Design of an impeller for a fire fighting pump

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Abstract. The article presents the design of a single-stage centrifugal pump impeller for use in a fire vehicle. For such a task, the set utility parameters were determined, and then several methods were used to determine the geometry and dimensions of a given impeller. The project uses A. Stepanoff’s formulas, correction of power consumption reduction (Pfleiderer coefficient), as well as the method of point selection of the width of the impeller cross-section in relation to the changing radius of the impeller. A graphical representation of the pump flow rate and pump head depending on the type of impeller and the pump characteristic speed has enabled convenient analysis in terms of assumed pump performance. The procedure adopted in the project did not require the use of advanced engineering tools. The obtained result in the form of geometry and dimensions of the designed result coincides with the parameters of the impellers used in fire pumps currently in use. In the last step of the project, a three-dimensional impeller model was developed, which is a good starting point for optimization by simulation and then experimental verification on a real object.

Keywords. centrifugal pump; pump impeller; firefighting pump; impeller project

1. Introduction
Designing a contemporary centrifugal pump means determining the dimensions and geometry of its individual elements. The method of choosing the right type of impeller can be simplified by comparing the types of impellers that can be used. Such a comparison can be built based on ranges of functional parameters suitable for given applications. This facilitates the selection of initial design parameters, and also clearly allows for specifying the limit values of parameters such as lifting height and expenditure for the designed device.

Issues related to designing and determining the geometry of centrifugal impellers of centrifugal pumps, which are manufactured for specific applications in industry or other branches of the economy [1], come down to the selection of the impellers of these pumps [2, 3]. At the design stage, account is taken of amendment by Pfleiderer [4] (power reduction factor), A. Stepanoff formula [5] and endurance hypotheses. The supporting method was used in the work on shaping the impeller blades by determining their geometry based on an analysis of the blade width for subsequent points on the increasing radius of the impeller. This method was introduced by Carl Pfleiderer in the 1930s.
[6], but it is still applicable in the design of centrifugal components for liquid pumps and compressors [7, 8]. The first step in this approach is to choose the right type of impeller based on the specific, obtainable and required operating parameters of the designed pump [9]. The required parameters are: nominal pump impeller speed, required flow rate at nominal impeller rotation speed and pump head. Based on these parameters, the so-called pump characteristic speed is calculated. Presenting in a graphical form the dependence of the change in the characteristic speed in relation to the flow rate and pump head will allow conducting a relatively simple and quick analysis allowing to choose the impeller that best meets the assumed parameters. The analysis will be carried out on the example of the planned impeller for use in an extinguishing car intended for use by fire protection units. Similarly, it is possible to perform such an analysis for pump impellers intended for other applications, from general-purpose pumps used in industry to pumps dedicated to specific technical devices or specific tasks.

2. Applying the method to a specific example

In order to choose the right type of impeller and, as a consequence, determine its performance parameters, dimensions and geometry, it is necessary to define the pump characteristic speed ($n_{HS}$).

The pump's characteristic speed is described as follows:

$$n_{HS} = \frac{n \cdot Q^{1/2}}{H^{3/4}}$$  \hspace{1cm} (1)

$n_{HS}$ – pump characteristic speed [-], $Q$ – flow rate [m$^3$/s], $H$ – pump head, [m], $n$ – impeller rotation speed [1/min].

Impellers with a given geometry for designed pumps are selected depending on the value of the characteristic speed. Individual impeller types are used for the following $n_{HS}$ values (Table 1):

| Value of $n_{HS}$ | Impeller type | Flow direction | Impeller model |
|-------------------|---------------|----------------|---------------|
| 10-30             | Impeller blades with single curvature | Centrifugal | ![Impeller image](image1.png) |
| 30-50             | Impeller blades with spatial curvature | Centrifugal | ![Impeller image](image2.png) |
| 50-80             | Helicoidal impeller blades | Centrifugal | ![Impeller image](image3.png) |
| 80-150            | Helicoidal or diagonal impeller blades | Centrifugal | ![Impeller image](image4.png) |

In the analysed example, the impeller was selected for the pump used in a fire truck used by fire protection units. A specific example requires a pump design that must meet strictly defined parameters.
as defined in the standard PN-EN 1846-1: 2011. For the selected vehicle, these parameters reach the following values:

- nominal flow rate - 0.1 m³/s [10],
- pump head - 89.09 m - The pump head was determined on the basis of the required pump suction lift and the required pump discharge pressure when operating at the nominal impeller rotation speed. The required suction lift specified for the pump designed for the application is 7.5 m. The required discharge pressure is 8 bar, which is 81.59 m. The total pump head is the sum of these values and equals 89.09 m. [10]
- nominal impeller rotation speed - 1512 (pump driven by the combustion engine of a truck, the take-off gear ratio is taken into account and engine operation is ensured in the range in which its highest power is achieved DTI13 EURO 6.

The presented example assumes the production of an impeller that meets the parameters required for a RENAULT K520 vehicle with a drive system 6x6 intended for use in firefighting. For fire pumps, the lowest flow rates found are around 0.0075 m³/s, while the pump heads in question reach 100 m. For operational reasons, the minimum pump heads are not less than 5 m. A graphical relationship has been drawn up taking into account all these parameters, illustrating areas of expenditure and pressures at which given impeller types are used. Due to the need to generate pump operating pressure reaching a value 8 bar, ranges were omitted in the chart, in which propeller impellers are used due to the low pump head values.

![Graph showing areas of application for different pump impellers depending on the pumps head value and the flow rate](image)

**Figure 1.** Areas of application for different pump impellers depending on the pumps head value and the flow rate [9].

The figure shows the various types of impellers described in the Table 1. Individual ranges marked in the drawing correspond to other types of impellers. In range A (marked in green) an impeller with single curvature should be used (n_HS values in the range of 10-30). Respectively further ranges: B- impeller with blades with spatial curvature, C- impeller with helicoidal blades, D- impeller with helical or diagonal blades. The dependencies presented in Figure 1 allow determining the possibility of changing the flow rate or pump head by changing the type of impeller used at the constant rotational speed of the engine driving the pump, and thus at the same constant pump speed. It is also possible to specify the limit values of the pump head relative to the flow rate and type of impeller or to determine the limit values of the flow rate relative to the pump head and type of impeller.

The n_HS point marked in Figure 1 corresponds to the characteristic speed specified for the impeller designed in this example (nominal flow rate - 0.1 m³/s, pump head - 89.09 m, nominal impeller rotation speed- 1512). For this impeller, the value of the characteristic speed calculated in the calculation method is: \( n_{HS} = 16.49 \). Thus, according to Figure 1 and Table 1, an impeller with single-curved blades was selected.
Figure 2 presents a diagram of the designed impeller and the values necessary to determine its geometry. The required values are: the nave diameter $d_p$, the outer diameter of impeller ingression $d_1$, the outfall diameter of impeller $d_2$, the blade width at impeller ingression $b_1$, the blade width at impeller outfall $b_2$, the impeller width $b$ and blade angle at ingestion $\beta_1$ and outfall $\beta_2$.

![Impeller diagram with marked values](image)

Figure 2. Impeller diagram with marked values necessary to determine the dimensions and geometry of the designed impeller.

In further calculations, it is necessary to determine the specific gravity of the medium that will flow through the impeller. For firefighting, the most commonly used medium is water. Water was accepted at a temperature of 16°C, the specific gravity of which is $\gamma = 9799$ N/m$^3$.

Based on the determined $n_{HS}$ value and the specific gravity of water, it is possible to determine the power demand necessary to drive the designed impeller. This will make it possible to check whether the engine intended for the drive has sufficient power reserve and will allow determining the minimum required diameter of the propeller shaft on which the impeller will be mounted. The ingestion diameter of the impeller directly affects its other dimensions, because the impeller inlet diameter is determined by the diameter of the nave mounted on the pump propeller shaft. This diameter was determined on the basis of the minimum power demand necessary to drive the pump impeller. Power demand was determined using the following equation:

$$P_n = \frac{\gamma \cdot Q \cdot H}{\eta} \cdot 10^{-3}$$

(2)

$P_n$ – demand for impeller driving power [kW], $\gamma$ – specific gravity of water (for a specified temperature) [N/m$^3$], $Q$ – flow rate [m$^3$/s], $H$ – pump head [m], $\eta$ – general efficiency of the pump, (the average value of the general efficiency was adopted after taking into account the efficiency components of centrifugal pumps $\eta = 0.8$) [9].

For the parameters specified above, based on equation 2, an engine with a power of at least $P_n = 110$ kW is required to drive the pump impeller. Based on the demand for drive power $P_n$, the minimum required diameter of the propeller shaft on which the impeller was mounted was determined. The Huber-von Mises-Hencky strength hypothesis was taken into account, which contained a reduction in the propeller shaft strength caused by a mortise for mounting the impeller nave:

$$d_{wnm} = \sqrt[3]{\frac{84.2 \cdot 10^6 \cdot P_n}{k_{sj} \cdot \eta \cdot n}}$$

(3)

$d_{wnm}$ – minimum diameter of impeller propeller shaft [m], $k_{sj}$ – material resistance coefficient for torsion, $n$ – impeller rotation speed [1/min].
For the production of the propeller shaft, high-quality constructional steel -E335 is provided, whose strength is $R_{\text{min}} = 590$ MPa, and the resistance coefficient for torsion is $k_{sj} = 75.0$ MPa. The minimum diameter of the propeller shaft determined from Equation 3 is 44 mm. Taking into account the additional bending stresses of the drive shaft and the need for a mortise, a propeller shaft with a diameter of 50 mm was chosen. Based on the propeller shaft diameter obtained, the impeller nave diameter was adopted, which is also the impeller ingestion internal diameter. The assumed nave diameter is $d_p = 70$ mm. Based on this, the minimum theoretical outfall diameter of the impeller ingestion was determined:

$$d_{1t} = \frac{4}{\pi} \left( \frac{Q}{\eta_v \cdot c_o} + \frac{\pi \cdot d_p^2}{4} \right)$$  \hspace{1cm} (4)

$Q$ – flow rate [m$^3$/s], $d_p$ – nave diameter [m] $\eta_v$ – pump volumetric coefficient [-] $c_o$ – the meridian velocity of the ingression water at the impeller blades [m/s]

In the next step, it was necessary to use A. Stepanoff’s formulas, which take into account the pump head, as well as the water ingression speed coefficients per impeller - $K_{cm1}$ and the water outfall speed from the impeller - $K_{cm2}$, allow determining the water ingestion and outfall speed. The coefficients were determined on the basis of Figure 3, which shows the course of variation of water ingestion and outfall velocity coefficients as a function of changing the pump characteristic speed - $n_{HS}$.

The following formulas were used, by A. Stepanoff:

$$c_{m1} = K_{cm1} \cdot \sqrt{2 \cdot g \cdot H}$$  \hspace{1cm} (5)

$$c_{m2} = K_{cm2} \cdot \sqrt{2 \cdot g \cdot H}$$  \hspace{1cm} (6)

On the basis of Equation 5, the relationship allowing the determination of the theoretical meridional velocity of water at the impeller ingestion – $c_{1t}$ was determined:

$$c_{1t} = 0.9 \cdot K_{cm1} \cdot \sqrt{2 \cdot g \cdot H}$$  \hspace{1cm} (7)

![Figure 3. The course of the coefficient values - $K_{cm1}$ and $K_{cm2}$ as a function of pump characteristic speed - $n_{HS}$ [9].](image)
By determining, using Figures 3, and Equations. 5 and 7, the parameter termed by Equation. 4, a value of 0.1798 m was obtained. Based on this, the outer impeller ingestion diameter \( d_1 = 0.18 \) m was assumed.

For the diameter \( d_1 \), after determining the actual ingestion section and taking into account the coefficient \( \eta_v \) (set on the basis of Figure 4), the actual water speed at the ingestion to the impeller \( c_1 = 4.88 \) m/s was determined. The volumetric efficiency coefficient of the pump \( \eta_v \) was set on the basis of the course of the variability of the value of this coefficient, which is presented in figure 4 below.

![Figure 4](image)

**Figure 4.** The value of the volumetric efficiency coefficient pumps - \( \eta_v \) as a function of pump characteristic speed - \( n_{HS} \) [9].

On the basis of the theoretical meridian velocity, using the following relationship, the actual water velocity at the impeller blade ingestion was determined:

\[
 u_1 = \frac{\pi \cdot d_1 \cdot n}{60} \tag{8}
\]

Value determined by Equation 8 is \( u_1 = 14.24 \) m/s.

Assuming the design angle of changing the direction of water flow in the impeller of \( \alpha_1 = 90^\circ \) (the angle between the axial inflow to the impeller and the radial outflow from the impeller) the total angle of the impeller blade at ingestion \( \beta_1 = 23^\circ \) and outfall \( \beta_2 = 27^\circ \).

Based on the following dependence and the assumed reduction of power consumption \( p = 0.36 \) (estimated value taking into account equation 11), the theoretical speed of water flowing out of the impeller was determined:

\[
 u_{2t} = \frac{c_{m2}}{2tg\beta_2} + \sqrt{\left(\frac{c_{m2}}{2tg\beta_2}\right)^2 + \frac{g \cdot H (1 + p)}{\eta_v}} \tag{9}
\]

The theoretical velocity determined is \( u_{2t} = 39.39 \) m/s. On its basis, the theoretical outfall diameter of the impeller was determined using Equation 10.

\[
 d_{2t} = \frac{60 \cdot u_{2t}}{\pi \cdot n} \tag{10}
\]

By determining the parameter referred to by Equation 10 using Figure3 and Equation 6, a value of 0.4978 m was obtained. Based on this, the outfall diameter of the impeller \( d_2 = 0.5 \) m was assumed. Then, through subsequent iterations, parameters determined by Equations. 11 and 12 were set. Additionally, by analogy with Equation. 8, but using the impeller outfall diameter, the value of the actual water velocity at the outfall of the impeller was determined, which is \( u_2 = 39.56 \) m/s.

Using the C. Pfleiderer method of determining the impeller width for all points-oriented radius with respect to the impeller, available impeller widths are provided in Figure 5.
For the designed impeller, a blade thickness of 16 mm at the ingression, a blade thickness of 18 mm at the outfall, and $Z = 6$ blades were assumed. Power consumption approximation factor (Pfleiderer coefficient) was estimated at the level of $p = 0.36$. $Z$ and $p$ values result from performing an iteration including Equations 11 and 12.

Determining the remaining water speeds at the outfall of the impeller, a method analogous to the ingression speeds was used, taking into account changes in relevant parameters, size and geometry. In the next step, the correctness of the assumed value of the Pfleiderer $p$ correction and the number of impeller blades $Z$ was checked using the following relationships:

$$
p = 1.3 \frac{\left(1 + \frac{\beta_2}{60 \varphi}\right)}{Z} - \frac{1}{1 - \left(\frac{d_1}{d_2}\right)^2} \quad (11)
$$

$$
Z = 6.5 \frac{d_2 + d_1}{d_2 - d_1} \sin\left(\frac{\beta_1 + \beta_2}{2}\right) \quad (12)
$$

Based on equations 11 and 12, the Pfleiderer correction value of 0.361 was determined. Given the difference between the assumed and determined value of the correction, the assumed value was considered to be correct. Based on Equation 12, the minimum number of blades $Z = 5.84$ was determined. The number of impeller blades must be an integer, so a number of blades of 6 was assumed. The assumed number of blades, according to Equation 12, was considered to be correct.

In order to determine the last required parameter, which is the impeller width $b$, the impeller narrowing coefficient - $\varphi$ was determined on the basis of the following relationships:

$$
\varphi = \frac{b_1}{b_2} \quad (13)
$$

$$
b = \frac{\varphi \cdot Q}{2 \cdot \pi \cdot r_w \cdot c_{m1} \cdot \eta_v} \quad (14)
$$

$Q$ – flow rate [m$^3$/s], $r_w$– inner radius of impeller ingression (0.90 [m]), $c_{m1}$– water ingression speed per impeller [m/s] $\eta_v$ – pump volumetric efficiency factor [-].

Using the relationships 13 and 14, the total impeller width was determined to be $b = 0.072$ m.

All the determined impeller parameters were used to make the impeller spatial model, which is shown in Figure 6. Additionally, the values of all the determined parameters together with their descriptions have been compiled in Table 2 below.
Figure 6. Model of the designed impeller with a partial cross-section showing the shape of the blade. Model made using SolidWorks software. The red arrow indicates the direction of rotation of the impeller.

Table 2. List of determined parameters defining the dimensions of the impeller

| Parameter | Explanation | Size  |
|-----------|-------------|-------|
| d_i       | The inner diameter of impeller ingress /diameter of impeller nave | 70 mm |
| d_1       | The outer diameter of the impeller ingestion | 180 mm |
| d_2       | The outer diameter of the impeller | 500 mm |
| b         | The total impeller width | 72 mm |
| b_1       | The width of the impeller blade ingestion | 38 mm |
| b_2       | The width of the outfall of the impeller blade | 18 mm |
| β_1       | The inclination angle of the impeller blade at the ingestion | 23° |
| β_2       | The inclination angle of the impeller blade at the outfall | 27° |

3. Conclusions
The determined dimensions and geometry of the impeller allow obtaining the assumed parameters of the pump intended for a fire vehicle. The work uses a method that does not require the use of advanced and expensive computer tools. The results obtained in the form of dimensions and geometry of the designed impeller coincide with the performance of real pumps used in fire vehicles. The designed impeller can be further subjected to optimization analysis using available engineering tools [11], followed by experimental verification on a real object.

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