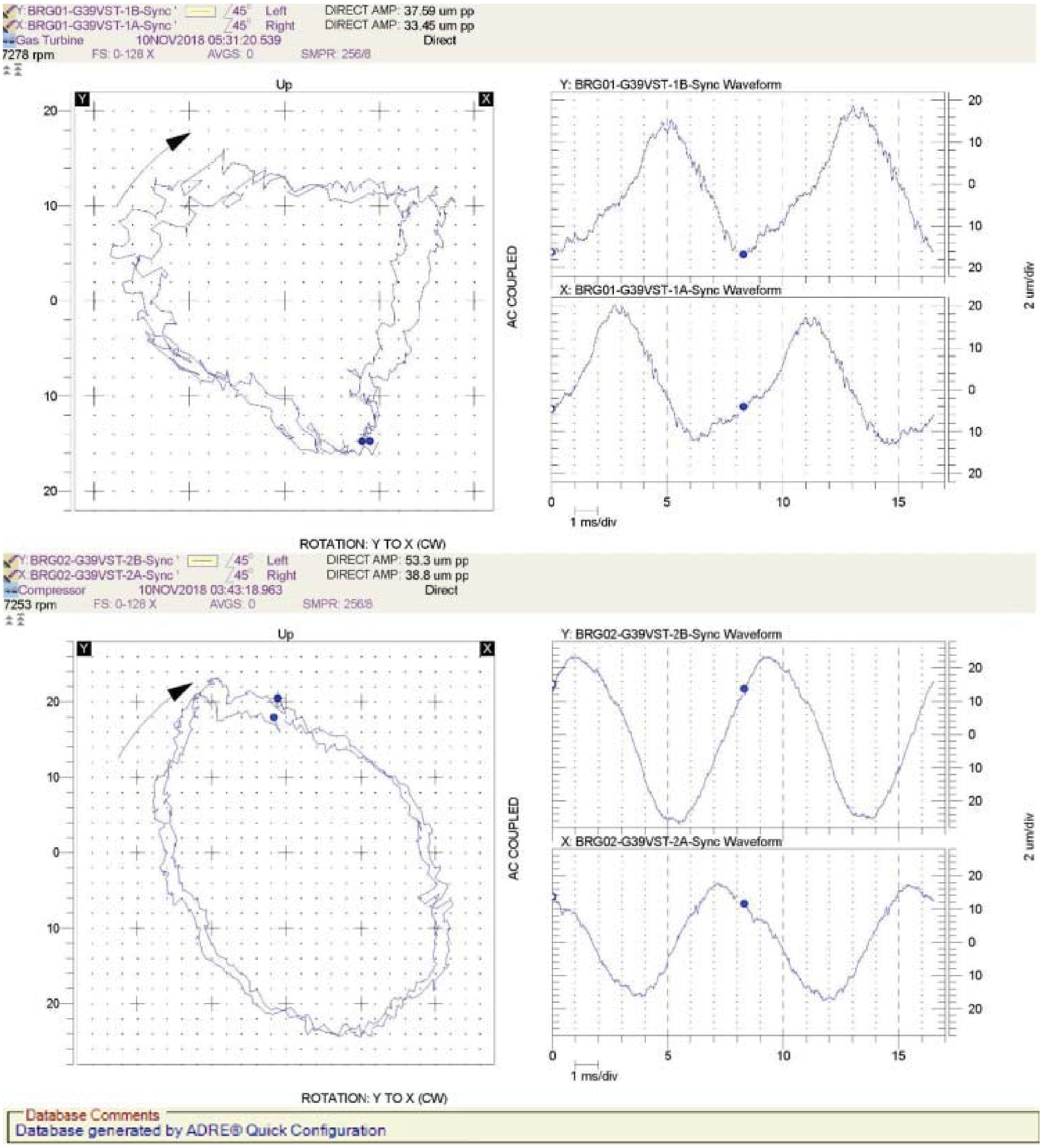


Fig. 10 Orbit plot of turbine bearings at 7278 rpms

Although coke formation is more of an aging phenomenon and be addressed by timely planning the maintenance of components exposed to lube oil circuit, there are some design phase measures to prevent this from happening. One technique is to introduce scavenge or vent lines in the gas path, this can significantly reduce deposits. Secondly, some coking tests have demonstrated powerful predictive capability and are essential during oil formulation and development phases of lubrication oils. The cooling and sealing air lines towards bearing housing also to be checked and blown away to remove any suspected choking.

## V. Learnings

Proper maintenance and operating practices can significantly affect the level of performance degradation & thus time between repairs or overhauls of a gas turbine. Based on the recent findings, reliability and maintenance team highlighted some gaps in the existing operational & maintenance practices. Recommendations are provided on how the operations can limit this kind of damage and deterioration of the gas turbines through proper O&M practices. Although maintenance & overhaul decisions are strongly linked with corporate production plans in a captive power plant but understanding performance degradation, as well as factors that influence degradation, can help in these decisions. Following recommendations were generated.

1) To switch the mode of gas turbines in between DROOP & ISO on regular intervals.
2) To maintain the cleanliness of air flow path, to insure the particle free air to compressor. This definitely improves the quality of cooling & sealing air been provided to different turbine components including bearing housings.
3) To blow the cooling & sealing air piping to remove any suspected choking in every available opportunity in order to insure the adequate flowrate.
4) To inspect & clean the lube oil resting points in every available opportunity especially at locations where temperatures are high like bearing housing.

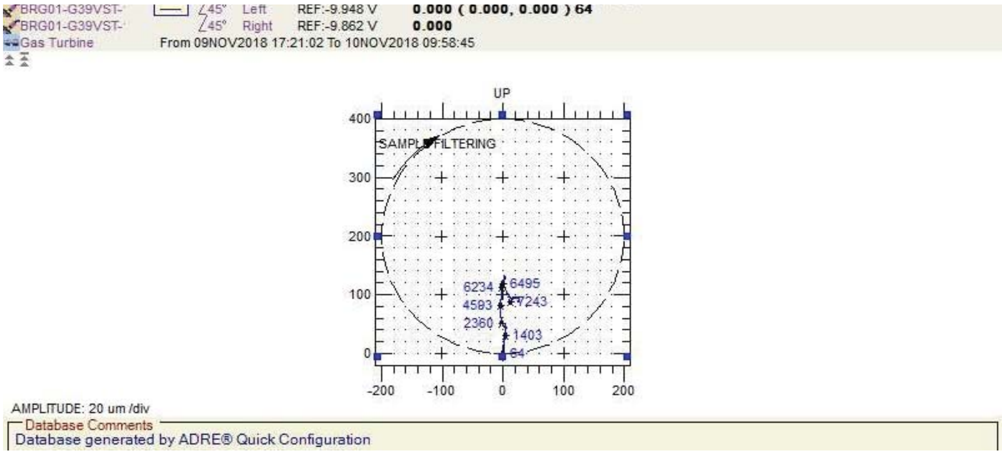


Fig. 11 Shaft Centre line of Bearing#01 after bearing replacement

Fig. 12 Physical texture of Coke puddles

Fig. 13 Shedding of coke deposits after getting rubbed

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