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FALL IN EFFICIENCY IN SYNTHESIS GAS COMPRESSOR DUE TO 'O' RING EXTRUSION (A CASE STUDY)

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1. ABSTRACT

The paper highlights the problem of fall in efficiency in high-pressure centrifugal compressor due to extrusion of 'O' ring. Generally such problem of fall in compressor efficiency is attributed to worn-out/damaged interstage labyrinth seals. But O-ring elastomers at times can be the cause of such severe problem resulting in huge energy loss. The compressor efficiency loss due to this problem can be as high as 25%

The paper details the job associated with troubleshooting, rectification, prevention and analysis of problem of low efficiency in Synthesis Gas Compressor at the National Fertilizers Limited, Panipat Unit, India.

2. INTRODUCTION

National Fertilizers Limited has four urea plants in India with a total ammonia capacity of 5400 MT/ day and urea capacity of 9100 MT/ day.

Panipat Plant of National Fertilizers Limited operates at capacity of 900MT of ammonia per day.

The Synthesis Gas Compressor in ammonia plant was manufactured by Nuovo Pignone, Italy design with three barrel type casings and supplied by BHEL, Hyderabad, India. It is driven by steam turbine of Siemens, Germany make and type EHNK-3. Refer Annexure I

2.1 Specifications:

2.1.1 Compressor:

Design: Nuovo Pignone, Italy

Type: Centrifugal, barrel type casings.

Supplier: BHEL, Hyderabad, India.

Speed: 11,100rpm.

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Stages → / Parameters ↓	1st Type: BCL-507	2nd Type: BCL-407a	3rd	Recycle
			Type: 2BCL-407b	
Gas Handled	Syn. Gas (H_2+N_2)			Recycle Gas
Capacity (Dry) Nm^3/Hr.	99,960			4,64,300
Suction Conditions:				
Pressure (kg/cm^2 abs)	37.5	80	146	219
Temperature (Deg.C.)	33	41	41	34
Discharge Conditions:				
Pressure (kg/cm^2 abs.)	81.6	148.5	221	235
Temperature (Deg.C.)	144	133	113	43.5

2.1.2 Turbine Specifications:

Make	Siemens, Germany
Type	Condensing-Extraction Third Generation Turbine
Model	EHNK-40/36/40-3
Supplier	BHEL, Hyderabad, India
Speed	11,100rpm
Power Rating	15983KW
Inlet Steam Condition	100 kg/cm^2, 480 Deg.C.
Extraction; Steam Condition	40 kg/cm^2

2.2 Sealing by 'O' ring elastomers in High pressure Centrifugal Compressor:

The casing of centrifugal compressor is either horizontal split or radially (vertical) split. As per API standard 617, the casing shall be radial (vertical) split, when partial pressure of hydrogen (at max. casing working pressure) exceeds 13.8 bar. Therefore all three-stage casings of Syn. Gas Compressor are of vertical split type or barrel type. In barrel type casing there is a diaphragm pack inside, containing rotor and labyrinth seals. These labyrinth seals are for inter-stage sealing. For sealing between suction and discharge sections formed between diaphragm outer surface and casing bore, there is 'O' ring. For Syn. Gas service generally silicon 'O' rings are used for static sealing. But as silicon is soft with low shore hardness of 50 to 60 (scale-A), hence back up ring of 1.0mm thick of Teflon is used for its reinforcement to prevent its extrusion. (Refer Sketch 2 and Exhibit 1 in Annexure I).

The rotor has seven impellers. The interstage sealing between these seven stages there are labyrinth seals of non-sparking material – Avional-14 (Aluminium alloy).

3. PROBLEM OF FALL IN EFFICIENCY

The second stage of Syn. Gas Compressor was running normally with efficiency of 63% (against design valve of 69.16%) at 10,060 RPM (106% plant load). It was on 20.1.2003, when the discharge pressure of second stage suddenly decreased to 134 kg/cm^2 from 145 kg/cm2. The temperature of gas at discharge of 2nd stage increased to 153^0 C from 127^0 C.

3.1 Decrease in Efficiency

The parameters were measured with standard gauges and polytropic efficiency came out to be 39.28%. (The compressor performance values are indicated in Annexure-II, against design values). The speed was increased by 450 RPM to maintain the machine load.

The outside temperatures of casing also increased.

The parameter remained sustained for about 10 days. The condition slowly started deteriorating further. The speed had to be increased to 10,740 RPM, so as to sustain the plant load. Anti-surge valve of Ist stage was opened by 6% as the discharge pressure of Ist stage was also increasing.

The bearing cap velocities on second stage also increased from 3.0mm/sec to 12.0mm/sec, though shaft vibrations were normal. Depletion of oil in lube oil console was also observed which was believed due to be from oil seals of second barrel.

The steam consumption had increased by 10 MT/Hr, on account of increased speed of compressor.

3.2 Prediction of exact cause

The fall in discharge pressure or ratio of discharge pressure to suction presser could be due to number of reasons:

Bypassing of gas in second stage, i.e. from discharge to suction

Clogging of suction strainer of second stage.

In case of bypassing of gas it could be at three locations: First is through interstage labyrinths, second is through 'O' ring between interstages and third could be through 'O' ring on diaphragm i.e. between diaphragm outside surface and casing bore, which seals suction and discharge sections. Generally, the problem of lower efficiency in centrifugal compressor is due to bypassing through interstage labyrinths and with similar symptoms except the rise in outside casing temperature. The rise was from 83°C to 100°C. This clearly eliminated all other reasons except the bypassing of gas through the 'O' ring between diaphragm other surface and casing bore.

It was decided to stop the compressor to take the job of inspection of internals of compressor. Prolonging with this problem was resulting into energy loss due to bypassing and moreover there were chances of erosion of casing bore or diaphragm outer surface at 'O' ring sealing area, due to high velocity of bypassing gas (at problem area).

3.3 Overhauling of 2nd stage of Compressor

The job of overhauling was taken up on 22.2.2003. A shutdown of plant was taken. The casing was removed, opened, and diaphragm pack was taken out. On opening of casing, complete 'O' ring of diaphragm along with its back up ring was found missing. Only about 10 inches length of 'O' ring and its back up ring was left at bottom end. (Refer Exhibit 2 and 3 in Annexure-III).

Lot of pitting was observed at seating area of 'O' ring on casing. (Refer exhibit – 4 in Annexure-III). The critical dimensions like bore of casing at 'O' ring seating and outside diameter of diaphragm at 'O' ring groove were measured.

Casing I.D.: 810.50 mm. Diaphragm O.D: 809.90 mm.

Width of 'O' ring groove: 9.4mm. Depth of groove: 5.9 mm

O ring thickness: 7 mm. Hence the diametrical clearance between Casing and Diaphragm: 0.6 mm

White metal of discharge end of HP oil seal ring was found to be severely rubbed.

Job done: Thorough cleaning of rotor, casing and diaphragm. All clearances of Lab. Seals (of interstage, balance drum and gas seals) were measured and found to be within permissible values. Clearance of both end journal bearings was found within permissible values. The diaphragm was assembled back with new parting plane 'O' ring.

3.4 Work done for Rectification of Problem

As cause of problem was the extrusion of 'O' ring, which happened due to lack of nip or squeeze (compression) in 'O' ring. This was due to increase in diametrical clearance between casing bore and outer dia of diaphragm, because of damage/pitting on seating area of casing bore. The clearance was 0.6 mm against maximum design value of 0.2 mm.

Hence the complete rectification meant replacement of casing with new casing. Or repair of damaged portion of casing. No spare casing was available and generally is not kept in inventory, being a high cost item with minimum probability of damage. The repair of damaged portion of casing involved lengthy activities like preheating for long time, undercutting of damaged portion, building up by welding or specialized metal spray and then precision machining on special CNC boring machine, maintaining the bore dimension within close tolerances. The casing was exposed to hydrogen gas for a very long time. Hence there are all chances of hydrogen diffusion in casing material. Moreover such pitting areas have lubrication oil soaked in it, over for a long period. This oil is present due to some seal oil going along with gas through floating type oil seal rings used for end sealing of compressor casing. This make preheating essential for longer time and welding process becomes difficult.

In fact this repair was a lengthy job involving minimum 7 to 8 days and hence resulting into a long shutdown of machine and plant, leading into a big production loss.

Hence after technical deliberations, it was planned to rectify the problem by increasing the nip or compression in 'O' ring with extra thickness of back up ring (Teflon ring), along with the 'O' ring. Originally as per design we had back up ring of 1.0mm thickness. We provided one more back up ring with thickness of 0.5mm along with 1.0 mm back up ring (towards low pressure side). Two rounds of Teflon tape were provided below the 'O' ring. Refer sketch-3 and exhibit-5 in Annexure–IV.

All above activities increased the nip or compression of 'O' ring.

The reason of damaged H.P. seal oil ring was vibrations in rotor, which was due to the instability caused by high flow of gas by bypassing gas. The system was insufficient to dampen the vibrations due to increased flow. The signatures of vibrations were also indicating vibrations at higher frequencies i.e. flow related vibrations. The diaphragm back was put inside the casing. Bearings and oil seals were assembled. New discharge side H.P. oil seal rings were used for assembly. The barrel casing was put back in position and aligned w.r.t. Ist and IIIrd stage casings.

Results after the overhaul:

After overhaul, the machine was rolled on 24th Feb. 2003. All parameters of 2nd Barrel were restored and near to design valves.

	Before overhaul	After overhaul
Pressure Ratio (P2/ P1):	1.55	1.7745
Polytropic efficiency:	39.28%	63.86%
Steam consumption:	208 MT/Hr.	198MT/Hr.

Saving in Steam: About 10 Tons/Hr.

The total time taken in the rectification job was 50 hours (about 2 days). Hence it avoided a long shutdown of 7 to 10 days if the casing repair job had been taken.

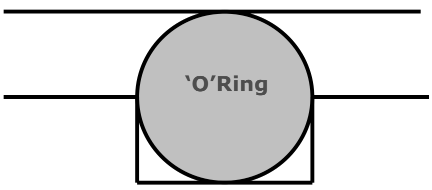
The compressor is performing satisfactorily after that. The casing repair job has been planned in coming planned plant annual turnaround, where we shall have ample time (float) to execute this long repair job.

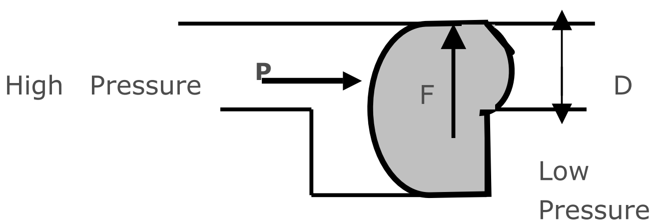
4. GENERAL DISCUSSIONS

Generally in case of fall in efficiency of compressor, the operation and maintenance Engineers of Centrifugal Compressor doubt on bypassing through interstage labyrinths. But 'O' ring elastomers can also play an important role and can be the cause of such high fall in efficiency of compressor. Hence assembly of 'O' ring and taking due precautions for 'O' ring (like centering of diaphragm), while putting and assembling the diaphragm pack in casing are very important activities in compressor overhauling process.

The Nip or Compression in 'O' ring:

The problem of 'O' ring extrusion in subject case was due to lack of nip or compression in 'O' ring due to increased diametrical clearance between diaphragm pack and casing. This was due to damage / pitting in casing bore at 'O' ring sealing area. This pitting has been observed in number of sites in such compressors, which have been operating for longer time of more than 20 years and have floating type oil seal rings for end sealing of compressor. There is always small carry over of oil along with the gas even with good condition seals. With the time this oil gets stagnated in area adjacent to 'O' ring of diaphragm. This stagnated oil results into fretting corrosion, causing pitting in such area. Such pitting with the time destroys the 'O' ring seating area on casing, thus 'O' rings loses its nip. This lack of nip in 'O' ring no more can generate sealing force, which should be essentially more than the pressure to be sealed.





D :Diametrical clearance

P: Pressure to be sealed

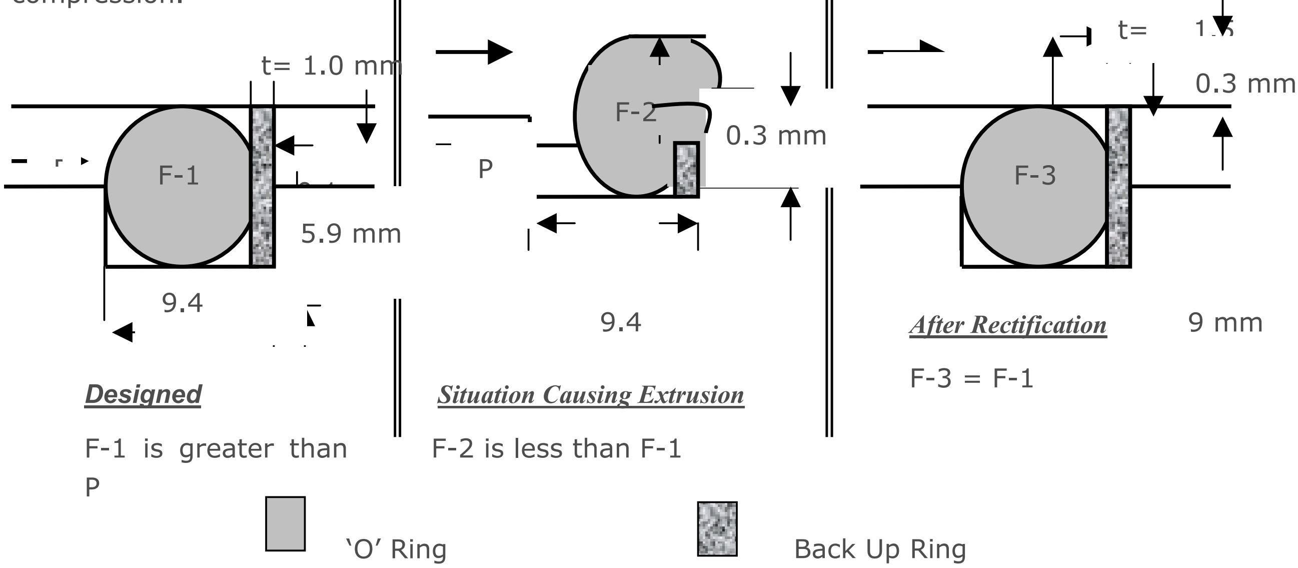
F: Sealing force

Hence in such situation, the pressure of gas from high-pressure side can easily extrude such 'O' ring and finds its way for bypassing towards low-pressure side. The actual compression or nip in 'O' ring should be 15% to 30% depending upon pressure to be sealed. There are standards like BS-4518 with various guiding factors and data relating to ‘O’ ring sealing. In the subject case 7mm dia of silicon ‘O’ ring should be compressed to reach dia 6.025mm to 6mm after assembly. As designed clearance between casing bore and diaphragm pack is 0.20mm (max.)

Designed casing I.D. in m.m.= $810^{+0.05}$

<u>Designed O.D. of diaphragm in m.m</u>. = $810^{-0.10}$

But due to increase in clearance to 0.6mm the ‘O’ ring could become 6.2mm after the compression.



We wisely increased the nip or compression of ‘O’ ring by providing extra back up ring, due to which the sealing force generated was more than the pressure to be sealed and hence was able to provide successful sealing without any extrusion.

Though this provided solution to problem within a very short shutdown of machine but chance cannot be taken to continue with this unconventional repair for a very long time. Hence the permanent solution involving casing repair shall be done during planned Annual Turnaround of the plant.

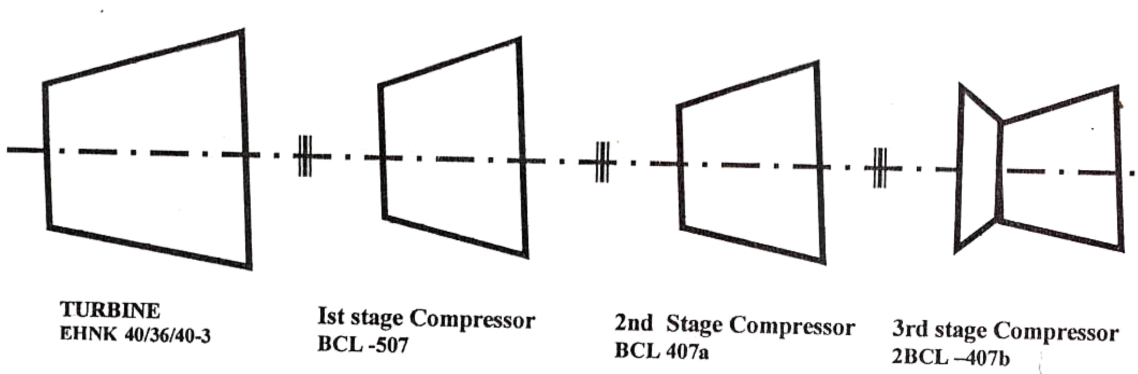
Trouble shooting of the problem in short time and arriving to optimum solution to the problem should always be desired approach of the maintenance engineers, specifically in today’s competent and challenging environment of high breakeven points of plants, economic recession, demanding high productivity and energy conservation.

REFERENCES

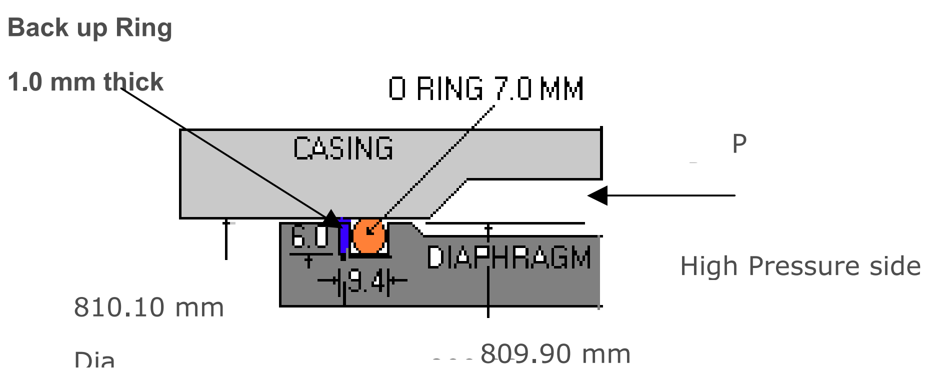
API Standard 617: For Centrifugal Compressor for General Refinery Service.

Seal and Sealing Handbook, by R.H. Warring.

ANNEXURE I



Sketch -1



Sketch -2

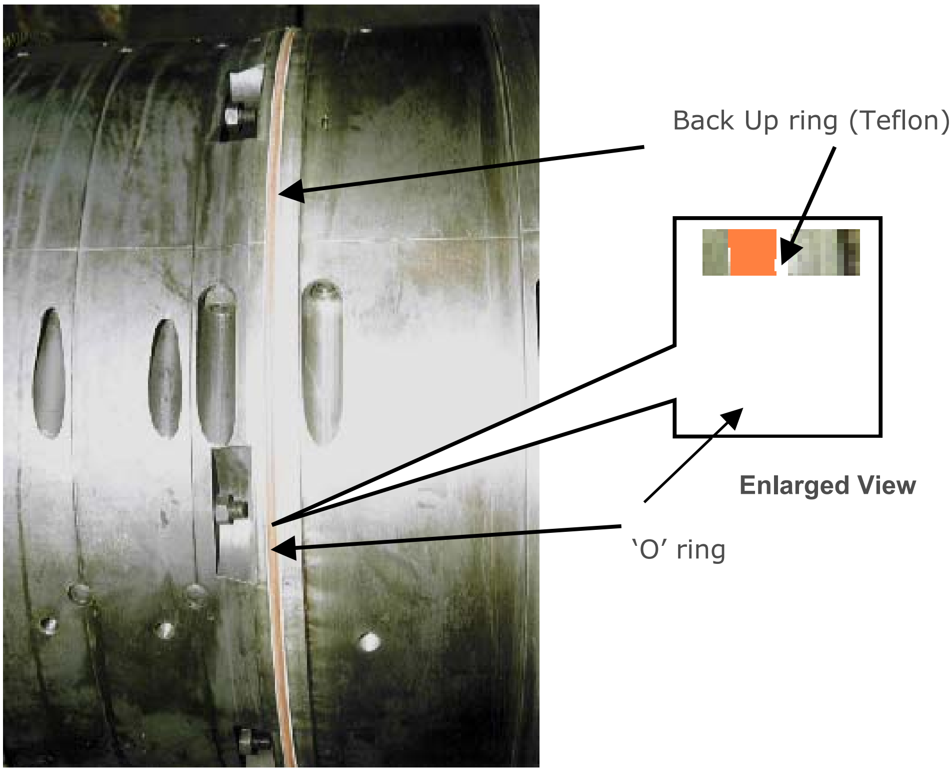


Exhibit -1

ANNEXURE II

DATA REGARDING PERFORMANCE OF COMPRESSOR and TURBINE DURING PROBLEM and AFTER THE RECTIFICATION, PLACED AGAINST THE DESIGN VALUES.

S.NO		UNIT	DESIGN (CALCULATED)					ACTUAL 27.02.2003				ACTUAL 21.01.2003		
							After the rectification				Parameters during the problem			
			I	II	III	IV	I	II	III	IV	I	II	III	IV
1	FLOW	KG/HR	38088			190773	42611	44574	40154	143076	42517	45830	40790	163283
2	P abs	KG/CM2												
	SUCTION		37.5	80	146	219	38.789	81.989	140.989	190.489	38.989	86.989	130.489	181.489
	DISCHARGE		81.6	148.5	221	235	84.489	145.489	192.989	198.489	88.189	134.989	182.989	193.489
3	P 2 / P 1		2.176	1.8563	1.514	1.0731	2.1782	1.7745	1.3688	1.042	2.2619	1.5518	1.4023	1.0661
4	T abs	DEG. C												
	SUCTION		306	314	314	307	302	309	306	306.5	295	309	309	301
	DISCHARGE		417	406	386	316.5	411	400	366	314	435	426	375	311
5	T 2 / T 1		1.3627	1.293	1.229	1.0309	1.3609	1.2945	1.1961	1.0245	1.4746	1.3786	1.2136	1.0332
6	K average		1.4018	1.4031	1.406	1.4285	1.402	1.4033	1.4058	1.4229	1.4016	1.4025	1.4051	1.4244
7	(K-1)/K		0.2866	0.2873	0.289	0.3	0.2868	0.2874	0.2887	0.2972	0.2865	0.287	0.2883	0.2979
8	(n-1)/n		0.3981	0.4154	0.498	0.4322	0.3959	0.4501	0.5703	0.5876	0.4758	0.7307	0.5725	0.5105
9	Comp. Efficiency (Poly)	%	72.00	69.16	57.95	69.4	72.44	63.86	50.62	50.58	60.22	39.27	50.36	58.37
10	HEAD (Poly)	M	28636	23340	15886	2357	28417	21413	11616	1427	29345	16388	12302	2005
11	POWER	KW	4127	3502	2844	1765	4553	4072	2510	1100	5644	5210	2714	1528
12	TOTAL POWER	KW		12238				12235				15096		
	(Compressor Shaft)													
13	STEAM FLOW	KG/HR		170000				201090				208410		
14	EXTRACTION FLOW	KG/HR		145000				156090				149910		
15	EXHAUST FLOW	KG/HR		25000				45000				58500		
16	STEAM RATE	KG/KWH		13.89				16.44				13.81		
17	OVERALL EFFICIENCY	%		96.51				69.34				71.91		

ANNEXURE III

EXHIBIT 2

Damaged 'O' ring of diaphragm

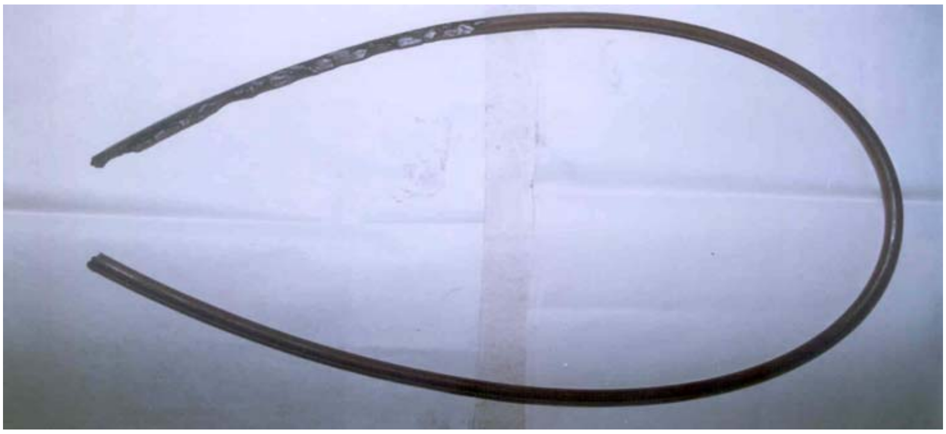


EXHIBIT 3

Damaged back up ring

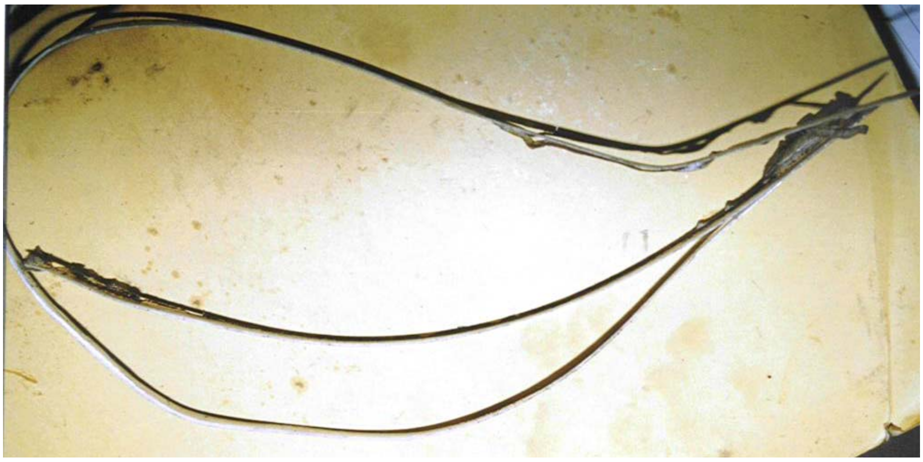
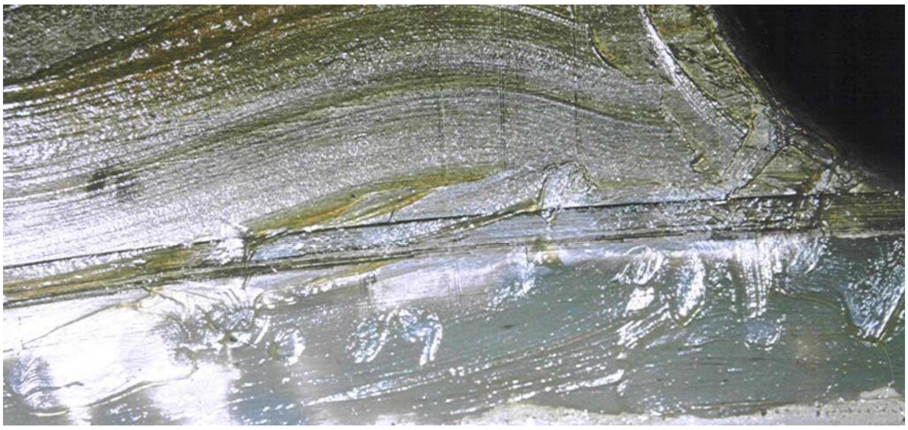


EXHIBIT-4

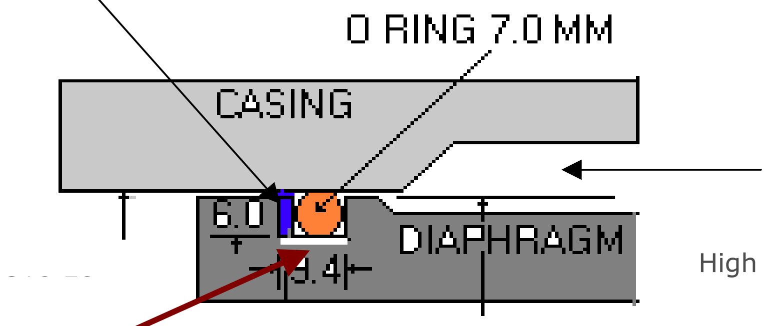
Pitting damage on casing at "O" ring seating



ANNEXURE IV

Back up Ring

1.0 = 0.5 = 1.5 MM thick



High Pressure side

809.90 mm

Teflon tape (2 rounds)

<u>Sketch 3</u>

'O' Ring

Back Up Ring

1.0 + 0.5 =1.5 mm

EXHIBIT 5

Fitting of New 'O' Ring and back up rings on Diaphragm