A PROJECT REPORT

ON

"FAILURE INVESTIGATION AND REDESIGN OF CRANKSHAFT AND CYLINDER OF RECIPROCATING PUMP"

Submitted by

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In partial fulfillment for the award of the Degree

Of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

UNDER THE GUIDANCE

Of

Prof. SHAKIL TADAVI

DEPARTMENT OF MECHANICAL ENGINEERING

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL,

NAVI MUMBAI – 410206

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KALSEKAR TECHNICAL CAMPUS NEW PANVEL

(Approved by AICTE, recg. By Maharashtra Govt. DTE,

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CERTIFICATE

This is to certify that the project entitled"**FAILURE INVESTIGATION AND REDESIGN OF CRANKSHAFT AND CYLINDER OF RECIPROCATING PUMP**"

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To the Kalsekar Technical Campus, New Panvel is a record of bonafide work carried out by him under our supervision and guidance, for partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering as prescribed by **University Of Mumbai**, is approved.

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Head of Department	**Principal**
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APPROVAL OF DISSERTATION

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Date: __________

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ABSTRACT

The crankshaft is an essential component in reciprocating pump, which is widely used in an automobile and reciprocating pumps. The most common modes of crankshaft failure are fatigue failure. During its operation, the crankshaft is always subjected to a cyclical load. Besides, the bending and shear load are also typical loads on the crankshaft. Due to the effect of these loads, the crankshaft fails and causes a substantial economic and loss of life. Therefore, predicting the more accurate design of the crankshaft is essential to save the economic and life losses. This report discusses several existing approaches to redesign of crankshaft. Various analytical and experimental approaches used to find the failure due to bending stress.. The review shows that crankshaft failures are the result of bending stress. The crack initiation can occur due to high-stress concentration or improper design of crankshaft . After successfully analyzing of bending stress we calculate the safe values for crankshaft design due to which failure of crankshaft gets reduced.

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1.1 Introduction to reciprocating pump

The main components of reciprocating pump are as follows:

1. Suction Pipe

2. Suction Valve

3. Delivery Pipe

4. Delivery Valve 5. Cylinder

6. Piston and Piston Rod

7. Crank and Connecting Rod

8. Strainer

9. Air Vessel

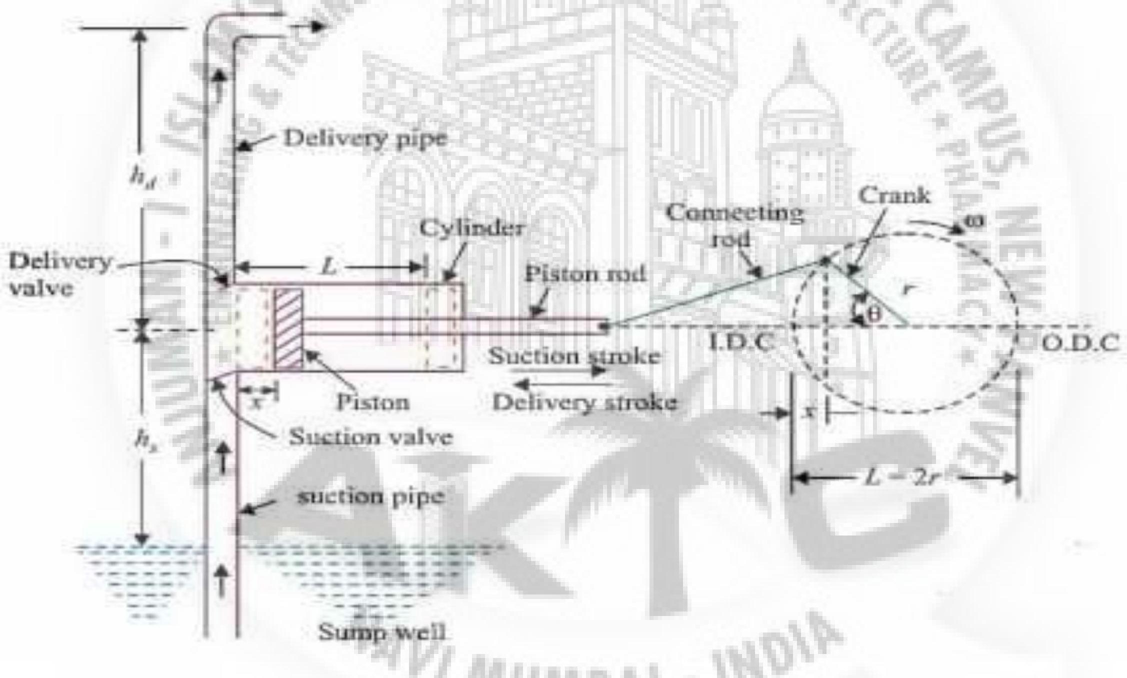

Fig. 1: reciprocating pump

- When the power source is connected to crank, the crank will start rotating and connecting rod also displaced along with crank.
- The piston connected to the connecting rod will move in linear direction. If crank moves outwards then the piston moves towards its right and create vacuum in the cylinder.
- This vacuum causes suction valve to open and liquid from the source is forcibly sucked by the suction pipe into the cylinder.
- When the crank moves inwards or towards the cylinder, the piston will move towards its left and compresses the liquid in the cylinder.

- Now, the pressure makes the delivery valve to open and liquid will discharge through delivery pipe.
- When piston reaches its extreme left position whole liquid present in the cylinder is delivered through delivery valve.
- Then again the crank rotate outwards and piston moves right to create suction and the whole process is repeated.
- Generally the above process can be observed in a single acting reciprocating pump where there is only one delivery stroke per one revolution of crank. But when it comes to double acting reciprocating pump, there will be two delivery strokes per one revolution of crank.

Advantages:

- No priming is needed in the Reciprocating pump compared to the Centrifugal pump.
- It can deliver liquid at high pressure from the sump to the desired height.
- It exhibits a continuous rate of discharge.
- It can work due to the linear movement of piston whereas the centrifugal pump works on the rotary velocity of the impeller.

Disadvantages:

- The maintenance cost is very high due to the presence of a large number of parts.
- The initial cost of this pump is high.
- Flow rate is less.
- Viscous fluids are difficult to pump.

Applications of reciprocating pump:

Reciprocating pump is mainly used for

- Oil drilling operations
- Pneumatic pressure systems
- Light oil pumping
- Feeding small boilers condensate return
- Water jet machine

1.2 Introduction to water jet machine

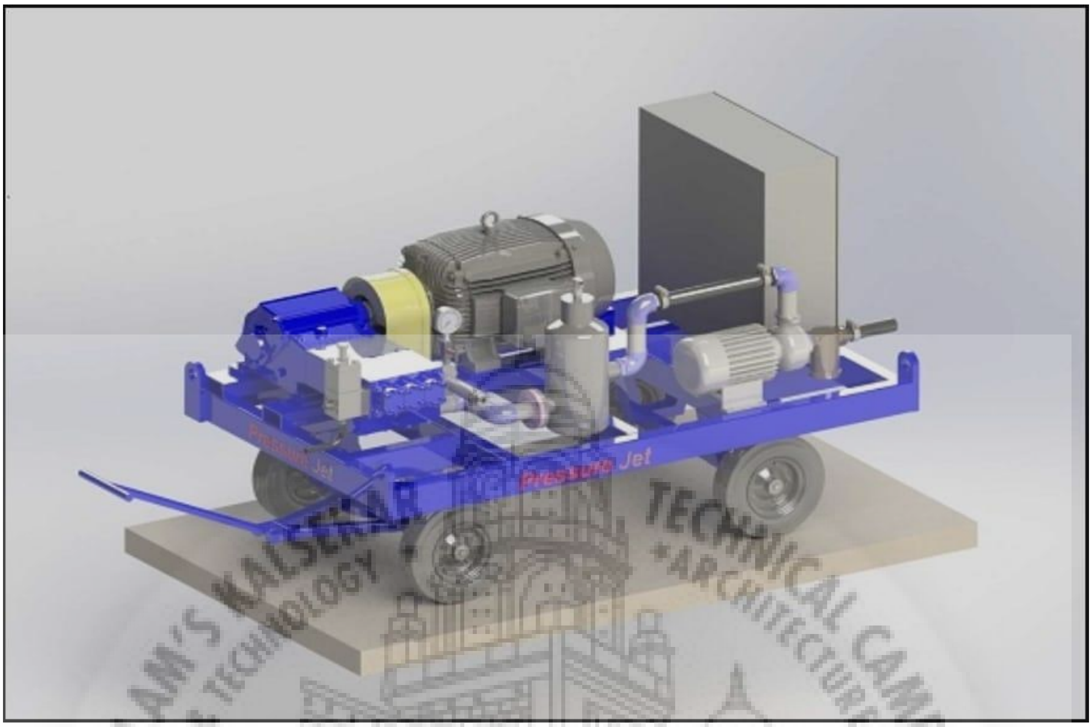

Fig.2: High Pressure Water Jet Cleaning Machine

Working :

High pressure waterjet cleaning, commonly known as hydro jet cleaning, is a widespread abrasive water discharge operation. As such, it is very successful because of its powerful blasting effect. In most cases, because of its efficiency, water jet cleaning will require only one operator for a given application.

Pressure and Flow both play key roles in efficient high pressure waterjet cleaning pumps and machines, and applications respond differently to each variable. Typically, hardened deposits respond better to higher pressures, and softer materials are best removed with higher flows. Each application is unique, though, and we can help you determine which combination of pressure and flow will work best for you. Aqua blasting is the method, by which a high pressure stream of water (450 bar or more) is used to remove old paint, rust, rubber, chemicals, or other heavy build up without causing damage to the surface below it.

This method is perfect for internal and external surfaces because the spray gun operator is able to access those “hard to reach” areas (such as pipelines). Also, because of its strong water spray, the user can keep a safe distance from the surface being cleaned. One of the major differences between high pressure water jet cleaning and other cleaning abrasives (such as silica sand, coal or smelter slag, metallic, synthetic, organic nut shells or fruit kernels) is the ability to contain, capture, filter and reuse the water. This eliminates waste water and contaminants after the cleaning. Extremely high pressure water jet cleaning (2800 bar or greater) are used for concrete cutting. If your requirement is high pressure, high water volume and high temperature output machine, then your answer is in heavy duty hydroblaster pressure washer. Due to the high removal and cleaning power, these high pressure water jet cleaning pumps pay for themselves after a very short period of time. High pressure water jet cleaning are ideal for specialized service providers, construction companies, renovators and for industry. high pressure water jet cleaning pumps consists of bare pump, electric motor and starter, pressure regulating valve, safety valve and pressure gauge. Other accessories include suction hose, delivery hose, foot valve/gun, nozzles, rigid lance and flexible lance.

Hydroblasting is a technique for cleaning external surfaces, which relies entirely on the sheer force of water from a pressurized source to achieve the desired cleaning effect on the intended surface. Abrasives, toxic and potentially harmful chemicals are not used in hydroblasting systems.

A highly pressurized and focused stream of water comes from a hydro blasting machine, which includes a pressure pump and the right nozzle.

A high pressure stream of water (pressure jet rangr: 450 bar to 1400 bar) is used to remove old paint, rust, rubber, chemicals, or other heavy build up without causing damage to the surface below it.

Industrial are a high pressure reciprocating plunger pump that uses high pressure water jet to remove mold, grime, dust, mud and dirt from surfaces and objects such as buildings, vehicles and concrete road surfaces. are also known as power pressure washer.

High Pressure Cleaners, also known as High Pressure Washers, are used for hundreds of various cleaning applications in almost all types of industries around the world. Our High Pressure cleaners are designed & manufactured with latest European technologies with superior quality materials. All spares are readily available with us to provide quick & efficient after sales service.

High pressure cleaners direct a high Pressure Stream or jet of water against a surface to clean, descale, degrease or prepare a surface.

The pumping unit is the pressure generator used to force out the carrier fluid at high velocity. Plunger-type high pressure cleaners pumps are commonly used to generate high pressure water for water jet cleaning and wet or water abrasive blasting for non-abrasive pressure cleaners or rinsing. Crankshaft driven plunger pumps are commonly used for pressure generation. Crankshaft driven plunger pumps may be more efficient in electrical energy costs compared to intensifiers.

Advantages:

Reduction of plant downtime, labour saving, plant protection, water conservation, no need of chemicals and non hazardous.

Applications:

For cleaning of rotary kiln, agitators, riser ducts, turbines, cooling towers, pump impellers filters and hoppers, mixers, ship hulls, conveyers, etc.

Applications of Hydro Blasting Machine :

Investment casting cleaning

Chloride removal from concrete

Corrosion removal from pilings

Historic cobblestone restoration

Marine surface preparation

Concrete surface cleaning

Tank and Vessel cleaning

Kiln Cleaning

1.3 Problem Statement

We got the project of HI-TECH WATER JETTING company. They have water jet machine of capacity 25000 psi which consists of a motor of 180 hp, water tank, reciprocating pump (3 cylinder single acting), rotary pump, pipes and nozzles. Motor is connected to reciprocating pump through a belt drive. As motor stars reciprocating pump also starts and a rotary pump is provided at suction of reciprocating pump to suck the water from tank and provide that water to reciprocating pump at a particular pressure

. Then reciprocating pump increases the pressure of water up to 25000 psi. But required pressure is 15000 to 20000 psi so pressure is controlled by bypass valve. Company provide their machine to some companies like company consists of boiler, sugar mill etc. For cleaning purpose.

The capacity of machine is 25000 psi and the failure of crankshaft and cylinder is occurring at the pressure 15000 to 20000psi. The frequency of failure is high these failures occurs sometimes in one week or sometimes in one month. Because of these failures company facing one more problem that is the material and manufacturing cost of the components are very high.

Fig. 3: Failure pattern of crankshaft

Fig. 4: Failure pattern of cylinder

Objectives:

- To prevent the failure of crankshaft and cylinder.
- To reduce the frequency of failure.
- To reduce the wastage of money by preventing the failure.

CHAPTER 2: LITERATURE SURVEY

Sr. No.	Journal / Title	Author	Description
1	Pump shaft failure	F Berndta , A. Von Bennekonb	Failure of shaft by corrosion and fatigue.
2	Engineering failure analysis	Amod Raut, Pratik Kudle,......	Water gets introduced into the fuel pump causing failures.
3	ASME	J. C. wachel	Formation of cavity due to high pulsation in the suction.
4	Engineering failure analysis	J.A. Becerra, M.Torres,...	Failure analysis of crankshaft of reciprocating compressor.
5	Engineering failure analysis	Michal sikora	Stress and failure analysis of the crank shaft of diesel engine.
6	Engineering failure analysis	Wouter Ost, Patrick De Baets	Failure of piston and pump shaft due to inadequate material used.

Table 1: Literature survey

1.Pump shaft failure case study in this paper author F Berndta , A. Von Bennekonb studutied that during operation, pump shafts usually suffer from degradation as a result of corrosion and/or mechanical degradation, usually in the form of fatigue failures. In many cases corrosion precedes fatigue failure and can actually accelerate the rate of failure. Pump shafts are generally exposed to the liquid being pumped either on a continual basis or at certain locations along the length of the shaft. Specialised sealing arrangements comprising sleeves and o-rings can be used to reduce the amount of liquid ingress, however, where these sealing systems are not implemented or where the integrity of these seals is compromised, damage to the shaft in the form of corrosion may occur.

2. Failure analysis of fuel pump use for diesel engine in transport utility vehicles in this paper author Amod Raut, Pratik Kudle studied that one of the important components for passenger vehicles or buses using fuel as diesel is fuel-injection pump. Fuel pump not only distributes the measured amount of fuel to the cylinder but also assists in controlling engine speed.

A typical fuel-injection system comprises of pressure generating system, motion transmitting system, control system, hydraulic delivery system and a system or unit to handle overload. The fuel-injection pump is indirectly driven by the crankshaft through gearing arrangement. Fuel injection system includes several components possessing motion relative to each other, however no separate unit is used for lubrication as fuel itself acts a lubricant during the fuel pump action. In present work a total of 89 fuel injection pumps were analyzed for different types of failures. Failure of pumps occurred at 70536 km far before the actual life of 200,000 km stated by original equipment manufacturer. Fuel pump failure is faced by all state transport systems and present work is conducted in one of the state central workshops utilized for regular maintenance and bus body building. The failure data collected is analyzed using various techniques and remedial action is devised which further affected the performance of fuel-injection pumps.

3. Analysis of vibration and failure problems in reciprocating triplex pumps for oil pipelines in this paper author J. C. Wachel studied that An analysis was made to identify the causes of vibration and failure problem with piping and reciprocating pump internals on an oil pipe line pump station. A fuel investigation was made to obtain vibration and pulsation over the entire range of plant operation condition. The data show that cavitation was present at nearly all operation condition due to high pulsation in the suction system .The discharge system experience high vibration and piping failure due to the ineffectiveness of accumulator. An acoustical analysis of the suction and discharge system was made to design the optimum acoustical filter system to alleviate the problems. The acoustical analysis were perform with the digital computer programme which predicts the acoustical resonant frequency and pulsation amplitude over the speed range.

4. Failure analysis of crankshaft of reciprocating compressor in this paper author J.A. Becerra, M.Torres studied that the first automobile to be equipped with air conditioning as we know it today appeared in 1939 (Packard) and this technology has been under constant development ever since. Indeed, today around 70% of new automobiles world-wide incorporate this system. In the case of buses almost all vehicles are fitted with this technology. One of the most important components in an air conditioning system is the compressor, which in many systems is a reciprocating volumetric compressor. Although crankshaft failure is not common in this type of equipment, when such an event does occur it could affect all of the components of the kinematic chain (connecting rod, cylinder head, etc.). An analysis of the failure of a reciprocating compressor belonging to a bus climate system is described here. The compressor consists of four cylinders (V arrangement) coupled to the diesel engine of the bus through a V belt. The compressor generally operates at a variable speed between 1000 and 2000 rpm. The location of the compressor, which is powered by the bus engine through a V belt, is shown in The compressor is switched on/off by an electromagnetic clutch located at the free end of the crankshaft. The most common cause of crankshaft failure is fatigue. In order for fatigue to occur, a cyclic tensile stress and crack initiation site are necessary. The crankshafts run with harmonic torsion combined with cyclic bending stress due to the radial loads of the cylinder pressure transmitted from the pistons and connecting rods – to which inertia loads have to be added. Although crankshafts are generally designed with high safety margins in order not to exceed the fatigue strength of the material, high cyclic loading and local stress concentration could lead to the formation and growth of cracks even when the fatigue strength is not exceeded in terms of average values. Pandey analysed failures in the crankshafts of 35 hp two cylinder engines used in tractors, where the fracture plane was located between the main bearing and the journal. The crack began to form at the crank-pin web region in a plane at around 45 with respect to the rotational axis. This crack showed typical fatigue failure with beachmarks. The stress related to the onset of fatigue was estimated to be 175 MPa, which is well below the tensile stress (around 680 MPa) of the nodular cast iron from which the crankshafts were made. Taylor et al. developed two fatigue experiments for a crankshaft of a four cylinder engine made of spheroidal graphite cast iron, which has a tensile strength of 440 MPa: one experiment was torsional and the other flexural. The crankshafts underwent torsional and flexural cyclic loading until failure and in both tests the same fracture angle of 45 with respect to the rotational axis was observed. The work described here concerns a methodology that allowed the cause of failure of a crankshaft to be established by considering both torsional and bending loads. The approach involved the evaluation of the von Mises stress at the crankshaft through dynamic analysis. This methodology is based on the results of a dynamic lumped model developed jointly with a finite element model.

5. Stress and failure analysis of the crank shaft of diesel engine in this paper author Michal sikora studied that the crankshaft is one of the main components of a piston engine. It transfers loads from connecting-rods (connected with pistons) on the clutch. In turbo diesel car engines the tendency to increase of engine power can be observed. It often causes a decrease of the fatigue life of engine components. Both the high power and large torsional moment of modern diesel engines cause a significant increase of operational stresses in the crankshaft. The failure analysis of the piston engine crankshaft was presented in several studies. A failure investigation of diesel engine crankshaft used in a truck has been conducted in study . The fracture occurred in the web between 2nd journal 2nd crankpin. The depth of the nitrided layer in the fillet region (close to the fracture) was determined by the scanning electron microscope and the hardness measurement, combined with nitrogen content analysis. Fractographic studies indicated that fatigue was dominant mechanism of crankshaft failure. The partial absence of nitrided layer may result from over-grinding after nitriding. In work the failure analyses of two crankshafts of diesel engines were performed. Both crankshafts were damaged shortly after major repair of the engine. The main reason of premature failure was wrong grinding process that originated small thermal fatigue cracks at center of the journals. Results of failure investigations of the diesel engine crankshafts were presented in work. The failure time of crankshafts varied between 30 h and about 700 h of engine operation. The crankshafts were induction hardened on pins and journals. In this operation the fillets were not hardened. An analysis indicated that cracks were initiated from the crankpin-web fillet region where the stress level was about 175 MPa. According to author, the main reason of premature fatigue failure of the crankshaft is typical fatigue mechanism which occurs in crank pin fillet at large amplitude of the stress. In study the failed crankshaft used in a truck was analyzed. Spectrum analysis, tensile test, hardness test, and metallographic examination revealed that the failed crankshaft material was ductile cast iron. The low hardness of the material in fillet region and presence of free graphite and nonspheroidal graphite into microstructure of the crankshaft were main reasons for decrease of its fatigue life. The results of an interesting experimental and numerical analysis of the crankshaft segment are presented in study. The normal stress distribution in fillet of the crankshaft section subjected to bending was first obtained using linear-elastic finite element analysis (FEA). The residual stress distribution into crankshaft fillet vicinity (induced by a fillet rolling process) was next determined by elastic–plastic FEA. Based on calculated normal stresses, two fatigue models used to calculate the fatigue limits on surface cracks and in-depth cracks were examined by an experimental data. A case study of a crankshaft catastrophic failure was presented in work. The crankshaft suffered a mechanical seizure on crankpin no. 2 after 3 years in service. It was repaired and after 30,000 km the vehicle had a catastrophic failure on the same crankpin. The macrograph of crankpin revealed that the crankpin was rectified and filled with a metal alloy for the same nominal diameter. According to authors, the catastrophic failure was a consequence of the inadequate repairing by a non-authorized manufacturer. The majority of mentioned above cases are related to fracture of the crankshaft in crank pin regions. This region could be indexed as critical. An interesting research, in which the crack was initiated in another region is described in work. The origin of the fatigue crack was the oil hole. Inspection of other oil holes showed another hole with sharp notch at the intersection of the oil hole and the journal surface. The fracture of the shaft was due to poor refurbishment of the journals by welding. The main objective of presented investigations is determination of stress state in the crankshaft during work of the engine. An additional aim of this work is to explain the failure reasons of the crankshaft of a diesel engine.

6. Failure of piston and pump shaft due to inadequate material used in this paper author Wouter Ost*,Patrick De Baets studied that For the application of glue during the production of cars a double-acting reciprocating pump is used to apply the glue (under pressure). The double-acting pump is driven by a pneumatic piston. The piston rods of both the pump and the pneumatic piston (in what follows the term shaft is used for both piston rods; the one of the pneumatic piston is termed ‘‘motor shaft” and the one of the pump is termed ‘‘pump shaft”) are connected by a coupling, which consists of two C-shaped halfcircular brackets placed in a holder. For the connection between both shafts and the coupling a groove is machined at the end of both shafts, over which the brackets are placed (see Fig. 1 for a schematic view). After about 1 year of service several failures occurred in both pump and motor shafts at the groove for the coupling. As standstill of the production line due to the unexpected failures of the shafts could not be tolerated, an inspection program using magnaflux (magnetic particle) was instigated by the user of the equipment. Periodically more shafts showed cracks and were taken out of service and replaced with new shafts. Two shafts (one motor and one pump shaft), which were found to be cracked using the magnaflux inspection, were investigated, and two unused shafts (one motor and one pump shaft) were also investigated. The aim of the investigation was to determine the cause of the failure, and to suggest a solution for this problem, preferably without affecting the geometry of the coupling.

CHAPTER 3: TOTAL COSTING

For cylinder:

Assuming the material cost will be 1600rs per kg and the weight of the cylinder is 7kg. Assuming that the raw material is of 15kg so the cost will be 24000rs. Assuming manufacturing cost is 6000rs so the total cost will be 30000rs per cylinder.

For crankshaft:

Assuming the material cost will be 200rs per kg and the weight of the crankshaft is 100kg assuming raw material is of 250kg so the cost will be 50000rs. Assuming manufacturing cost is 10000 so total cost will be 60000rs per crankshaft.

CHAPTER 4: METHODOLOGY

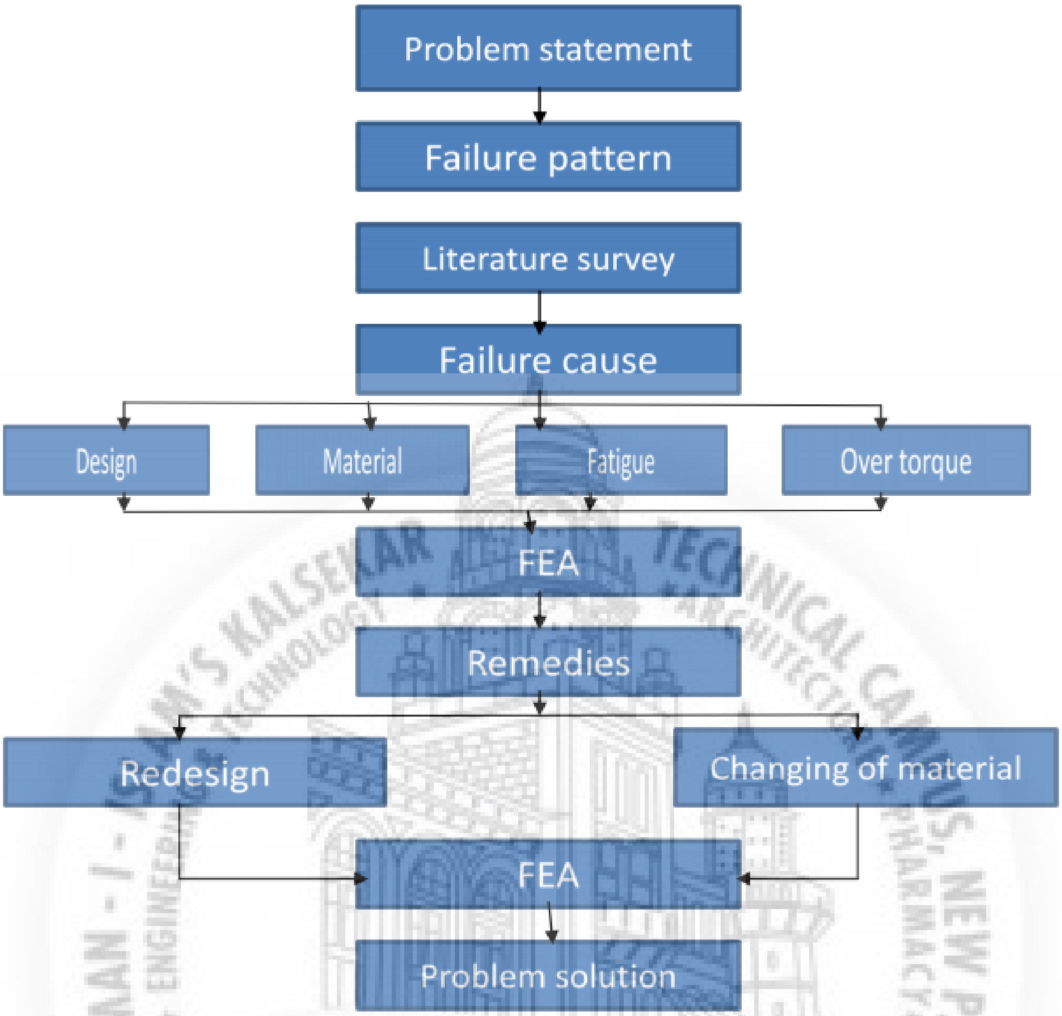

Fig. 5: Methodology flow chart

- To solve our problem we will follow this flow chart. The first element is problem statement which has been discussed previously. Then second is failure pattern, we have seen some failure pattern.
- Third is literature survey, we did some literature survey discussed previously, in which authors have shown some failure pattern, we will compare those failure pattern to our pattern.
- Then we will find some failure causes like improper design, improper material selection or defect in chemical properties of material, fatigue failure and over torque.
- By considering these all parameters we will find out the forces and torque acting on crankshaft and cylinder. Then we will analyse old design in FEA software.
- Then we will find remedies to our problem, according to that we will redesign our component or change the material, then analyse our new design in FEA software like that we will get solution.
- Then we will create 3d model of new design and give it to company so that they can manufacture the crankshaft according to new design.

CHAPTER 5: CALCULATION AND DESIGN

5.1: Calculation

- **Speed of motor:**

N1=Nm=1485rpm

- **speed of crankshaft:**

N2=Nc=?

[(D/d)=(N1/N2)×n].............psg 7.61

D: diameter of pulley mounted on motor

D= 23 inch =584.2mm

d:diameter of pulley mounted on crankshaft

d=6.5 inch =165.1mm

n=0.98......assumed

[(584.2/165.1)=(1485/N2)×0.98]

$\boxed{\text{N2=Nc=411.280 rpm}}$

- **motor is transmitting power 180hp:**

P=180 Hp =134.226kw

- **Torque at crank shaft:**

P=(2πNT/60)

134.226×10 3 =[(2π×411.280×T)/60]

$\boxed{\text{T=3116.52 Nm}}$

$\boxed{\text{T=3116.52×10\^{}3 Nmm}}$

- Pressure inside cylinder

$\boxed{\text{p= 20,000 psi =137.895mpa}}$

5.2: Redesign Of Crankshaft

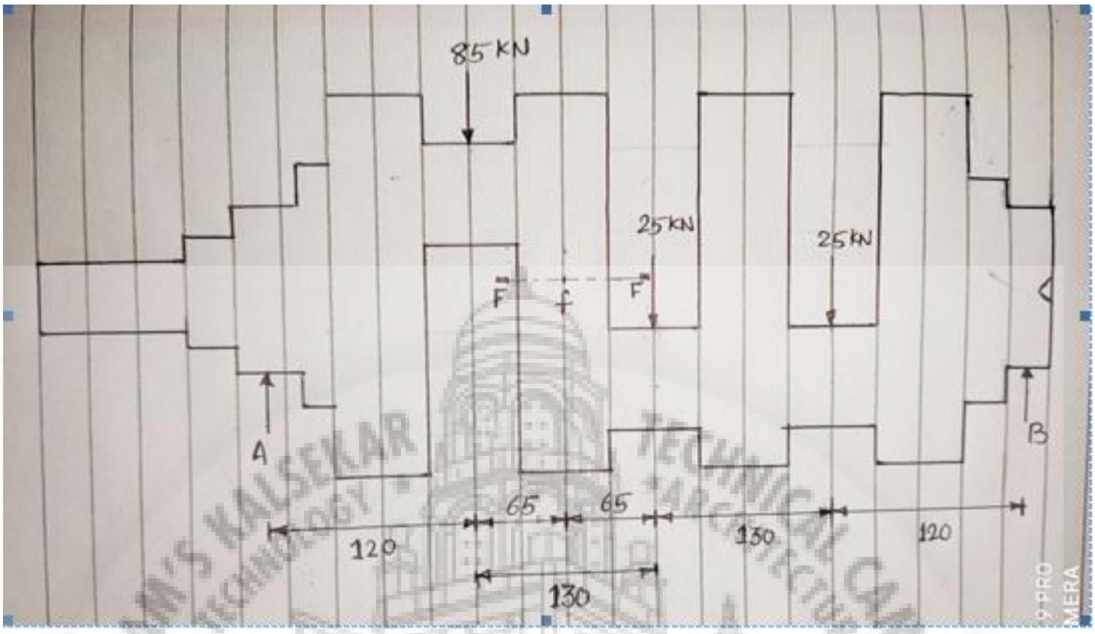

Fig. 6: Crankshaft

Material of crankshaft:

- **MSEN8**

Yield stress = 290mpa

FOS = 4

Tensile stress = 72.5mpa

Bending stress = 94.25mpa

Calculating reaction force at bearing:

RA + RB=135KN

MA=0

-RB×500+25000×380+25000×250+85000×100

RB=51900N

RA=831000N

Calculating bending moment at failure section:

Mf=-51900×315+25000×195+25000×65

Mf= 9848.5 KNmm

bending stress= (M×y)/I

bending stress = ((9848.5×1000)×25)/((235×50^3)/12))

bending stress = 100mpa

bending stress calculated > bending stress design

hence design is failed.

If we increase the thickness from 50mm to 52mm then bending stress is,

bending stress/y= M/I

bending stress= [((9848.5 × 10^3)×25)/((235×52^3)/12)]

bending stress= 92.99mpa

so,

bending stress $_{\text{calculate}}$ < bending stress $_{\text{design}}$

hence design is safe.

5.3: Result

- Safe bending stress = 94.25mpa
- Thickness of crank web (old design) = 50mm
- Bending stress induced in crank web for 50mm thickness = 100mpa
- Calculated bending stress > safe bending stress hence design failed.
- Thickness of crank web (new design) = 52mm
- Bending stress induced in crank web for 52mm thickness = 92.99mpa
- Calculated bending stress < safe bending stress hence design is safe.

CHAPTER 6: CONCLUSION

- It is conclude that the crankshaft design was not applicable for the pressure 25000 psi which is failing due to bending stress.
- so we calculated the bending stress induced at the failure section. Which is exceeding the safe stress.
- so we have increased the thickness of crank web and find the bending stress for that thickness and we found that stress was inducing less than the safe stress.
- As we increase the thickness of crank web the bending stress induced in it reduces.

CHAPTER 7: REFERENCES

[1.] Rincle Garg, Sunil Baghla, “Finite element analysis and optimization of crankshaft”, International Journal of Engineering and

Management Reaserch, vol-2,Issue-6,ISSN: 2250-0758, Pages:26-31, December 2012.

[2.] C.M Balamurugan, R. Krishnaraj, Dr.M.sakhivel, K.kanthavel, Deepan Marudachalam M.G, R.Palani, “Computer Aided

modelling and optimization of Crankshaft”, International Journal of scientific and Engineering Reaserach, Vol-2, issue-8,

ISSN:2229-5518, August-2011.

[3.] Gu Yingkui, Zhou Zhibo, “Strength Analysis of Diesel Engine Crankshaft Based on PRO/E and ANSYS”, Third International

Conference on Measuring Technology and Mechatronics Automation, 2011.

[4.] Abhishek choubey, Jamin Brahmbhatt, “Design and Analysis of Crankshaft for single cylinder 4-stroke engine”, International

Journal of Advanced Engineering Reaserch and studies, vol-1, issue-4, ISSN:2249-8974, pages: 88-90, July-sept 2012.

[5.] R.J Deshbhratar, Y.R Suple, “ Analysis and optimization of Crankshaft using FEM”, International Journal of Modern Engineering

Reasearch, vol-2, issue-5, ISSN:2249-6645, pages:3086-3088, Sept-Oct 2012.

[6.] Farzin H. Montazersadgh and Ali Fatemi “ Stress Analysis and Optimization of Crankshafts Subjected to Dynamic Loading”,

AISI, August 2007.

[7.] Hicks, E. J. and Grant, T. R., "Acoustic Filter

Controls Recip Pump Pulsation," The Oil and

Gas Journal, January 15, 1979, pp 67-73.

[8.] Ludwig, M., "Design of Pulsation Dampeners for

High Speed Reciprocating Pumps," Division of

Transportation, American Petroleum Institute

Vol 36 [V] 1956, pp 47-54.

[9.] Sparks, C. R. and Wachel, J. C., " Pulsations in

Centrifugal Pumps and Piping Systems,"

Hydrocarbon Processing July 1977,

pp 183-189.

[10.] Wachel, J. C. and Szenasi, F. R., "Vibration and

Noise in Pumps," Pump Handbook, 1st Edition,

McGraw-Hill, 1976, pp 9-87 to 9-97.

[11.] Miller, J. E., "Liquid Dynamics of Reciprocating

Pumps - Parts 1 and 2," The Oil and Gas

Journal, April 18, 1983

[12.] Pandey RK. Failure of diesel-engine crankshafts. Eng Fail Anal 2003;10:165–75.

[13.] Taylor D, Ciepalowiz AJ, Rogers P, Devlukia J. Prediction of fatigue failure in a crankshaft using the technique of crack modelling. Fatigue Fract Eng Mater Struct Ltd. 1997;20:13–21.

[14.] Becerra Villanueva JA, Metodología para el estudio de las causas de rotura de cigüeñales en motores de combustión interna alternativos y compresores alternativos. Aplicación en un modelo de mantenimiento predictivo. PhD Dissertation. Universidad de Sevilla; 2007.

[15.] Rahnejat H. Multi-body dynamics. UK: Professional Engineering Publishing; 1998.

[16.] Rezeka SF, Henein NH. A new approach to evaluate instantaneous friction and its components in internal combustion engines, SAE Paper 840179; 1985

[17.] Wegst C, Wegst M. Stahlschlüssel. 20th ed. Marbach: Verlag stahlschlussel West GmbH; 2004.

[18.] Zahavi E. Life expectancy of machine parts: fatigue design. Boca Raton: CRC Press; 1996.

[19.] Pilkey WD. Peterson's stress concentration factors. 2nd ed. New York: John Wiley & Sons; 1997.

[20.] Matek W, Muhs D, Wittel H, Becker M, Jannach D. Roloff/Matek Machineonderdelen: tabellenboek. 3rd ed. Schoonhoven: Academic Service; 2001 [in Dutch].