ISSN: 2319-5967
ISO 9001:2008 Certified
International Journal of Engineering Science and Innovative Technology (IJESIT)
Volume 2, Issue 2, March 2013

Root Cause Analysis to Identify the Problem Causing Turbine Failure

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Abstract— The root cause analysis method consists of three steps: target problem detection, root cause detection and corrective action innovation. In this paper a root cause analysis method to identify a turbo-driven pump failure case occurring in leading petrochemical company is presented. The turbo-driven pumps are one of the primary equipment that pumps steam into the boiler in order to generate the required amount of power to run the plant. The root cause is identified using a 'Symptom' based approach of the Root Cause Analysis method where in the symptoms are first focused. Various industrial engineering tools are used and finally a feasible and acceptable solution is identified.

Index Terms—**Root Cause Analysis, Moisture Deposition, Turbine, Symptom, Rust, Maintenance.**

I. INTRODUCTION

The key for effective problem prevention is detecting the causes of a problem, followed by a structured investigation of the problem to identify which underlying causes need to be fixed. Root cause analysis (RCA) is a method that is used to address a problem or non-conformance, in order to find the root cause of the problem. It is used to correct or eliminate the cause, and prevent the problem from recurring. Resolution of customer complaints and returns; disposition of non-conforming material (scrape or repair) via the material review process; corrective action plans resulting from internal and customer audits etc. are the traditional applications of root cause analysis. Root cause is the fundamental breakdown or failure of a process which, when resolved, prevents a recurrence of the problem. Root cause analysis is therefore a systematic approach to get to the true root cause of process problem. Most of the industrial cases in chemical industry root cause analysis have aimed to lower the defect rates by preventing the causes of the most typical types of the defects. The RCA method has proved promising and resulted in high quality corrective actions. This paper presents a turbo-driven pump failure issue onto which the root cause analysis method will be applied to identify the root cause of the problem. The remainder of this paper is organized as follows. The problem description is given in section 2. The section 3 describes the methodology and section 4 gives the results and discussions. The concluding remarks are given in section 5.

II. PROBLEM DESCRIPTION

The captive power plant of the petrochemical division has three boilers of capacity 60 tonnes/hr. each. Steam is generated at 110 kg/cm^2 abs. and 510^{o}C. During normal conditions, two boilers are operated, providing a total of 110.227 tonnes of steam per hour. During periods of maximum power requirement, all three boilers are put to operation, producing 151.116 tonnes of steam per hour. Even though the capacity of plant is 16MW, only 8MW of power is generated for daily use. Full capacity of plant is utilized when there is shortage of power from the electricity board supply. For producing 15MW of power, steam requirement is 129 tonnes per hour. During normal operation (for 8MW), steam requirement is 96.5 tonnes per hour. There are 3 turbo driven feed pumps and 2 motor driven feed pumps to facilitate the supply of steam into to the primary turbine. Usual procedure for generating 8MW of power is by running a single turbo driven and a motor driven turbine during normal conditions. But during power failure or if the nearby plant is in need of additional steam, all the turbo driven pumps are to be used in order to generate the required power and steam. During such situations it is necessary for all the pumps to be functioning properly in order to avoid a drop in power output or, in certain cases, to avoid production loss. Hence these 5 turbines play an important role in the production line. The problem is identified in one of these pumps which is turbo driven. The company was in need of additional steam as the primary turbine in the neighboring plant was shut down for maintenance, but since this turbo driven pump was showing abnormal vibrations, it had to be stopped for inspection. Hence, over all, there was shortage of steam and power. A single stage impulse steam turbine is used to

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run this pump. The turbine is an APE Belliss type SSA single stage steam turbine of impulse design with two rows of moving blades and one row of fixed blade. The steam enters the turbine through a set of nozzles and after expansion in the nozzles, steam passes into the first row of moving blades. The fixed row of blades reverse the steam flow, directing it back into the second row of moving blades and then to the exhaust. The rotor assembly is carried out by two oil lubricated journal bearing of sleeve type with replaceable liners and comprising of steel shells with white metal lining. The axial thrust is taken up by four point angular contact ball thrust bearing. The turbine shaft is sealed where it passes through the casing by a Carbon ring gland, consisting of carbon rings backed by corrosion resistant springs. A leak off connection for leakage steam and water is provided. Turbine is equipped with an emergency valve and a separate governing valve. Opening of the governing valve is controlled by a direct acting hydraulic package governor. The problem with the pump was an abnormal vibration and had to shut it down for inspection. The vibration was observed at the turbine casing side, which means there was fault at the turbine side. Hence the casing was removed. The problems detected were:

- Material removal from shaft surface due to high vibration
- Broken carbon ring
- Rust on turbine blades
- Deposition of rust particles on the impeller surface leading to unbalance and vibration
- Rust particles present in the lubrication oil

The shaft, impeller and the carbon rings had to be replaced with a new one immediately since the pump had to be in running condition as soon as possible. But the root cause to this problem was left unidentified. The premature rusting of internal parts leading to high vibration had to be investigated in order to find the root cause to this problem. Turbine Specification is given in table I.

Table I: Turbine specification

Power	580KW
Inlet Steam Pressure	40-45 ATA
Inlet Steam temperature	267-410°C
Exhaust Steam Pressure	5 ATA
Speed	5000 rpm
Over speed trip setting	5500 rpm
Maximum exhaust	9.176 Kg/hr

III. METHODOLOGY

Symptom is defined as a characteristic sign or indication of the existence of an undesirable situation or a bodily disorder. It is a phenomenon or circumstance accompanying a disorder and serving as evidence of the disorder. The primary symptom identified was the abnormal vibration. In the symptom based approach, the first step is data collection based on the primary symptom. Every machine is unique and has a unique characteristic behavior. In order to find out the root cause of a disorder, it is important to know the history of the machine through its log book. Through guided brainstorming the various causes of vibration and various types of failure were identified. The working process to identify the causes is shown in Fig. 1. After the brainstorming sessions, final problems identified were premature failure of journal bearing liners; material removal from the surface of the shaft; increasing trend in vibration on the impeller side; soft foot on the pump side; unbalance due to presence of foreign particles; aging of various parts in the turbine assembly; lack of skilled labors; lack of proper equipment to do maintenance; and lack of raw materials due to insufficient fund.

The maintenance log book of the turbine was also referred in order to go through the machine history. Several problems related to equipment, machine aging, material removal, raw material shortage and lack of skilled labor issues were mentioned in the log book.

The cause and effect diagram (Fig. 2) was drawn to list out problems causing the failure of the turbo-driven pump. Here the problems were categorized into fine main sections: material, equipment, machine, labor and aging. These five sections form the side bone leading to the center bone, which points towards the head representing the end effect ‘Failure of Turbine’. Each side bones have secondary bones which represent the problems or the causes within the primary bone. The various reasons for failure of the turbo driven pump are also listed in table II.

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ISO 9001:2008 Certified
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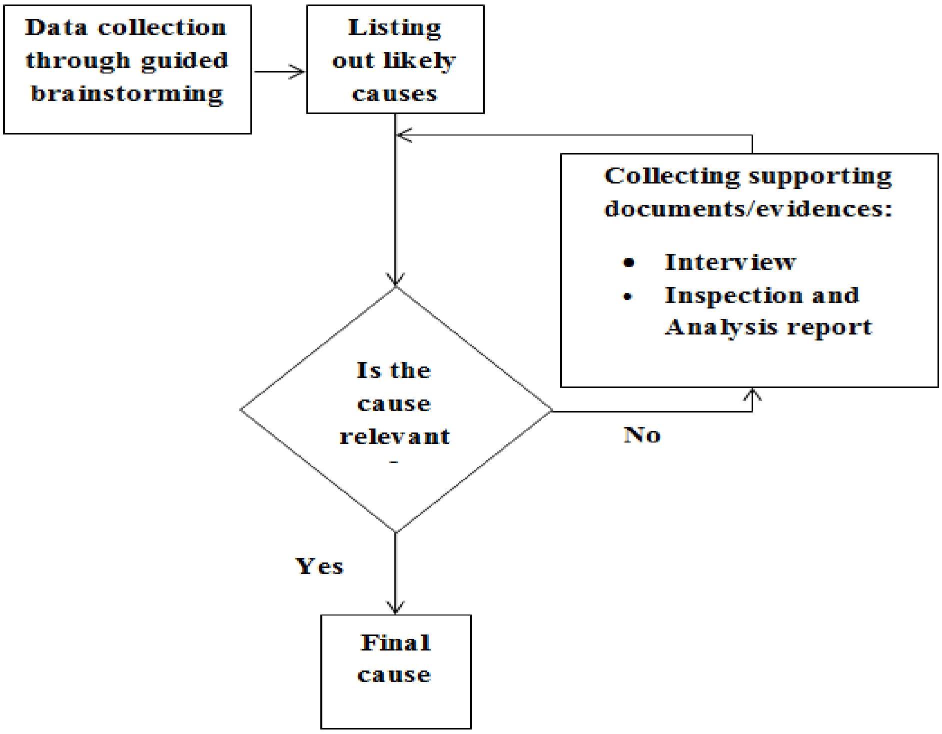

Fig. 1 working process to identify final causes

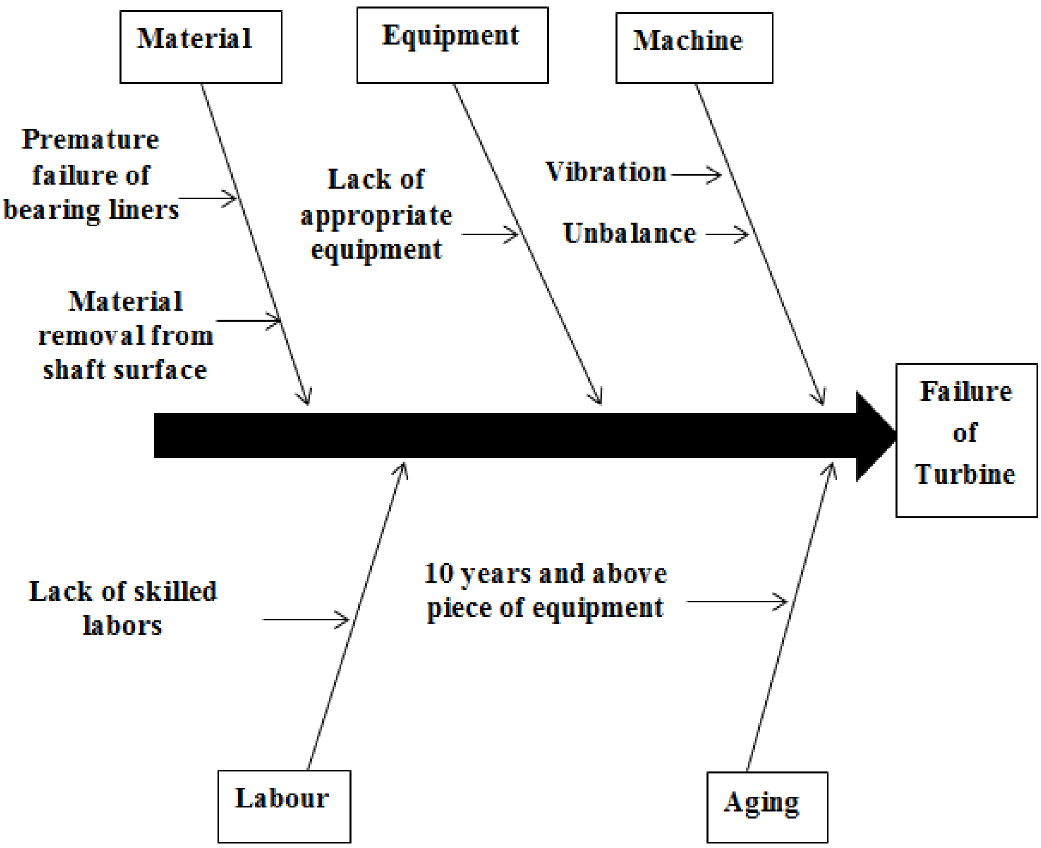

Fig. 2 General cause and effect diagram

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Table II: Reason for failure of turbo-driven pump

Observations	Occurrence
Vibration	45
Misalignment	28
Unbalance	34
Emergency trip valve activated	1
Soft foot	10

A Pareto chart (Fig. 3) was constructed in order to highlight the major problem causing the failure of turbo-driven pump and it is clear that vibration and unbalance together contributes to 66.94% of the problem.

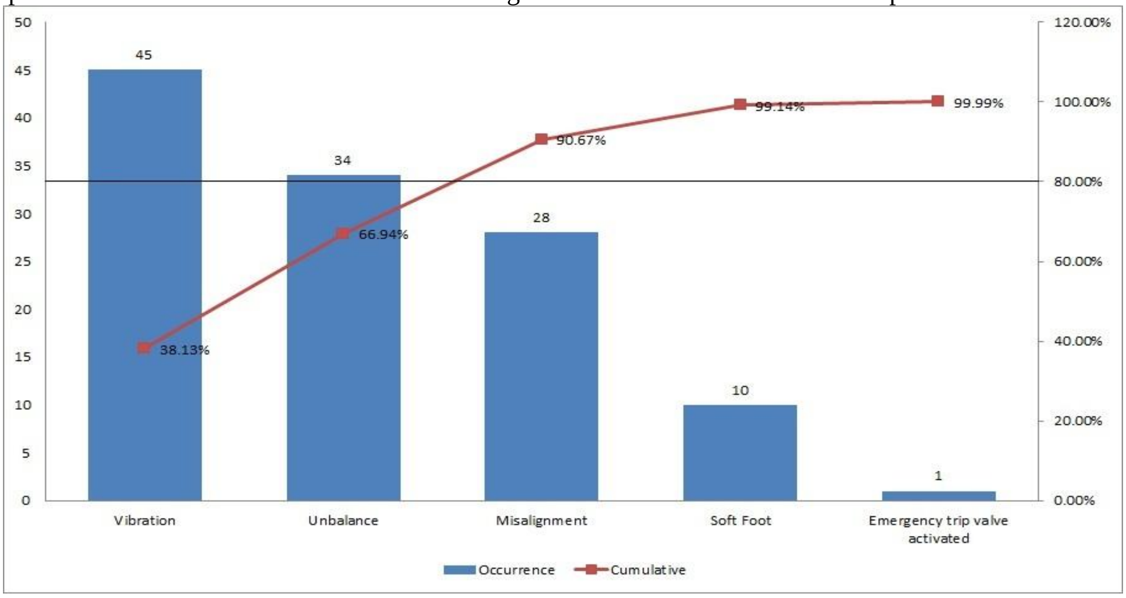

Fig. 3 Pareto chart

Cause and effect diagram is drawn to list out the problems causing vibration and unbalance (Fig. 4).

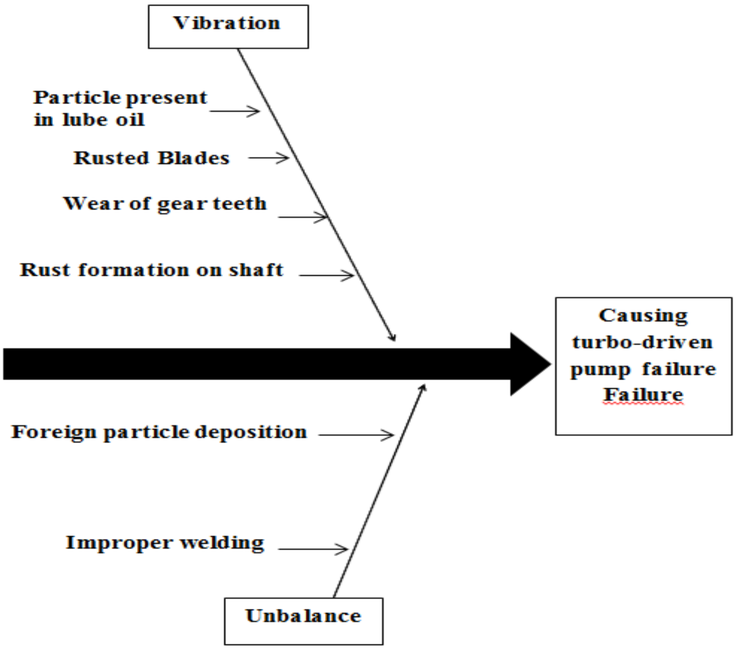

Fig. 4 Cause and effect diagram for vibration and unbalance

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The reasons for vibration and unbalance are listed in table III.

Table III: Observation table considering machine failure

Problem Description	Occurrence
Foreign particle in lube oil	4
Rusted blades	23
Wear of gear teeth	7
Rust formation on shaft	24
Foreign particle deposition	19
Improper welding	2

A Pareto chart was constructed using the data from table III which is shown in Fig. 5.

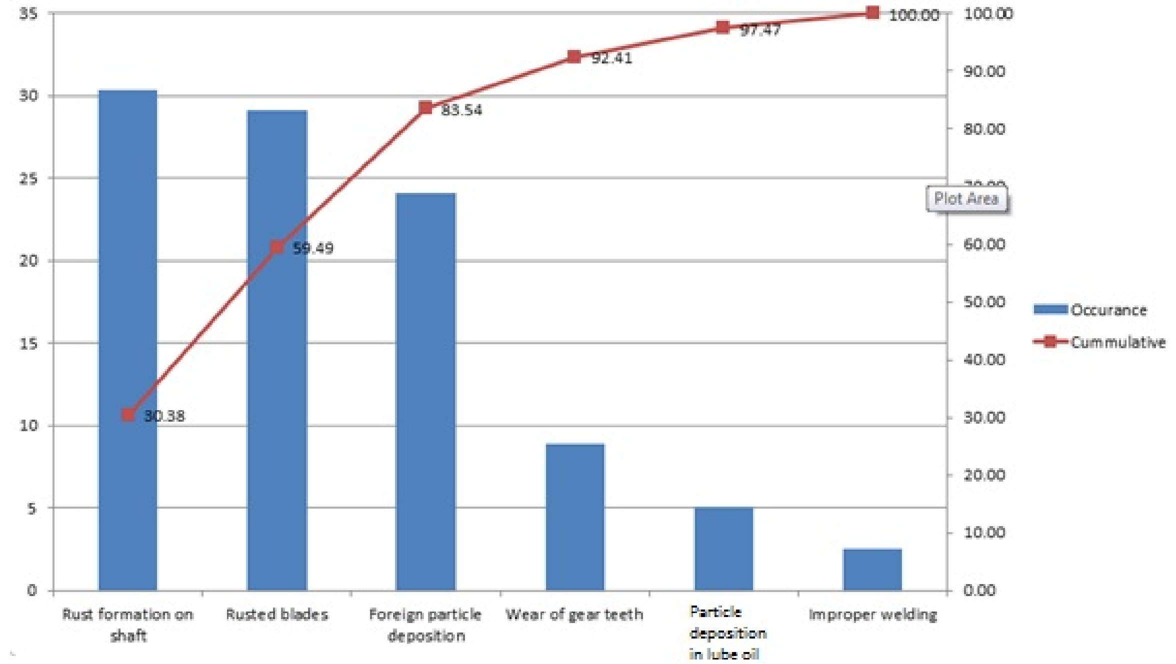

Fig. 5 Pareto chart considering machine failure

From the Pareto chart drawn (Fig. 5) it is clear that the problem caused due to rust issues are more. Rust formation on shaft, Rusted blades and foreign particle deposition together contributes to 83.54% of the problems. The foreign particles were found to be rust particles. These rust particles deposited on the surface of the shaft and blades causing improper balance and heavy vibration leading to failure of various parts of the turbine. Moisture deposition is the primary reasons causing the rust formation. Study was conducted in order identify the possible ways of moisture deposition and rust formation. Inside the turbine a gland box is used to prevent the entry of steam to the critical parts of the turbine. The primary part that prevents steam from entering the bearing and other compartment is this gland box. Hence there is a possibility that the gland box has failed. Further study was conducted on the carbon ring gland box. The carbon ring gland box is one of the most inexpensive and efficient gland box [1]. The gland box used in this turbine utilises tenon joint carbon rings. The problem with these carbon rings is that, the coeffiecent of thermal expansion is half of that of the steel shaft. Which means it will take more time to set when compared to the shaft [3]. Due to this problem, the ring is designed with the clearance to cop up with this issue. But that alone will not do. It is to be made sure that proper servicing and maintenance must be carried out in order to ensure the proper functioning and long lasting life of the part. The general practice with carbon ring glands. relative to removing the steam that does leak through the gland, is to bleed the leakage off the shaft through a leaf-off, and to depend one two or three carbon rings to prevent excessive amounts of steam to get past the last ring. Anything that will elevate the pressure in the gland at the leak-off location such as, restrictive leak-off piping, will increase the leakage past the gland and raise the dewpoint where the air is drawn into the bearing housing. Vaccuum removal at the leak-off, obtainable with eductors or gland condensers, is not generally considered necessary with carbon rings. Nevertheless, these methods would be practical except for the added installation and operating expenses.

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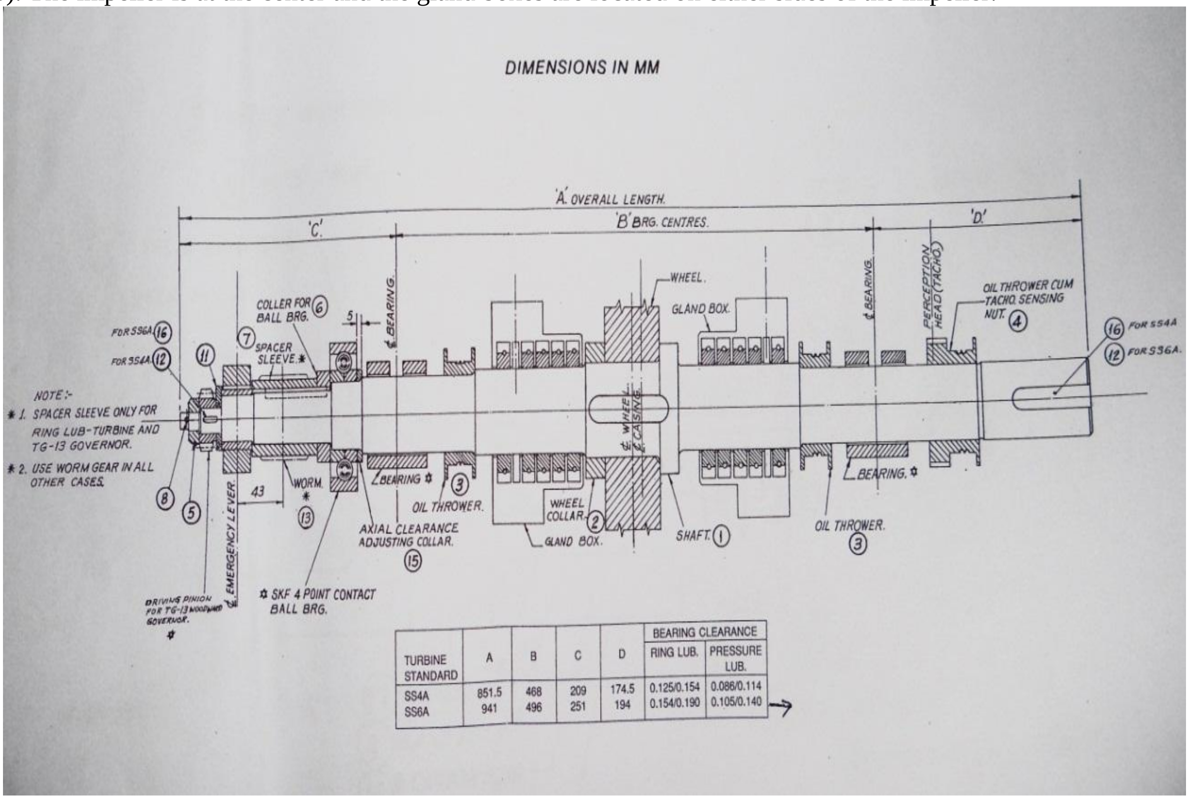
The carbon ring takes approximately 12 hours to set under 140°C temperature and a speed of 2500-3500 rpm. Else the ring will fail to prevent leakage. Also use of new carbon rings on old shaft and viz versa is not recommended since the clearance will be different. If a ring is replaced, the clearance must be caliberated for the present shaft. The proper procedure must be followed for its proper functioning. From the maintanence log book it is clear that the servicing and maintenance of the machine was improper. The gland ring must not be aloud to cool down. If at all it happens, then before start up, it must be ensured that the gland is maintained at a temperature of 140°C for at least 3 hours. Later after start up, due to increase in temperature, speed and pressure the ring will be set properly. Hence the possibilty of malfunction of carbon ring was identified. The main shaft assembly is shown below (Fig. 6). The impeller is at the center and the gland boxes are located on either sides of the impeller.



TURBINE STANDARD	A	B	C	D	BEARING CLEARANCE RING LUB.	BEARING CLEARANCE PRESSURE LUB.
SS4A	851.5	468	209	174.5	0.125/0.154	0.086/0.114
SS6A	941	496	251	194	0.154/0.190	0.105/0.140

Fig. 6 Main shaft assembly of the turbo-driven pump

IV. RESULTS AND DISCUSSIONS

The various possible solutions to the current problem were found as follows:

a) Alternative for carbon ring box:

The alternative for carbon ring gland box is a carbon labyrinth gland box which is a hybrid model consisting of the labyrinth design installed with carbon rings. The labyrinth design is used in high speed high temperature turbines. Coupling the labyrinth design with the carbon ring seal will ensure proper sealing. The process of installation and initial start-up will take 10 days. But considering the company’s situation, it is not possible for them to install this setup as it is not economical and practical for them. The current layout of the turbine is in such a way that the carbon gland cannot be replaced with a new design. It also involves design of labyrinth carbon gland box based on the specifications of the turbine which will cost more than just replacing the ring every time.

b) Alternative for tenon joint carbon ring:

The alternative for tenon joint carbon ring (Fig. 7) is the wedge rings (Fig. 8). The wedge rings can be designed based on the specification of the shaft and the gland box. Replacing the current carbon rings with the wedge rings will help improve the sealing efficiency.

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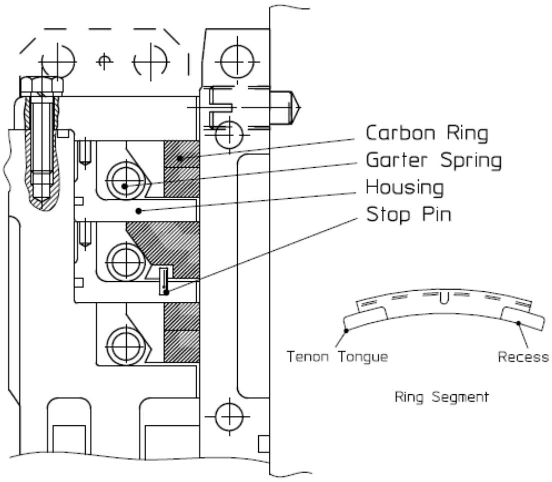

Fig. 7 Tenon joint carbon ring (arranged in series) [7]

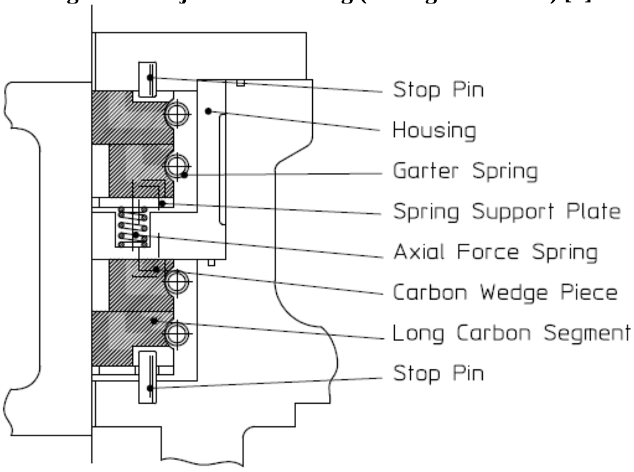

Fig. 8 Carbon wedge rings (arranged in series) [7]

The wedge type rings shown in fig. 8 typical of carbon gland ring arrangements installed at many hydroelectric power stations. Each ring is made up of segments. A garter spring seated in a groove around the outside circumference of the ring holds the assembly of segments and wedges together. Wear on the carbon is taken up by the action of the spring and water pressure continually forcing the segments into contact with the shaft, while the wedges move out-wards. Thus wearing of the carbon segment takes place while they still maintain an effective seal. Considering the current financial capabilities of the company, the solution found must be cheap and efficient.

Technical Feasibility

The wedge type rings are available in the market that fits the specifications given by the petrochemical company. The comparison of technical specifications of the current tenon joint carbon ring and the proposed wedge type carbon ring [9] is shown below.

Table IV: Comparison of tenon type and wedge type carbon ring

Specification	Tenon joint carbon ring	Wedge carbon ring
Tensile Strength (MPa)	8-11	8-17
Compressive strength (MPa)	50-90	70-110
Temperature range (^{o}C)	260 - 450	265 – 550
Working Speed (rpm)	Up to 5000	Up to 7000
Friction Coefficient	0.310	0.149
Life (hours)	10000	50000
Grade	LINK CY2T	LINK CY2WA (higher grade than LINK CY2T)

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ISO 9001:2008 Certified
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Table IV shows that the wedge carbon ring is a better alternative to the tenon joint carbon ring. Therefore replacing tenon joint carbon rings with wedge carbon rings is technically feasible.

Economic Feasibility:

a) The Labyrinth carbon gland box available in the market cost Rs.19,00,530/- approximately, inclusive of installation and labour charge [6]. It is 10% larger than the carbon ring gland box. Installation takes 20-25 days since the entire layout must be changed to accommodate the labyrinth gland box. The company's annual maintenance shutdown duration is 10-15 days. According to their current economic condition they cannot afford to lose more than 15 days of shutdown. Hence the installation of labyrinth gland box is not economically feasible.

b) The wedge carbon rings provided by the company costs Rs.320 per piece [7]. It takes 8-10 days for installation of the wedge carbon ring which fits into the maintenance shutdown duration of 10-15 days.

Loss to the company due to the use of tenon type carbon ring is calculated below:

The cost of a single carbon ring	=	Rs.120
Total number of carbon rings	=	8 Nos.
Number of turbo-driven pumps	=	3 Nos.
Total cost of replacing carbon rings every year	=	Rs.120x8x3
	=	Rs.2880/-
Loss to company due to turbine failure	=	Rs.19,25,280/-
Repair cost of turbine per annum	=	Rs.12,16,674/10 years
	=	Rs.1,21,667.4
Manufacturing and labour charge	=	Rs.1,30,247/-
Hence total loss due to turbine failure per annum	=	Rs.21,77,194.40

The profit when replacing tenon joint carbon ring with wedge carbon rings is calculated below:

Cost of modifications (including labour charge)	=	Rs. 2482
Total cost of wedge carbon rings	=	Rs.320 x 8
	=	Rs.2560
Cost of single seal housing	=	Rs. 14,800
Number of seal housings	=	2 Nos.
Total cost of implementing seal housing	=	2 x 14,800
	=	Rs. 29,600
Labour charge	=	Rs. 11,450
Cost of installation in one turbo-driven pump	=	2482 + 2560 +29,600 + 11,450
	=	Rs. 46,092
Total cost of implementation (in all 3 turbo driven pumps)	=	3 x 46,092
	=	Rs. 1,38,276/-
Increase in profit during the year of implementation	=	21,77,194.40 – 1,38,276
	=	Rs.20,38,918.40/-

Therefore replacing tenon joint carbon rings with wedge carbon rings is economically feasible as well.

V. CONCLUSION

A root cause analysis was conducted successfully to identify the problem causing the failure of a turbo-driven pump in a petrochemical company. Industrial engineering tools like Pareto chart and the cause and effect diagram were used to identify the root cause.. Recommendations were also given to eliminate the root causes of the failure identified. Feasibility study was conducted to identify the feasibility of recommendations.

REFERENCES

[1] William L Coleman, "Water contamination in steam turbine", Proceedings to the Seventeenth Turbo-Machinery Symposium, Huston, Texas, 1993.

[2] Heinz P. Bloch, Fred K. Geitner, "Major Equipment Maintenance and Repair", Gulf Publishing Company, Second Edition, Volume 4, 1997.

[3] http://reliabilityweb.com/index.php/articles/Eliminating_Moisture_Damage_to_Bearings_on_Critical_Steam_Turbines/ (Assessed on 11.10.2012).

[4] Otakar Jonas and Lee Machemer, "Steam Turbine Corrosion and Deposits Problems and Solutions", Proceedings to Thirty-Seventh Turbo machinery Symposium, 2008.

ISSN: 2319-5967
ISO 9001:2008 Certified
International Journal of Engineering Science and Innovative Technology (IJESIT)
Volume 2, Issue 2, March 2013
[5] Philip Dowson, Derrick Bauer and Scot Laney, "Selection of materials and material related processes for centrifugal compressors and steam turbines in the oil and petrochemical industry", Proceedings to Thirty-Seventh Turbo machinery Symposium, 2008.
[6] http://www.weiku.com/products/4964411/Gland_Rings_Packaging_And_Bearings.html, (Assessed on 07.10.2012).
[7] http://www.morganamt.com/gb/customer-service/literature/, (Assessed on 10.10.2012).
[8] http://books.google.co.in/books?id=IpBaybUJ5T8C&pg=PA343&lpg=PA343&dq=carbon+ring+gland+box&source=bl&ots=WIBr_aa7mB&sig=QVT8gLLfxzvqa2YKUTtyueKK6NI&hl=en&sa=X&ei=7SJLUYLrIoHKrAeDpIHgAQ&ved=0CHoQ6AEwCQ#v=onepage&q=carbon%20ring%20gland%20box&f=false (Assessed on 07.10.2012)
[9] https://www.google.co.in/url?sa=t&rct=j&q=&esrc=s&source=web&cd=4&cad=rja&ved=0CEwQFjAD&url=http%3A%2F%2Findustrial-supply.exportersindia.com%2Fcarbon-products%2Fcarbon-rings.htm&ei=7SJLUYLrIoHKrAeDpIHgAQ&usg=AFQjCNGidUOmWDne_RcANOXKtu0BU52cFQ&sig2=83aFGxTA3T0bgMrhM0YgnA&bvm=bv.44158598,d.bmk, (Assessed on 11.10.2012).