The Analysis of the Influence of the Initial Impeller on the Discharge and the Delivery Head of High Speed Pump with Radial Blades

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Abstract: The article discusses a theoretical analysis of fluid flow through the initial impeller which is mounted at the inlet to the centrifugal impeller in the impeller pump. The article presents the results of laboratory tests for the influence of various initial impellers in the impeller pump with an open-flow impeller and rectilinear radial blades on the pump operating parameters and the course of performance characteristics. The research has been performed at the rotational speed from 3000 to 7000 rpm.

Key words: high-speed pump, open-flow impeller, radial blades, initial impeller, performance characteristics

1 Introduction
Observing the development of impeller pumps design, we can notice the tendency for increasing rotational speed [5]. The increase of nominal rotational speed for a particular pump allows reduction of the overall dimensions and its weight. Moreover, the increase of rotational speed causes the increase of specific speed and allows to widen the range for the use of such an impeller pump, i.e. it facilitates reaching of delivery head at a small discharge [4, 6, 7, 10, 16]. Thus, today we can design an impeller pump with parameters, which have been feasible so far with the use of positive pumps [11, 12].

Using open flow impellers with radial blades may have positive influence on forces acting on centrifugal pump impeller [3, 15].

Hence, it can be observed that in the case of single-stage pumps using impellers with blades of single curvature or rectilinear radial blades which were operating at increased rotational speed, some pumps use initial impellers while others do not have such impellers.

During the research performed on the impeller pumps operating at higher rotational speed, we decided to conduct a research on the influence of the initial impeller on the performance parameters of a single stage pump with radial blades. The results of the research are presented in this article.

2 Model Pump
Laboratory tests were conducted on a specially designed model pump. The cross-section of a fragment of the pump model pump is shown on Figure 1.

It is a single stage impeller pump with centrifugal impeller and rectilinear radial blades. The inlet to the centrifugal impeller is equipped with initial impeller shaped as one-turn endless screw with a constant hub diameter and a constant outer diameter. The tests have been performed using three different initial impellers with different screw line pitch. For comparison, the measurement of performance parameters of the pump without initial impeller was conducted, with a nut mounted instead of the initial impeller. In order to maintain similar conditions at the inlet to the centrifugal impeller,
the nut used has the same shape as the shape of the initial impeller hub.

Different initial impellers and the impeller nut are shown on Figure 2, the major dimensions of initial impellers are shown on Figure 3, while Figure 4 presents the major dimensions of the nut.

Radial gap between the outer diameter of the initial impeller and the casing is 0.001 mm.

Due to the decrease of the axis thrust, the centrifugal impeller was an open-flow impeller with remains of back disk, which enabled joining the blades with the impeller hub [1, 2, 13, 14, 17]. The pump shaft was mounted within angle ball bearing. The pump has a spiral casing. The pump has mechanical sealing.

The test utilised an centrifugal impeller with the following dimensions:
- the diameter of the blade at the inlet to the impeller \( d_1 = 0.06 \) m
- the outer diameter of the impeller \( d_2 = 0.18 \) m,
- the width of the blades at the inlet \( b_1 = 0.018 \) m,
- the width of the blades at the outlet \( b_2 = 0.009 \) m,
- the number of the blades \( z = 12 \)

The centrifugal impeller is shown on Figure 5.

### 3 Theoretical discharge of the initial impeller

If we do not take into account the impact of the liquid friction against the walls of the blade profile of initial impeller and assume a very little gap between the outer diameter of the initial impeller and the housing, the theoretical discharge of the initial impeller can be calculated from the following formula

\[
Q_{th} = \frac{\Pi}{4 \cdot 60} \cdot n \cdot (d_2^2 - d_1^2) \cdot (t_s - s_s)
\]  

(1)

With the adopted method of initial impeller realisation, when the wall thickness has been measured perpendicularly to it (fig. 6), the wall thickness measured parallel to the axis equals:

\[
s_s = \frac{s}{\cos \alpha}
\]  

(2)

\[
\alpha = \arctan \frac{t_s}{2 \cdot \pi \cdot r}
\]  

(3)
As the above facts prove, the wall thickness measured parallel to the axis changes with the radius

\[ s_s = f(r) \]  

(4)

while the radius changes within the limits of

\[ \frac{d_{i1}}{2} \leq r \leq \frac{d_{i2}}{2} \]  

(5)

Thus, the basic flow rate can be defined with the following formula

\[ dQ_{th} = 2\pi \cdot r \cdot \left(t_s - \frac{s}{\cos\alpha}\right) \cdot \frac{n}{60} \cdot dr \]  

(6)

Considering that

\[ \cos\alpha = \frac{s}{\sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2}} \]  

(7)

after substitution of (7) for (6) we get

\[ dQ_{th} = \frac{\pi}{30} \cdot n \cdot \left(t_s - s \cdot \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2}\right) \cdot r \cdot dr \]  

(8)

The total volumetric flow rate equals

\[ Q_{th} = \frac{\pi}{30} \cdot n \int_{r_{i1}}^{r_{i2}} \left(t_s - s \cdot \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2}\right) \cdot r \cdot dr \]  

(9)

where

\[ r_{i1} = \frac{d_{i1}}{2} \quad r_{i2} = \frac{d_{i2}}{2} \]  

(10)

After integration we finally get \( Q_{th} \) [m³/s]

\[ Q_{th} = \omega \left\{ \frac{t_s}{2} \left(r_{i2}^2 - r_{i1}^2\right) - \frac{s}{2} \left[f_1 + f_2 + f_3 + f_4\right] \right\} \]  

(11)

where

\[ \omega = \frac{\Pi \cdot n}{30} \]  

(12)

\[ A_i = \frac{t_s}{2\pi} \]  

(13a)

\[ f_1 = r_{i2} \sqrt{r_{i2}^2 + A_i^2} \]  

\[ f_2 = -r_{i1} \sqrt{r_{i1}^2 + A_i^2} \]  

(13b)

\[ f_3 = A_i^2 \ln \left(r_{i2} + \sqrt{r_{i2}^2 + A_i^2}\right) \]  

\[ f_4 = -A_i^2 \ln \left(r_{i1} + \sqrt{r_{i1}^2 + A_i^2}\right) \]

The total volumetric flow rate equals

\[ Q_{th} = \int_{r_{i1}}^{r_{i2}} \left(t_s - s \cdot \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2}\right) \cdot r \cdot dr \]  

(11)

Calculated theoretical discharge \( Q_{th} \) [dm³/s] of initial impellers under examination depending on the rotational speed, has been presented in the table 1.

**Table 1**

<table>
<thead>
<tr>
<th>Impeller</th>
<th>WS1 [dm³/s]</th>
<th>WS2 [dm³/s]</th>
<th>WS3 [dm³/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>n [rpm]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1000</td>
<td>0.1346</td>
<td>0.3223</td>
<td>0.5093</td>
</tr>
<tr>
<td>2000</td>
<td>0.2691</td>
<td>0.6446</td>
<td>1.0185</td>
</tr>
<tr>
<td>3000</td>
<td>0.4037</td>
<td>0.9669</td>
<td>1.5278</td>
</tr>
<tr>
<td>4000</td>
<td>0.5382</td>
<td>1.2892</td>
<td>2.0370</td>
</tr>
<tr>
<td>5000</td>
<td>0.6728</td>
<td>1.6115</td>
<td>2.5463</td>
</tr>
<tr>
<td>6000</td>
<td>0.8074</td>
<td>1.9337</td>
<td>3.0556</td>
</tr>
<tr>
<td>7000</td>
<td>0.9419</td>
<td>2.2560</td>
<td>3.5648</td>
</tr>
</tbody>
</table>

The model pump is shown on Figure 7, and the pump without the casing at the inlet side is shown on Figure 8.
4 The course of the research

All tests performed to define the influence of the initial impellers on the pump operating parameters used the same centrifugal impeller, with constant gaps before and behind the impeller. The initial impellers were changed and in order to maintain the same conditions at the inlet, the sleeves were changed accordingly (the length of the suction stub pipe).

The parameters’ measurements were performed in accordance with the recommendation of the standard PN-EN ISO 5198:2002 (Centrifugal, mixed flow and axial pumps - Code for hydraulic performance tests - Precision class) [18].

Operation parameters readings were performed to define the shape of the curves $H = f(Q)$. The readings were taken at the rotating speed $n = 3000, 5000, 6000$ and $7000$ rpm.

In addition, the pressure, among another things, was measured behind the initial impeller.

The readings were taken at different discharges i.e. from $Q = 0$ (the valve at the delivery side closed) until cavitation occurred.

The delivery head of the pump was defined by a formula:

$$H_p = \frac{P_2 - P_1}{\rho \cdot g} + \frac{w_2^2 - w_1^2}{2 \cdot g} + z_m$$  \hspace{1cm} (14)

While the measurements were being taken the rotating speed would change slightly and this was the reason why operating parameters were corrected to the nominal rotational speed by formulas:

$$Q = Q_p \cdot \frac{n_{nom}}{n}$$  \hspace{1cm} (15)

$$H = H_p \cdot \left(\frac{n_{nom}}{n}\right)^2$$  \hspace{1cm} (16)

Figure 9 shows the views of test bench.

As the analysis of tests results proves, the discharge, during which cavitation occurs, is much greater than the theoretical discharge of the initial impeller. The difference of pressures after and before the initial impeller forces additional flow between the blade profile of initial impeller and in the gap between the housing and the outer diameter of the initial impeller.

5 Volumetric flow rate between blade profile of initial impeller forced between the blade profile of initial impeller

The cross-sectional area of flow between the profile can be defined with the following formula:

$$A_s = (r_{r_2} - r_{s_2}) \cdot (t - s)$$  \hspace{1cm} (17)

where

$$t = \frac{t_i}{\cos \alpha}$$  \hspace{1cm} (18)
Considering that

\[
\alpha = \arctan \frac{t_s}{2 \cdot \pi \cdot r}
\]  

(19)

\[
t = t_s \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2}
\]  

(20)

Hence

\[
A_{sl} = (r_{s2} - r_{s1}) \cdot \left[ t_s \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot r}\right)^2} - s \right]
\]  

(21)

Because \( \alpha = f(r) \), surface area calculated for a mean radius

\[
r_s = \frac{r_{s2} + r_{s1}}{2}
\]  

(22)

Shall amount to

\[
A_{sl} = (r_{s2} - r_{s1}) \cdot \left[ t_s \sqrt{1 + \left(\frac{t_s}{2 \cdot \pi \cdot (r_{s2} + r_{s1})}\right)^2} - s \right]
\]  

(23)

Mean velocity of flow

\[
c = \frac{Q_{sl}}{A_{sl}}
\]  

(24)

Radius component of velocity forced by difference of pressures after and before the initial impeller

\[
c_r = c \cdot \cos \alpha
\]  

(25)

Whereas axis component of velocity forced by difference of pressures

\[
c_o = c \cdot \sin \alpha
\]  

(26)

### 6 Volumetric flow rate through the gap

Simultaneously, the difference of pressures after and before the initial impeller forces additional axial flow in the gap between the housing and outer diameter of the initial impeller.

Axial velocity which results from rotational motion of initial impeller

\[
c_{ow} = t_s \cdot \frac{n}{60}
\]  

(27)

Axial component of mean velocity at outer diameter of the initial impeller

\[
c_{oz} = c \cdot \sin \alpha_z
\]  

(28)

where

\[
\alpha_z = \arctan \frac{t_s}{2 \cdot \Pi \cdot r_{s2}}
\]  

(29)

Total axial velocity at outer diameter of the initial impeller

\[
c_{oz} = c_{oz} + c_{ow}
\]  

(30)
If we assume that for \( r_{ob} \) the component of axial velocity
\[
e_{ax} = 0 \quad (31)
\]
while for \( r_s \) the component of axial velocity equals axial velocity in the initial impeller at its outer diameter
\[
e_{ax} = c_{ac} \quad (32)
\]
volumetric flow rate in the gap can be calculated

Basic flow rate through the ring-shaped surface with the central point in the axis of rotation equals
\[
dQ = 2 \cdot \pi \cdot r \cdot e_{ax} \cdot dr \quad (33)
\]

Assuming boundary conditions for axial velocity distribution defined in (31) and (32), and assuming that in the gap some flow occurs, which axial component has linear velocity profile, axial velocity in the function of radius is expressed by the following relationship:
\[
e_{ax} = f(r) = A_2 \cdot r + B_2 \quad (34)
\]
where
\[
A_2 = -\frac{e_{ax}}{r_{ob} - r_{s2}} \quad (35)
\]
\[
B_2 = e_{ax} \left(1 + \frac{r_{s2}}{r_{ob} - r_{s2}}\right) \quad (36)
\]

Flow rate through the gap is defined by the following formula:
\[
Q_{sc} = 2 \cdot \pi \int_{r_{s2}}^{r_{ob}} \left(A_2 \cdot r + B_2\right) \cdot r \cdot dr \quad (37)
\]

Eventually the flow rate through the gap amounts to
\[
Q_{sc} = \frac{2}{3} \cdot \pi \cdot A_2 \cdot \left(r_{ob}^3 - r_{s2}^3\right) + \pi \cdot B_2 \cdot \left(r_{ob}^2 - r_{s2}^2\right) \quad (38)
\]
The total flow rate consists of:
- theoretical discharge of initial impeller,
- flow rate between blade profile of initial impeller, which results from the difference of pressures after and before the initial impeller,
- flow rate in the gap between the housing and the outer diameter of the initial impeller.
\[
Q = Q_{th} + Q_{sl} + Q_{sc} \quad (39)
\]

For selected flow rates, velocity components and flow rate components were calculated. The results of the calculations are shown in the table 2.

7 The influence of flow between blade profile of initial impeller on the whirl at the inlet to the centrifugal impeller

The additional flow between the blade profile of initial impeller forced by the difference of pressures causes the radius component of velocity, which results from this flow, to trigger off initial whirling of liquid before it reaches the blades of the centrifugal impeller.

Calculations prove that in case of initial impeller WS2 at additional volumetric flow rate between blade profile \( Q_{sl} = 0.8918 \text{ dm}^3/\text{s} \), at the rotational speed \( n = 6000 \text{ obr/min} \), the velocity between the blade profile of impeller equals \( c = 3.833 \text{ m/s} \). For outer diameter of initial impeller \( \alpha_z = 9.49^\circ \), radial component equals \( c_{rz} = 3.781 \text{ m/s} \). Peripheral velocity for the outer diameter of initial impeller equals \( u_{w2} = 12.566 \text{ m/s} \). Absolute radial velocity (at the outer diameter of initial impeller) calculated with the formula
\[
c_{rw2} = u_{w2} - c_{rz} \quad (40)
\]
equals \( c_{rw2} = 8.785 \text{ m/s} \).

This is initial whirl related to the outer diameter of the initial impeller. Peripheral velocity (transportation velocity) at the inlet to the centrifugal impeller equals \( u_1 = 18.850 \text{ m/s} \).

Thus the velocity of initial whirl \( c_{rw2} = 8.785 \text{ m/s} \) is lower than the peripheral velocity of blades at the inlet into the centrifugal impeller \( u_1 = 18.85 \text{ m/s} \) and has not a decisive effect on the impactless inflow of liquid into the centrifugal impeller.
8 The results of the laboratory research

The tests results are presented in the form of graphs. Figure 10 shows the performance characteristics of pump $H = f(Q)$ at various rotational speeds, with the utilisation of different initial impellers (three version of initial impellers: WS1, WS2 and WS3) and with the pump operating without initial impeller (instead of an initial impeller a nut was used to tighten the centrifugal impeller, marked WS0).

![Fig. 10. Performance characteristic of model pump](image1)

![Fig. 11. Performance characteristic of the initial impellers](image2)

Figure 11 shows the dependency of an initial impeller delivery head on the discharge $H_w = f(Q)$ for different initial impellers with the pump operating at $n = 7000$ rpm. The initial impeller delivery head was defined as the difference of pressure head behind and before the initial impeller.

Table 2. Velocity components and flow rate components for selected discharges $Q$
at rotational speed $n = 6000$ rpm

<table>
<thead>
<tr>
<th>Initial impeller WS1</th>
<th>$Q$ (dm$^3$/s)</th>
<th>1.0000</th>
<th>1.5000</th>
<th>2.0000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{th}$ (dm$^3$/s)</td>
<td>0.8074</td>
<td>0.8074</td>
<td>0.8074</td>
<td></td>
</tr>
<tr>
<td>$Q_{sl}$ (dm$^3$/s)</td>
<td>0.1189</td>
<td>0.5923</td>
<td>1.0657</td>
<td></td>
</tr>
<tr>
<td>$Q_{sz}$ (dm$^3$/s)</td>
<td>0.0737</td>
<td>0.1003</td>
<td>0.1269</td>
<td></td>
</tr>
<tr>
<td>$c$ (m/s)</td>
<td>1.255</td>
<td>6.253</td>
<td>11.251</td>
<td></td>
</tr>
<tr>
<td>$c_{oz}$ (m/s)</td>
<td>0.104</td>
<td>0.521</td>
<td>0.937</td>
<td></td>
</tr>
<tr>
<td>$c_{ow}$ (m/s)</td>
<td>1.050</td>
<td>1.105</td>
<td>1.050</td>
<td></td>
</tr>
<tr>
<td>$c_{oc}$ (m/s)</td>
<td>1.154</td>
<td>1.571</td>
<td>1.987</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Initial impeller WS2</th>
<th>$Q$ (dm$^3$/s)</th>
<th>2.5000</th>
<th>3.0000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{th}$ (dm$^3$/s)</td>
<td>1.9337</td>
<td>1.9337</td>
<td></td>
</tr>
<tr>
<td>$Q_{sl}$ (dm$^3$/s)</td>
<td>0.4134</td>
<td>0.8918</td>
<td></td>
</tr>
<tr>
<td>$Q_{sz}$ (dm$^3$/s)</td>
<td>0.1529</td>
<td>0.1745</td>
<td></td>
</tr>
<tr>
<td>$c$ (m/s)</td>
<td>1.777</td>
<td>3.833</td>
<td></td>
</tr>
<tr>
<td>$c_{oz}$ (m/s)</td>
<td>0.293</td>
<td>0.632</td>
<td></td>
</tr>
<tr>
<td>$c_{ow}$ (m/s)</td>
<td>2.100</td>
<td>2.100</td>
<td></td>
</tr>
<tr>
<td>$c_{oc}$ (m/s)</td>
<td>2.393</td>
<td>2.732</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Initial impeller WS3</th>
<th>$Q$ (dm$^3$/s)</th>
<th>3.5000</th>
<th>4.0000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{th}$ (dm$^3$/s)</td>
<td>3.0556</td>
<td>3.0556</td>
<td></td>
</tr>
<tr>
<td>$Q_{sl}$ (dm$^3$/s)</td>
<td>0.2337</td>
<td>0.7141</td>
<td></td>
</tr>
<tr>
<td>$Q_{sz}$ (dm$^3$/s)</td>
<td>0.2107</td>
<td>0.2303</td>
<td></td>
</tr>
<tr>
<td>$c$ (m/s)</td>
<td>0.612</td>
<td>1.871</td>
<td></td>
</tr>
<tr>
<td>$c_{oz}$ (m/s)</td>
<td>0.149</td>
<td>0.455</td>
<td></td>
</tr>
<tr>
<td>$c_{ow}$ (m/s)</td>
<td>3.150</td>
<td>3.150</td>
<td></td>
</tr>
<tr>
<td>$c_{oc}$ (m/s)</td>
<td>3.299</td>
<td>3.605</td>
<td></td>
</tr>
</tbody>
</table>
As the graphs on Figure 11 show, when a pump is operating at a higher discharge than the nominal discharge of the initial impeller, vacuum appears behind the initial impeller and cavitation occurs in the pump. Figure 12 shows an example of the relationship of pressure head behind the initial impeller WS2 (before the centrifugal impeller) depending on the discharge for various rotational speeds.

![Graph showing pressure head dependence on discharge]

**Fig. 12.** The dependence of pressure head on the discharge behind the initial impeller WS2

### 9 Conclusions

On the basis of the results analysis for the tests performed the following conclusions may be drawn:

1) The performance characteristics curve of a pump with rectilinear radial blades is flat and practically speaking the delivery head does not change when the discharge level is changed (at constant rotating speed).

2) The delivery head reached by the initial impeller is very small in comparison to the delivery head of a centrifugal impeller. While the pump is in operation, up to a certain discharge, the initial impeller has no significant impact on the pump delivery head, and when a certain discharge level is exceeded it hinders the inflow of liquid to the centrifugal impeller, which causes cavitation.

3) The highest discharge up to the moment when cavitation occurred was gained by the pump without initial impeller. The centrifugal impeller itself produces a very high vacuum on suction.

4) It seems that the use of initial impellers in single-stage pump with impeller with rectilinear radial impellers is unjustified. The only reason, which supports the use of initial impellers, may be a need to limit the pump discharge (by means of causing cavitation) and to avoid overload of the motor.

### Acknowledgements

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### Notation schedule

- $A_{sl}$: cross-sectional area between the blade profile of the initial impeller, m$^2$
- $b_1$: the width of the blades at the inlet, m
- $b_2$: the width of the blades at the outlet, m
- $c$: velocity, m/s
- $c_0$: axial component of liquid velocity between the blade profile of the initial impeller forced by difference of pressures, m/s
- $c_{oc}$: total axial velocity between the blade profile of the initial impeller, m/s
- $c_{oz}$: axial component of liquid velocity in the gap, m/s
- $c_{oxz1}$: axial component of liquid velocity at the inner diameter of the gap, m/s
- $c_{oxz2}$: axial component of liquid velocity at the outer diameter of the gap, m/s
- $c_{ow}$: axial velocity results from rotational speed of the initial impeller, m/s
- $c_{oz}$: axial component of liquid velocity at the outer diameter of the initial impeller forced by difference of pressures, m/s
- $c_{rz}$: axial component of liquid velocity between the blade profile of the initial impeller forced by difference of pressures, m/s
- $c_{rw2}$: radial component of absolute liquid velocity at the outer diameter of the initial impeller, m/s
- $c_{rz2}$: radial component of liquid velocity at the outer diameter of the initial impeller forced by difference of pressures, m/s
- $d$: diameter, m
- $d_1$: the diameter of the blade at the inlet to the centrifugal impeller, m
- $d_2$: outer diameter of the centrifugal impeller, m
- $d_{in}$: hub diameter, inner diameter of the initial impeller, m
- $d_{out}$: outer diameter of the initial impeller, m
- $g$: acceleration of gravity, m/s$^2$
- $H$: pump delivery head, m
- $H_p$: measured pump delivery head, m
- $H_{iw}$: initial impeller delivery head, m
- $n$: rotational speed, rpm
- $n_{nom}$: nominal rotational speed, rpm
- $p_1$: pressure at the inlet to the pump, Pa
- $p_2$: pressure at the outlet of the pump, Pa
- $Q$: pump discharge, m$^3$/s
- $Q_p$: measured pump discharge, m$^3$/s
Q_{sl} – fluid flow rate between the blade profile of
the initial impeller forced by difference of
pressures, m$^3$/s

Q_{sz} – fluid flow rate in the gap, m$^3$/s

Q_{th} – theoretical discharge of the initial impeller,
m$^3$/s

r – radius, m

r_{ob} – inner radius of the housing, m

r_{s1} – hub radius, inner radius of the initial
impeller, m

r_{s2} – outer radius of the initial impeller, m

s – wall thickness of the initial impeller, m

s_s – wall thickness of the initial impeller
measured parallel to the axis, m

t – pitch of initial impeller screw line, m

t_s – pitch of initial impeller screw line measured
parallel to the axis, m

u_1 – peripheral velocity of the blade at the inlet
to the centrifugal impeller, m/s

u_{w2} – peripheral velocity at the outer diameter of
the initial impeller, m/s

w_1 – mean flow-speed of liquids at the inlet to
the pump, m/s

w_2 – mean flow-speed of liquids at the outlet of
the pump, m/s

z – number of the blades

z_m – level difference between delivery
manometer and suction manometer, m

α – inclination angle of screw line of the initial
impeller, grad

α_z – inclination angle of screw line at outer
diameter of the initial impeller, grad

ρ – liquid density, kg/m$^3$

ω – angular speed of the initial impeller, rad/s

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