A New Method to Calculate Centrifugal Pump Performance Parameters for Industrial Oils

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ABSTRACT

Pumping of oil instead of water using centrifugal pumps causes rapid increase in the hydraulic losses which results significant reduction in head and efficiency. Therefore, deriving analytical methods to calculate variation of pump performance parameters versus working fluid viscosity is very important. In the present study, a novel method is proposed to calculate the head (H), efficiency (η) and input power (P in) based on the loss analysis for pumps using industrial oils. A computer code is developed based on represented method and the results of this method are compared with experimental results for a centrifugal pump of type KWP K-Bloc65-200. The results show good agreement between analytical and experimental methods. Finally, using such computer code, diagrams of head, efficiency and input power versus working fluid viscosity are plotted. The results show an interesting point known as “sudden rising head” which is observed experimentally and numerically in literatures.

Keywords: Centrifugal pump; Loss analysis; Analytical method; Industrial oil.

NOMENCLATURE

A Area (m²)
b width of channel in meridional section (m)
c absolute velocity (m/s)
C eq roughness equivalence factor
c f friction coefficient
d diameter (m)
d a arithmetic average of diameters at impeller (m)
d w geometric average of diameters at impeller (m)
e vane thickness (m)
f geo geometrical factor
f H correction factor for head
f Q impeller eyes per impeller
f Q correction factor for flow rate
f K roughness influence on disk friction
f η correction factor for efficiency
f ηb correction factor for hydraulic efficiency
g acceleration of gravity (m/s²)
H head (m)
q* flow rate referred to flow rate at BEP
r radius (m)
Re Reynolds number
S w axial distance between impeller shrouds and casing
u circumferential velocity (m/s)
ε equivalent sand roughness (m)
γ slip factor
η efficiency
η h hydraulic efficiency
η vol volumetric efficiency
η η head coefficient
ρ density (kg/m³)
Q s leakage flow rate through seal at impeller inlet
ν kinematic viscosity (m²/s)
τ blade blockage factor
ω angular rotor velocity (1/s)
1. INTRODUCTION

Hydraulic losses in all components of pump are made up by friction and turbulent dissipation losses. Friction losses are depended on working fluid viscosity and therefore pump performance parameters are highly affected by this parameter. It should be noted that this is because the flow regime of highly viscous fluids tends to be laminar, roughness of impeller side walls and roughness of hydraulic channels is of little importance for operation with viscous fluids. Due to importance of working fluid viscosity, there have been great researches done to study viscous fluid pumping (Nemdili and Hellmann 2007; Li 1999, 2000, 2002, 2004, 2008, 2010; Gulich 2003). Most of these studies are based on experimental works which are time-consuming and costly.

Li et al. (Li and Hu 1997) studied effect of viscosity of fluid on performance of centrifugal pumps, experimentally. In their studies, oil with different kinematic viscosities was used. Results of such studies show that by increasing viscosity, head and efficiency in different operational conditions decrease and input power increases. They also represented correction factors for calculating pump performance parameters when the pump works with viscous fluids. In this way, mechanical, volumetric and disk friction losses are considered in calculating hydraulic efficiency. A computer program was developed for computing the pump performance parameters such as head, efficiency and input power for viscous fluid pumping. Results of this new method were verified by comparing with the experimental results. Diagram of head versus fluid viscosity are also derived using such computer program. The interesting point known previously as “sudden rising head” can be seen on such diagram.

2. CALCULATION OF PUMP PERFORMANCE PARAMETERS FOR VISCOUS FLUID PUMPING

In this section, performance parameters of centrifugal pump such as head (H), efficiency (η) and input power (P_w) for viscous fluid pumping will be derived using new method based on loss analysis.

Geometries of the typical centrifugal pump impeller have been shown in Fig. 1.

Hydraulic, mechanical, volumetric and disk friction losses are considered in this loss analysis, separately. Friction coefficient is the most important parameter in hydraulic losses. The friction coefficient is depended on flow regime and relative roughness. Transition from laminar to turbulent flow occurs at the critical Reynolds number Re_ww which can be calculated as follow
where \( Tu \) is turbulence intensity and in the present study is considered as 0.1 (Ladouani and Nemdili 2009). According to the value of \( Re_{cri} \) flow regime will be determined. Reynolds number can be calculated from

\[
Re = \frac{u_2 d_{2b}}{v}
\]  
(2)

where \( d_{2b}, u_2 \), and \( v \) are impeller outlet diameter, circumferential velocity in impeller outlet and viscosity of fluid, respectively.

Friction coefficient of the centrifugal pumps can be written as

\[
\epsilon_{f,turb} = \frac{0.136}{(-\log(0.2 \frac{E}{r_{2b}})^{1.15}} \text{ Re} > Re_{cri}
\]  
(3)

\[
\epsilon_{f, lam} = \frac{2.65}{Re^{0.875}} \left( 2 \frac{2}{8Re^{+} 0.016} + 1.328 + \sqrt{Re} \right)
\]

0.01 ≤ Re ≤ Re_{cri}

(4)

where \( E \) denotes the equivalent sand roughness and is defined as

\[
E = \frac{6 \epsilon_{CLA}}{C_{eq}}
\]  
(5)

where \( C_{eq} \) and \( \epsilon_{CLA} \) represent the equivalence factor and center line average roughness, respectively. \( C_{eq} \) is considered as 2.6 (Gulich 2008) and \( \epsilon_{CLA} \) is also considered as 100 \( \mu \)m according to the experimental results (Shojaeefard et al 2012).

The best method to achieve mechanical losses is to use documentation of the manufacturer of the mechanical equipment. If more accurate information is not available the mechanical losses in its BEP can be estimated using equation below.

\[
\frac{P_m}{P_n} = 0.0045 \left( \frac{Q_{ref}}{Q} \right)^{0.4} \left( \frac{n_{ref}}{n} \right)^{0.3}
\]  
(6)

In this equation \( Q_{ref} \) and \( n_{ref} \) are considered as 1 m\(^3\)/s and 1450 rpm, respectively. The mechanical efficiency of large pumps is around 99.5% or even above (Gulich 2008).

Disk friction losses are caused by shear stresses in boundary layers on solid structures. These are dependent on Reynolds number and surface roughness which have significant effects on the thin boundary layers and attached flow in comparison with decelerated or separated flow. Since Reynolds number decreases, disk friction losses will be increased. Disk friction effect on efficiency is important especially in low rotational speed. The disk friction losses equation can be written as

\[
K_{RR} = \frac{\pi r_2^2}{2ReS_{ax}} + \frac{0.02}{Re^{0.2}} \frac{1 + \frac{S_{ax}}{r_2}}{1 + \frac{S_{ax}}{r_2}} f_R
\]  
(7)

where \( S_{ax} \) is geometric parameter which is considered as 0.05 \( d_{2b} \). Also, \( f_R \) indicates roughness influence on disk friction which has been extracted in (Gulich 2008). Using experimental data such as head and volumetric flow rate of water at the BEP for each impeller geometry, head coefficient and specific speed for water can be determined (Gulich 2008).

\[
\psi_{BEP} = \frac{2gH_{BEP}}{(r_{2b} \alpha)^2}
\]  
(8)

\[
n_{BEP} = \sqrt{\frac{Q_{BEP}}{f_q}}
\]  
(9)

\( f_q \) denotes impeller eyes per impeller. For impellers with one entrance \( f_q \) is equal to one (Gulich 2008).
Disk friction power is written as

$$P_{fr} = \frac{K_{fr} \rho \omega^3 R^5 f_{geo}}{\eta_f^2 \beta_{2B} f_q}$$ \hspace{1cm} (10)

where geometric factor $f_{geo}$ for radial impellers is considered as 1.22 (Gulich 2008). Density of water ($\rho$) is also considered as 1000 $\frac{Kg}{m^3}$ at 25°C.

Volumetric efficiency can be evaluated by following formula

$$\eta_{vol} = \frac{Q}{Q_f + Q_{SP} + Q_E}$$ \hspace{1cm} (11)

where $Q$ indicates volumetric flow rate, $Q_f$ is leakage through the annular seal at the impeller inlet and $Q_{vol}$ is the volumetric flow through a device for axial thrust balancing flow. According to the reference (Gulich 2008), $Q_f$ and $Q_{vol}$ can be considered equal, approximately.

Hydraulic losses consist of mixing and friction losses in all components between suction and discharge nozzles affect hydraulic efficiency. Mixing losses are the result of non-uniform distribution of velocity. These losses are mostly depended on Reynolds number.

Flow deceleration causes a thick boundary layer which makes non-uniform velocity distribution. In non-uniform flow, momentum exchange between streamlines will be increased because of different eddies with various scales. Water hydraulic efficiency at BEP can be calculated as following equation.

$$\eta_{vol,w} = \frac{\eta_w}{1 - \eta_w \left(\frac{P_{fr}}{P_w} + \frac{P_{SP}}{P_w} + \frac{P_E}{P_w}\right)}$$ \hspace{1cm} (12)

Hydraulic correction factor is defined as following equation which indicates ratio of viscous fluid hydraulic efficiency to water

$$f_{ph} = \frac{\eta_{vol,w}}{\eta_{vol,w} - 1} = \frac{\frac{\xi_{R,W}}{\psi_{BEP}}}{1 - \frac{\xi_{R,W}}{\psi_{BEP}}} - 1$$ \hspace{1cm} (13)

$\xi_{R,W}$ can be found according to equation stated in reference (Gulich 2008).

Multiplying hydraulic efficiency correction factor by water hydraulic efficiency at BEP, viscous fluid efficiency in this point is achieved in the form of:

$$\eta_{h,w} = f_{ph} \eta_{h,w}$$ \hspace{1cm} (14)

Head of centrifugal pump at BEP for viscous fluid can be written as

$$H = \frac{\eta_{h,w} \frac{H^2}{g} \left(\frac{C_{2m}}{\eta_2 \tan \beta_{2B}} \tau \frac{\gamma}{2}\right)}{2}$$ \hspace{1cm} (15)

where $\eta_h$, $\gamma$, $\tau$ and $\beta_{2B}$ are the hydraulic efficiency, slip factor, blade blockage factor and the blade outlet angle , respectively. $u_2$ and $C_{2m}$ stand for circumferential and meridional components of absolute velocity at impellers outlet.

Blade blockage and slip factors can be calculated from equations (16) and (17).

$$\tau_2 = \left(1 - \frac{z_2}{\tau f_{2B} \sin \beta_{2B}}\right)^{-1}$$ \hspace{1cm} (16)

$$\gamma = \frac{f_1}{1 - \frac{\sin \beta_{2B}}{\lambda}}$$ \hspace{1cm} (17)

where $k_w$ denotes the influence of impellers inlet diameter on slip factor. Value of coefficient $f_j$ for radial impeller is considered as 0.98(Gulich 2008). The absolute velocity will be calculated from equation (18),

$$c_m = \frac{Q}{\pi d_b}$$ \hspace{1cm} (18)

To calculate $C_{2m}$ at BEP, it is necessary to calculate volumetric flow rate of viscous fluid in this point which vary by variation of kinematic viscosity. To calculate volumetric flow rate of viscous fluid according to the kinematic viscosity divergence, a new method is proposed and employed in the is calculated from present study. At the first step, head correction factor

$$f_{H,BEP} = \frac{H_{BEP,v}}{H_{BEP,w}}$$ \hspace{1cm} (19)

where $H_{BEP,v}$ and $H_{BEP,w}$are the head of the centrifugal pump at BEP for viscous fluid and water, respectively and superscript i denotes the step number. It is needed to obtain the value of $H_{BEP,w}$ for each pump with specific impeller, experimentally. $H_{BEP,v,i}$ is the value calculated at the step (4) , but it is required to assume an initial value for this parameter. In this way, initial estimation of the $Q_{BEP,v,i}$ is considered equal to value of the water volumetric flow rate at BEP. The value of $C_{2m}$ is then calculated from equation (18). Using obtained values for $Q_{BEP,v,i}$ and $C_{2m}$ the value of $H_{BEP,v,i}^0$ can now be extracted by equation (15).

At the 2nd step, volumetric flow rate for viscous fluid pumping $Q_{BEP,v}$ is calculated by

$$Q_{BEP,v} = Q_{BEP,w} f_{H,v,BEP}$$ \hspace{1cm} (20)

where $f_{H,v,BEP}$ was obtained from step (1). $Q_{BEP,v}$ is value of the water volumetric flow rate of the pump at BEP which is obtained experimentally. It should be noted that to derive equation (20) it is supposed that $f_{H,BEP} = f_{q,BEP}$ (Gulich 2008).

At the 3rd step, meridian component of the outlet absolute velocity $C_{2w}$ is obtained from equation (18)
as
\[ e_{2m}^{i} = \frac{Q_{BEP,v}^i}{\pi d_{2b} L_{2}} \]  (21)
where \( Q_{BEP,v}^i \) was calculated at the step (2).

At the 4th step, new value for viscous fluid head is calculated from equation (15) as
\[ H_{BEP,v}^{i+1} = \frac{H_{BEP,v}^i}{f_{H,BEP}} \]  (26)
where \( f_{H,BEP} \) was obtained at step (3).

At the last step, the error value of the viscous fluid head is examined using following criteria
\[ H_{i+1} - H_i \leq \delta \]  (23)
where \( \delta \) is the user defined reference value considered as 0.01 in this paper. If the value of \( H_{i+1} - H_i \) is bigger than \( \delta \), then a return to the step (1) and consideration \( H_{i+1} \) will be occurred, else the algorithm will be finished and value of viscous fluid head, volumetric flow rate, head correction factor and volumetric flow rate correction factor at BEP can be obtained from
\[ H_{BEP,v} = H_{i+1} \]  (24)
\[ Q_{BEP,v} = Q_i \]  (25)
\[ f_{H,BEP} = \frac{H_{BEP,v}^i}{H_{BEP,v}^i} \]  (26)
\[ f_0 = \frac{Q_{BEP,v}^i}{Q_{BEP,v}^i} \]  (27)

After computing \( H_{BEP,v}^i \) and \( f_{H,BEP} \), it is possible now to calculate head of the pump for viscous fluid pumping at other working conditions (part load and over load). In this way, the following equation can be used
\[ H_v = H_{w} \cdot f_H \]  (28)
where \( H_v \) is head of centrifugal pump based on water volumetric flow rate achieved experimentally. \( f_H \) is the head correction factor which is obtained from (Gulich 2008)
\[ f_H = 1 - (1 - f_{H,BEP})(q^*)^{0.75} \]  (29)
where \( q^* \) is \( \frac{Q_v}{Q_{BEP,w}} \).

volumetric flow rate of viscous fluid in any working condition can be obtained by
\[ Q_v = f_0 \cdot Q_{w} \]  (30)
where \( f_0 \) stands for flow rate correction factor that will be independent from water volumetric flow rate. Pump efficiency for viscous fluid pumping is obtained using equation
\[ \eta_v = f_q \cdot \eta_w \]  (31)
where \( \eta_w \) is pump efficiency for water handling which is obtained experimentally. \( f_q \) stands for efficiency correction factor that can be written as
\[ f_q = \left( \frac{Re_{mod}}{Re_{mod}} \right)^{\gamma} \]  (32)
where \( \gamma \) is considered as 0.7. \( Re_{mod} \) is represented in reference(Gulich 2008). Equation (32) is based on the loss analysis and \( f_q \) is independent of pump performance condition.

Useful power transferred to the viscous fluid can be written as
\[ P_v = \rho_v \cdot H_v \cdot Q_v \cdot g \]  (33)
where \( g \) is acceleration of gravity and \( \rho_v \), \( H_v \), \( Q_v \) stand for density, head and volumetric flow rate of viscous fluid, respectively. Density of each working fluid can be obtained from ASTM D7042 according to the fluid kinematic viscosity at 25°C.

Input power as an important parameter can be achieved by dividing useful power transferred to fluid by pump efficiency, which will be defined as:
\[ P_{in} = \frac{P}{\eta_v} \]  (34)

The procedure of proposed method is summarized in Fig. 3.

3. EXPERIMENTAL PROCEDURE

In order to evaluate the new method represented in section (2) some experimental tests are carried out by the authors. Tests programs and comprehensive explanations can be found in reference (Shojaeefard et al 2012). The pipe of the rig was made of stainless steel with inner diameter of 80 mm. The tank net volume was 2400 liters. The operation condition was controlled by a gate valve on the discharge pipe. A digital pressure transmitter gage was used at the pump inlet and outlet pipes to measure the inlet and outlet pressures accurately. Baffle plates were used for damping the disturbance of discharged fluid. Test setup schematic is shown in Fig. 2. KWP K-Bloc65-200 pump made by Pumpiran Company is used for experimental tests. This centrifugal pump has single axial suction and vane less volute casing. The pump test is driven by a three-phase AC electric motor, which its rated power is 5.5 KW and speed is 1450 rpm.

The performance parameter of centrifugal pump for water and oil with viscosity of 1 and 43 CSt for three different geometries of impeller (A, B and C) under steady conditions were measured. Specifications of impellers A, B and C are depicted in table (1).

The experimental results for six experiments at best efficiency point are indicated in table (2).
Fig. 2. Centrifugal pump test setup.

Fig. 3. Flow chart of computer code.
Table 1 Specification of the tested impellers A, B and C

<table>
<thead>
<tr>
<th>Impeller Type</th>
<th>Impeller parameters</th>
<th>β₁B</th>
<th>β₂B</th>
<th>b₁</th>
<th>b₂</th>
<th>d₁B</th>
<th>d₂B</th>
<th>e₁</th>
<th>e₂</th>
<th>c</th>
<th>eCLA</th>
<th>ZLa</th>
</tr>
</thead>
<tbody>
<tr>
<td>A (Original impeller)</td>
<td>Dimensions</td>
<td>30</td>
<td>27.5</td>
<td>34</td>
<td>17</td>
<td>109</td>
<td>103</td>
<td>3.5</td>
<td>3.5</td>
<td>3.5</td>
<td>100</td>
<td>6</td>
</tr>
<tr>
<td>B</td>
<td></td>
<td>30</td>
<td>30</td>
<td>42</td>
<td>21</td>
<td>109</td>
<td>103</td>
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<td>3.5</td>
<td>3.5</td>
<td>100</td>
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<tr>
<td>C</td>
<td></td>
<td>30</td>
<td>32.5</td>
<td>34</td>
<td>17</td>
<td>109</td>
<td>103</td>
<td>3.5</td>
<td>3.5</td>
<td>3.5</td>
<td>100</td>
<td>6</td>
</tr>
</tbody>
</table>

Table 2 Experimental and analytical results achieved in three different impeller geometries for various viscosities

<table>
<thead>
<tr>
<th>Test number</th>
<th>Impeller Type</th>
<th>Geometry</th>
<th>Physical</th>
<th>Head</th>
<th>Power consumption</th>
<th>Total efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>β₁B</td>
<td>b₂</td>
<td>ρ</td>
<td>ν</td>
<td>Experimental</td>
<td>Analytical</td>
</tr>
<tr>
<td>1</td>
<td>A</td>
<td>27.5</td>
<td>17</td>
<td>1000</td>
<td>1</td>
<td>12.7</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>27.5</td>
<td>17</td>
<td>875</td>
<td>43</td>
<td>12.45</td>
</tr>
<tr>
<td>3</td>
<td>B</td>
<td>30</td>
<td>21</td>
<td>1000</td>
<td>1</td>
<td>12.8</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>30</td>
<td>21</td>
<td>875</td>
<td>43</td>
<td>12.11</td>
</tr>
<tr>
<td>5</td>
<td>C</td>
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<td>17</td>
<td>1000</td>
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<td>13.3</td>
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<td>C</td>
<td>32.5</td>
<td>17</td>
<td>875</td>
<td>43</td>
<td>12.15</td>
</tr>
</tbody>
</table>

4. RESULTS AND DISCUSSIONS

A computer program has been developed for performing the new method presented in section (2). The code has been developed using MATLAB R2009a. To verify the proposed method, the results of this work are compared with the experimental results. Analytical results for two viscosities of 1 and 43 CSt for three different impeller (A, B and C) are shown in table (2). Comparing experimental and analytical results shows that these results are in agreement with each other.

Figures 4, 5 and 6 show the performance curves (H-Q, η-Q, P-Q) of centrifugal pump for pumping fluids with viscosities of 1, 35, 86, 119 and 228 centistokes for three different impeller geometries A, B and C, respectively, extracted by the new method presented in this paper. When the highly viscous fluid is pumped by centrifugal pump, the head and efficiency of the pump decrease and simultaneously input power increases with increase in fluid viscosity due to increase in hydraulic losses and skin friction.

Fig. 5. The performance curve of centrifugal oil pump with different viscosities for impeller "B".

Fig. 6. The performance curve of centrifugal oil pump with different viscosities for impeller.

Figure 7 illustrates pumps head variation versus working fluid viscosities for three different impeller geometries at BEP. It can be seen in this figure that decreasing trend of the curve is changed and a sudden rise in head occurs at viscosity of 115 centistokes. For ν ≥ 115 centistokes, Reynolds number (see equation (2)) will be smaller than critical Reynolds number (see equation (1)) and thus flow regime changes to laminar. It is obvious from equation (4) that, in laminar regime, roughness of
impeller walls and hydraulic channels has no significant effects on the friction coefficient.

Moreover, in this case, increase in Reynolds number results in friction coefficient reduction. Furthermore, for \( \nu \geq 115 \) CST the flow of the boundary layers on the impeller walls and hydraulic channels is in the hydraulically smooth region which causes reduction in friction coefficient, too. It is evident that reduction in \( c_f \) results in increasing of \( \eta_{sh} \) and \( H \) (see equations (13) and (15)). This phenomenon namely sudden-rising head has been observed in reference (Li 2000, 2008) previously. Sudden-rising head also can be seen in \( H-Q \) curves (Figs. 4, 5 and 6) between \( \nu=86 \) CST and \( \nu=119 \) CST.

Figs. 8 and 9 represents the variation of maximum efficiency and input power with fluid viscosity for three different geometries, respectively. It is clear from fig. 8 that, the slope of \( \eta - \nu \) curve decreases with increase in viscosity of fluid. It is because that, according to equation (32) a sharp fall in efficiency correction factor occurs when viscosity is changed from 1 to 35 centistokes, otherwise for \( \nu \geq 35 \) CST increasing in viscosity causes less fall in this factor. Furthermore, as shown in fig. 9 a steep slope is seen in the range of 100 CST to 120 CST. The steep slope of \( P_{in} \) can be justified using the sudden increase in the head that is shown in fig. 7. According to equation (33), as the head increases suddenly, \( P_{in} \) shows a significant growth.

5. CONCLUSION

In this paper an analytical investigation into the performance of centrifugal pumps for viscous fluid pumping has been carried out. Considering the comparison between the performance of centrifugal pump with different impeller during pumping water and oil, the following conclusions can be made:

- The results obtained from experimental and analytical methods have satisfactory agreement which shows accuracy and reliability of new proposed method based on loss analysis.
- According to figure \( \eta - Q \) for three different impellers, the best efficiency point shifts to lower flow rates at higher hydraulic losses.
- Comparing the performance curve of centrifugal pump for three different impeller geometries according to Figs. 4, 5 and 6 illustrates that geometry B have got a priority over two other geometries of A and C for pumping viscous fluid. This is because of increasing the impeller passage width from 17 to 21 mm which increases the head and hydraulic efficiency due to reduction of the friction losses. Also the centrifugal pump performance with the impeller passage width equal to 21 mm, at outlet blade angle of 30° is improved in comparison with 27.5° and 32.5°. This is due to reduction of the dissipation arising by vortex formation in impeller passage when the pump handles viscous liquid.
- Figure \( H - \nu \) shows that in viscosity of 115 centistokes the sudden rising head is occurred.
- According to figure \( P_{in} - \nu \), the slope of increase of \( P_{in} \) with the growth of \( \nu \) in geometry B is smaller than other geometries due to the reduction of the hydraulic loss in this geometry.

REFERENCES


with Centrifugal Pumps-Part 2. *World pumps.*


