# FAILURE ANALYSIS OF CENTRIFUGAL PUMP SHAFT USING ANSYS IN PETROCHEMICAL INDUSTRIES

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***Abstract -*** **Pumps are commonly used to transport fluids such as water, sewage, petroleum, and petrochemical products. The pumps can be divided into two general categories, namely dynamic pumps and displacement pumps. The energy, in a displacement pump such as a diaphragm pump, is added to the pumping medium periodically while the medium is contained in a set volume. The capacity of a pump is defined based on the pressure head (in meters) and the maximum delivery flow rate at a specific speed of the shaft. The pump (51WP1) shaft used to pump the water from the cooling tower to cool all the required parts used in this refinery is breaking non-periodically every month or once in two or three months at the edge of the impeller eye. Due to this situation there is a need to make shutdown frequently in the refinery and in turn spending crores of money and frequent replacement of shaft is done. In order to minimise the money spent and shaft replacement, we were asked to find a solution for this problem. Failure analysis was done on several aspects which may cause failure of the shaft.**

***IndexTerms -: Centrifugal pump, ANSYS, Petrochemical products, Impeller Eye***

## I. INTRODUCTION

The transfer of rotational energy from one or more driven rotors called impellers. Fluid enters the rapidly rotating impeller along its axis and is cast out by centrifugal force along its circumference through the impeller's vane tips. A centrifugal pump converts rotational energy, often from a motor, to energy in a moving fluid. A portion of the energy goes into kinetic energy of the fluid. Fluid enters axially through eye of the casing, is caught up in the impeller blades, and is whirled tangentially and radially outward until it leaves through all circumferential parts of the impeller into the diffuser part of the casing. The fluid gains both velocity and pressure while passing through the impeller. The doughnut-shaped diffuser, or scroll, section of the casing decelerates the flow and further increases the pressure. It is important to note that the water is not pushed radially outward by centrifugal force (non-existent force), but rather by inertia, the natural tendency of an object to continue in a straight line (tangent to the radius) when traveling around circle. This can be compared to the way a spin-cycle works in a washing machine.

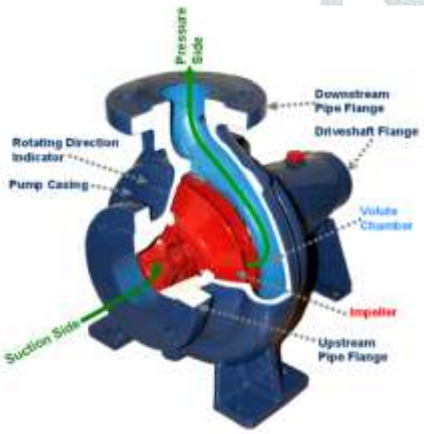
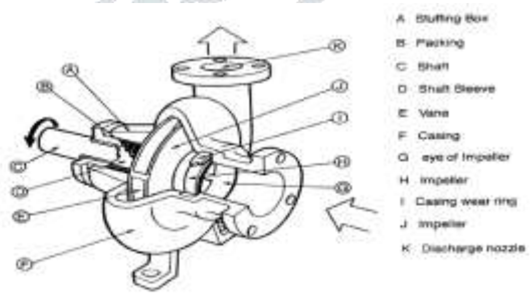



Fig.1 Typical Centrifugal Pump




Main parts of the Centrifugal Pump

Each centrifugal pump is made of hundreds of parts. There are a few components that virtually every centrifugal pump has in common. These components can be subdivided into the wet end and the mechanical end.The wet end of the pump includes those parts that determine the hydraulic performance of pump.The two primary wet ends are the impeller and casing. In some cases the first radial bearing can be water lubricated. In this case also bearing can belongs to wet ends.The mechanical end includes those parts that support the impeller within the casing. The mechanical end of the pump includes the pump shaft, sealing bearings and shaft sleeve.

Rui Yu, Jinxiang Liu presented a paper on Failure analysis of centrifugal pump impeller(2018).This paper analyzed the looseness failure between impeller and shaft. In order to determine the cause of the failure, the torque produced by external loads was calculated theoretically. The torque capacity of interference connection was analyzed using finite element method (FEM). The basic mechanical properties of different impeller were tested and applied to the FEM model to find out the influence of mechanical properties on interference connection between impeller and shaft. The analysis results indicate that interference connection condition is greatly affected by mechanical properties. The torque capacity of interference connection decreases with the mechanical properties decrease. The maximum torque transmitted by interference connection being lower than resultant resistance torque is the main reason of looseness failure between impeller and shaft.

GDas, A.N Sinha, S.K Mishra(Pathak), D.K. Bhattacharya presented a journal on Failure analysis of counter shafts of a centrifugal pump(1998).This paper analyses the premature failure of two counter shafts used in centrifugal pumps for lifting slurry has been carried out. Chemical analysis, microstructural characterisation, fractography, hardness measurement, tensile and Charpy impact tests were used for the analysis. The chemical compositions for the shafts were as per recommendation. The microstructure of one of the shafts was ferritic–pearlitic and its mechanical properties were inferior to the recommended values. For the other shaft the microstructure was tempered bainite; although the impact energy satisfied the specification, the other properties (hardness, UTS) were inferior. It was concluded that the improper heat treatment was the prime cause for the premature failure of the shafts.Adam Adamkowski, Adam Henke, Mariusz Lewandowski presented a paper on Resonance of torsional vibrations of centrifugal pump shafts due to cavitation erosion of pump impellers(2016).

In this paper, the selected results were gathered during investigation of centrifugal pumps used in a sea water cooling system of one of Diesel power stations are presented in the paper. The main goal of research was to explain the reason of occurring fractures in pump shafts. The investigation has shown that the fractures were caused mainly by the resonance between pump shaft torsional natural vibrations and those following from the pressure fluctuations related to the frequency of the shaft rotational speed and the number of impeller blades. The resonance occurred as a result of intense erosion of pump impellers derived mainly from cavitation phenomenon that caused about 20% of the impeller mass decrease. The scope of the investigation has covered among others: erosion damage recognition, tests of the investigated pumps operating conditions, spectral analysis of pressure fluctuation generated by the pump blade system as well as strength analysis of the pump shaft and the frequencies of its natural bending and torsional vibrations. F.Berndt, A.vanBennekom presented a paper on Pump shaft failures — A compendium of case studies(2001).

This paper presents a collection of pump shaft failures that have been encountered during the consulting activities at the University of the Witwatersrand and the Plant Infrastructure and Pipelines Centre at the Council for Scientific and Industrial Research (CSIR).This research article presents a methodology to identify the critical component of a centrifugal pump and analyze its reliability. Failure analysis of major components of the pump was done, risk priority number was calculated and critical component was identified. The critical component was considered to be composed of two sections. Its reliability was analyzed by mathematical modeling of the first section. A finite element model of the component was developed and stress analysis was done using finite element method, in order to assess the reliability index of the second section. Total reliability of the component was calculated by multiplying the two reliability indices.

*1.1* POSSIBLE CAUSES OF PUMP SHAFT FAILURE

a. Metallurgical / Manufacturing Process Flaws b. Vibration c. Imbalance

d. Misalignment e. Fluid properties

f. Hydraulic Shocks g.Reverse Flow

1.3 Failure Analysis of a Centrifugal Pump Shaft

Pumps are commonly used to transport fluids such as water, sewage, petroleum, and petrochemical products. The pumps can be divided into two general categories, namely dynamic pumps and displacement pumps. In a dynamic pump, such as a centrifugal pump, energy is added to the pumping medium continuously and the medium is not contained in a set volume. The energy, in a displacement pump such as a diaphragm pump, is added to the pumping medium periodically while the medium is contained in a set volume. The pump is driven by a prime mover that is either an engine or an electric motor. The capacity of a pump is defined based on the pressure head (in meters) and the maximum delivery flow rate at a specific speed of the shaft.

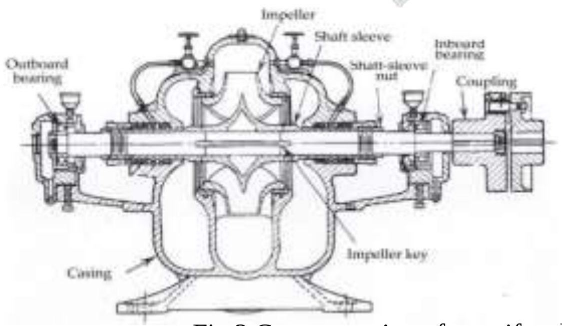



Fig.3 Cut out section of centrifugal pump

Fig.4 Impeller

Fig.5.Shaft

It was reported that in this refinery, the pump (51WP1) shaft used to pump the water from the cooling tower to cool all the required parts used in this refinery is breaking unperiodically every month or once in two or three months at the edge of the impeller eye. Due to this situation there is a need to make shutdown frequently in this refinery and inturn spending crores of money and frequent replacement of shaft is done.In order to minimise the money spent and shaft replacement, we were asked to find a solution for this problem. Failure analysis was done on several aspects which may cause failure of the shaft. Failure analysis which is done on most possible causes of centrifugal pump (51WP1) shaft failure is given below.

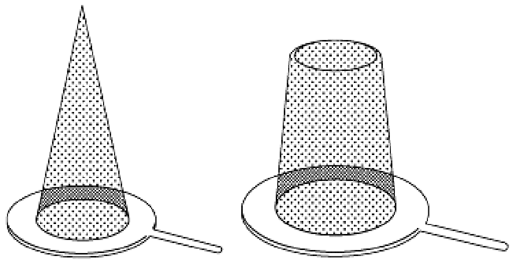


Fig.6 Strainer

As the failure analysis of shaft on manufacturing or metallurgical processes has been ruled out, the second failure analysis on the imbalance of centrifugal pump is done. Normally imbalance creates problems when the pump is running. It is a source of vibration.

Impeller imbalance is mainly caused by the following factors:

- New, never balanced impeller
- Trimmed and not balanced impeller
- Foreign object stuck in vanes
- Vanes bent or out of plane
- Balance holes plugged
- Product build up on impeller

## II. MATERIALS AND METHODS

2.1 Failure analysis on Mechanical design of centrifugal pump shaft:

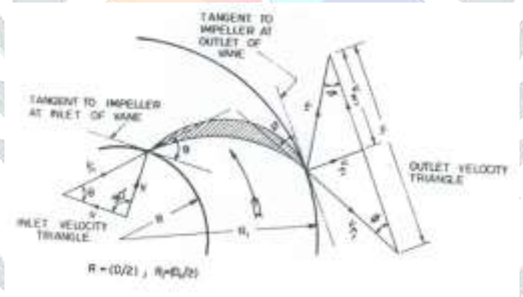



Fig. **7** Velocity Diagram

N= speed of the impeller in RPM

D1= Diameter of impeller at inlet

u1= Tangential velocity of impeller at inlet

D2= Diameter of impeller at outlet

u2= Tangential velocity of impeller at outlet

V1= Absolute velocity of water at inlet

Vr1= Relative velocity of water at inlet

α= Angle made by absolute velocity (V1) at inlet

θ= Angle made by relative velocity (Vr1) at inlet

V2= Absolute velocity of water at outlet

Vr2= Relative velocity of water at outlet

β= Angle made by absolute velocity (V2) at outlet

φ= Angle made by relative velocity (Vr2) at outlet

2.2 PARAMETERS KNOWN:

Width of impeller (B2) = 105mm = 0.105m

Inner Diameter of impeller (D1) = 530mm = 0.530m

Outer Diameter of impeller (D2) = 640mm = 0.640m

Weight of the impeller = 256kg

Speed of the impeller (N) = 1000rpm

Material of the impeller = Cast steel

Weight of the shaft = 147kg

Material of the shaft = EN24

Material of the key = Mild steel

Discharge of the water (Q) = 5800m3/hr = 1.6111 m3/s

2.3 CALCULATIONS:

2.3. Tangential Velocity of the impeller at inlet (u1) = (πD_1 N)/60

= (π*0.53*1000)/60

=27.75 m/sec

2.4 Tangential Velocity of impeller at Outlet (u2) = (πD_2 N)/60

= (π*0.64*1000)/60 = 33.51 m/sec

2.5 To find the velocity of flow at inlet:

Discharge of water, Q = Area of inlet * Velocity of flow at inlet = 1.6111/2 = π/4(0.532 – 0.1112)*Vf1

Vf1 = 3.81 m/sec

2.6 To find the velocity of flow at outlet:

Vf2 = Q/(πD_2 B_2 )

= 1.6111/(π*0.64*0.105)

= 7.63 m/sec

2.7 To find absolute velocity of water at outlet:

Tan φ = V_(f_2 )/u_(2-V_(w_2 ) )

Tan 〖22〗^0=7.63/(33.51-V_(w_2 ) )

Vw2= 14.62 m/sec

2.8 To find the Weight of the water:

W= ρ*g*Q

W= 1000*9.81*1.6111

W= 15804.891 N/sec

2.9 Workdone by impeller on water per second:

= W/g*Vw2*u2

=15804.891/9.81*14.62*33.51

= 789303.98 Watts

2.10 Calculations of Efficiencies:

a. Manometric efficiency ($^{\eta}$man)

= (gH_m)/(Vw2*u2)

= (9.81*45.8)/(14.273*33.493)

$^{\eta}$man = 0.939 = 93.9%

b. Mechanical efficiency ($^{\eta}$mech) =

(workdone )/(shaft power)

c. Shaft power (S.P) = VIcosφ

= 6600*82*cos(0.9)*1.732

= 937242 watts

d. Mechanical efficiency ($^{\eta}$m)

= 783463/937242

$\eta_m$ =83.5%

e. Overall efficiency ($\eta_o$) =$\eta$ man* $\eta$m

= 0.93*0.835

= 0.776

$\eta_o$ = 77.6%

2.11 Calculation of maximum shear stress acting on the centrifugal pump shaft:

a. Parameters Known:

Power = 789.303 KW

Shaft diameter = 111 mm

Shaft weight = 147kg = 1470 N

Impeller weight = 256kg = 2560 N

Impeller width = 105 mm

Length of the shaft = 2110 mm

Speed of the shaft = 1000 rpm

Shaft material = EN24

b.Properties of the EN24 material shaft:

Tensile strength = 745 MPa

Yield strength = 470 MPa

Permissible shear stress (τmax) = 0.5 * yield strength

= 235 MPa

The given shaft is subjected to both twisting and bending. Hence the shaft will be designed based on the twisting moment and bending moment.

2.12 TO FIND TORQUE:

P = 2πNT/60

Torque (T) =(p*60)/2πN

= (783* 〖10〗^3*10)/(2 π *1000)

T = 7480*103 N-mm

Taking moment about A,

(1470 * 1055) + (2560 * 1849) = (RB * 2110)

RB= 2978.35 N

RA + RB = 1470+2560

RA + 2978.33 = 4030

RA= 1051.67 N

Taking Moment about C,

MC = (RB * 1055) – (2560 * 794)

MC = (2978.33 * 1055) – (2560 * 794)

MC= 1109498.15 N-mm

Taking Moment about D,

MD = RB * 261

MD = 2978.33 * 261

MD= 777344.13 N-mm

Equivalent moment,

Meq = √Mc2 + MD2

= √ ((1109498.15)2 + (999344.13)2)

Meq = 1354914.009 N-mm

Now,

$Teq = \pi/16 * \tau * D3$

Where, $Teq = \sqrt{} T2 + Meq2$

$= \sqrt{} ((7480*103)2 + (1354714.007)2)$

$Teq = 7601687.315$ N-mm

Therefore,

$7601637.315 = \pi/16 * \tau * (1113)$

$\tau = 28.322 * 106$ N/mm2

$\tau = 28.322$ MPa$<\tau$max (235 MPa)

The maximum shear stress acting on the shaft is lesser than that of the permissible shear stress.

Therefore, the mechanical design of the shaft is safe.

## III. RESULTS AND DISCUSSION

3.1 MECHANICAL DESIGN ANALYSIS USING SOFTWARE: ANSYS

The maximum shear stress acting on the shaft which is found using the software is 2.45 MPa and that is lesser than that of the permissible shear stress 235 MPa.Therefore, it can be concluded that the mechanical design of the shaft is safe.

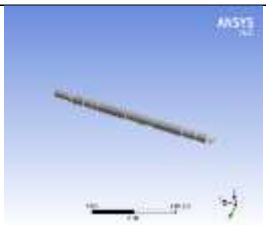


Fig.8Shaft 3D model

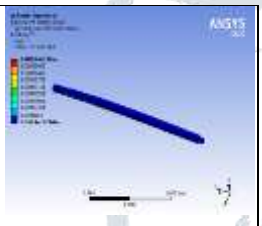


Fig.9Equivalent elastic strain

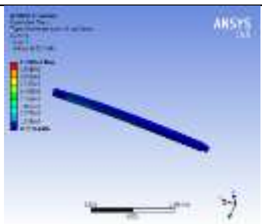


Fig.10Equivalent stress

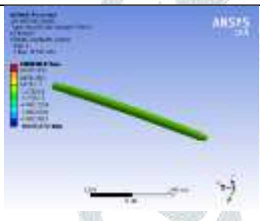


Fig.11Shear elastic strain

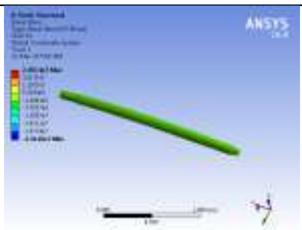


Fig.12Shear stress

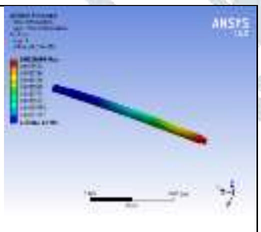


Fig.13Total deformation

## IV. CONCLUSION

Serious damage can be caused by hydraulic forces. For example, when a check valve slams shut, interrupting the flow of fluid, a massive shock wave results. This shock wave reverses flow and travels back downstream. When the shock wave collides with a pump, assuming the shock is strong enough, the shaft could bend or break instantly or over time.The reverse flow causes a pump's impeller and shaft to turn backwards and the pump suddenly kicks on which causes stress and makes the shaft to break.

From Bernoulli's equation, it is found that, as both the suction and delivery pipe diameters of the centrifugal pump are same in size, the velocity of the water after crossing the impeller increases and thus causing a vacuum in the impeller area. In order of to fill the vacuum, the water which is about to leave the outlet pipe reverses and thus causing a hydraulic shock at the impeller eye.

In order to avoid this, the design of the pipe has to be changed. Either the area of the delivery pipe has to be decreased or the area of the suction pipe has to be increased. Here it is advisable to increase the suction pipe diameter from 24 inch to 32 inch so that there will be a uniform flow of water inside the centrifugal pump and avoids hydraulic shocks and vibrations and in turn stops the shaft from premature failure.

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