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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 (m^2)
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_b	arithmetic average of diameters at impeller (m)
d_m	geomet ric 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_R	roughness influence on disk friction
f_η	correction factor for efficiency
$f_{\eta h}$	correction factor for hydraulic efficiency
g	acceleration of gravity (m/s^2)
H	head (m)
q^*	flow rate referred to flow rate at BEP
r	radius (m)
Re	Reynolds number
s_{ax}	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
ψ	head coefficient
ρ	density (kg/m^3)
Q_{sp}	leakage flow rate through seal at impeller inlet
ν	kinematic viscosity (m^3/s)
τ	blade blockage factor
ω	angular rotor velocity ($1/s$)

K_{RR}	disk friction coefficient
k_W	influence of impeller inlet diameter on slip factor
n	rotational speed (rpm)
n_q	specific speed
P_{in}	input power (w)
P_m	mechanical power losses (w)
P_{RR}	disk friction power (w)
P_u	useful power transferred to fluid
Q	flow rate $\left(m^3/s\right)$
Q_E	flow rate through axial thrust balancing device
ω_s	universal specific speed
β_B	blade angle

Abbreviations, subscripts, superscripts

1	impeller blade leading edge
2	impeller blade trailing edge
BEP	best efficiency point
CLA	center line average
crit	critical number
i	step number
imp	impeller
Lam	laminar flow
m	meridional component
mod	modify
Tu	turbulent level
turb	turbulent flow
w	water
ν	oil

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 such as head, efficiency and volumetric flow rate when the pump works with viscous fluid instead of water. Correction factors are obtained using experimental data as: $K_\eta = \dfrac{\eta_\nu}{\eta_w}$, $K_H = \dfrac{H_\nu}{H_w}$, $K_Q = \dfrac{Q_\nu}{Q_w}$, where Q, H and η are volumetric flow rate, head and efficiency, respectively and subscripts ν and w represent viscous and water fluids. Similar works have been performed by Gulich et al. (Gulich 1999, 2003) to investigate fluid viscosity effects on the centrifugal pump performance. Shojaeefard et al. (Shojaeefard et al 2006, 2007, 2012) did vast researches to optimize centrifugal pump geometry for viscous fluid pumping. These researches include experimental and numerical studies of impeller roughness, number of impeller blades, passage width, blade outlet angle and type of impeller (closed or semi open). They studied centrifugal pumps performance using water and oil with different viscosities and evaluated role of fluid viscosity in operation of centrifugal pump.

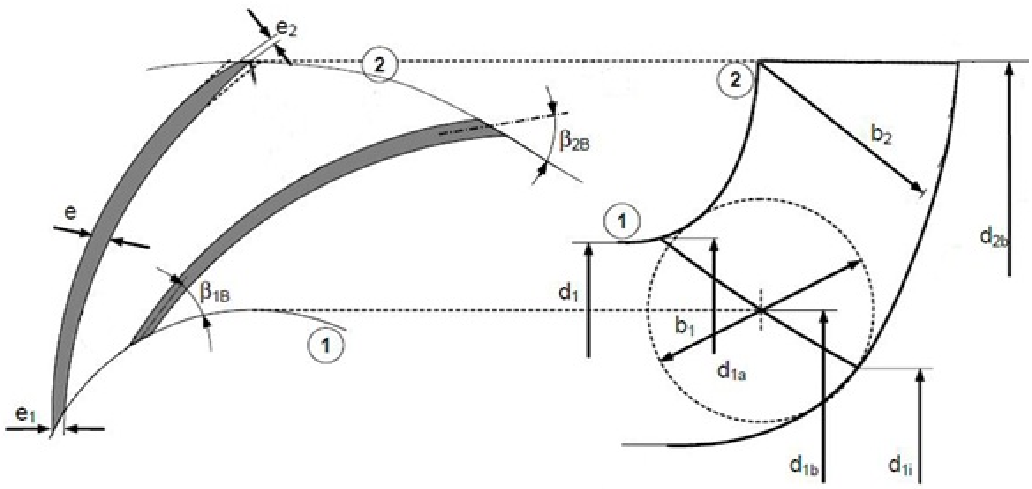
Most of previous researches were done experimentally or numerically. In this paper, unlike previous researches, analytical method based on loss analysis is represented for calculating centrifugal pump performance parameters when the pump works with viscous fluids. In this way, mechanical, volumetric 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_{in}), 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_{crit} which can be calculated as follow



(a) **(b)**

Fig. 1. a) A view of the modeled impeller b) Meridian plane of the impeller

$$Re_{crit} = \frac{3\times10^6}{1+10^4.Tu^{1.7}} \tag{1}$$

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_{crit} flow regime will be determined. Reynolds number can be calculated from

$$Re = \frac{u_2.d_{2b}}{2\nu} \tag{2}$$

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

Friction coefficient of the centrifugal pumps can be written as

$$c_{f,turb} = \frac{0.136}{(-\log(0.2\frac{\varepsilon}{r_{2b}}+\frac{12.5}{Re}))^{2.15}} \quad Re > Re_{crit} \tag{3}$$

$$c_{f,lam} = \frac{2.65}{Re^{0.875}} - \frac{2}{8Re+\frac{0.016}{Re}} + \frac{1.328}{\sqrt{Re}}$$

$$0.01 \le Re \le Re_{crit} \tag{4}$$

where ε denotes the equivalent sand roughness and is defined as

$$\varepsilon = \frac{6.\varepsilon_{CLA}}{C_{eq}} \tag{5}$$

where C_{eq} and ε_{CLA} represent the equivalence factor and center line average roughness , respectively. C_{eq} is considered as 2.6 (Gulich 2008) and ε_{CLA} is also considered as 100 µm 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.

$$\frac{P_m}{P_u} = \frac{0.0045}{\eta}.(\frac{Q_{ref}}{Q})^{0.4}.(\frac{n_{ref}}{n})^{0.3} \tag{6}$$

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 depended 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

$$K_{RR} = \frac{\pi.r_2}{2Re.S_{ax}} + \frac{0.02}{Re^{0.2}}.\frac{1+\frac{S_{ax}}{r_{2b}}}{1+\frac{S_{ax}}{2r_{2b}}}.f_R \tag{7}$$

where S_{ax} is geometric parameter which is considered as $0.05\cdot d_{2b}$. 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).

$$\psi_{BEP} = \frac{2g.H_{BEP}}{(r_{2b}.\omega)^2} \tag{8}$$

$$n_q = \frac{n\sqrt{\frac{Q_{BEP}}{f_q}}}{H_{BEP}^{0.75}} \tag{9}$$

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

$$\frac{P_{RR}}{P_u}=\frac{K_{RR}.\rho.\omega^3.r_{2b}{}^5\cdot f_{geo}}{n_q{}^2.\psi_{BEP}^{2.5}\cdot f_q} \tag{10}$$

where geometric factor f_{geo} for radial impellers is considered as 1.22 (Gulich 2008). Density of water (ρ) is also considered as 1000 $\frac{Kg}{m^3}$ at 25^{0c}.
Volumetric efficiency can be evaluated by following formula

$$\eta_{vol}=\frac{Q}{Q+Q_{SP}+Q_E} \tag{11}$$

where Q indicates volumetric flow rate ,Q_{SP} is leakage through the annular seal at the impeller inlet and Q_E is the volumetric flow through a device for axial thrust balancing flow. According to the reference (Gulich 2008), Q_{SP} and Q_E 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.

$$\eta_{h,w}=\frac{\eta_w}{\eta_{vol,w}\left[1-\eta_w\left[(\frac{P_{RR}}{P_u})+\frac{P_m}{P_u}\right]\right]} \tag{12}$$

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

$$f_{\eta h}=\frac{\eta_{h,v}}{\eta_{h,w}}=1-\frac{\xi_{R,W}}{\psi_{BEP}}\left(\frac{c_{f,v}}{c_{f,w}}-1\right) \tag{13}$$

$\frac{\xi_{R,W}}{\psi_{BEP}}$ 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:

$$\eta_{h,v}=f_{\eta h}.\eta_{h,w} \tag{14}$$

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

$$H=\frac{\eta_h.u_2^2}{g}\left(\gamma-\frac{c_{2m}}{u_2.\tan\beta_{2B}}.\tau_2\right) \tag{15}$$

where η_h , γ , τ and β_{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 meridian components of absolute velocity at impellers outlet .

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

$$\tau_2=(1-\frac{z_{la}.e}{\pi.d_{2b}.\sin\beta_{2B}})^{-1} \tag{16}$$

$$\gamma=f_1\left(1-\frac{\sqrt{\sin\beta_{2B}}}{z_{la}^{0.7}}\right)k_w \tag{17}$$

where k_w denotes the influence of impellers inlet diameter on slip factor. Value of coefficient f_1 for radial impeller is considered as 0.98(Gulich 2008). The absolute velocity will be calculated from equation (18),

$$c_m=\frac{Q}{\pi\cdot d_b\cdot b} \tag{18}$$

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

$$f_{H,BEP}^i=\frac{H_{BEP,v}^i}{H_{BEP,w}} \tag{19}$$

where $H_{BEP,v}^i$ 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 $Q_{BEP,v}^0$ is considered equal to value of the water volumetric flow rate at BEP. The value of c_{2m}^0 is then calculated from equation (18). Using obtained values for $Q_{BEP,v}^0$ and c_{2m}^0 the value of $H_{BEP,v}^0$ can now be extracted by equation (15).

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

$$Q_{BEP,v}^i=Q_{BEP,w}\cdot f_{H,BEP}^i \tag{20}$$

where $f_{H,BEP}^i$ was obtained from step (1). $Q_{BEP,w}$ 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_{2m} is obtained from equation (18) as

$$c_{2m}^{i} = \frac{Q_{BEP,\nu}^{i}}{\pi . b_2 . d_{2b}} \quad (21)$$

where $Q_{BEP,\nu}^{i}$ was calculated at the step (2).

At the 4th step, new value for viscous fluid head is calculated from equation (15) as

$$H_{BEP,\nu}^{i+1} = \frac{\eta_h . u_2^{2}}{g}\left(\gamma - \frac{c_{2m}^{i}}{u_2 . \tan\beta_{2B}}\tau_2\right) \quad (22)$$

where c_{2m}^{i} was obtained at step (3).

At the last step, the error value of the viscous fluid head is examined using following criteria

$$H_{i+1} - H_i \le \delta \quad (23)$$

where δ 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 δ, then a return to the step (1) and consideration $(H_i)_{new} = H_{i+1}$ 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

$$H_{BEP,\nu} = H_{i+1} \quad (24)$$

$$Q_{BEP,\nu} = Q_i \quad (25)$$

$$f_{H,BEP} = \frac{H_{BEP,\nu}}{H_{BEP,w}} \quad (26)$$

$$f_Q = \frac{Q_{BEP,\nu}}{Q_{BEP,w}} \quad (27)$$

After computing $H_{BEP,\nu}$ 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

$$H_\nu = H_w . f_H \quad (28)$$

where H_w 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)

$$f_H = 1 - (1 - f_{H,BEP}) \cdot (q^*)^{0.75} \quad (29)$$

where q^* is $\frac{Q_w}{Q_{BEP,w}}$.

volumetric flow rate of viscous fluid in any working condition can be obtained by

$$Q_\nu = f_Q . Q_w \quad (30)$$

where f_Q 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

$$\eta_\nu = f_\eta \cdot \eta_w \quad (31)$$

where η_w is pump efficiency for water handling which is obtained experimentally. f_η stands for efficiency correction factor that can be written as

$$f_\eta = (\mathrm{Re}_{mod})^{\frac{-19}{\mathrm{Re}_{mod}^{\,y}}} \quad (32)$$

where y is considered as 0.7. Re_{mod} is represented in reference(Gulich 2008). Equation (32) is based on the loss analysis and f_η is independent of pump performance condition.

Useful power transferred to the viscous fluid can be written as

$$P_u = \rho_\nu . H_\nu . Q_\nu . g \quad (33)$$

where g is acceleration of gravity and , H_ν , Q_ν 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^{oC}.

Input power as an important parameter can be achieved by dividing useful power transferred to fluid by pump efficiency, which will be defined as:

$$P_{in} = \frac{P_u}{\eta_\nu} \quad (34)$$

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).

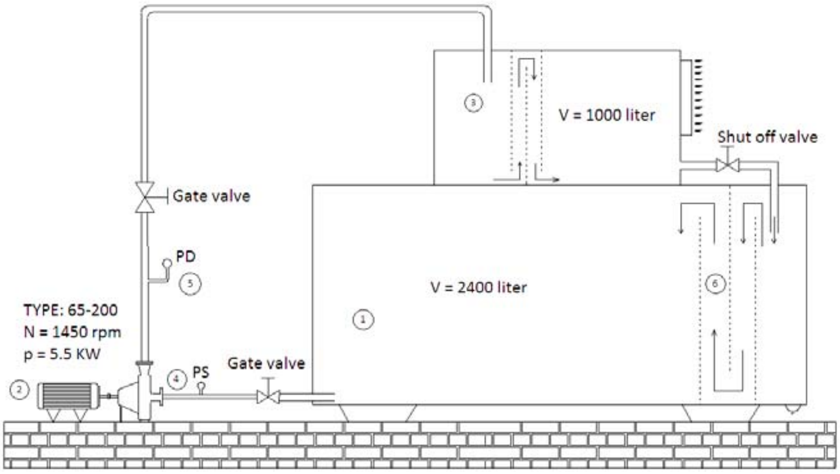


Fig. 2. Centrifugal pump test setup.

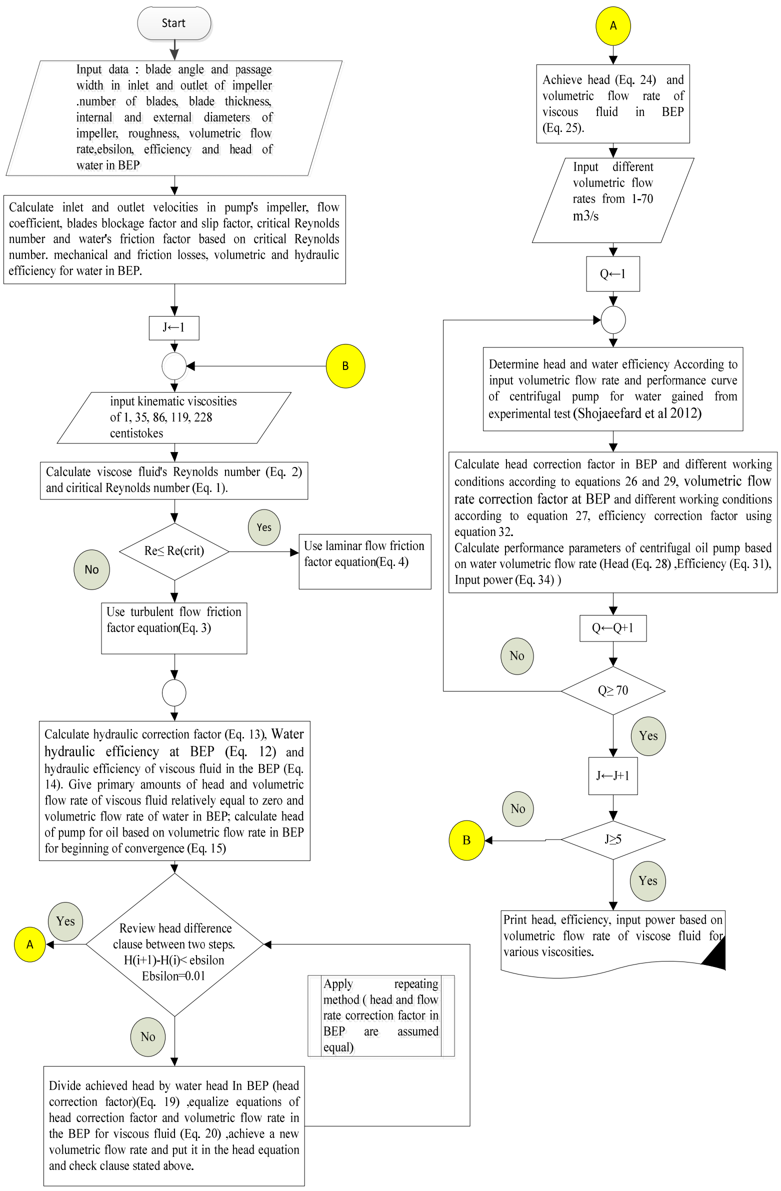


Fig. 3. Flow chart of computer code.

Table 1 Specification of the tested impellers A, B and C

Impeller Type	parameters	β_{1B}	β_{2B}	b_1	b_2	d_{1b}	d_{2b}	e_1	e_2	e	ε_{CLA}	Z_{la}
	Dimensions	Degree	Degree	mm	mm	mm	mm	mm	mm	mm	μm	--
A (Original impeller)		30	27.5	34	17	209	103	3.5	3.5	3.5	100	6
B		30	30	42	21	209	103	3.5	3.5	3.5	100	6
C		30	32.5	34	17	209	103	3.5	3.5	3.5	100	6

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

Test number	Impeller Type	Geometry		Physical		Head			Power consumption			Total efficiency		
		β_{2B}	b_2	ρ	ν	Experimental	Analytical	Error (%)	Experimental	Analytical	Error (%)	Experimental	Analytical	Error (%)
1	A	27.5	17	1000	1	12.7	13.11	3.22	2.8	2.51	10.35	60.8	58.9	3.12
2	A	27.5	17	875	43	12.45	12.73	2.24	3.48	3.07	11.78	40.5	45	11.11
3	B	30	21	1000	1	12.8	13.35	4.29	3.2	2.5	21.87	60.5	59.48	1.68
4	B	30	21	875	43	12.11	12.94	6.85	3.94	3.14	20.3	48.5	48.34	0.32
5	C	32.5	17	1000	1	13.3	12.77	3.98	2.6	2.38	8.4	59.8	57.89	3.19
6	C	32.5	17	875	43	12.15	12.38	1.89	3.257	3.06	6.04	42.25	43	1.77

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_{in} -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.

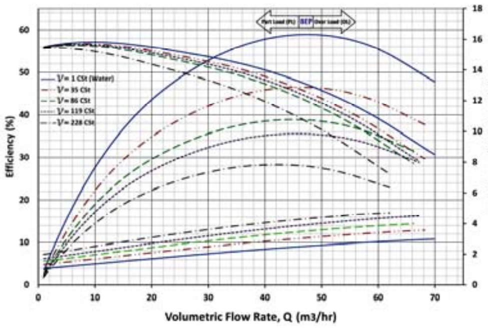


Fig. 4. The performance curve of centrifugal oil pump with different viscosities for impeller "A".

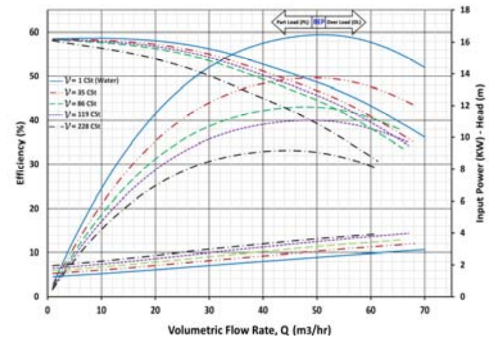


Fig. 5. The performance curve of centrifugal oil pump with different viscosities for impeller "B".

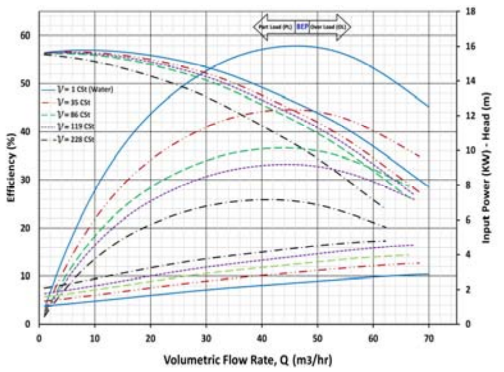


Fig. 6. The performance curve of centrifugal oil pump with different viscosities for impeller.

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 $\nu \geq 115$ 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 $f_{\eta h}$ 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 ν=86 CSt and ν=119 CSt.

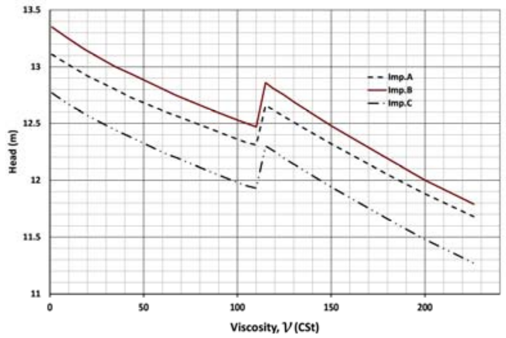


Fig. 7. H-ν curve of centrifugal pump for three different geometries at BEP.

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_u shows a significant growth.

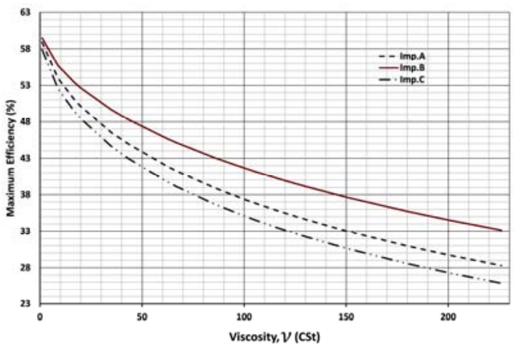


Fig. 8. $\eta - \nu$ curve of centrifugal pump for three different geometries at BEP.

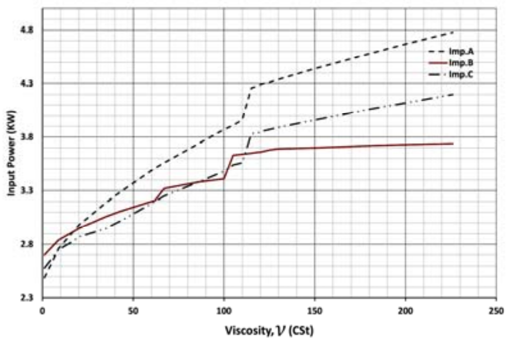


Fig. 9. $P - \nu$ curve of centrifugal pump for three different geometries at BEB.

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^{o} is improved in comparison with 27.5^{o} and 32.5^{o}. 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 ν in geometry B is smaller than other geometries due to the reduction of the hydraulic loss in this geometry.

REFERENCES

Gulich, JF. (2003). Disk friction losses of closed turbomachine impellers. *Forsch Ingenieurwes,* 68,87-95.

Gulich, J. F. (1999). Pumping Highly Viscous Fluids with Centrifugal Pumps-Part 1. *World pumps*.

Gulich, J. F. (1999). Pumping Highly Viscous Fluids with Centrifugal Pumps-Part 2. *World pumps*.

Gulich, J. F. (2003). Effect of Reynolds number and surface roughness on the efficiency of centrifugal pumps. *ASME Journal of Fluid Engineering* 125(4),670-679.

Gulich, J. F. (2008). Centrifugal pumps. *Springer-Verlag, Berlin Heidelberg*, ISBN 978(3),540-73694-3

Ladouani A. and A. Nemdili (2009). Influence of Reynolds number on net positive suction head of centrifugal pumps in relation to disk friction losses", *Forsch Ingenieurwes* 73,173-182.

Li, W. G. (1999). Effects of viscosity of fluids on centrifugal pump performance and flow pattern in the impeller. *International Journal of Heat and Fluid Flow* 21,207-212.

Li, W. G. (2000). The Sudden-Rising Head Effect in Centrifugal Oil Pumps. *World Pumps*.

Li, W. G. (2002). Experimental investigations into the performance of a commercial centrifugal oil pump. *World Pumps* 26-8.

Li, W. G. (2004). A Method for Analyzing the Performance of Centrifugal Oil Pumps. *ASME Journal of Fluid Engineering* 126,482-485.

Li, W. G. (2008). Performance computations of centrifugal pump handling viscous oils. *Dra Irr Mach*.

Li, W. G. (2010). Blade Exit Angle Effects on Performance of a Standard Industrial Centrifugal Oil Pump. *Journa l of Applied Fluid Mechanics* 4:105-119.

Li, W. G., Z. M. Hu (1997). An Experimental Study on Performance of Centrifugal Oil Pump. *Fluids Mach* 25(2), 3-7.

Nemdili, A. and D. H. Hellmann (2007). Investigations on fluid friction of rotational disks with and without modified outlet sections in real centrifugal pump casings. *Forsch Ingenieurwes* 71,59-67.

Shojaeefard, M. H., M. B. Ehghaghi and F. A. Boyaghchi (2006). Experimental study and three dimensional numerical flow simulation in a centrifugal pump when handling viscous fluids. *IUST Int J Eng Sc* 17(3-4), 53-60.

Shojaeefard, M. H. and F. A. Boyaghchi (2007). Studies on the influence of various blade outlet angles in a centrifugal pump when handling viscous fluids. *American Journal of Applied Science* 4(9),718-724

Shojaeefard, M. H, M. Tahani, M. B. Ehghaghi, M. A. Fallahian and M. Beglari (2012). Numerical study of the effects of some geometric characteristics of a centrifugal pump impeller that pumps a viscous fluid. *Computers&Fluids* 60,61-70.