Hydraulic efficiencies of impeller and pump obtained by means of theoretical calculations and laboratory measurements for high speed impeller pump with open-flow impeller with radial blades

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***Abstract*—**The article discusses the results of measurements of parameters of a high speed impeller pump with open-flow impeller having radial blades. The method of calculating the hydraulic efficiency of the pump and an impeller was proposed on basis on laboratory measurements. Using the results of measurements of pressure in the space around the open-flow impeller with radial blades the hydraulic efficiency of pump and impeller were calculated.

***Key words*—**high speed pump, hydraulic efficiency, open-flow impeller, radial blades.

I. INTRODUCTION

There is increasing interest in pumps operating at high rotational speed. Steady progress in the field of construction materials, bearings systems and seals made possible to construct the pumps operating at increasingly higher rotational speeds.

Increasing the rotational speed of centrifugal pumps causes increasing the delivery head obtained from one stage of the pump and thus make possible to reduce overall dimensions and weight of the pump. The effect of increasing the rotational speed is increased pressure in the space around the impeller [1] and an increase an axial thrust acting on the pump impeller [2] [3]. For this reason, it is preferable to use of open-flow impeller, instead of closed one.

In the case of open-flow impeller with curvilinear blades, grows problems associated with large tensile stress and strain of the blade.

Under the influence of centrifugal mass forces the blade "straightens" and increasing the outer diameter of the impeller. In addition, the blade is twisted. This necessitates to use enough large distance of the tongue of the spiral casing from the outer diameter of the impeller, which increases during pumps' work and increasing of the axial gap between the casing wall at the side of the pump inlet and the edge of the blade [4].

Therefore in the high-speed pumps with open-flow impellers is preferable to the use of radial (rectilinear) blades.

Geometry of flow (velocity field) and pressure field in the case of such an impeller are different than in the impeller of traditional geometry [5] [6]. Therefore it was decided to analyze the hydraulic efficiency of the open-flow impeller with radial blades. For this purpose, the theoretical analysis of flow carried out for such an impeller.

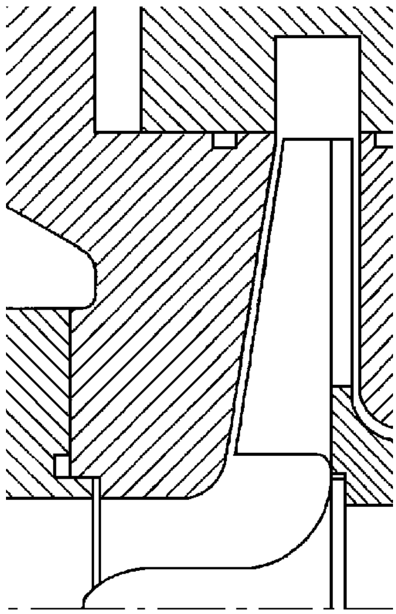

Fig. 1. Cross section of the pump flow system

Using the results of laboratory measurements of the pump working at high rotational speed, hydraulic efficiency of the impeller and hydraulic efficiency of the pump were calculated.

The article contents the results of the analyses.

II DESCRIPTION OF THE PUMP DESIGN

Laboratory tests and theoretical analysis were conducted on a specially designed pump. Cross-section of the pump flow system is shown in Fig. 1. Fig. 2 shows view on pump flow system.

It is a single-stage centrifugal pump with horizontal axis.

The pump impeller is open-flow. Due to construction (joining the blades to a hub) and strength reasons the impeller has remains of back disk. There are axial gaps between the blades of impeller and pump casing. The impeller blades are radial with decreasing width. Inlet to the impeller is axial.

The design of pump allows to use different initial impellers [7] and radial impellers with various outer diameters, diameters of the beginning of blades, blade widths and different axial gaps before and after the impeller [8]. Analyzed researches concerns the pump without the initial impeller.

Fig. 2. View of the pump flow system. The front wall of the casing and impeller nut were dismantled

The pump has a spiral casing and a diffuser.

Shaft at the exit from the casing of the pump has a mechanical seal.

The model pump makes possible to conduct researches at rotational speed up to n = 12000 rpm.

During the tests was used centrifugal impeller with the radial blades with the following dimensions:

- diameter of blades beginning $d_1 = 0.06$ m,
- outer diameter of the impeller $d_2 = 0.18$ m,
- width of the blades at the inlet $b_1 = 0.018$ m,
- width of the blades at the outlet $b_2 = 0.009$ m,
- blade thickness s = 0.005 m
- impeller disc thickness f = 0.005 m
- number of blades z = 12.

The impeller is shown in Fig. 3.

Fig 3. The impeller of the pump – view

The axial gap between the front casing wall and the impeller blades was $s_1 = 0.0015$ m.

The axial gap between the rear casing wall and the impeller was $s_2 = 0.0015$ m.

III LABORATORY TESTS

The aim of the laboratory tests was to determine the flow characteristics of the pump. Fig. 4 shows the pump on the laboratory stand.

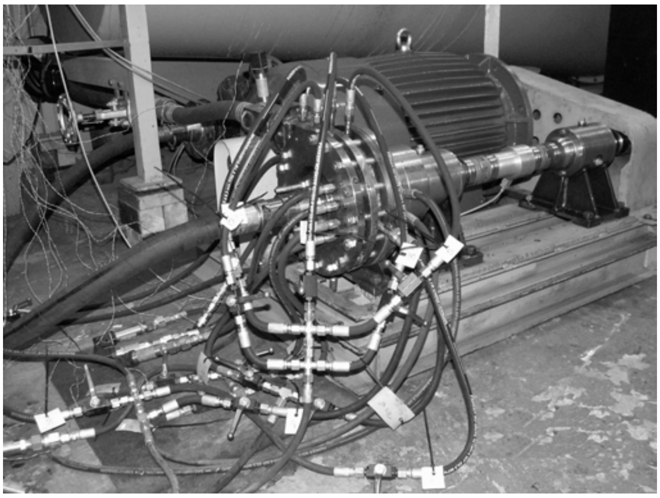

Fig 4. Pump on a laboratory stand

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*) [9].

During tests the power at the pump input (applied to the pump shaft) was measured using torquemeter mounted between drive (gear) and pump (Fig. 5) Power consumption was therefore measured directly on the shaft of the pump.

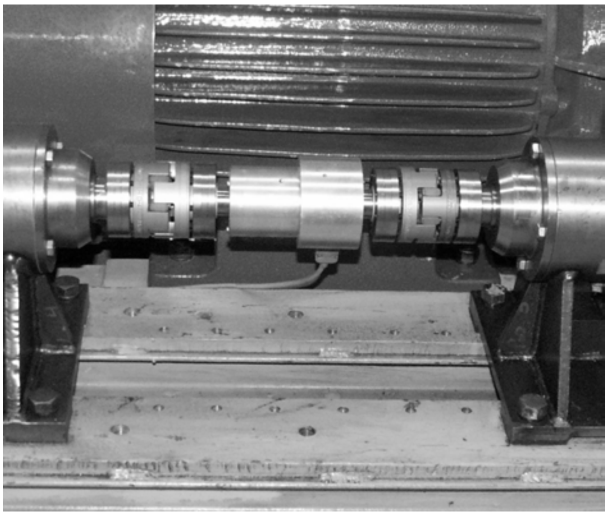

Fig. 5. Torquemeter mounted between the pump and gear

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

The readings were taken at different discharges i.e. from Q = 0 (the valve at the delivery side closed) until cavitation occurred [10].

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

$$H_p = \frac{p_t - p_s}{\rho \cdot g} + \frac{c_t^2 - c_s^2}{2 \cdot g} + z_m \tag{1}$$

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

$$Q = Q_p \cdot \frac{n}{n_{nom}} \tag{2}$$

$$H = H_p \cdot \left(\frac{n}{n_{nom}} \right)^2 \tag{3}$$

The total pump efficiency by measurement can be determined by the balance of power

$$\eta = \frac{P_u}{P} = \frac{Q \cdot \rho \cdot g \cdot H}{P} \tag{4}$$

In addition during the tests the pressure at various points of the pump flow system were measured:

- on the casing front wall before the inlet to the impeller (at the radius r_L = 0.0026 m)
- from the constructional reasons it was not possible to measure pressure at the impepper outlet (on radius r_2 = 0.09 m).Therefore on the casing wall before the impeller preasures were measured in three points with the radiuses r_C = 0.0060 m; r_B = 0.00725 m; r_A = 0.0085 m. Location of pressure measurement points shown in Fig. 6.

Values of the pressure were converted to the pressure heads according to the formula:

$$H_i = \frac{p_i}{\rho \cdot g} \tag{5}$$

Analyzing the results of the pressure readings on the front wall before the impeller it was found, that in each case the pressure dependence of the radius p = f(r) in the range from r_C = 0.060 m to r_A = 0.085 m is linear. Therefore, the pressure within the outer radius of the impeller was determined using linear extrapolation.

Fig. 7 shows the dependence of pressure on the radius in the space before impeller for the rotational speed n = 7000 rpm.

Fig. 8 to 11 shown the delivery head of the pump and the pressure heads in characteristic points of the pump flow system depending on discharge for various rotational speeds.

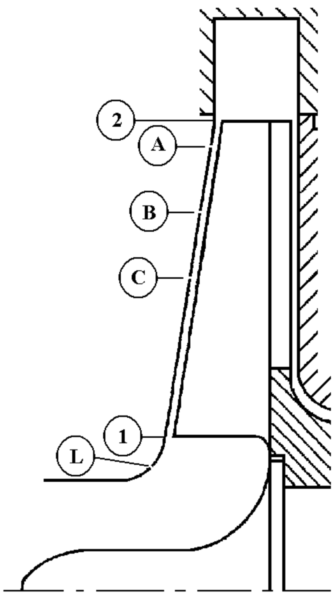

Fig. 6. Denotation of characteristic points of a flow system

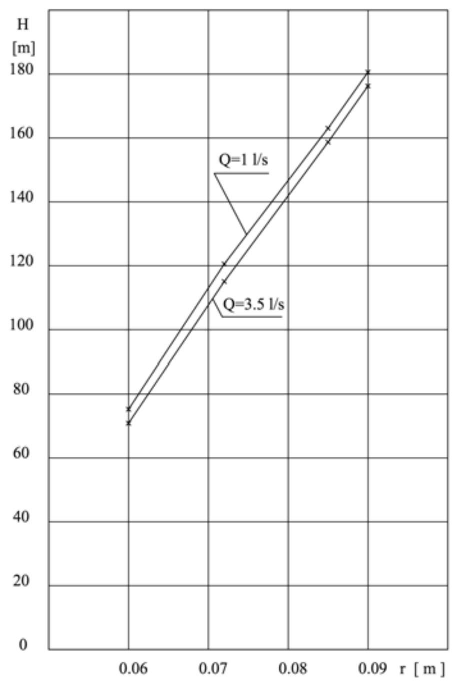

Fig. 7. Dependence of pressure on the radius in space before the impeller at rotational speed n = 7000 rpm

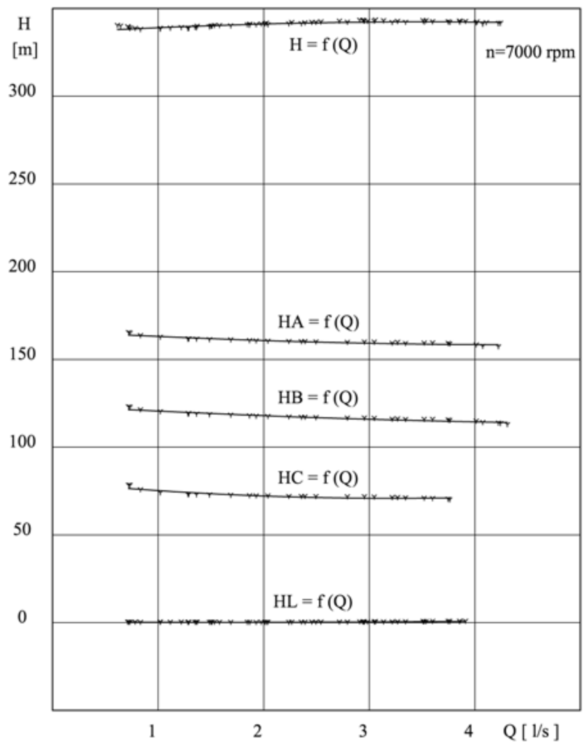

Fig. 8. Performance characteristics of the pump and dependence of pressure head on pump discharge in characteristic points of the pump flow system, n = 7000 rpm

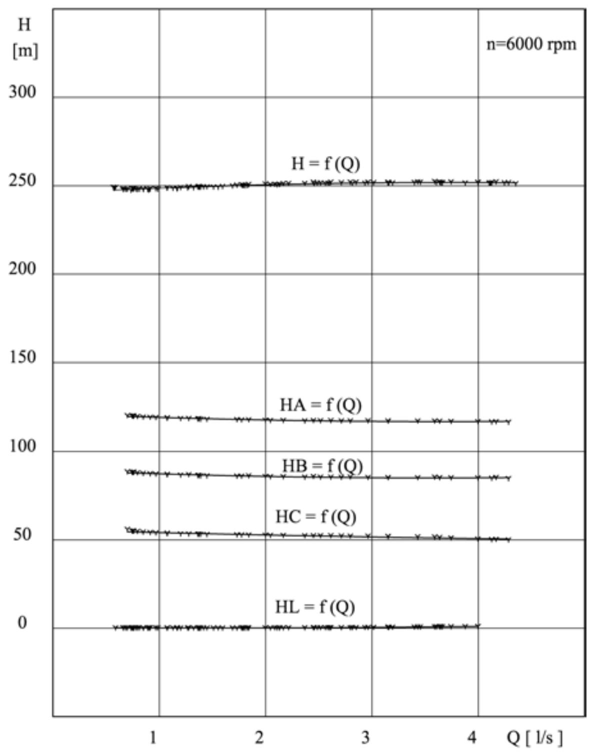

Fig. 9. Performance characteristics of the pump and dependence of pressure head on pump discharge in characteristic points of the pump flow system, n = 6000 rpm

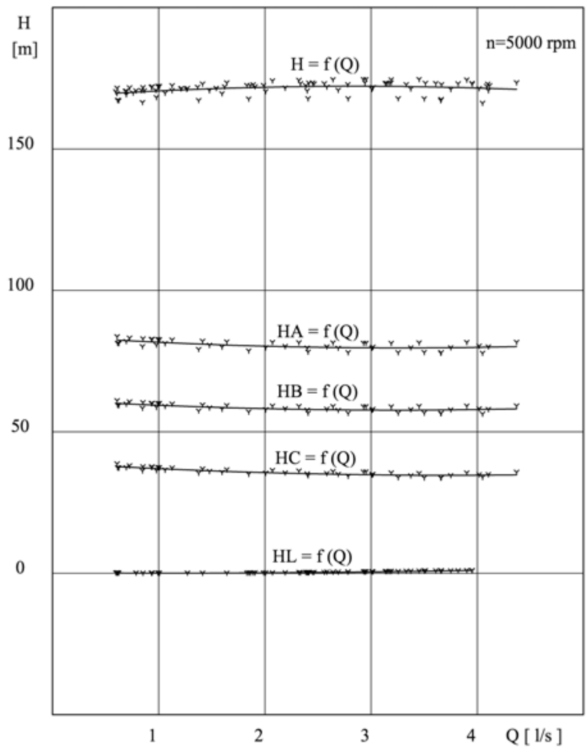

Fig. 10. Performance characteristics of the pump and dependence of pressure head on pump discharge in characteristic points of the pump flow system, n = 5000 rpm

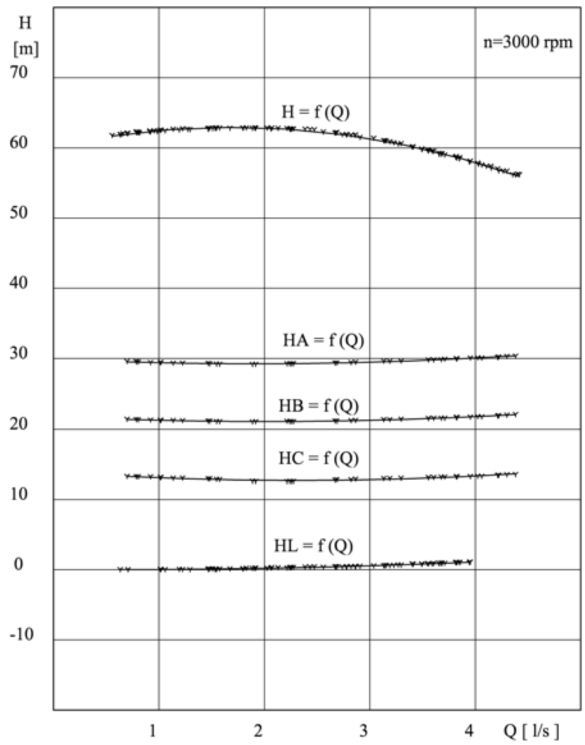

Fig. 11. Performance characteristics of the pump and dependence of pressure head on pump discharge in characteristic points of the pump flow system, n = 3000 rpm

Using the results of the tests an attempt to determine the hydraulic efficiency of the pump and hydraulic efficiency of the impeller was made.

IV THEORETICAL ANALYSIS OF THE HYDRAULIC EFFICIENCY OF THE PUMP

Analysed centrifugal pump is unusual because:
- operates at high rotational speed,
- the impeller of the pump is open-flow, which is beneficial due to lower axial thrust,
- impeller has a radial blades, which is beneficial for technological and strength reasons,
- number of impeller blades is relatively large,
- fields of sections of the spiral casing is much greater than is apparent from the method of constant angular momentum.

Therefore decided to carry out an analysis of the hydraulic efficiency of the pomp and the impeller .

Hydraulic efficiency of the pump is determined by the formula

$$\eta_h = etah = \frac{H}{H_{th}} \tag{6}$$

Centrifugal pump is a flow machine, hence using the one-dimensional flow model to analyze the phenomena of energy conversion one can use fundamental Eulerian equation for Turbomachinery [11] [12].

Euler's equation takes the form

$$H_{th} = \frac{u_2^2 - u_1^2}{2 \cdot g} + \frac{w_1^2 - w_2^2}{2 \cdot g} + \frac{c_2^2 - c_1^2}{2 \cdot g} \tag{7}$$

Peripheral velocities

$$u_1 = \omega \cdot r_1 \tag{8}$$

$$u_2 = \omega \cdot r_2 \tag{9}$$

Relative velocities are defined by formulas

$$w_1 = \frac{Q_{th}}{\frac{\Pi \cdot d_1 \cdot b_1}{\varphi_1} + \Pi \cdot d_1 \cdot s_1} \tag{10}$$

$$w_2 = \frac{Q_{th}}{\frac{\Pi \cdot d_2 \cdot (b_2 + f)}{\varphi_2} + \Pi \cdot d_2 \cdot (s_1 + s_2)} \tag{11}$$

Coefficients of narrowness are defined by formulas:
at the inlet

$$\varphi_1 = \frac{t_1}{t_1 - s} \tag{12}$$

at the outlet

$$\varphi_2 = \frac{t_2}{t_2 - s} \tag{13}$$

Pitch of the blades at the inlet

$$t_1 = \frac{\Pi \cdot d_1}{z} \tag{14}$$

Pitch of the blades at the outlet

$$t_2 = \frac{\Pi \cdot d_2}{z} \tag{15}$$

Between the relative velocities and absolute velocities occurring compounds

$$w_1 = \frac{c_{m1}}{\sin \beta_1} \tag{16}$$

$$w_2 = \frac{c_{m2}}{\sin \beta_2} \tag{17}$$

As for impellers with radial blades, there is

$$\beta_1 = 90° \tag{18}$$

$$\beta_2 = 90° \tag{19}$$

so

$$w_1 = c_{m1} \tag{20}$$

$$w_2 = c_{m2} \tag{21}$$

Absolute velocities are defined by formulas

$$c_1 = \sqrt{u_1^2 + w_1^2} \tag{22}$$

$$c_2 = \sqrt{u_2^2 + w_2^2} \tag{23}$$

V THEORETICAL ANALYSIS OF HYDRAULIC EFFICIENCY OF THE IMPELLER

Transforming the Euler's equation, the theoretical delivery head of the pump can be calculated using the formula [13]:

$$H_{th} = \frac{c_{u2}^2 - c_{u1}^2}{2 \cdot g} + \frac{u_2^2 - u_1^2}{2 \cdot g} + \frac{w_{u1}^2 - w_{u2}^2}{2 \cdot g} \tag{24}$$

Theoretical delivery head of pump can be expressed as the sum of the static (potential) delivery head H^{st}_{th} and dynamic delivery head H^{dy}_{th}

$$H_{th} = H_{th}^{st} + H_{th}^{dy} \tag{25}$$

Static pressure head is caused by centrifugal forces and reduction of the relative velocity:

$$H_{th}^{st} = \frac{u_2^2 - u_1^2}{2 \cdot g} + \frac{w_{u1}^2 - w_{u2}^2}{2 \cdot g} \tag{26}$$

Static delivery head height can be measured as the hydrostatic pressure at the outlet of the impeller.

Dynamic delivery head is due to the existing the difference of peripheral component of absolute velocities.

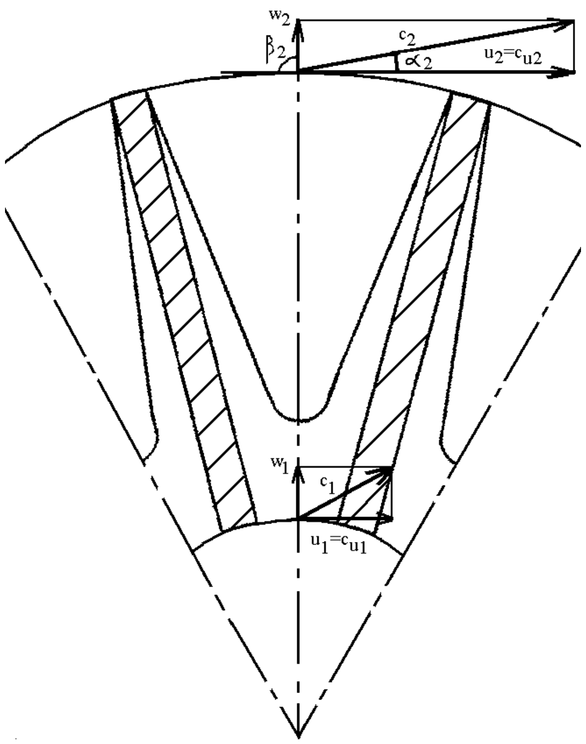

Fig. 12. Velocity triangles at the inlet and at the outlet of the pump impeller

$$H_{th}^{dy} = \frac{c_{u2}^2 - c_{u1}^2}{2 \cdot g} \tag{27}$$

These dependencies are valid under the assumption that the impeller have an infinite number of blades with infinitely small thickness.

Actual impeller, of course, does not meet those assumptions.

Fig. 12 shows the velocity triangles at the inlet and outlet of the pump impeller.

As can be seen, in the case of application of radial blades, in order to ensure non-thump inflow of liquid to the impeller should apply high initial whirl of the liquid (pre-rotation) prior to inflow into the impeller. The tested pump did not have an initial impeller or guide-wheel prior to the inflow and initial whirl has not occurred [14].

Peripheral velocities

$$u_1 = \omega \cdot r_1 \tag{28}$$

$$u_2 = \omega \cdot r_2 \tag{29}$$

From the velocity triangles results

$$\beta_1 = 90^o \tag{30}$$

$$\beta_2 = 90^o \tag{31}$$

therefore

$$c_{u1} = u_1 \tag{32}$$

$$c_{u2} = u_2 \tag{33}$$

$$w_{u1} = 0 \tag{34}$$

$$w_{u2} = 0 \tag{35}$$

VI THE CALCULATION OF THE HYDRAULIC EFFICIENCY OF THE PUMP AND THE IMPELLER ON THE BASIS OF PRESSURES MEASUREMENTS

Hydraulic efficiency of pump is determined by relation (6).

Using the results of the measurements and the theoretical dependence (Ch. IV) the hydraulic efficiency of the pumps working at different discharges and different rotational speeds were calculated.

Fig. 13 shows the hydraulic efficiencies of pump for different rotational speeds.

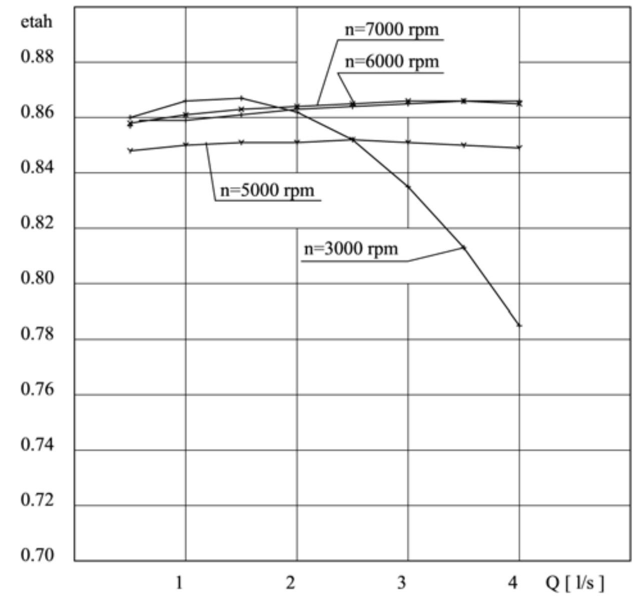

Fig. 13. Pumps' hydraulic efficiency calculated for different rotational speeds

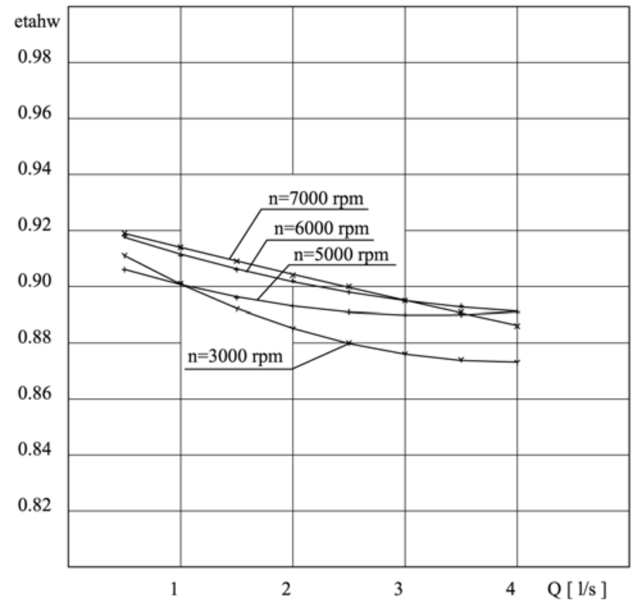

Fig. 14. Impellers' hydraulic efficiency calculated for different rotational speeds

Using the theoretical formulas (Ch. V) the static delivery head if the impeller H^{st}_{th} working at different discharges and different rotational speeds were calculated.

Static delivery head is equal to the pressure difference after and before the impeller. This difference can be calculated from the pressure measurements inside the pump

$$H^{st}_{pom} = H_2 - H_L \tag{36}$$

Comparing the values obtained from measurements with the values calculated theoretically, it is possible to determine the hydraulic efficiency of the impeller

$$\eta_{hw} = etahw = \frac{H^{st}_{pom}}{H^{st}_{th}} \tag{37}$$

Fig. 14 shows hydraulic efficiencies of the impeller calculated for different rotational speeds.

VII CONCLUSIONS

On the basis of completed research and analysis can be stated as follows:

- at high rotational speed pump has obtained a large delivery head,
- because the blade angle at outlet from the impeller β_2 is wide, liquid flowing out the impeller has large absolute velocity c_2 and dynamic delivery head of the impeller is large. The kinetic energy of the liquid was converted to pressure in spiral case and in the diffuser,
- obtained from measurements the characteristics of a delivery head of the pump and pressure distributions inside the pump have a straight course, i.e. the head of pressure almost no changes with the pump discharge,
- hydraulic efficiency of the impeller calculated from measurements is high (about 90%). These efficiency slightly decreases with increasing discharge and increase slightly with increasing pump rotational speed,
- hydraulic efficiency of the pump calculated from measurements is high (about 86%). These efficiency slightly increases with increasing discharge and increase slightly with increasing pump rotational speed,
- calculated hydraulic efficiency of the impeller decreases slightly with increasing discharge whereas the hydraulic efficiency of the pump increases. This means that the spiral casing and the diffuser of the pump pumps work more effectively at larger discharges. Spiral casing because it has a comprehensive cross-sections much greater than is apparent from the method of constant angular momentum.

The analysis confirms the advisability of the construction of pumps operating at higher rotational speed. Flow system with of the open-flow impeller with radial blades has a satisfactory hydraulic efficiency. The advantages of the application of an open-flow impeller is smaller axial thrust.

NOTATION SCHEDULE

b_1	-	width of the impeller blades at the inlet, m
b_2	-	width of the impeller blades at the outlet, m
c_1	-	absolute velocity at the impeller inlet, m/s
c_2	-	absolute velocity at the impeller outlet, m/s
c_s	-	mean flow-speed of liquid at the inlet to the pump, m/s
c_t	-	mean flow-speed of liquid at the outlet of the pump, m/s
c_{m1}	-	radial (meridional) component of absolute velocity at the impeller inlet, m/s

c_{m2} - radial (meridional) component of absolute velocity at the impeller oulet, m/s
c_{u1} - peripheral component of absolute velocity at the impeller inlet, m/s
c_{u2} - peripheral component of absolute velocity at the impeller outlet, m/s
d_1 - diameter of impeller blades beginning, m
d_2 - outer diameter of the impeller, m
f - thickness of the impeller disk, m
g - acceleration of gravity, m/s^2
H - pump delivery head, pressure head, m
H_2 - pressure head at the pump impeller outlet, m
H_A - pressure head before the impeller at point A, m
H_B - pressure head before the impeller at point B, m
H_C - pressure head before the impeller at point C, m
H_L - pressure head before the impeller inlet, m
H_p - measured pump delivery head, m
H_{th} - theoretical delivery head of pump, m
$H^{dy}{}_{th}$ - theoretical dynamic delivery head of pump, m
$H^{st}{}_{pom}$ - static (potential) delivery head of pump calculated based on measurements, m
$H^{st}{}_{th}$ - theoretical static (potential) delivery head of pump, m
n - rotational speed, rpm
n_{nom} - nominal rotational speed, rpm
p - pressure, Pa
p_s - pressure at the pump inlet, Pa
p_t - pressure at the pump outlet, Pa
P - power at the pump input (applied to the pump shaft), W
P_u - efective power, W
Q - pump discharge, m^3/s
Q_p - measured pump discharge, m^3/s
r - radius, m
r_1 - radius of impeller blades beginning, m
r_2 - outer radius of the impeller, m
r_A - radius of the point A of pressure measurement on the casing front wall, m
r_B - radius of the point B of pressure measurement on the casing front wall, m
r_C - radius of the point C of pressure measurement - on the casing front wall, m
r_L - radius of the point of pressure measurement on the casing front wall before the inlet to the impeller, m
s - blade thickness, m
s_1 - axial gap between the front casing wall and the impeller blades, m
s_2 - axial gap between the rear casing wall and the impeller, m
t_1 - pitch of the blades at the impeller inlet,
t_2 - pitch of the blades at the impeller outlet,
u_1 - peripheral velocity of the blade at the inlet to the impeller, m/s
u_2 - peripheral velocity of the blade at the outlet of the impeller, m/s
w_1 - relative velocity at the impeller inlet, m/s
w_2 - relative velocity at the impeller outlet, m/s
w_{u1} - peripheral component of relative velocity at the impeller inlet, m/s
w_{u2} - peripheral component of relative velocity at the impeller outlet, m/s
z - number of impeller blades,
z_m - level difference between delivery manometer and suction manometer, m
α_1 - angle of absolute velocity at the impeller inlet, ^{o}
α_2 - angle of absolute velocity at the impeller outlet, ^{o}
β_1 - blade angle at inlet, ^{o}
β_2 - blade angle at outlet, ^{o}
η - efficiency of the pump
η_h (etah) - hydraulic efficiency of the pump,
η_{hw} (etahw) - hydraulic efficiency of the impeller,
ρ - liquid density, kg/m^3
φ_1 - coefficient of narrowness at the impeller inlet,
φ_2 - coefficient of narrowness at the impeller outlet,
ω - angular speed of the impeller, rad/s

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