ABSTRACT
The optimal selection of pumping systems for viscous fluids is very important in order to reduce the running costs. In this paper, a selecting method based on laboratory investigations on the performance of centrifugal pumps when handling viscous fluids and on diagrams of performance correction factors; is suggested.

The ratio of specific speeds of water and the considered viscous mediums is nearly equal to the unity, and the obtained diagrams of performance correction factors versus the Reynolds number allow a clear and a rapid selection of pumps.

From the specific speed required by the system and based on these results, the optimal pump is selected, and the optimal diameter of the network pipes is deduced. If the rate flow is very different from the required by the system, a new pump is adapted by the calculation of its new rotational speed. If the diameter of the system pipe is imposed, a nearly pump must be selected and adapted by varying its rotational speed. The final optimal selection will takes into account the amortization of the costs of the components of the system.

KEYWORDS
Selection, Centrifugal pump, Pumping system, Viscous fluid

NOMENCLATURE

\begin{align*}
  b & \quad [\text{mm}] \quad \text{impeller width} \\
  D & \quad [\text{mm}] \quad \text{impeller diameter} \\
  f & \quad [-] \quad \text{performance correction factor} \\
  H & \quad [\text{m}] \quad \text{head} \\
  ks & \quad [-] \quad \text{singular head loss coefficient} \\
  L & \quad [\text{mm}] \quad \text{pipe length}
\end{align*}

\begin{align*}
  n & \quad [\text{RPM}] \quad \text{rotational speed} \\
  n_q & \quad [\text{min}^{-1}] \quad \text{specific speed} \\
  Q & \quad [\text{m}^3/\text{s}] \quad \text{flow} \\
  \Re & \quad [-] \quad \text{Reynolds number} \\
  u & \quad [\text{m/s}] \quad \text{peripheral velocity} \\
  \nu & \quad [\text{m}^2/\text{s}] \quad \text{cinematic viscosity} \\
  \beta & \quad [\text{°}] \quad \text{blade angle}
\end{align*}

Subscripts and Superscripts

\begin{align*}
  \text{opt} & \quad \text{optimal} \\
  \text{st} & \quad \text{static} \\
  \text{v} & \quad \text{viscous} \\
  w & \quad \text{water} \\
  2 & \quad \text{outlet}
\end{align*}

ABBREVIATIONS

\begin{align*}
  \text{cSt} & \quad \text{centiStockes} \\
  \text{PC} & \quad \text{Parabolic Curve} \\
  \text{SL} & \quad \text{Straight Line}
\end{align*}

1. INTRODUCTION

Optimal selection of pumping systems for viscous fluids is very important in order to reduce the running costs.

Users of centrifugal pumps working in network pipes for viscous fluids are confronted with two problems during the calculation for system selection:

- The first concerns the determination of performance correction factors of pumps available in the market.
- The second concerns the laws to use to bring into focus the operating points of systems: Are they the same of those used for water transport? Must corrections be made? How to choice the optimal pumping system? How to adapt the operating points?

In this study and to find responses to these questions, a selecting method based on laboratory investigations...
on the performance of centrifugal pumps when handling viscous fluids and on diagrams of performance correction factors; is suggested.

2. RATIO OF SPECIFIC SPEEDS

On the basis of laboratory investigations on the performance of a centrifugal pump with a specific speed \( n_q = 20 \text{min}^{-1} \) when handling two kinds of oil with a viscosity of 75cSt and 646cSt, the calculation of the ratio of specific speeds of the viscous oil to the water provides values between 0.9 and 1.1; namely near to the unity.

The result is shown in the Figure 1, when the Reynolds number increases, this ratio tends to the unity. The value of the Reynolds number \( Re = 10^4 \) can be considered as reference.

![Figure 1: Ratio of specific speeds viscous fluid / water versus Reynolds number](image)

Figure 1: Ratio of specific speeds viscous fluid / water versus Reynolds number

Analytical calculation confirms the tendency of this ratio to the unity, hence:

\[
\begin{align*}
    n_{qv} &= n \cdot \frac{Q_v^{0.5}}{H_v^{0.75}} \\
    n_{qw} &= n \cdot \frac{Q_w^{0.5}}{H_w^{0.75}} \\
    Q_w &= \frac{Q_w}{f_Q} \\
    H_w &= \frac{H_w}{f_H}
\end{align*}
\]

In practice, this preliminary result can be used during the preliminary selection of the category of the centrifugal pump, which can satisfy the requirement of the discharge system represented by \( H_v \) and \( Q_v \) when considering \( n_{qv} = n_{qw} \).

Therefore:

\[
\frac{n_{qv}}{n_{qw}} = \left( \frac{1}{f_H} \right)^{0.75} \quad \text{with} \quad f_Q < f_H < 1
\]

3. PERFORMANCE CORRECTION FACTORS

Experimental results and design coefficients of the impellers allow to represent the performance correction factors of the tested pump that has a design angle \( \beta_2 = 23^\circ \). The ratio of outlet diameter to outlet width of the impeller \( D_2/b_2 \) is varied as shown in Table 1.

<table>
<thead>
<tr>
<th>( n_q ) (min(^{-1}))</th>
<th>20</th>
<th>25</th>
<th>35</th>
<th>45</th>
</tr>
</thead>
<tbody>
<tr>
<td>( D_2/b_2 )</td>
<td>21</td>
<td>17.5</td>
<td>11</td>
<td>7.5</td>
</tr>
</tbody>
</table>

Using diagrams of performance correction factors established by KSB [2], the following third curves are presented in the Figure 2 below.

It’s another manner of representation different from KSB-diagrams, since it uses the Reynolds number.

Figure 2 shows that when the specific speed increases, the performance correction factors decrease. The reductions are more important for the efficiencies, for flows and at the end for heads. The variation of head and flow correction factors versus Reynolds number is not important, which means that head and flow laws can be applied for a same viscosity when the difference of rotation speeds and diameters is not great.

This observation is confirmed especially when a value of Reynolds number is greater than \( 10^6 \). On the other hand, the efficiency correction factors are very pronounced, and are nearly linear at a value of the Reynolds number of \( 10^6 \). The power law can not be applied in any case.

The representation of the performance correction factors versus Reynolds number facilitates, for the users, the preliminary selection of pumps proposed by the manufacturers.

The user gives the head and the required flow by the viscosity and the system, takes the rotational speed and the diameter of the proposed suggested pumps from manufacturers and makes his own choice.

For this, it is recommended to centrifugal pump manufacturers to represent the performance correction factors versus Reynolds number.
4. OPTIMAL SELECTION OF PUMPING SYSTEM

In a classical discharge system (suction pipe, pump and discharge pipe), the data are the static head $H_{st}$ and the head losses. The set is considered to be the resistance $R$ of the system. In this resistance curve, the required flow $Q_{rv}$ of the considered medium is fixed, the required head $H_{rv}$ of the medium is deduced, this corresponds to the point 1 on the Figure 3.

With regard to the supply, they are the hydraulic parameters of the pumps: $n, D, n_q, H(Q)$ and $\eta(Q)$.

The intersection point between supply and demand represents the operating point, hoping that it is the optimal point.

To find again this great demanded point in the case of viscous fluids, the following method is suggested:

- Compute the specific speed $n_q$ required by the system for a fixed rotational speed $n$ of the proposed pump.
- With the assumption that $n_{qv} = n_{qw}$, pumps with the class $n_{qw}$ are selected. Between these pumps, many impeller diameters are proposed. The Reynolds number is determined for each diameter using:

$$Re = \frac{u_2 \cdot D_2}{\nu} \quad (3)$$

- Determine the values of correction factors for each diameter using the diagram:

$$f_H, f_Q, f_\eta = f(Re)$$
as shown in Figure 4.

- For each case evaluate the terms $Q_{rv}$ and $H_{rv}$ as:

$$Q_{rv} = \frac{Q_{rv}}{f_{rv}} \quad (4)$$
\[ H_{rw} = \frac{H_{rv}}{f_H} \]  

(5)

That corresponds to point 2 in Figure 3.

- Select the best pump, that corresponds to a small difference between optimal values of head and flow \( H_{opt} \) and \( Q_{opt} \) and \( H_{rw} \) and \( Q_{rw} \) of the pump; that means minimal values \((H_{opt} - H_{rw})\) and \((Q_{opt} - Q_{rw})\).
- Evaluate the efficiency of the system as:

\[ \eta_s = \eta_{opt} \cdot f_\eta \]  

(6)

Figure 3 represents this ideal case where the point 2 is on the curve Head-Flow \( H_w(D) \) for water and the point 1 on the corrected curve for the viscosity \( H_v(D) \).

If the difference \((Q_{opt} - Q_{rw})\) is very important as shown in Figure 5, the point 2 is not on the curve \( H_w(D) \), or if the system operating flow \( Q_{f3} \), which is determined by the intersection point 3 between the system resistance \( R \) and the corrected curve \( H_v(D) \); is so farther of the required flow \( Q_{rv} \) (more as 10 %), it is better to suggest another diameter of the network pipe.

The optimal choice of pumping systems takes into account the operating costs created mainly by the pump and the amortization costs caused by the long conducts in the case of pipe lines.

4.1. Adaptation by diameter reduction

In previous studies [6], it was found that for a certain range of rotational speed and diameter, the affinity laws of heads and flows are respected for the same viscosity, but not between the different viscosities.

With regard to powers, the affinity laws are not respected. In all the cases the efficiencies are not preserved.

The adaptation by diameter reduction is avoided, because it reduces highly the Reynolds number (the diameter is raised to the power of 2 during the calculation of Reynolds number), this reduces highly the pump performances for a considered viscosity.

Nevertheless when the operating flow is very important in regard to the required flow, the reduction of the diameter is necessary, with a limit of 5% between initial diameter and reduced diameter, to minimize performance reductions.

Figure 6 represents the adaptation by diameter reduction.

This method consists on:

- Selecting a pump having \( H_w(D) \), which is nearest to the point 2.
- Plotting corrected characteristic of the pump \( H_v(D) \) using correction factors.
- Drawing a straight line \( SL \) between origin and required point 1:

\[ SL = \left( \frac{H_{rv}}{Q_{rv}} \right) \cdot Q \]  

(9)

Figure 5: Verification of operating point

The optimal diameter of the pipe is evaluated as follows:

\[ D_{Opt} = \frac{(4.074 \cdot k_s \cdot L \cdot Q_{Opt} \cdot f_\eta)^{0.25}}{(H_{Opt} \cdot f_H \cdot H_{St})^{0.25}} \]  

when \( Re < 2000 \)

And

\[ D_{Opt} = \frac{(0.024 \cdot L \cdot V^{0.25} \cdot (Q_{Opt} \cdot f_\eta)^{0.75})^{0.21}}{(H_{Opt} \cdot f_H \cdot H_{St})^{0.21}} \]  

when \( 2000 < Re < 100000 \)

Note that it is better to use greater diameters but in the limit of their amortization costs.
Finding the capacity $Q_3$, which is the intersection of the straight line $SL$ with the curve $H_v(D)$.

- Evaluating a new diameter $D_1$ using the flows law, because it is applied for a certain range of diameters between points 1 and 3.
- Verifying the percent of diameter reduction, which must be little than 5 %.
- Evaluate the efficiency of $Q_w$ (point 4) on the curve $H_w(D)$ using corrections as:
  \[ Q_w = \frac{Q_3}{f_{q_3}} \quad \text{and} \quad \eta_w = \frac{\eta_3}{f_{\eta}} \]  \hspace{1cm} (10)

- Evaluating the efficiency of required flow $Q_{rw}$ (point 1) on the curve $H_v(D_1)$ using a new correction as:
  \[ \eta_{rw} = \frac{\eta_w}{f_{\eta}} \]  \hspace{1cm} (11)

4.2. Adaptation by variation of rotational speed

Same procedure applied to diameter reduction is used to the variation of rotational speed, see Figure 7, but with two major differences:
- The variation of the rotational speed can be till more or less than 20 %.
- The point 3 is obtained using the intersection between the parabolic curve $PC$ passing through the origin and the required point 1:
  \[ PC \equiv \frac{H_{pc}}{Q_{pc}}, Q \]  \hspace{1cm} (12)

It is better, when it is possible to change the rotational speed, to select the pump having a performance different from the required one, and after that to adapt this pump by increasing the rotational speed.

Therefore it is to indicate that these methods can be applied in a range between $0.8xQ_{opt}$ and $1.2xQ_{opt}$ (see Figure 8), because the efficiencies of centrifugal pumps do not differ greatly from optimal efficiencies in this range.

5. CONCLUSION

This study has indicated that the optimal selection of pumping systems for viscous fluids is simplified when:
- The ratio of specific speeds viscous fluid / water is near to the unity.
- The affinity laws of heads and flows are applied for a same viscosity.
- The correction factors are represented versus the Reynolds number.

In these cases the selection and the adaptation of the pumping systems are the same as those used for water, by using necessary corrections as mentioned in this paper.

![Figure 7: Variation of rotational speed](image)

![Figure 8: Adaptation Area](image)
REFERENCES


