PERFORMANCE ANALYSIS OF BOILER FEED PUMP

ABHISHEK KUMAR, SHRIKUL SHRIVASTAVA, PRABHAT KUMAR MISHRA & SURYANSH GUPTA

Research Scholar, Department of Mechanical Engineering, Galgotias University, Uttar Pradesh, India

Research Scholar, Department of Mechanical Engineering, Galgotias University, Uttar Pradesh, India

Research Scholar, Department of Mechanical Engineering, Galgotias University, Uttar Pradesh, India

Research Scholar, Department of Mechanical Engineering, Galgotias University, Uttar Pradesh, India

ABSTRACT

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 enough pressure to overcome steam pressure developed by boiler. In the present study of design of boiler feed pump; there are various parameters that has great influence on the pump performance. The blade exit angle and blade inlet angle have significant effect on the discharge and the efficiency of the centrifugal pump. A MATLAB program is being formed which basically design all the essential pump parameters by taking input data obtained from plant. Volute casing curve at number of blades equal to 6 is generated and finally performance curves between blade inlet angle and discharge, blade exit angle and discharge, blade inlet angle and overall efficiency, Pump Head and efficiency are being constructed and verified by the help of data from plant and various research papers.

KEYWORDS: *Centrifugal Pumps, Impeller Shaft, Cavitation, Boiler Feed Water Pump, Energy Efficiency, Vane Profile, Volute Casing, Discharge, Inlet Blade Angle, Blade Exit Angle, Overall Efficiency, Blade Single Curvature, Diffuser, Impeller Eye Diameter, Hub Diameter, Corrosion, Sealing*

Received: Apr 05, 2020; **Accepted:** Apr 25, 2020; **Published:** Jul 21, 2020; **Paper Id.:** IJMPERDJUN2020401

1. INTRODUCTION

Fluid machinery is a direct application of subject fluid mechanics which we have been studying in our course. This subject and hence its application have a prime importance in power generation, model prototype analysis as well as in aerodynamics. Direct influence of concept of Newtonian physics which we all have read in our lower classes makes this subject more interesting and appealing.

A pump that has been widely used in industry is the most typical type of fluid machinery that transforms machinery energy into fluid pressure and kinetic energy via impellers. A centrifugal pump, the most common type of pumps, has been used in industrial areas, such as water, sewage, drainage, and the chemical industry. Accordingly, numerous studies have been performed for the designs of various models of centrifugal pumps. Due to the needs of the industry, optimization using mechanical concepts has recently been studied in order to make higher-efficiency pumps with higher heads. An impeller, among all of the components of the pump, has the biggest influence on performance, since fluid flow in the pump generates energy through it. Therefore, an accurate analysis

is essential to optimize variables that affect the performance of the pump.

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.

Therefore, in the present study, in the design of boiler feed pump there are various parameters that has great influence on the pump performance. The blade exit angle and blade inlet angle have significant effect on the discharge and the efficiency of the centrifugal pump. A MATLAB program is being formed which basically design all the essential pump parameters by taking input data obtained from plant. Volute casing curve at number of blades equal to 6 is generated and finally performance curves between blade inlet angle and discharge, blade exit angle and discharge, blade inlet angle and overall efficiency, Pump Head and efficiency are being constructed and verified by the help of data from plant and various research papers.

2. PROBLEM DESCRIPTION & METHODOLOGY:

Design Data of Boiler Feed Pump for Thermal Power Plant:

2.1. Design Input Data:

- Discharge (Q) = 23.7 lit/s
- Head to be developed = 324 m
- Rotational speed (N) = 6170 rpm
- Water temp. = 100⁰ C
- Power available for steam turbine for drive = 142 hp
- Leakage loss = 10 %

2.2. Pump Design:

Design parameters:

- Discharge (Q)
- Head (H)
- Speed of rotation (n) {n ≈ 1500 to 3000 rpm}
- Discharge through the impeller pump:

Design discharge (Q) = $\frac{Q_{total}}{k}$; Q_{total} = final discharge

k = no. of suction to the impeller

Head to be developed (H) = $\frac{H_{total}}{i}$; i = no. of stages in the pump

- Total head: It is the head to be overcome by pump = suction head + delivery head

H_s = it is head to be overcome on to suction side of pump

= suction left + friction losses + velocity head + losses

$$= h + \frac{4fL_s V^2}{2gD_s} + \frac{V_s^2}{2g} + (0.3 \text{ to } 0.4 \text{ m of work})$$

H_d = delivery head to impart by impeller liquid to the reservoir

= vertical left + friction losses in delivery pipe + velocity head imparted to water

$$= h + \frac{4fL_d V_d^2}{2gD_d} + \frac{V_d^2 - V_g^2}{2g}$$

- Specific speed of pump:

$$n_{sp} = \frac{n\sqrt{Q}}{H^{\frac{3}{4}}}$$

$\approx 11 \text{ to } 38$ (a radial flow pump with single curvature blades)

- if $n_{sp} < 15$ try to have a multistage pump so that solve of η (efficiency) gets improved.

Leakage losses, disc friction losses tend to get reduced

- if $n_{sp} < 30$ a double suction impeller is preferred
- Leakage losses: these leakage losses are generally taken 3 to 10% of shrouded type (covering on both sides)

$$Q' = (1.03 \text{ to } 1.10) Q$$

Q' = discharge being handled by impeller

$$Q' = \frac{Q}{\eta_{vol}}; \eta_{vol} = \text{volumetric efficiency (.85 to .95)}$$

Empirical relation:

$$\frac{1}{\eta_{vd}} = 1 + .68 \eta_{sq}^{-\frac{2}{3}}$$

- A pump with lower η_{sq} - higher leakage losses, less volumetric η
- A pump with high η_{sq} - lower leakage losses, high volumetric η
- Power required to drive the pump: shaft power is power input at the pump shaft coupling

$$P_{s\Box} \leq P_{em} \text{ (power output of electric motor)}$$

$$P_{s\Box} = \frac{wQH}{75\eta}$$

$$\text{overall efficiency } \eta_o = \eta_{vol} \times \eta_{\Box yd} \times \eta_{mec\Box}$$

- η_o varies between 38% in case of low sp. Speed multistage type of pump with very low flow rate.

- 90 % in case of high sp. Speed, single stage pumps with large flow rates.
- η_o (specific speed, type of pump, pump size quality of manufacture)
- η_o is maximum with single stage, single suction, radial flow pump running with sp. Speed

Reason: short length of the liquid particles inside the pump small no. of change in direction of flow.

- Hub diameter (D_h) : d_h – hub diameter at inlet d_h' – hub diameter back

$$d_h(\text{at inlet}) = (1.3 \text{ to } 1.4) \text{ (c.m.)}$$

$$d_h'(\text{at the back}) = (1.35 \text{ to } 1.5) \text{ (c.m.)}$$

- Impeller eye diameter (d_o): d_o – axial velocity with which the liquid enter into the impeller

$$d_o = \sqrt{\frac{4\theta'}{\pi c_0} + d_h'^2}$$

- Impeller inlet diameter (D_i): $D_i > d_o$.
- most commonly, $D_i: (1.05 \text{ to } 1.10)d_o$
- same time, $D_i = d_o + (5 + 10)\text{mm}$
- peripheral velocity at inlet, $u_1 = \frac{\pi D_i n}{60}$
- Impeller blade inlet angle (β_1):

From velocity diagram-

$$\tan \beta_1 = \frac{u_1}{v_{f1}}$$

2.3. Design of Diffuser:

- Diffuser angle ' δ ' –

$$\delta = \arcsin \left(\frac{v_{f2}}{v_2} \right) = f(n, s, q)$$

- Velocity of diffuser outlet (v_2) –

$$v_2 = 1 \text{ to } 2 \sqrt{2gH}$$

- Diffuser outlet diameter (D_2) –

$$Q = \left(\frac{\pi}{4} \right) D_2^2 v_2$$

- Length of diffuser –

$$\tan(\delta/4) = \frac{D_2 - D_1}{2L}$$

D_1 = diameter of the angle whose area is equal to largest casing of volute (at 360°)

l =length of diffuser

2.4. Design of Blade Single Curvature:

Selection of vanes-

- Backward curved vanes- Energy is less, outlet velocity is low. $\eta_{\text{higher}} (\beta_2 < 90^\circ)$
- Straight vane- used only in small pumps whose economy is only considered.

$\beta_1 < 90^\circ, \beta_2 < 90^\circ \eta = 55-65\%$

- Radially ending vane- superior to straight vane but cannot compete with backward curved vanes.
- Forward curved vanes- High energy transfer, High outlet velocity, low efficiency, not used in practice.

2.5. Vane Design:

- circular arc method
- point to point method.

2.6. Construction of Vane Profile:

Having now found the dimensions of impeller like hub diameter, inlet diameter Breadth of the impeller at the inlet and the outlet, outside diameter and the vane Angles, the vane profile or the curves must be generated.

The vane profile for the impeller of the pump considered as model in this work Is a multiple curvature profile with four different radii of curvature with four different centres as shown in the figure.

The Following are the Steps Involved in Tracing the Vane Profile:

- The inlet and the outlet circles are drawn.
- Two axes of reference, one vertical and one horizontal, are drawn.
- In order to trace the profile with four radii of curvature, four more circles, that is point 1 to 6 are drawn at equal intervals on the axis. The curve is drawn through A, B, C, D and E based on the positions G, H, I, J and K.
- From the point where inlet circle meets the horizontal axis, a line at an angle of inlet vane angle (20.9°) is drawn to the length of the radius of curvature of the first arc.
- An arc is drawn with the end point of this line as the centre and with the corresponding radius of curvature and passing through the previous centre.

This is the Procedure is Followed Till the Four Arcs are Drawn.

Vane profile constructed manually from the design input data taken from plant which is mentioned above:

Scale- 1m=100 cm, 1unit=2 cm.

Figure 1

Figure 2

Now, for designing following steps are followed:

Figure 3

MATLAB CODING & RESULTS:**MATLAB Coding:**

```

Q=input('Q1');

% Q is discharge in litre per second

H=input('H1');

% Head in metre

N=input('N1');

% N in rpm

WaterTemp=input('WT');

Psh=input('PA');

% Psh stands for power available

Leakageloss=input('LL');

Q2=Q/1000;

% Q2 is in cubic metre per second

NewQ=(Leakageloss*(Q/1000))+ (Q/1000);

d=957;

% d is density in kg/cubic metre

n0=0.7;

% n0 is overall efficiency

g=9.81;

% g is acceleration due to gravity

w=6.55;

% calculating power required

PowReq= (d*Q2*g*H)/(736*n0);

fprintf('Power required %fhp \n',PowReq);

% calculating shaft diameter

Sys=450;

% here Sys is taken for shaft material forged steel in kilogramforce per

% square centimetre

Dsh= (((360000*Psh)/(Sys*N))^(1/3))*10;

```

```

fprintf('Shaft diameter %fmm \n', Dsh);

Dsh1=120*((Psh/N)^(1/3));

fprintf('To check shaft diameter in both shear and bending,the diameter is %fcm \n',Dsh1);

% calculating hub diameter

Dh=1.3*Dsh1;

fprintf('Hub diameter is %fmm \n',Dh);

Dhb=1.5*Dsh1;

fprintf('The hub at the back has diameter %fmm \n',Dhb);

Kcm1=0.115;

Cm1=Kcm1*((2*g*H)^(1/2));

fprintf('Kcm1 from graph is 0.115 \n');

c0=0.95*Cm1;

A0=(NewQ/c0)*10^6;

Ah=(pi/4)*Dh*Dh;

A=A0+Ah;

d0=((4*A)/pi)^(1/2);

fprintf('inlet eye diameter is %fmm \n',d0);

D1=d0+10;

fprintf('Blade inlet dia is %fmm \n',D1);

u1=(pi*D1*N)/60000;

B1=atand(Cm1/u1);

B11=B1+3;

fprintf('Blade inlet angle is added with angle of incidence so as to improve efficiency \n');

% we can take angle of incidence between 2 to 6 degree

s1 =3;

fprintf('Taking angle of incidence is %fmm \n',s1);

B111=B11+s1;

fprintf('Blade inlet angle %fdeg \n',B11);

fprintf('taking blade thickness 5mm \n');

```



```

fprintf('Enter Number of blades to be assumed, remember it should be in between 5 to 8 \n');
z=input('Z');
t1=(pi*D1)/z;
s1=5;
Su1=s1/sind(B11);
fi=(t1-Su1)/t1;
b1=(NewQ/(pi*D1*Cm1*fi))*10^6;
fprintf('Inlet breadth is %fmm \n',b1);
nm=0.9;
nv=0.9;
nh=n0/(nm*nv);
Hth=H/nh;
fprintf('Theoretical head is %f \n',Hth);
B2 =27;
fprintf('Blade outlet angle is %f \n',B2);
r1dr2=0.5;
y=(1.2)*(1+(sind(B2)))*(r1dr2);
cp=2*(y/z)*(1/(1-(r1dr2)^2));
Hthin=Hth*(1+cp);
u2=(w/(2*(tand(B2))))+sqrt(((w/(2*(tand(B2))))^2)+(g*Hthin));
D2=((60*u2)/(pi*N))*1000;
z1=6.5*((D2+D1)/(D2-D1))*sind((B11+B2)/2);
z2=round(z1);
m=D1/D2;
ifz==z2
fprintf('Your assumption is correct, No. of blades are %d \n',z);
fprintf('Impeller outlet diameter is %f \n',D2);
elseifm<3.5
y=(1.2)*(1+(sind(B2)))*(m);

```

```

cp=2*(y/z)*(1/(1-(m)^2));
Hthin=Hth*(1+cp);
u2=(w/(2*(tand(B2))))+sqrt(((w/(2*(tand(B2))))^2)+(g*Hthin));
D2=((60*u2)/(pi*N))*1000;
z1=6.5*((D2+D1)/(D2-D1))*sind((B1+B2)/2);
z2=round(z1);
if z==z2
fprintf('Your assumption is correct, No. of blades are %d \n',z);
fprintf('Impeller outlet diameter is %f \n',D2);
else
fprintf('your assumption of no. of blades is incorrect, enter again \n');
end
else
fprintf('unknown error');
end
R=(1-((g*Hth)/(2*u2^2)));
fprintf('Degree of reaction is %f \n',R);
Kcv=0.46;
% Kcv = an emperical coefficient
Ht=9.144;
Cv=Kcv*sqrt(2*g*Ht);
fprintf('mean constant velocity of flow in the volute is %fm/s \n',Cv);
A=11.7;
fprintf('Diffuser angle based on the calculated value of Cv is %f \n',A);
for a= [0.1,0.2]
Cd=a*sqrt(2*g*Ht);
fprintf('Velocity at diffuser Outlet is %fm/s \n',Cd);
end
Cd=2.25;

```

```

fprintf('Assuming velocity of diffuser outlet is %fm/s \n',Cd)

Qd=0.00456;

% Qd is diffuser discharge

Dd=sqrt((4*Qd)/(Cd*pi))*1000;

fprintf('Diffuser outlet diameter is %fmm \n',Dd);

Av=(Qd/Cv);

fprintf('Area of volute casing is %f \n',Av);

Dv=(sqrt((Av*4)/pi))*1000;

fprintf('Diameter of circle whose area is equal to largest casing of volute (at 360 degree) is %fmm \n',Dv);

Ld=((Dd-Dv)/(2*tand (A/2)));

fprintf('length of diffuser is %fmm \n',Ld);

Cs=1.77;

fprintf('Assuming Velocity of Suction Pipe is %fm/s \n',Cs);

Qs=0.00456;

As=(Qs/(Cs))*10^4;

fprintf('area of suction pipe is %fcm^2 \n',As);

Ds=sqrt((As*4)/pi);

fprintf('diameter of suction pipe is %fcm \n',Ds);

Ad=((Qd*10^4)/Cd);

fprintf('area of delivery pipe is %fcm^2 \n',Ad);

Cdp=((Qd*10^4)/Ad);

fprintf('velocity of flow in delivery pipe is %fm/s \n',Cdp);

% Design of diffuser vanes

b2=7;

Cu2=62.05;

Cm2=6.55;

Zd=z+1;

zd=floor(Zd);

Cp=0.26;

```

```

Cu3=Cu2/(1+Cp);
C3=((Cm2)^2 +(Cu3)^2)^(0.5);
a3=atand(Cm2/Cu3);
fprintf('Number of diffuser vanes are assumed to be as %d \n',zd);
S4=3;
fprintf('Vane thickness is assumed as %dmm \n',S4);
fprintf('Radial clearance b/w impeller blades & diffuser vanes is taken as 3.5mm \n');
Rc=3.5;
K3=1.3;
D4=D2+ 2*(Rc);
t4=(pi*D4)/Zd;
a4=atand(K3*(1.15)*tand(a3));
Su4=S4/sind(a4);
fprintf('Angle of inclination of diffuser vane is %fdeg \n',a4);
Kcv=0.485;
Cv=Kcv*(2*g*H)^(0.5);
Av360=NewQ/Cv;
A4=(Av360/Zd)*10^6;
fprintf('Inlet area of diffuser passage is %fsq mm \n',A4);
b3=1.1*b2;
fprintf('Diffuser width is %f mm \n',b3);
e4=A4/b3;
fprintf('Height of inlet vane passage %fmm \n',e4);
D5=1.4*D2;
fprintf('Outer diameter of diffuser ring %fmm \n',D5);
ht=(D5-D2)/2;
fprintf('Height of diffuser passage %fmm \n',ht);
L=4*e4;
e5=1.6*e4;

```

```
Del=2*(atand((e5-e4)/(2*L)));
```

```
fprintf('Average angle of divergence of passage %fdeg \n',Del);
```

3. RESULTS

3.1. Volute Casing:

Volute casing is profile is being generated by the help of MATLAB program that uses input data, which is mentioned under problem description, having discharge 23.7 lit/s.

Figure 5 shows A different volute casing profile is generated when the discharge is $Q=36 \text{ m}^3 / \text{hr}$, which is being taken from Effects of Blade Discharge Angle, Blade Number and Splitter Blade Length on Deep Well Pump Performance.

Figure 4

Figure 5

3.2. Plot between head (m) and discharge (lit/s):

The curve in the graph clearly states that as pump head increases, pump discharge decreases if number of blades are taken as constant i.e. $Z=6$ in our study.

Figure 6

3.3. Plot between blade exit angle(β_2) and discharge (lit/s):

The curve shows that on increasing blade exit angle pump discharge increases

Figure 7

3.4. Plot between blade inlet angle and discharge:

According to the graph, increase in blade inlet angle increases pump discharge.

Figure 8

3.5. Plot between inlet blade angle and overall efficiency:

The plot shows that increase in inlet blade angle improves pump's overall efficiency.

Figure 9

4. CONCLUSIONS

Volute casing can be obtained at different discharge and head values, which implies that at different angles there is unique discharge. Here volute casing profile is generated by using data that is discharge from research paper by the help of generated MATLAB code.

For investigation of the impeller blade exit angle effect and blade inlet angle effect on the performance of the centrifugal pump various blade exit and inlet angles are examined, and variation in discharge of impeller pump is seen. The main conclusions can be drawn from our study are:

- By keeping blade number constant that is $Z=6$, if pump head increases, pump discharge increases.
- Increase in blade exit angle leads to increase in pump discharge.
- Increase in blade inlet angle leads to increase in pump discharge.
- Increasing blade inlet angle improves pump overall efficiency.
- Even trial and error methods can also have an impact on the performance of the impeller pump. Some research papers have shown that increasing the inlet angle may improve the overall efficiency to some extent and discharge improves too, which is clearly being verified by the results of our investigation.

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