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SHAFT OIL BARRIER SEALS CAN CAUSE HIGH VIBRATION. DO YOU KNOW ? (a)

(A Case Study of High Vibrations in Syn Gas Compressor, its Diagnosis and Correction)

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1. Abstract

The paper deals with high vibrations at low frequencies, due to shaft oil barrier seals. Generally such vibrations are associated with problems at journal bearings. But at times the shaft oil barrier seals used for end sealing of rotor in the centrifugal compressor, act as lightly loaded bearing thus causing high vibrations at lower frequencies, and hence severe load limitations on the machine. Proper fitting of oil barrier seals with guidelines and modifications can eliminate the occurrence of such vibrations.

The paper details the work associated with identifying, correcting and steps taken to prevent similar occurrences of the problem of high vibrations caused by shaft oil barrier seals in syn gas compressor at the National Fertilisers Limited, Panipat Unit.

2. Introduction

National Fertilizers Limited, Panipat Unit operates at capacity of 900MT of ammonia per day and 1550MT of urea per day. Its ammonia plant has five high speed turbo machines:-

1) The oxygen gas compressor is from Demag Germany while its drive steam turbine is from AEG KANIS, Germany.
2) Air and nitrogen compressors and their respective drive steam turbines are supplied from MES Japan.
3) Ammonia refrigeration compressor and its drive steam turbine are by BHEL Hyderabad.
4) The synthesis gas compressor was supplied by Nuovo Pignone, Italy and the design with three barrel type casings were supplied by BHEL, Hyderabad. It is driven by Siemens steam turbine type EHNK-3 (Refer Annexure-I). All compressor stages and its drive turbine are equipped with Bentley Nevada Vibration Monitoring System, with two radial vibration probes in two planes in each radial bearing, one probe each for axial displacement and one electromagnetic speed pick up probe at steam turbine.

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2.1 Compressor Specifications

Design : Nuovo Pignone, Itlay
Type : Centrifugal, barrel type casings.
Supplier : BHEL, Hyderabad.
Speed : 11,100rpm.

Stages	1st BCL-507	2nd BCL-407a	3rd BCL-407b	
Gas Handled	Syn. Gas(H2+N2)	Recycle Gas		
Capacity(Dry)Nm3/Hr	99.96	4,64,300		
Suction Conditions:				
Pressure (kg/cm2 abs)	37.5	80	146	219
Temperature (Deg.C.)	33	41	41	34
Discharge Conditions:				
Pressure (kg/cm2 abs.)	81.6	148.5	221	235
Temperature (Deg.C.)	144	133	113	43.5

2.2 Turbine specifications

Make	Siemens, Germany
Type	Condensing-Extraction Third Generation Turbine
Model	EHNK-40/36/40-3
Supplier	BHEL, Hyderabad
Speed	11,100rpm
Power Rating	15983KW
Inlet steam condition	100 kg/cm2, 480 Deg.C.
Extraction; steam condition	40 kg/cm2

2.3 Oil barrier type oil sealing system

The sealing system with oil barrier shaft seal in the centrifugal compressor (synthesis gas compressor) is well known to us. Oil barrier seals are provided to prevent leakage of gas to atmosphere, along the shaft in centrifugal compressors. These seals contain two or more bushes with close clearances with the shaft. These bushes are called sealing rings, which are free floating on the shaft. These seal rings are kept apart with a spring and oil is injected between these bushes at slightly higher pressure than that of the gas to be sealed, so there is slight transfer of oil into the compressor. The leaked oil is called sour oil. This sour oil enters traps (separator) which separate gases and oil. The gas separator is either vented or recycled to suction of compressor through an orifice (with bypass line for this orifice). Refer Annexure II.

3. Problem Faced

The second stage compressor BCL-407a was overhauled in June 2001 due to problem of fall in polytropic efficiency (from 60.4% to 47.7%), though the shaft vibrations on all the stages were normal. The compressor was opened. The diaphragm ‘O’ ring was found to be damaged, which was main cause of bypassing of gas, thus resulting into low efficiency. The rotor had grooving and rubbing at impeller eye and their spacers. The rotor was replaced with spare rotor. The assembly was made with spare repaired rotor, new ‘O’ rings, new labyrinths and new oil barrier seal rings. The clearance of new H.P. seal ring was 0.06mm (designed value of 0.06 to 0.08 mm). The compressor was aligned with 1st stage and 3rd stage compressor, very near to recommended designed values.

<u>On the restart of machine, high vibrations appeared at both ends of the compressor at full load of machine.</u>

The values of shaft vibrations as indicated on Bentley Nevada monitor were as follow;

<u>Suction end:</u> <u>3.5 mils peak to peak</u>

<u>Discharge end:</u> <u>4.0 mils peak to peak</u>

The pressure differential between reference gas and seal oil pressure was found hunting.

The vibration signatures were taken (by instrument CSI, model: 1900). Peak component was occurring at 0.42 to 0.45 times the RPM. Kindly refer Annexure I, for vibration signatures. Generally such low frequency vibrations, below frequency of 1 x RPM (0.42 to 0.45 times RPM) are associated with oil whirl phenomenon. It was felt that the problem was of Oil whirl occurring at journal bearings. Hence the oil pressure to journal bearings (both ends) were changed along with temperature as a remedy for problem of oil whirl, but no change in vibrations was observed. Hence chances of oil whirl appearing in journal bearings were eliminated.

The high vibrations caused severe limitation in plant load. The machine was kept running at low load with close watch on parameters and vibrations.

4. <u>Diagnosis of Problem</u>

As the chance of oil whirl in bearing was omitted, the probable area of concern could be floating oil barrier seal rings as well (either side). The oil whirl can occur on these close clearance seal rings also, when H.P. seal ring (with low clearance of 0.06mm to 0.08mm) starts acting as lightly loaded bearing. This can happen when floating action of seal ring is restricted. During the overhauling we had replaced the rotor with repaired one that was balanced and alignment of 2nd stage with 1st stage and 3rd stage was perfect, near to design values. This could have further decrease the preloading and resulting in enhancement of oil whirl phenomenon.

4.1 Decrease in vibrations

Fundamentally oil seal phenomenon can be omitted by varying the oil condition i.e. pressure and temperature (viscosity). But it is difficult to change the seal oil inlet pressure to seals, as it needs change of oil level in overhead seal oil tank. A change of one-meter level in overhead seal oil tank only result in very small change in oil pressure to oil seal (say by 0.1 kg/cm2).

Hence it was tried to change the seal oil flow to seal by varying the opening of bypass valve ‘Vb’ of orifice of gas vent from seal oil (trap) separator of HP seal sour oil to first stage suction (refer Annexure II). With this the vibration suddenly decreased to 2.9 mils at discharge end and 2.8 mils at suction end, from valves of 4.0 mils and 3.6 mils respectively.

This confirmed that the cause of oil whirl was oil barrier seal rings. The bypass valve ‘Vb’ was then closed as its opening was bypassing the gas from second stage to first stage suction causing unloading of machine.

5. Corrective Action

It was planned to inspect oil barrier seals assemblies of both ends of 2nd stage compressor. To check seals from acting as lightly loaded bearing following actions were planned:

- To reduce clearance of journal bearings
- To increase the clearance of HP seal ring with the rotor shaft. New seal rings were kept ready for replacement on available opportunity. These new HP seal rings were scrapped so as to maintain clearance with respect to the shaft, on higher side (0.09 to 0.1mm) in place of design clearance of 0.06 to 0.08mm. That time the compressor was running with HP seal rings with clearance of 0.06mm at both ends, which were fixed during overhauling of compressor in June, 2001.

The job of inspection of seal rings and journal bearings were taken in July, 2001. The seals rings and journal bearings were opened. The observations were as follows.

- Suction side journal bearing pads had rubbing. Removal of babit material at one edge was observed (refer Annexure III).
- The discharge and suction end oil barrier seal rings were found to be rubbed and deep pitting was found on the babit metal (refer Annexure III)
- The edge of the slot on the leaf spring was found digging inside its resting face on HP seal ring.

6. Final Solution of Problem

To prevent the restricting of floating action of HP seal ring, all concerned areas were taken care of as follows: a)The journal bearings were replaced with new bearings with lower clearance. b) The new HP seal ring with higher clearance of 0.09mm to 0.1mm was fixed. A chamfer of 45 degrees was provided at oil inlet ends of HP seal ring. c)The assembly of seal rings were made with leaf spring positioning in such a way that slot on leaf spring did not rest on face of seal ring,(i.e., it was set in air). Kindly refer Annexure IV for correct and incorrect position of leaf spring.

7. Status on Restart of Machine

The machine was restarted and loaded. All the vibrations on 2nd stage were within permissible limit i.e. 0.5 mils at suction end and 0.3 mils at discharge end. The shaft vibration signatures were taken and the peak component, which was earlier occurring at 42% to 45% of shaft RPM disappeared.

8. General Comments

Oil whirl is an abnormal phenomenon, which is associated with pressure lubricated sleeve bearings, or, more precisely associated with any pressure lubricated shaft constraint, operating at relatively high speed. Oil whirl vibration is often quite severe. A pure oil whirl generally occurs at about 42 to 45 percent of running speed frequency. It occurs when the bearings are lightly loaded. The journal (shaft) is travelling at rotor speed and bearing is traveling at zero speed. The oil which is being sheared continuously is travelling at less than one-half speed due to slippage.

It is right moment to refer here the description of "Oil Whirl" by John Sohre, the turbo machinery specialist, *" If the bearing and hence the oil in the bearing is lightly loaded the oil carries the shaft about in the bearing much like surfboard which is caught in the wave. If, however the shaft is heavily loaded, e.g., as a battleship, it is not sensitive to oil film forces".*

It is generally observed that the rotating machinery users, in case of low frequency vibrations doubt oil whirl at the journal bearings. But it is important to know that 'Oil Whirl' can set in any continuous shaft constraint, which is lightly loaded. (i.e., not necessary at only journal bearings). It is what the subject case study was about. The seal rings were acting as lightly loaded bearing and causing oil whirl, as the babit metal in seal rings acts as continuous constraint without any break or lobe in it. The probable reasons were tight clearance of HP seal ring, lack of flexibility in seal rings due to digging of leaf spring end edges (Refer Annexure V).

In such a case firstly we are to identify the exact location of occurrence of oil whirl: whether at journal bearings or at other shaft continuous constraints, i.e., to eliminate and distinguish between two situations so as arrive at correct cause of vibrations.

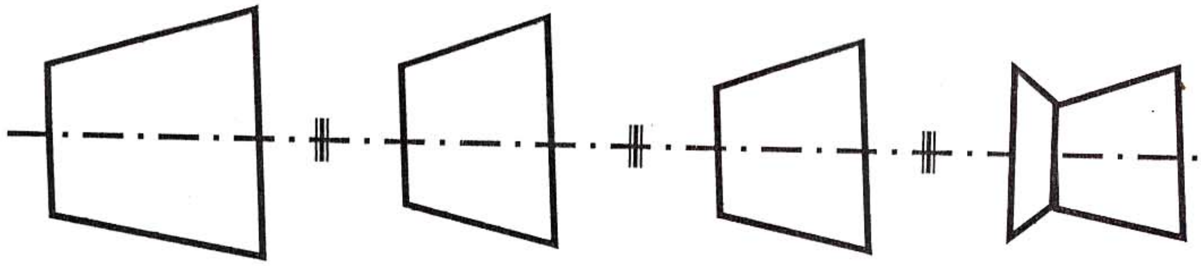
9. Acknowledgements

The Author acknowledges the efforts and contribution of all team members of Mechanical Maintenance of Ammonia Plant of National Fertilizers Limited, Panipat Unit (India) in trouble shooting and execution of rectification job. Above all the continuous motivation and the expert guidance of Shri P.S. Grewal, Chairman and M.D., Shri, S.S. Dandona, G.M., and Shri, R.K. Dixit, G.M., National Fertilizers Limited, India.

10. Reference

The Practical Vibration Primer by *Charles Jackson*

ANNEXURE - I



TURBINE
EHNK 40/36/40-3

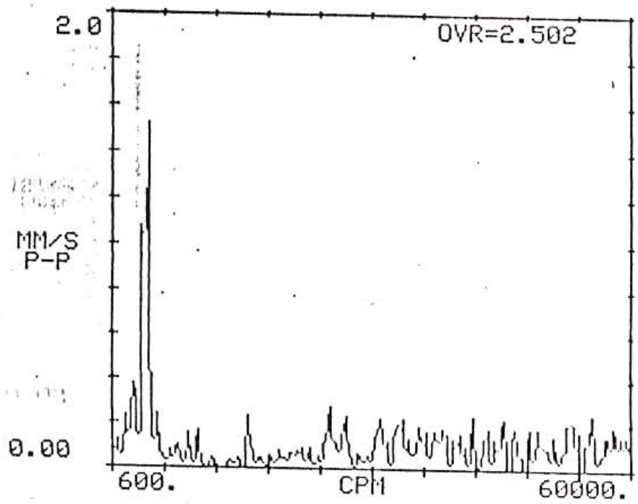
Ist stage Compressor
BCL -507

2nd Stage Compressor
BCL 407a

3rd stage Compressor
2BCL –407b

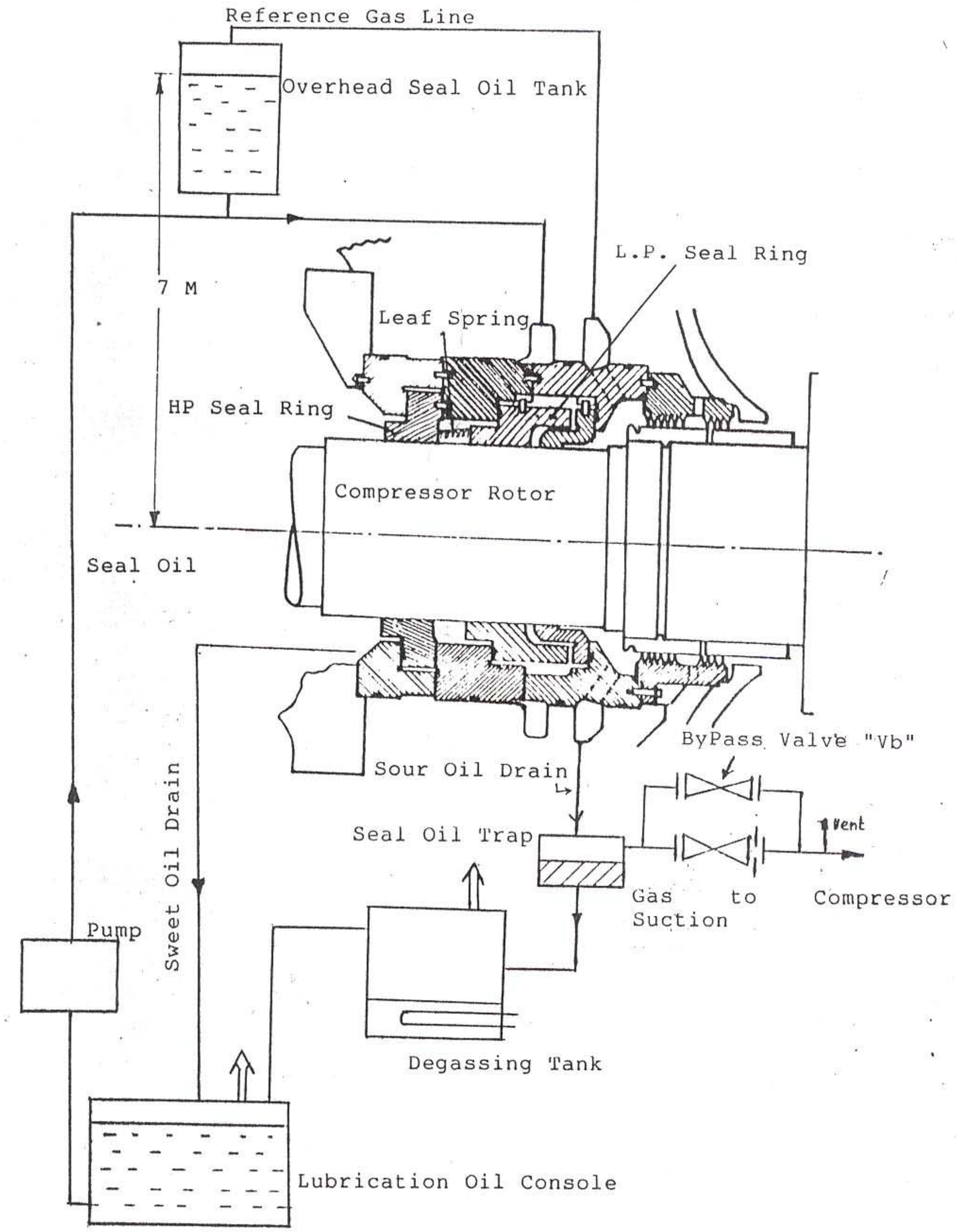
SIGNATURES OF SHAFT VIBRATION (2nd Stage)

DATA_DESCRIPTION: CSI MODEL 1900 ANALYSIS REPORT
DATE/TIME: 30.06.01



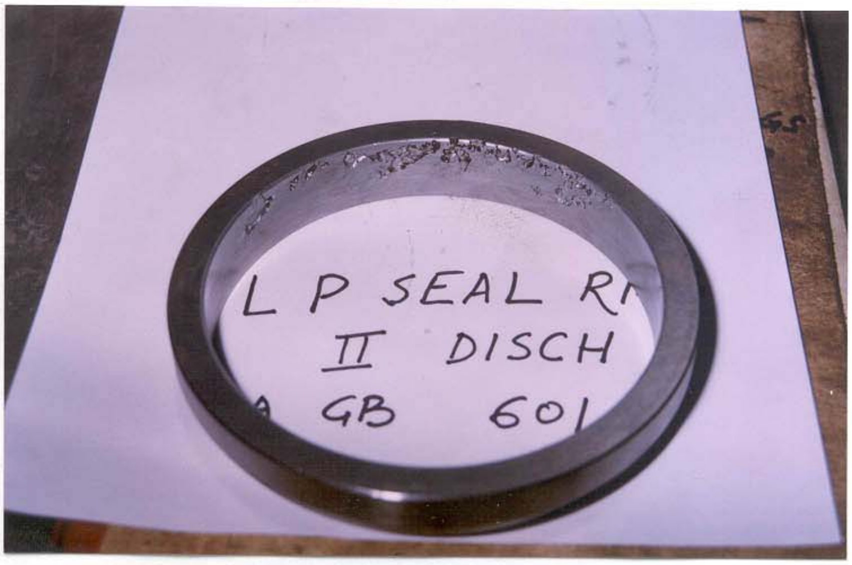
FREQ	PEAK	FREQ	PEAK	FREQ	PEAK	FREQ	PEAK
2100.	0.2368	16252.	0.2355	33796.	0.2367	46389.	0.1905
3131.	0.4246	25377.	0.3120	35788.	0.1965	48189.	0.1951
3900.	1.079	27176.	0.2553	38529.	0.1947	52800.	0.2038
4697.	1.645	31096.	0.2401	41619.	0.2420	55414.	0.2563
5700.	0.2428	33300.	0.1948	45176.	0.2447	58020.	0.2084

ANNEXURE –II

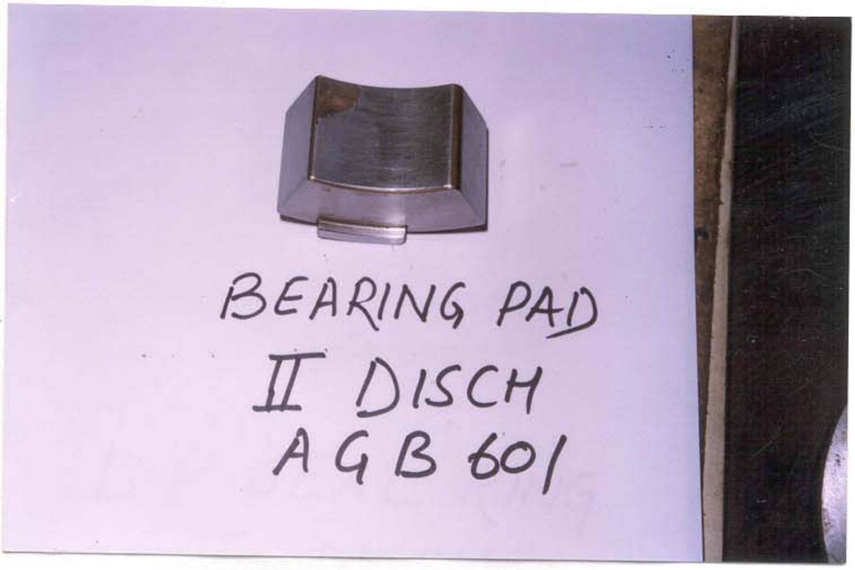


OIL BARRIER TYPE SHAFT SEALING SYSTEM

ANNEXURE-III



OIL BARRIER SHAFT SEAL RINGS
(Showing rubbing & deep pitting)



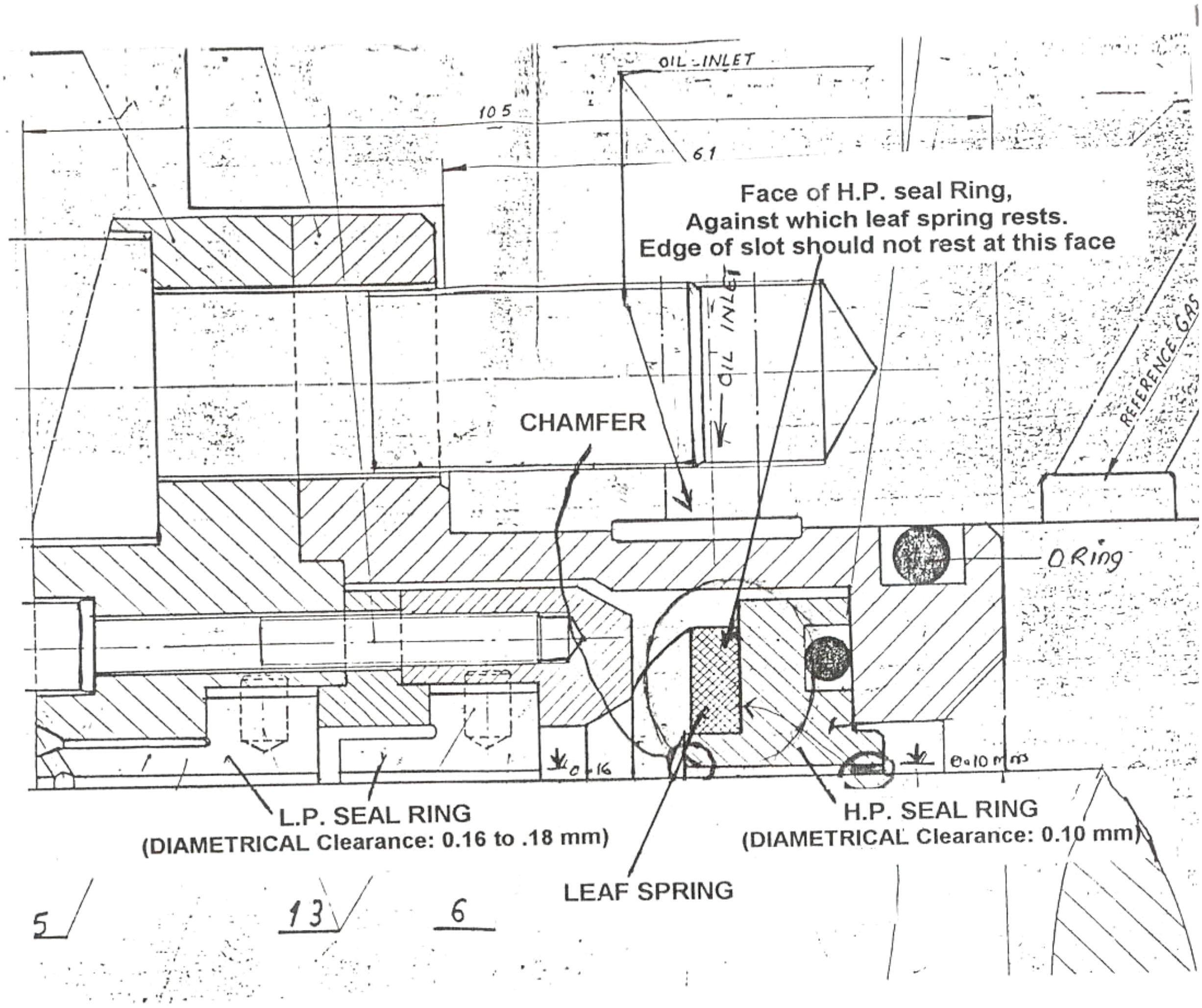
DAMAGED JOURNAL BEARING PAD
(Removed Babit metal at corner)

ANNEXURE-IV

WRONG SEAL ASSEMBLY
(Edge of the Leaf spring resting on seal ring face)

SEAL ASSEMBLY IN RIGHT WAY
(Edge of the Leaf spring NOT resting on seal ring face)

ANNEXURE –V



OIL BARRIER SEAL ASSEMBLY
(MODIFICATIONS FOR OIL WHIRL AVOIDANCE)