Identification and Elimination of High Vibration Caused by Misalignment Induced Oil Whirl without Stoppage of the Machine

A case study

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ABSTRACT

The problem of high vibration in single stream machines is a crucial issue which requires thorough investigation before a decision is taken for any corrective action. Continuous running of machine with high vibrations can cause internal rub and fatigue failure. However, stoppage of the machine without diagnosis of the root cause may not solve the problem in a single attempt. Multiple stops and start-up of the single stream machine to solve vibration problems can lead to heavy financial losses and may also result catastrophic failure of vital components of the machine. The present case study reports a systematic investigation and successful elimination of high vibration caused by misalignment induced oil whirl of Synthesis Gas Compressor rotor in its running condition.

KEYWORDS: Subsynchronous, spectrum, oil-whirl, hydrodynamic, transient, waterfall-plot, ammonia, urea, synthesis gas compressor

INTRODUTION

Synthesis Gas compressor is considered to be most critical single steam machine in Ammonia-Urea fertilizer plants. National Fertilizers Limited (NFL), Nangal is a gas based fertilizer plant with installed capacity of 950 MT/day of Ammonia and 1450 MT/day of Urea. Synthesis gas compressor G-1701 installed at NFL Nangal is Nuovo Pignone, Italy design compressor with three barrel type casings i.e. BCL 507, BCL407a and 2BCL406b supplied by BHEL, Hyderabad. The compressor is driven by Siemens make Model EHG40/25/50-3 HP and WK32/28 LP steam turbine. The main design parameters of the compressor are figured in **Table 1**. The complete Synthesis gas compressor-turbine train **(Figure 1)** is equipped with Bently Nevada 3500 series machinery monitoring system for continuous online vibration and axial position monitoring of each rotor. Rack 3500 is further backed up with GE state of the art vibration diagnostic platform *System-1*. Various steady state and transient (during stop and start) shaft movement and vibration plots like trend plot, polar plot, orbit plot, time wave form, shaft centre line plot, bode plot, cascade plot, waterfall plot etc. can be seen on *System-1* terminal for current and any historical period selected by the user. Synthesis gas compressor always remained area of critical attention for maintenance engineers due to problems like high vibration in any one of the 4/5-rotor of the machine, high seal oil consumption, oil leakages, problems related to turbine speed and extraction control system etc. High vibration is considered to be one of the most challenging problems because of limited availability of time for diagnosis of root cause before the machine trips or is forced to stop. The present case study details the work associated with identification of root cause and elimination of high vibration of the tune of 78 microns in 2nd barrel rotor of Synthesis gas compressor with careful corrective actions based upon detailed analysis using precise back up data.

BRIEF DESCRIPTION OF THE PROBLEM

Synthesis compressor 2nd Barrel BCL-407a was overhauled in Feb-2016. Before plant tripping on 04.08.2016 due to power failure, synthesis gas compressor was running with 2nd barrel shaft vibrations within acceptable limits i.e. upto 26 micron on drive end (DE) side and 44 micron on non-drive end (NDE) side, against alarm value of 60 microns and trip value of 82 microns. After start-up of Syn Gas Compressor on 05.08.2016, vibration of 2nd barrel increased to 72~78 microns and the vibration amplitude was fluctuating with change in plant load as shown in trend plot **(Figure 2)** from 11.08.16 to 15.08.16. To avoid tripping of Syn gas compressor on high vibration, plant load was restricted to 95% to find the root cause of vibration before stoppage of the machine for any corrective action.

PRELIMINARY INVESTIGATION

As per normal practice of preliminary investigation of high vibrations, various operating parameters, vibration trends and vibration spectrum of different bearings were studied. The following observations were noted down.

1. All operating gas parameters of the compressor were in normal operating range.
2. Vibration amplitude of the 1st and 3rd barrel rotors were well below alarm limits.
3. Overall vibration amplitude of the 2nd barrel rotor was running above the alarm limits even at reduced plant load. DE side overall amplitude of vibration was approaching to trip value of 82 microns and on NDE side just above the alarm limit of 60 microns.
4. The vibration level of 2nd barrel was increasing with increase in load on the compressor, whereas there was no appreciable change in vibration level of 1st and 3rd barrel rotors with change in load on the compressor.
5. The predominant vibration peak in the DE side vibration spectrum **(Figure 3)** was at a Subsynchronous frequency of 0.42~0.43 multiple of the rotor speed (highest peak at 0.42-0.43 X in the spectrum).
6. Bearing hosing vibration of all the bearings was normal. Maximum vibration velocity was below 1.5 mm/sec as a set of cap vibrations recorded in **Table 2**.
7. Lube oil temperature of the machine was 49 deg C, which was slightly on the higher side due to peak summer conditions.

ACTIONS BASED UPON PRELIMINARY INVESTIGATION

The sub-synchronous peak at a frequency of 0.42-0.43X in the vibration spectrum indicated that there is some fluid induced instability and most probably oil whirl in the hydrodynamic bearings or in floating oil seal rings of the compressor. In the oil whirl condition a thick oil film starts moving in a swirling motion around the journal of the rotor. As the journal is moving at the rotor speed and the bearing inner surface remain under steady state the oil between the journal and the bearing inner surface rotates at speed approximately 42 to 45 % of the rotor speed. The whirling motion of the oil causes shaft to vibrate at a frequency of 0.42~0.48 of the rotor speed which is known as the oil whirl. The normal practice to eliminate the oil whirl or to reduce the effect of oil whirl on vibration amplitude is to change oil pressure or temperature in the forced feed hydrodynamic bearings. Due to hot climate conditions temperature of the oil in the lube oil header was running at 49 deg C which was already at higher end of the recommended normal oil temperature range of 40 to 50 deg C. So to avoid any other problem by increasing the oil temperature beyond recommended values, it was decided to decrease oil temperature by about 7-8 deg C. Lube oil temperature of 42 deg C was achieved by circulating available cold raw water through lube oil cooler with an existing provision for operating the syn gas compressor lube oil cooler on raw water in place of cooling water. This exercise resulted decrease in vibration amplitude by about 8-10 microns but even then as the compressor load was increased marginally beyond 95%, shaft vibration of 2nd barrel DE side again approached the trip value. Variation in oil pressure was neither feasible nor appropriate during running because of possibility of oil pressure going excessively low causing bearing failure due to specific construction of oil regulators in radial bearing oil line. Apparently, at this stage only option was to take shutdown for about 3-days to inspect the bearings and floating oil seal rings of the compressor. However, as with this decision production loss of 3-days and huge energy loss was evident, it was decided to carry out a detailed diagnosis of the problem for identification of the root cause before any corrective action.

DIAGNOSIS OF THE PROBLEM

To investigate root cause of the problem, various steady state and transient vibration related plots of 2nd barrel were intensely analyzed. The average shaft centre line plot of DE bearing **(Figure 4)** plotted for transient condition during start up on 05.08.2016, indicated that DE side rotor shaft had lifted by approximately 155 microns (0.155 mm) from its bottom steady state condition which is almost equal to the bearing clearance of 0.16 mm. Ideally shaft lift during start up should be near to half of the bearing clearance and the average shaft centre line of the shaft should finally settle in 4th quadrant in the bearing clearance circle for anticlockwise rotation. Average shaft centre line plot of NDE bearing **(Figure 5)** indicated that the shaft was near to the centre of the bearing after start-up. The shaft lift almost equal to the bearing clearance kept during assembly on DE side was clearly indicating that after start up on 05.08.2016 bearing clearance had increased to some extent and the rotor was no more supported on the normally desired thin oil film in the radial bearing. This is a condition of unloading of a radial bearing. Hence the excessive shaft lift from the bearing bottom i.e. unloaded radial bearing was permitting a thick oil film to move around the rotor journal in a swirling motion causing high vibrations in the rotor at oil whirl frequency of 0.42-0.43 of rotor speed. As there was no upward force for the rotor causing upward lift in the bearing clearance on DE side, it was concluded that misalignment alongwith slight increase in bearing clearance are the only predominant factors casing excessive lift of the rotor in one of the two bearings of the same shaft. Hence, detailed diagnosis of the problem had established that oil whirl related high vibration in 2nd barrel rotor on DE side is caused by unloading of the radial bearing due to misalignment.

CORRECTIVE ACTION

Diagnosis of the problem based upon intensive analysis of various vibration related plots revealed that the root cause of the high vibration is misalignment of 2nd barrel rotor w.r.t. 1st barrel rotor coupled with slight increase in clearance during stop and start of the compressor. Based upon details of initial bearing clearance and rotor lift during star up, it was established that to support the rotor shaft on the bearing surface with nominal oil film thickness, it is required to lift the bearing shell by approx. 0.08-0.10 mm. Normally, this would require adding shims between compressor barrel front end pedestal & compressor casing lugs. However, this too was not possible without shutting down the machine because after insertion of shims, it would become necessary to check the alignment between 1st and 2nd barrel Rotor shafts.

After exhausting all the conventional means to overcome the problem, a detailed brain storming session coupled with an intensive thought process, clinched an idea to raise the compressor barrel front pedestal without stopping the machine. Based upon height of the pedestal and coefficient of expansion of the pedestal material, it was calculated that if temperature of the DE side supporting pedestals is increased by 15 deg C, then the drive side bearing shell will be lifted by approximately 0.10 mm

With complete back up of data and calculations, on 16.08.2016, both (left and right) drive end pedestals of the 2nd barrel were slowly and equally heated from the existing average temperature of 65 deg C to 80 deg C with a carefully regulated supply of a small quantity of steam in both the pedestal boxes. During heating process the lift of both left and right the pedestals was accurately measured with a dial gauge **(Figure 6)** and continuous temperature measurement was ensured with thermo-gun. As the pedestal lift approached 0.08 mm, average temperature increase of both left and right pedestals by about 12 deg C, the vibration level on DE side of the rotor came down from 72-78 microns to 22-24 microns **(Figure 2)**. This vibration level was well within normal acceptable vibration limits. The machine was kept under continuous close observation for full 24 hrs with adequate amount of steam outflow to maintain temperature of pedestals at approximately 75-80 deg C. Now there was no noticeable change in vibration amplitude of the machine even with increase in load on the machine. The waterfall plot **(Figure 7)** shows that subsynchronous vibration component at 0.42-0.43X has been eliminated altogether after heating DE side pedestals to the desired temperature. Thereafter it was decided to run the machine at pedestal temperature of 75-80 deg C with help of closely monitored heating steam. With round the clock close watch on the pedestal temperature & availability of heating steam by the shift-duty operating staff, The machine was successfully kept running at the desired load till annual shutdown of the plant in March-2017. On opening of the machine in March-2017 shutdown, alignment reading between 1st and 2nd barrel verified our investigation of misalignment causing lifting of the 2nd barrel rotor on DE side. In idle condition the 2nd barrel rotor was found 0.38 mm (dial indicator reading 0. 76mm) down w.r.t. 1st barrel rotor against protocol of value of 0.26 mm (dial indicator reading 0.52 mm) and the DE side bearing clearance was found 0.18mm against design value of 0.13-0.17mm. Though gear couplings are capable to take care of this kind of misalignment, however in case light weight rotors when bearing clearances are on the higher side, even small misalignment in the train of rotors can cause fluid instability at an intermediate bearing having higher clearance. As a long term solution to the problem, during ATA-2017, radial bearing clearances were adjusted to the design values and the idle condition alignment was corrected to the protocol values.

CONCLUSION

High vibration in single stream machines is a key area of concern for process plant engineers, which can result in severe implications on productivity and profitability of the process plants. Comprehensive diagnosis of the root cause can reduce and sometimes completely eliminate downtime of the plant due to vibration problems. On line corrective actions need utmost care and reliable basis to avoid any kind of failure in the single stream machines. With the knowledge and usage of latest machinery diagnostic systems like System-1, it is now possible to identify the root cause of vibration so that corrective action can be planned accordingly. With systematic diagnostic approach coupled with dependable data and analytical tools, NFL- Nangal managed solve the alarming problem of high vibration of synthesis gas compressor 2nd barrel shaft without stoppage of the machine.

Table 1: Design Parameters of Synthesis Gas Compressor					
Parameter	**Unit**	**BCL 507**	**BCL 407a**	**2BCL 406b**	
		1st STAGE	**2nd STAGE**	**3rd STAGE**	**RECYCLE**
Capacity	Nm^3/h	99852	99852	99852	448244
Weight Flow	Kg/h	38024	38024	38024	201517
Inlet Pressure	ATA	38	80	146	214
Discharge Pressure	ATA	81.6	154	218	231.5
Inlet Temperature	Deg C	40	41	41	26
Discharge Temperature	Deg C	152	137	107	35
Speed	RPM	10870	10870	10870	10870
Coupling Power	KW	4285	3880	2800	1750

Table 2:Bearing housing vibration at the time of increased shaft vibration of 2nd Barrel						
	Displacement (microns)			**Velocity (mm/sec)**		
	H	**V**	**A**	**H**	**V**	**A**
1st Barrel BCL507 DE	10	9	10	1.5	1	1.5
1st Barrel BCL507 NDE	9	8	10	1	1	1.5
2nd Barrel BCL 407a DE	8	10	10	1	1.5	1.5
2nd Barrel BCL 407a NDE	10	10	9	1	1.5	1.5
3rd Barrel 2BCL406b DE	10	9	10	1.5	1	1.5
3rd Barrel 2BCL406b NDE	8	10	10	1	1.5	1.5

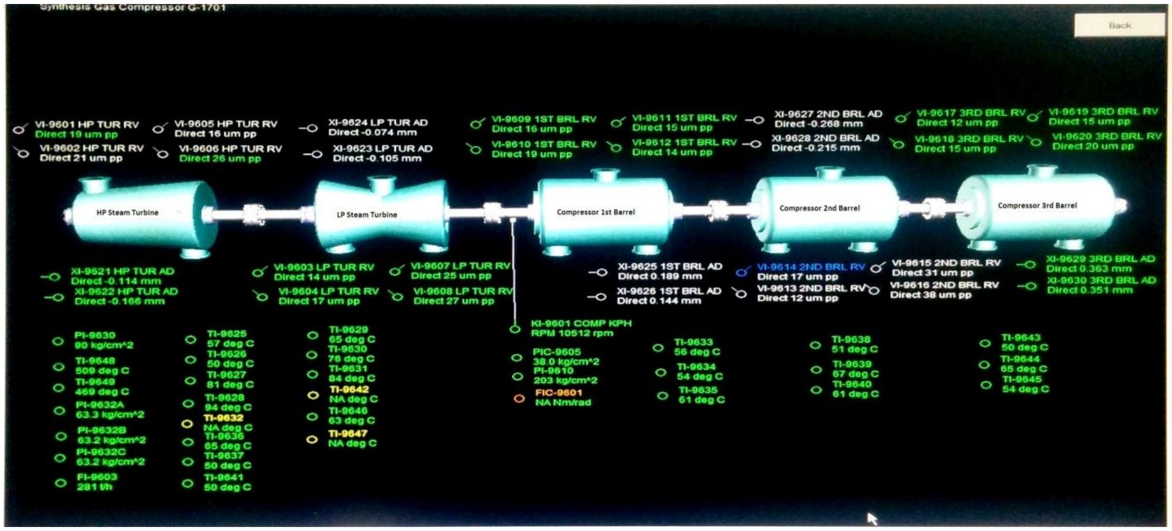

Figure 1: Schematic view of synthesis gas compressor train and its drive turbines

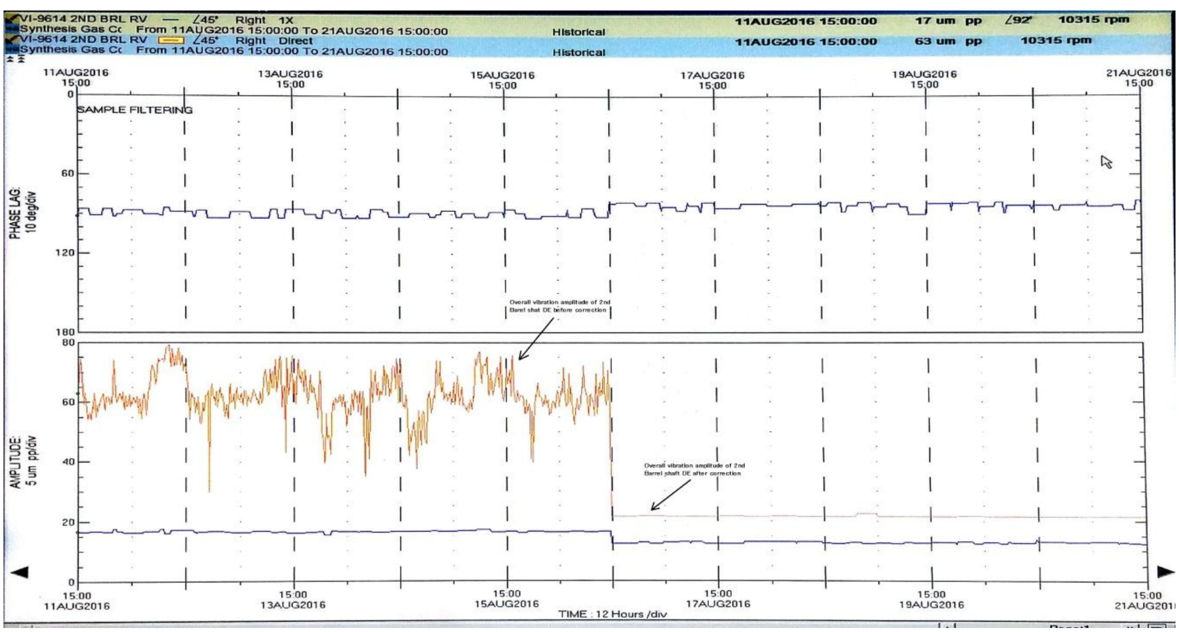

Figure 2: Vibration trend of 2nd barrel after startup on 05.08.16

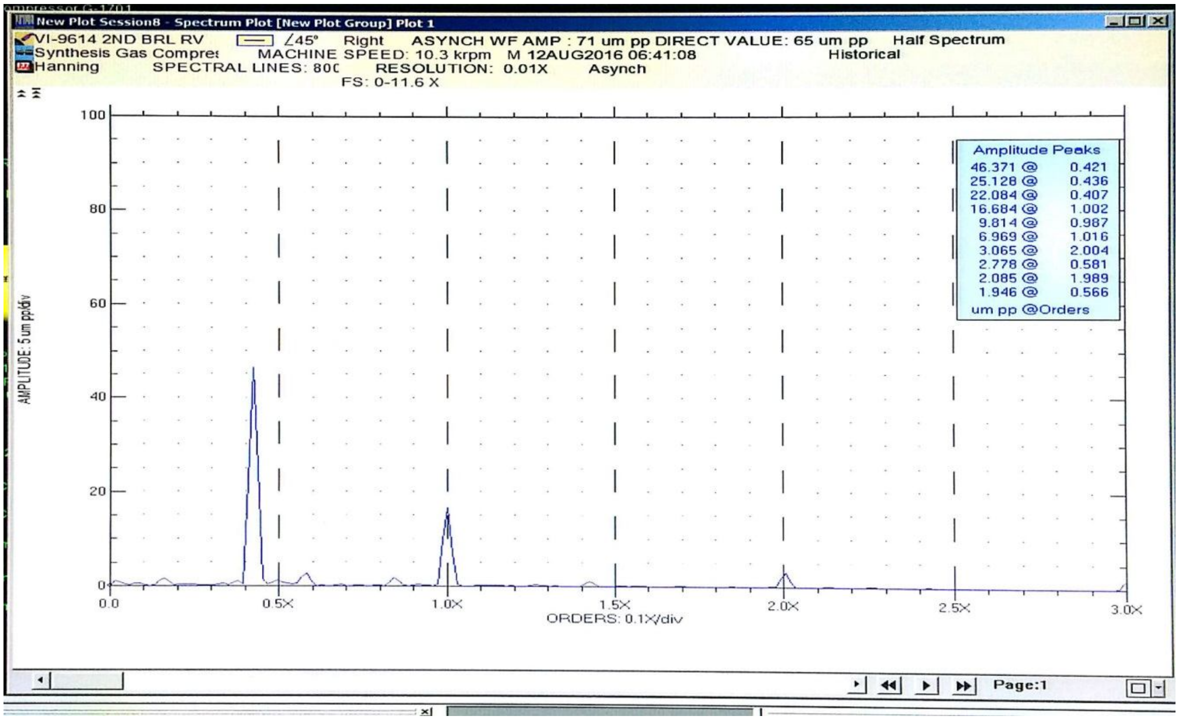

Figure 3: DE side shaft vibration spectrum showing high peal at 0.42~0.43X

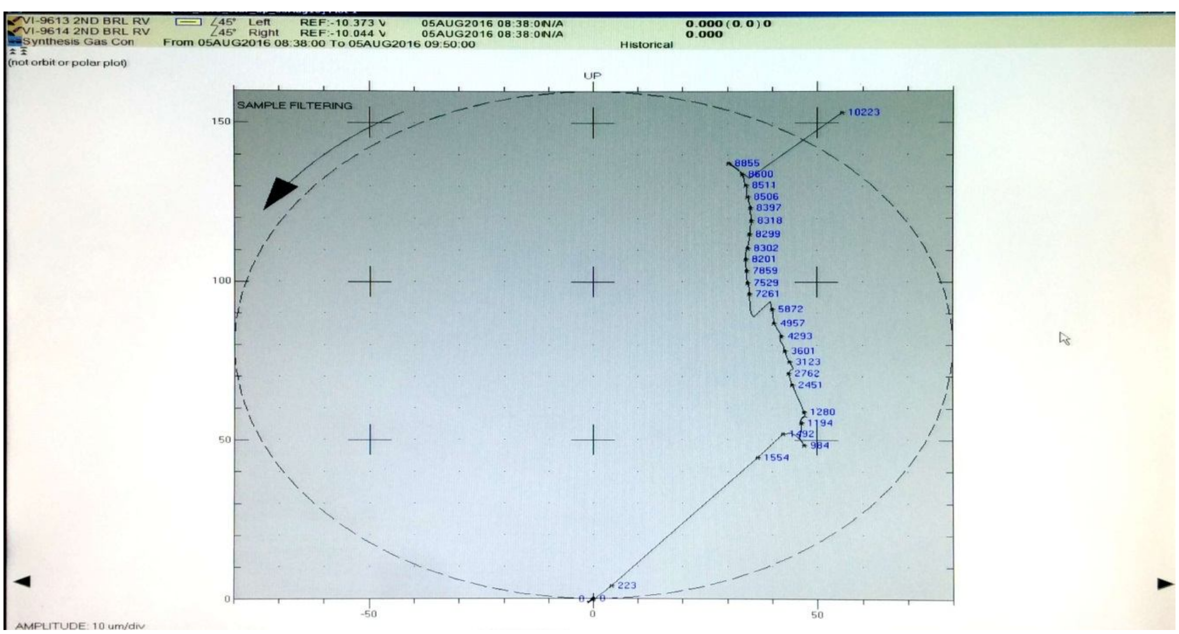

Figure 4: 2nd Barrel rotor DE bearing shaft centre line plot during start up on 05.08.2016

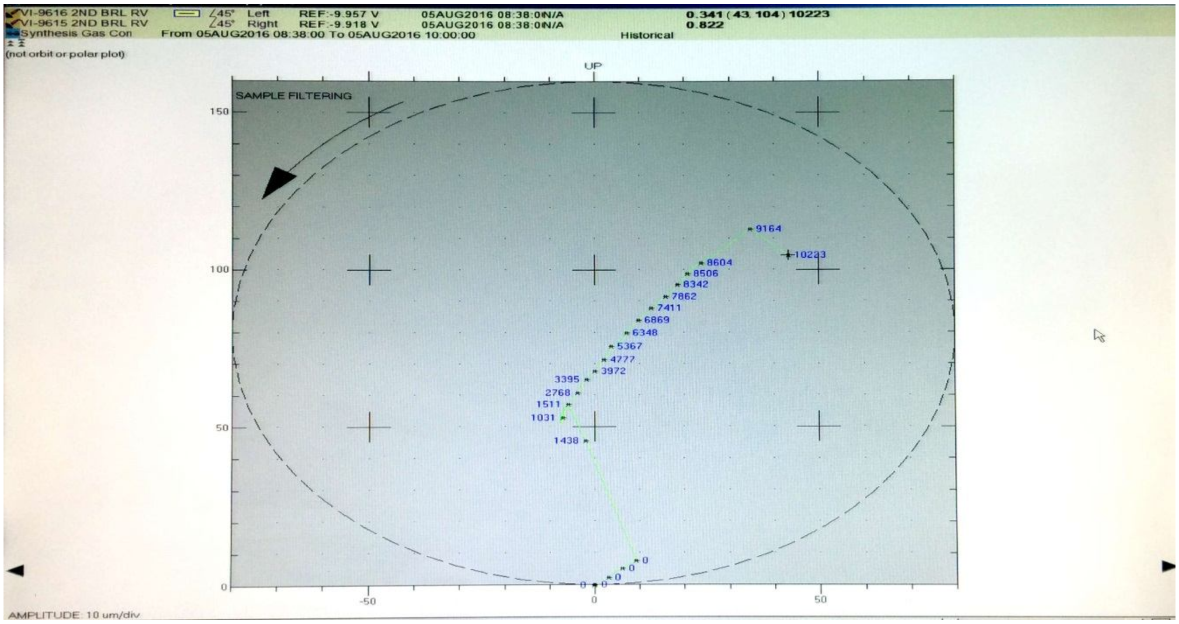

Figure 5: 2nd Barrel rotor NDE bearing shaft centre line plot during start up on 05.08.16

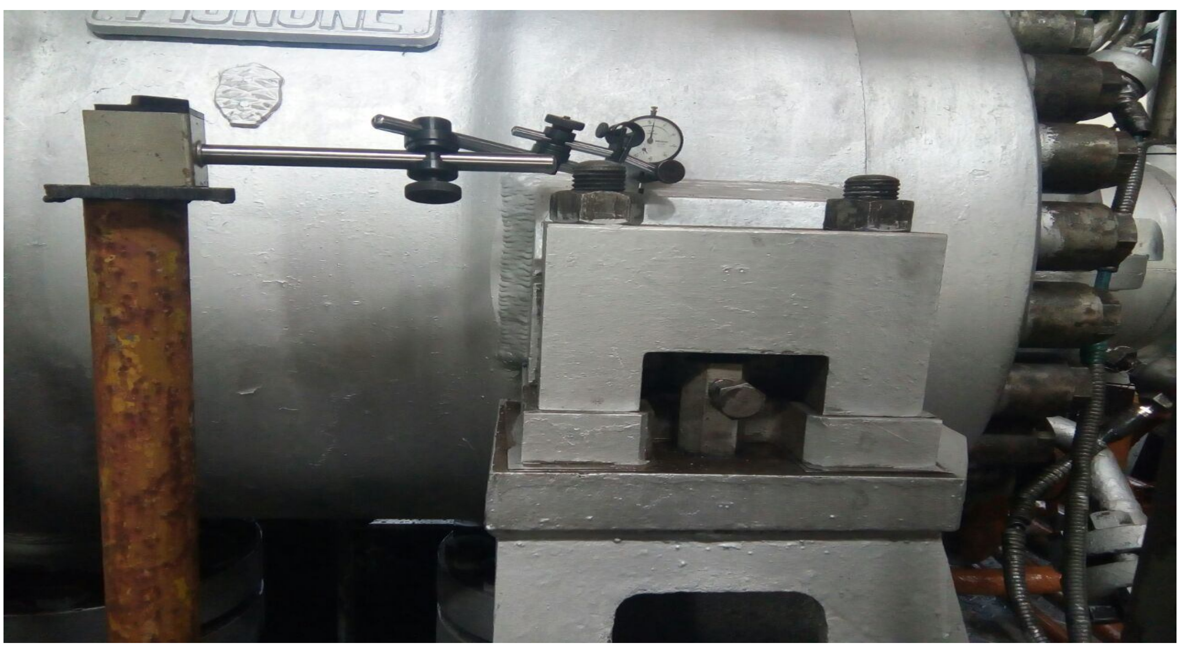

Figure 6: Measurement of 2nd Barrel DE pedestal lift during heating process

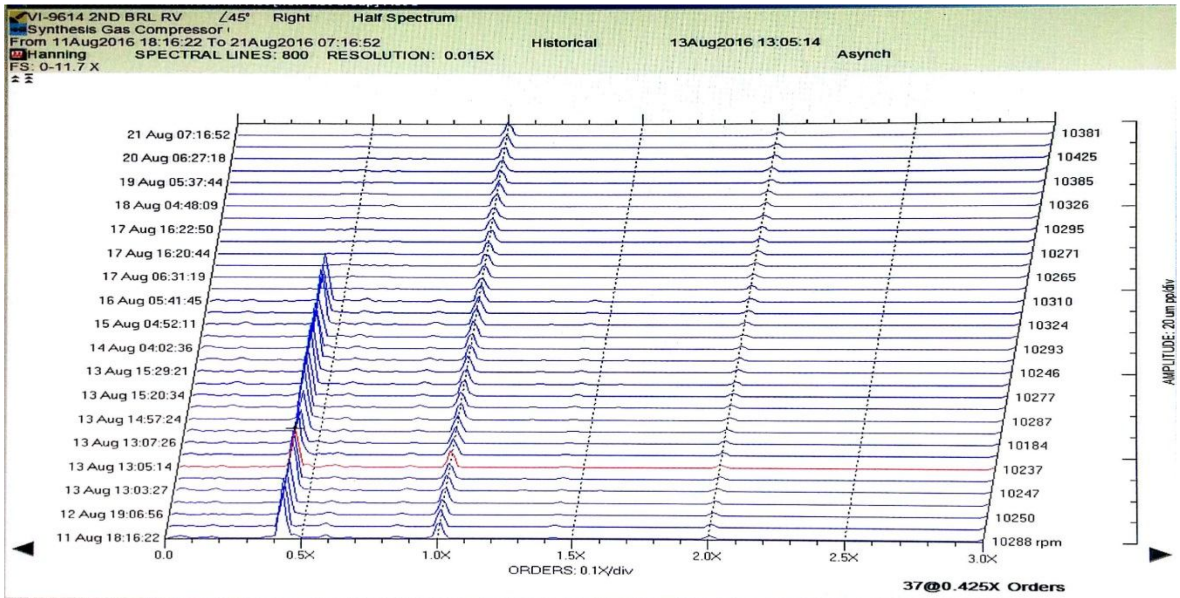

Figure 7: Waterfall plot showing elimination of 0.42~0.43X component after correction

THE FERTILISER ASSOCIATION OF INDIA

ENSURING FOOD SECURITY

T/44
15th June, 2017

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Dear Sir,

We thank you for contributing an article on "Identification and Elimination of High Vibration Caused by Misalignment induced Oil Whirl without Stoppage of the Machine - A Case Study" for publication in the June, 2017 Technical Special issue of Indian Journal of Fertilisers. A copy of the Journal is enclosed.

We have pleasure in enclosing a cheque no. 256673 dated 14.06.2017 for Rs. 2500/- drawn on Canara Bank, New Delhi as an honorarium.

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Please acknowledge the receipt.

Thanking you,

Yours faithfully,

Manish Goswami
Manish Goswami
Chief (Technical)

Encl.: As above