Design and Development of Boiler Feed Water Pump

Kiran Shinde¹, Amol N. Gaikwad², Abhimanyu K. Chandgude³

¹,² Dr.D.Y.Patil School of engineering, Lohgaon, Pune, India.
³ MIT Academy of Engineering, Pune, India

Abstract— Pump is energy absorbing rotodynamic machinery transporting fluid from one place to another, being key parameter and heart of any industrial process plant as well as thermal and nuclear power plant, also used for dewatering and irrigation purpose. Boiler feed water pump feeds condensed return water against high steam pressure produces by the boiler. BFP normally are centrifugal type pump containing a wide number of operating parameters, operator requirement make this system to work within parameter functioning by the manufacturer on high efficiency regimes.

As pump is centrifugal type pump, water enters axially through impeller eye and exit radially. Generally electric motor is used as a prime mover to run the pump. In the present study we design and analysis of boiler feed pump having a flow of 138 m³/hr under a head of 632 m at 3550 RPM and operating temperature range is 200° 15 degree Celsius has been taken up. In these design judgmental task is to set up an high head with in four stages. The various pump parameter obtained from design is developed using 3D modeling software Pro-E, and analysis is carried out by using Ansys, CFD software module.

Keyword- boiler feed, efficiency, head rpm, impeller eye

I. INTRODUCTION

Pumps are energy-consuming devices being used in many industrial applications and in particular in dewatering, handling water & other fluids and in agricultural, the efficiency improvement of pumps has significant meaning to energy conservation. The entire project work focus is deal with the performance & efficiency of pump which directly save power. In this project we design a multi-stage centrifugal pump which complies to various aspects require to US boiler feed water pump market.

Feed pumps are an essential subsystem of boilers used in industrial process plants and called as boiler feed pump (BFP). Normally, BFP is high pressure unit that takes suction from condensate return system and can be of the centrifugal type pump. In centrifugal pump, water enters axially through the impeller eyes and exits radially. Generally, electric motor is used as prime mover to run the feed pump. To force water into boiler, the pump must generate sufficient pressure to overcome steam pressure developed by boiler.

There is no rigorous procedure to be followed in designing a pump. Lot of approaches have been developed and, although each has a slightly different method of calculation, the broad underlying principles of all are similar. The velocity limitations and proportions are also there to which it requires to adhere; but these may be exceeded in certain instances to meet competition with regard to cost or performance.

The usual design is based upon a certain desired head and capacity at which the pump is operated most of time. In design of centrifugal pump, the parts to be designed are: shaft, impeller, vane, casing, and...
selection of bearing. To design these parts different methodologies can be obtained through literature survey. From the given conditions, the specific speed is obtained. According to required head, the flow rate and from specific speed, pump of double volute, doubles suction and single stage type is selected. The minimum shaft diameter can be obtained by using maximum shear stress theory. Impeller and vane are designed according to methodology provided by Church. To design the vane empirical relations are used. API standard is used to design the volute and for bearing selection.

II. LITERATURE REVIEW

A wide variety of centrifugal pump types have been constructed and used in many different applications in industry and other technical sectors. However, their design and performance prediction process is still a difficult task, mainly due to the great number of free geometric parameters, the effect of which cannot be directly evaluated. The significant cost and time of the trial and error process by constructing and testing physical prototypes reduces the profit margins of the pump manufacturers. For this reason CFD analysis is currently being used in the design and construction stage of various pump types. The rotor stator interaction can be also studied with the aid of CFD, the use of which reduces significantly the new pump development costs. The average reduction is estimated to 65% during 2005. The numerical simulation can provide quite accurate information on the fluid behavior in the machine [1].

The present works discuss the comparison between the analytical and experimental performance curve of the high pressure boiler feed pump - a multistage centrifugal pump. The hydraulic loss model stated by J.F. Gulich is widely used for the analytical performance prediction of the single stage centrifugal pumps. In the present work, this methodology is implemented for the analytical performance prediction of the 10-stage industrial centrifugal pump. The given pump is installed as HPBFP (High Pressure Boiler Feed Pump) to carry water from the generator to boiler. The H-Q (Head versus Discharge) analytical performance characteristics curve of the pump is compared with the experimental performance characteristics curve. The deviation between the analytical and experimental curve is between 2-13% for the discussed range of discharge [2].

III. PROBLEM STATEMENT AND OBJECTIVES

To develop higher head with high efficiency by decreasing the number of stages that is currently used. The present Boiler Feed Pump range has got limitation on head developed per stage resulting to more number of stages. This is a trigger point for this development.

At a growing technology & awareness user requires compact size of pump & its assembly (especially in developed countries). Parts interchangeability pump require to meet the various standard (API-610, HIS).

Objectives:
• Reducing the number of stages by increasing head per stage by improving performance of pump.
• Existing BB4 range expansion to cover requirement of BFW pump.
• Generating the hydraulics which can comply with various standards (country specific).
• Design of Compact size pump complying with requirement.
• Pump suitability for 60 Hz@3550 rpm

IV. SCALING FACTOR
As per ANSI 1.3-2012 Affinity Law,

\[
\frac{Q_p}{Q_m} = \left( \frac{D_p}{D_m} \right)^3 \times \frac{N_p}{N_m}
\]

\[
138 = \left( \frac{D_p}{D_m} \right)^3 \times \frac{3550}{2980}
\]

\[
\frac{D_p}{D_m} = 1.052
\]

\[
\frac{H_p}{H_m} = \left( \frac{D_p}{D_m} \right)^2 \times \left( \frac{N_p}{N_m} \right)^2
\]

\[
158 = \left( \frac{D_p}{D_m} \right)^2 \times \left( \frac{3550}{2980} \right)^2
\]

\[
\frac{D_p}{D_m} = 1.052
\]

V. MODELLING OF PARTS

A. The Dimensions of Pump Rotor Shaft

Dimension of shaft is as follows:

Overall length is 1052 mm, diameter is 65 mm.

The material properties of steel

The material is ASTM 434 TP 4140, which is having all these characteristics.

<table>
<thead>
<tr>
<th>TABLE I Material Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
</tr>
<tr>
<td>Young’s modulus</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
</tr>
<tr>
<td>Density</td>
</tr>
<tr>
<td>Yield strength</td>
</tr>
<tr>
<td>Tensile strength</td>
</tr>
</tbody>
</table>

B. Dimension of Rotor

Gross Weight of Impeller m = 7 Kg
Total Weight in Newton = 7 x 9.81
W = 68.67 N

Maximum diameter of impeller = 293 mm
Minimum diameter of impeller = 255 mm
Number of blades of impeller = 6
Angle of blade = 12°
Impeller exit width = 12 mm

**Fig. 2. Modelling of Impeller**

CAD software like PRO-E is higher end software which is feature based solid modeling systems. PRO-E is used for modeling of IMPELLER.

**C.3D Modeling of Inlet and Outlet Casing.**

**Fig. 3. ‘3D’ Views of Inlet Casing**

Pressure-containing parts are generally made thicker than required for handling a noncorrosive liquid so that full pumping capability will be maintained even after the loss of some material to the corrosive medium.

Thickness of inlet and outlet casing as per ASME pressure vessel section-8,

\[ t = \left( \frac{P * D}{200 * S * E} - 1.2 * P \right) + 3 \]
D. 3D modeling of Guide vane

![Guide vane](image1)

**Fig. 4. ‘3D’ Views of guide vane**

Guide vane is designed to give path to the liquid which is coming out from impeller and discharges liquid to interstage casing.
- Internal diameter = 65mm
- Number of exit port = 8

E. Assembly of Pump

![Assembly of Pump](image2)

**Fig. 5. ‘3D’ Assembly of Pump**

IV. FINITE ELEMENT ANALYSIS

ANSYS (Workbench) software is used to mesh the solid model. CAD model which is in STEP format is imported to ANSYS to carry out the analysis.

A. Analysis of Rotor Shaft Assembly

a. Meshing of assembly

The conventional model which was developed in PRO-E software has to be meshed for analysis. For this ANSYS (workbench) software is used. It is a high-performance finite element pre-processor that provides a highly interactive and visual environment to analyze product design performance. With the
broadest set of direct interfaces to commercial CAD and CAE systems. The solid tetrahedron elements are used to generate the meshing of the control Arm.

![Image](image1.png)

**Fig. 6. Baseline Meshing Of Rotor And Shaft Assembly**

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Description</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Number of Nodes</td>
<td>53002</td>
</tr>
<tr>
<td>2</td>
<td>Number of Elements</td>
<td>26694</td>
</tr>
<tr>
<td>3</td>
<td>Element Size</td>
<td>Maximum 15 mm, Minimum 5 mm</td>
</tr>
</tbody>
</table>

**B. Design parameters**

In case of pump in actual running conditions forces acting on it are of dynamic in nature and changes as per discharge and head load conditions. In order to make preliminary analysis steady state operating conditions are assumed.

**b. Static Structural of baseline model**

![Image](image2.png)

**Fig.7. Equivalent static structural stress on rotating assembly**

End of the shaft at driving side connected to the coupling followed by constraint by the roller support of bearings. Changes in cross section at stuffing box housing produces stress concentration.
Maximum stress occurring at the area is less than the shaft material yield strength of 550 MPa since design is out of danger. Also blue shade indicates the minimum stress in that area.

c. Elastic strain generated due to rotational velocity

![Elastic strain due to rotation.](image)

Rotor shaft is mounted on bearing support which is connected to motor through the coupling due to the rotation of shaft and rotor stress generated within the material. strain generated at the area of red shade is much lower than the critical range of material failure.

C. ANALYSIS RESULT OF IMPELLER

a. maximum stress plot

After Finite Element analysis on ANSYS workbench 18.0 following results have been find out. The displacement contour plots of impeller are shown in the below figure 10.

![Maximum Stress Plot](image)

b. Deformation at the eye and tip of the impeller

At the eye of impeller leakages from the clearance area create stresses on the outer area of impeller; also inlet to the eye of a impeller generates stress inside area of eye. This equal and opposite stress weak the material at end section. Here material used for the impeller has higher yield strength as stress generated due to highest load.
As per maximum shear stress theory, the maximum equivalent stress observed in the model is lower than the material yield strength 512 MPa; here we notify that design is safe.

c. Elastic strain at volute casing due to outlet pressure load

Deflection in impeller is far less than the allowable limit. Deflection in casing is 37µ, which is less than limit specified by API 610.

VII. CRITICAL SPEED OF ROTOR SHAFT ASSEMBLY

The speed at which shaft and rotor vibrate violently in the direction perpendicular to axis of rotation said that critical speed. Rotational velocity and concentrated load on the shaft produces some frequency if rational load frequency matches with the natural frequency of shaft and rotor then operation of the system no longer will be good, in other words whole system may collapse.

For the proper functioning of the system it is crucial to finding the natural frequency of assembly.
Using the Dunkerley’s method estimating the critical speed of the rotor shaft assembly which is 92.6 Hz i.e. 5557.6 RPM.
Pump rotational frequency is 59.8 Hz that is 3550 RPM since to avoiding the embalncing of rotor and shaft pump should rotate below the critical speed of shaft.

VIII. EXPERIMENTAL SETUP

IX. TEST RESUL AND DISSCUSSION.

1. Head v/s Discharge and power v/s discharge test

Fig..14. Test result flow Vs capacity and flow Vs power
2. Efficiency v/s Discharge and N.P.S.H. v/s Discharge test

![Graph showing efficiency vs discharge and N.P.S.H. vs discharge](image)

**Fig.15. Test Result flow Vs efficiency and flow Vs NPSH**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Existing pump</th>
<th>New prototype pump</th>
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<tbody>
<tr>
<td>Flow (Usgpm)</td>
<td>440.20</td>
<td>610.90</td>
</tr>
<tr>
<td>Pump head (feet)</td>
<td>1320.13</td>
<td>2074.81</td>
</tr>
<tr>
<td>Pump efficiency (%)</td>
<td>66.5</td>
<td>68.0</td>
</tr>
<tr>
<td>Scale Factor</td>
<td>Base</td>
<td>1.050</td>
</tr>
<tr>
<td>Pump input (Kw)</td>
<td>164.75</td>
<td>351.41</td>
</tr>
<tr>
<td>Specific Gravity</td>
<td>1.000</td>
<td>1</td>
</tr>
<tr>
<td>Pump speed (rpm)</td>
<td>2980</td>
<td>3550</td>
</tr>
<tr>
<td>Stages Required</td>
<td>7</td>
<td>4</td>
</tr>
<tr>
<td>Motor rating</td>
<td>220</td>
<td>450</td>
</tr>
<tr>
<td>NPSHr (feet)</td>
<td>15</td>
<td>23</td>
</tr>
<tr>
<td>Specific Speed (Ns)</td>
<td>800</td>
<td>800</td>
</tr>
</tbody>
</table>

**TABLE III COMPARISON OF OLD AND NEW MODEL**

**X. CONCLUSION**

As deflection and stress of model is within the range Thus, the design of the model is safe after running pump at 3550 rpm and with enlarged scaled factor 1.05. Result obtained in the final test of the pump is satisfactorily with increase in efficiency by 1.5% to reference. Using the modern state of art composite material weight of the impeller and shaft will reduce considerably it optimizes the operation of pump effectively.

**XI. ACKNOWLEDGMENT**

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4. Christian Allerstorfer, Centrifugal pump, Dep. of design and conveying technology MU Leoben.


Mr. Kiran Shinde currently studying in M.E(Design) final year in Dept. Mechanical Engineering Dr.D.Y.Patil School of engineering, Lohgaon, Pune, India

e-mail: shinde.k33@gmail.com, currently working on design on boiler feed water pump with compact design & energy efficient pumps, he has published one research paper in international journal.

Prof. Amol B. Gaikwad (Co-Guide) currently working as Professor (PG studies) Dept. Mechanical Engineering, Dr. D. Y. Patil School of Engineering, Lohgaon, Pune, India. He has more than 6 years of teaching experience. He has published many technical papers in national and international journal.

Prof. Amol N. Patil currently working as Professor (Guide & M.E coordinator) Dept. Mechanical Engineering, Dr. D. Y. Patil School of Engineering, Lohgaon, Pune, India. He received the M.E. Degree in Design. He has more than 7.5 years of teaching experience and 6.5 years of Industrial experience. His interest are focused on structural vibrations and Finite Element Analysis. He has published many technical papers in national and international journal.