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The Influence of Wetting of Flow Passage Surfaces in Pumps-as-Turbines on Their Energetic Characteristics

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***Abstract**: This article describes the procedure of consideration for wettability of flow passage surfaces of hydropower equipment upon numerical solution of 3D problems of hydrodynamic analysis using commercial software. The experimental studies performed in Moscow Power Engineering Institute (MPEI) and devoted to the influence of wettability of internal surfaces of Du25, Du50, Du65 and Du80 pipes on hydraulic loss are systemized. The interrelation between wetting angle, equivalent sand roughness, and generalized indicator, roughness coefficient, has been determined. FlowVision software was used for verification of the proposed procedure and the influence of wettability of single elements of flow passage in M29 pump-as-turbine in pump and turbine modes on its energetic characteristics was predicted.*

Keywords: *pumped hydroelectric energy storage (PHES), pump-as-turbine, hydraulic loss, hydrophobic coating, flow passage, wetting angle, numerical simulation.*

I. INTRODUCTION

Modern hydroengineering complex of Russia is comprised of 100 operating power plants with capacity more than 10 MW and 3 PHES. Total installed capacity of hydropower units in Russia is about 49.86 GW (7th position in the world) [1].

Improvement of energy efficiency and environmental safety of electric energy production cycles and development of renewable energy resources, including hydropower, are nowadays the main trends of development of energy industry. While 3D methods of hydrodynamic analysis are being developed using *Ansys CFX*, *Fluent*, *StarCCM+*, *FlowVision*, *Numeca* and other software, prediction of energy characteristics of hydraulic machines is not very difficult, however, the issues of development of integrated procedures of their design and optimization of energy performances are still urgent. In the frames of joint Russian–Czech studies by MPEI (Moscow, Russia) and Sigma Group a. s. (Lutin, Czech Republic) in cooperation with VUT (Brno Technical University, Czech Republic) optimization approaches were developed for design of flow passages and pumps-as-turbines for hydraulic assemblies in Russia (heads $H \leq 20$ m) and Czech Republic (heads $H > 20$ m) [2]. The developed approaches were verified and adjusted on the basis of M29 pump-as-turbine (Table 1) using Sigma automated multimachine system oriented at operation conditions in PHES in Russia and Central Europe.

TABLE I. SPECIFICATIONS OF M29 PUMP-AS-TURBINE

n, rpm	Q, l/s	H, m	η_{max}, %	N, kW
Turbine mode				
1,250	54	60.0	78	22.5
Pump mode				
1,500	40	55	68	22.00

It was proposed to integrate biomimetics principles into design and optimization algorithms of hydraulic machines as a trend of development of these optimization approaches aiming at production of integrated solutions for power engineering industry. In particular, the solutions using hydrophobic coatings based on the lotus effect were considered as the most promising to improve energy characteristics of hydraulic turbines and pumps-as-turbines [3]. At present, certain amount of predictions and experimental data is available [4] for pump units of minor specific speed allowing to forecast variation trends of their energy performances and to predict increase in efficiency upon hydrophobization of specific working units. In addition, a set of studies was performed on canonic regions, such as pipe and plate. This experience can be partially applied to the field of hydraulic turbines and pumps-as-turbines. Nevertheless, a wide scope of issues related with parametric analysis is still unclarified. In particular, the influence of roughness R_z, R_a or equivalent roughness $\Delta_{eq.}$, as well as the wetting angle Θ of flow passage surfaces on energy (the head H and efficiency η) and cavitation properties of hydraulic units in pump and turbine modes is not considered. Herewith, solutions for wide range of $\Theta = 30 \div 150°$ (hydrophilicity/neutral surface/hydrophobicity) should be considered. Such experimental studies are rather labor and resource consuming. Application of software for 3D hydrodynamic analysis seems to be the most promising and cost efficient, however, it should be based on well-developed procedures allowing to consider for the aforementioned wettability properties of flow passage surfaces in computational model.

This article describes a procedure to solve this problem and results of its verification as exemplified by a M29 pump-as-turbine.

II. PROCEDURE OF CONSIDERATION FOR WETTABILITY OF FLOW PASSAGE SURFACES UPON NUMERIC SOLUTIONS TO THE PROBLEMS OF HYDRODYNAMIC ANALYSIS USING FLOWVISION SOFTWARE

In order to simulate interaction of liquid (flow) and solid phases (walls of flow passage presented by streamlined stationary or moving bodies), the computational model applies Wall boundary condition. According to the proposed procedure, the wettability properties during numerical simulation are considered by variation of equivalent roughness preset in the Wall boundary conditions (BC). This can be implemented by systemization of experimental results by MPEI of the influence of wettability of internal surfaces of Du25, Du50, Du65, and Du80 pipes on the hydraulic loss [5].

On the basis of such systemization, the following equation can be derived: $k_{\Delta_{eq.}} = f(\Delta_{eq.neutr.}, \Theta)$, which determines the interrelation of wetting angle Θ, equivalent sand roughness $\Delta_{eq.neutr.}$, when agreement with experiment is achieved in turbulent mode for pipes with neutral surface ($\Theta \approx 70 \div 90°$), and generalized indicator: coefficient of roughness $k_{\Delta_{eq.}} = \frac{\Delta_{eq.hydroph.}}{\Delta_{eq.neutr.}}$, where $\Delta_{eq.hydroph.}$ is a certain predicted equivalent roughness, when agreement with experiment is achieved on hydrophobized surface with wetting angle Θ. $\Delta_{eq.hydroph.}$ and $k_{\Delta_{eq.}}$ are predicted by the Darcy–Weisbach equation (1).

$$h_{hl} = \lambda \frac{L}{D} \frac{v^m}{2g} \tag{1}$$

where $L = 1$ m is the pipe length (in all experiments by MPEI the pipes of the same length were used); $1 \leq m \leq 2$ is the exponent determining the flow regime and the interrelation between hydraulic loss and flow rate; D is the pipe diameter; g is the acceleration of gravity; v is the flow rate; λ is the coefficient of hydraulic friction determined by the Shifrinson (2) or the Altschul (3) equation.

$$\lambda = 0.11 \left(\frac{\Delta_{eq.}}{D}\right)^{0.25}, \text{ at } \begin{cases} \mathrm{Re} < 560 \frac{D}{\Delta_{eq.}} \\ 1.5 < m \leq 2 \end{cases} \tag{2}$$

$$\lambda = 0.11 \left(\frac{68}{Re} + \frac{\Delta_{eq.}}{D}\right)^{0.25}, \text{ at } \begin{cases} \mathrm{Re} > 560 \frac{D}{\Delta_{eq.}} \\ m = 2 \end{cases} \tag{3}$$

where $Re = \frac{vD}{\nu}$ is the Reynolds number, where ν is the coefficient of kinematic viscosity ($\nu = 10^{-6}$ m²/s for water at 20°).

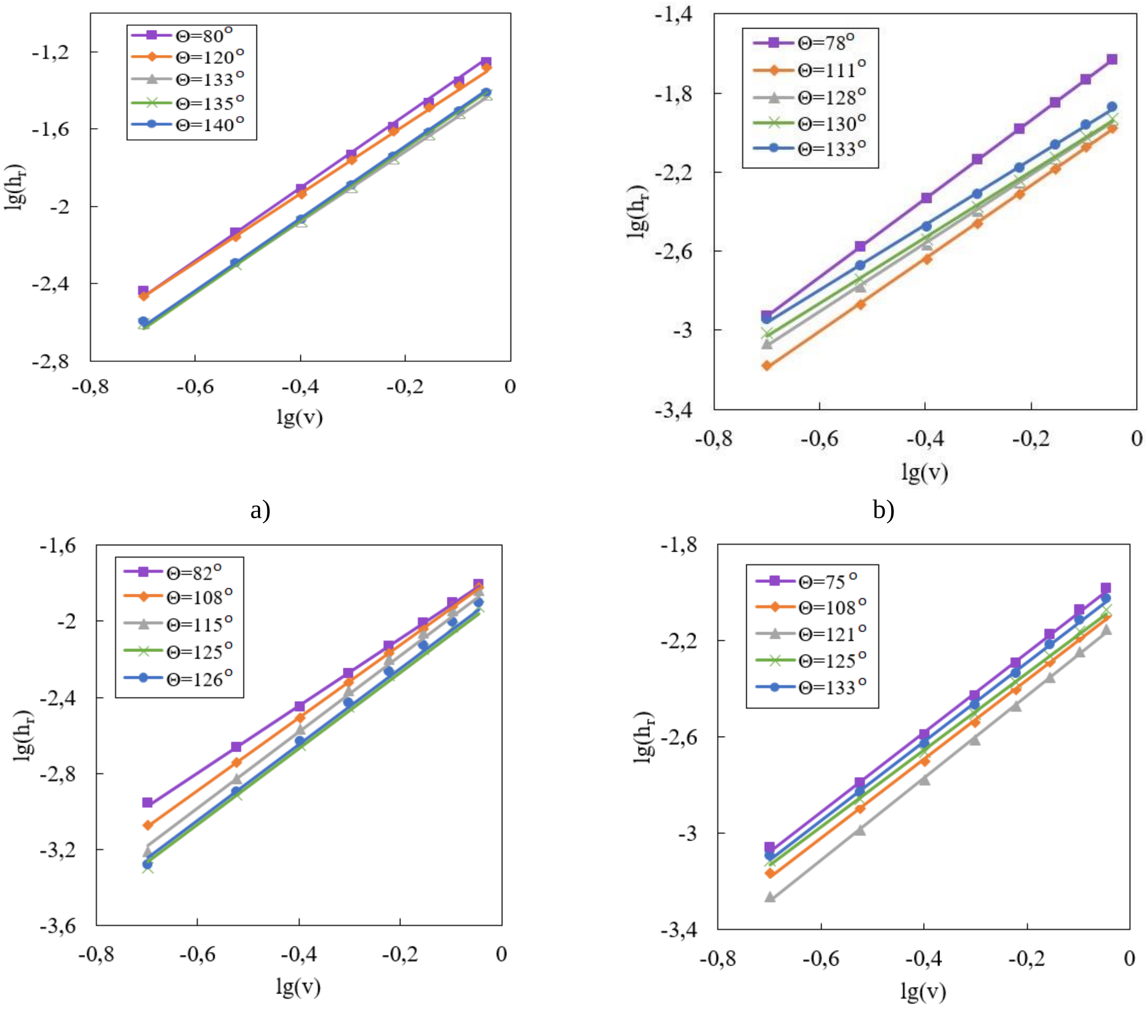
Then, with consideration for Eqs. (1), (2), and (3) with known hydraulic friction loss as a function of flow regime upon streamlining of neutral $h_{hl.neutr.} = f(v)$ and hydrophobized $h_{hl.hydroph.} = f(v)$ surfaces, it is possible to predict $\Delta_{eq.neutr.}$, $\Delta_{eq.hydroph.}$, and $k_{\Delta_{eq.}}$ by Eq. (4) and to determine $k_{\Delta_{eq.}} = f(\Delta_{eq.neutr.}, \Theta)$ by approximation of a set of points in 3D coordinates:

$$\begin{aligned} &\frac{h_{hl.neutr.}}{h'_{hl.hydroph.}} \\ &= v^{(m_{neutr.} - m_{hydroph.})} \left(\frac{\lambda_{neutr.}}{\lambda_{hydroph.}}\right) \end{aligned} \tag{4}$$

where $m_{neutr.}$ and $m_{hydroph.}$ are the exponential variables for neutral and hydrophobized surfaces. These indicators are determined analytically by finding logarithm of Eq. (1) taking the form of $\mathrm{LG}(h_{hl}) = f(\mathrm{LG}\,(v))$ (Fig. 1); $\lambda_{neutr.}$ and $\lambda_{hydroph.}$ are the coefficients of hydraulic friction of neutral and hydrophobized surfaces determined by Eqs. (2) or (3) depending on flow regimes; $h'_{hl.hydroph.} = f(v)$ is the hydraulic loss determined by Eqs. (1) and (4) as a function of flow rate upon streamlining of hydrophobized surface, maximum close to the experimental one, that is, $h'_{hl.hydroph.} = h_{hl.hydroph.}\big|_{\delta(h_{hl})_{rms} \to 0}$, $\delta(h_{hl})_{rms}$ is the root mean square deviation of predictions from experimental data determined by Eq. (5):

$$\delta(h_{hl})_{rms} = \frac{\sum_{i=1}^{n} \delta(h_{hl})_i{}^2}{n} \tag{5}$$

where n is the number of flow rate regimes used for determination of experimental evaluation of error; $\delta(h_{hl})_i = \frac{(h_{hl_i} - h'_{hl_i})}{h_{hl_n}}$ is the relative error in the i-th flow rate regime.



a) b)

c) d)

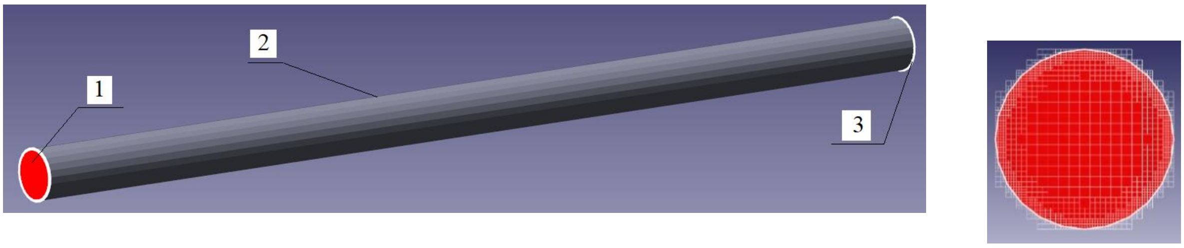
Fig. 1 Experimental studies of pipes with various wetting angles of flow passage surfaces: a) – Du25; b) – Du50; c) – Du65; d) – Du80

III. RESULTS

A. Computational studies on the basis of Du50 pipe with various wetting angles of flow passage

The computational model of Du50 pipe is illustrated in Fig. 2a, where 1 is the Inlet BC with constant pressure $p = 100$ kPa, 2 is the Wall BC with Logarithmic law wall function, 3 is the Inlet BC with preset flow rate for each regime in the range of $v = 0.2 \div 0.9$ m/s.

With consideration for the recommendations [6]-[10], the grid viscosity was analyzed at first, i.e. error estimation of numerical simulation at various density of the computational grid (Table 2 and Fig. 2b). During the predictions the pressure was monitored (Fig. 2c) at the outlet of Du50 pipe, as well as the flow rate field in the flow passages.



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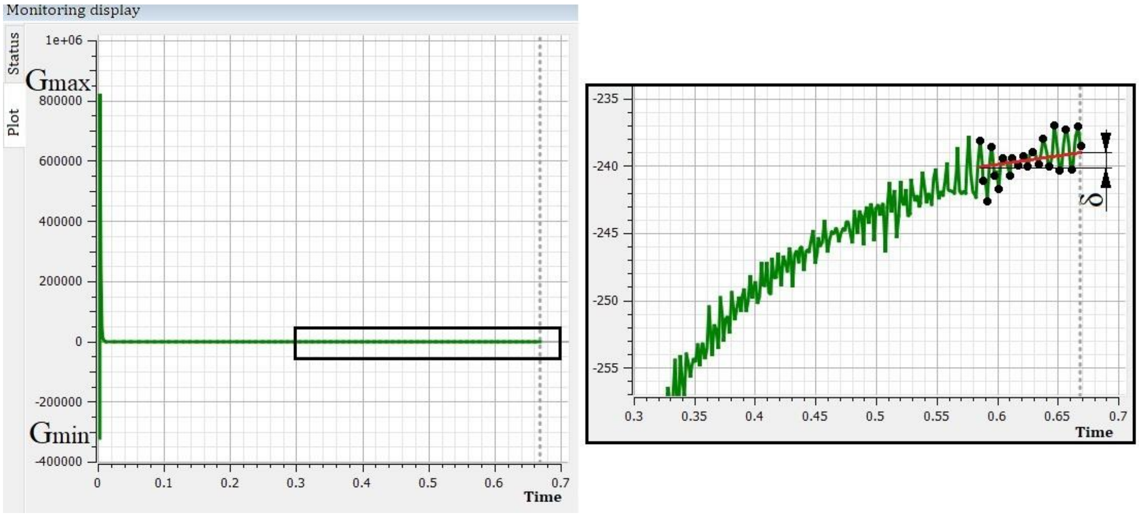


c)

Fig. 2 Computation model of Du50 pipe: a) BC; b) computation grid with adaptation in near wall region; c) termination of computations in *FlowVision*

The main criterion of error evaluation was the root mean square deviation $\delta(h_{hl})_{rms}$ of hydraulic loss predicted by *FlowVision* from experimental values in the region of simulated flow rate regimes. $\delta(h_{hl})_{rms}$ was determined by Eq. (5).

During the grid viscosity analysis, the influence of such factors as prediction method of equivalent roughness, adaptation rate of computation grid, and turbulence model on the error of computation model was determined.

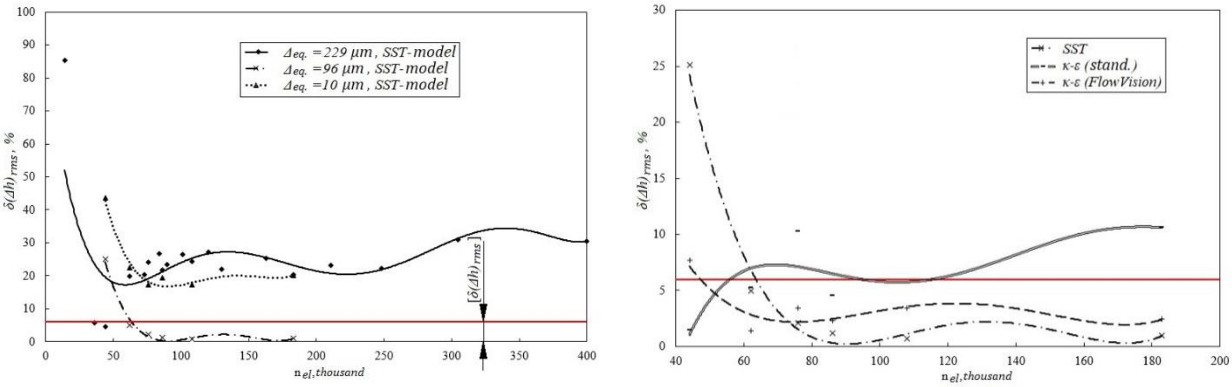
TABLE II. PROPERTIES OF COMPUTATION GRID OF DU50 PIPE MODEL

Grid No.	n_{el}, thousands	Main grid			Adaptation	
		n_X	n_Y	n_Z	Number of adaptation levels of computation grids	Number of layers of adaptation levels (array, starting from the 1st adaptation level)
1	14	5	5	100	1	[2]
2	36	10	10	100	1	[2]
3	44	10	10	100	1	[3]
4	62	15	15	100	1	[2]
5	73	17	17	100	1	[2]
6	76	15	15	100	1	[3]
7	84	15	15	100	1	[4]
8	86	20	20	100	1	[2]
9	90	17	17	100	1	[3]
10	101	17	17	100	1	[4]
11	108	20	20	100	1	[3]
12	120	20	20	100	1	[4]
13	130	20	20	150	1	[2]
14	163	20	20	150	1	[3]
15	183	25	25	150	1	[2]
16	211	28	28	150	1	[2]
17	248	30	30	150	1	[2]
18	305	17	17	100	2	[3, 2]
19	400	17	17	100	2	[3, 3]

As mentioned above, the equivalent roughness can be predicted by the Shifrinson (2) or the Altschul (3) equation depending on the flow rate regime. As exemplified by preliminary evaluation for Du50 pipe, the values of $\Delta_{eq.neutr.}$ obtained by Eqs. (2) and (3) varied by 2.39 times equaling in average to $\Delta_{eq.neutr.} = 229$ µm ($D/\Delta_{eq.neutr.} = 218.3$) and $\Delta_{eq.neutr.} = 96$ µm ($D/\Delta_{eq.neutr.} = 520.8$), respectively. According to recommendations for the considered region of experiments with Du50 pipe characterized by the flow rates of $v = 0.2 \div 0.9$ m/s and the Reynolds numbers $Re \approx 10{,}000 \div 45{,}000$, $\Delta_{eq.neutr.}$ should be predicted by the Altschul equation ($10\frac{D}{\Delta_{eq.neutr.}} < Re < 560\frac{D}{\Delta_{eq.neutr.}}$), which was also confirmed by predictions in *FlowVision* (Fig. 3) performed during analysis of grid viscosity for three values of roughness in the Wall BC: $\varDelta_{eq.neutr.} = 229$ μm, $\varDelta_{eq.neutr.} = 96$ μm, as well as $R_z = 10$ μm, measured by profile meter before experimental studies. As seen in Fig. 3a, Eq. (2) overestimated the equivalent roughness, at which the model had higher error $\delta(h_{hl})_{rms} \approx 30.5\%$, whereas at $\varDelta_{eq.neutr.} = 96\ \mu m$, determined by Eq. (3), the agreement with experiment was achieved with the error of $\delta(h_{hl})_{rms} \approx 1.0\%$. In addition, on the basis of predictions, it was established that adaptation of computational grid in some cases could lead to increased error of computational model. Thus, for instance, in the course of analysis of grid viscosity on the basis of Du50 pipe, one- and two-level adaptation of the computational grid was performed in wall region covering from 1 to 4 layers of the main grid. Upon error evaluation of computational model, cumulative number of computational cells including adaptation were considered. As demonstrated by the predictions, in the case of two-level adaptation the accuracy of the computational model sharply decreased by 7%, which could be attributed to decrease in automatically determined calculation step in time with increase in density of computational grid, as well as to increase in the oscillation amplitude of the monitored parameter (in this case: pressure at pipe outlet) in the frames of computation cycle without explicitly preset initial conditions. The latter is related with peculiarities of implementation in *FlowVision* of Eq. (6): stop of predictions based on error of monitored parameter can lead to early termination of predictions and erroneous results (see Fig. 3a).

$$\frac{\delta}{G_{max} - G_{min}} < \varepsilon \qquad (6)$$

where ε is the acceptable prediction error of monitored parameter G (during predictions: $\varepsilon = 10^{-6} \div 10^{-7}$); G_{max} and G_{min} are the maximum and the minimum values of monitored parameter in the frames of computation cycle; δ is the variation of monitored parameter upon linear approximation of the last predictions for the last n steps (see Fig. 2c).



a) b)

Fig. 3 Grid viscosity: a) consideration for prediction of equivalent roughness and adaptation degree of computation grid; b) consideration for the influence of computation turbulence model

Taking this into account upon further predictions, the variants of computational models with low grid density ($n_{el} < 62$ thousand), characterized by very low accuracy, were not considered, as well as the computation models with two-level grid adaptation ($n_{el.} > 183$ thousand).

The influence of turbulence model was analyzed with consideration for this recommendation on the basis of three most popular models according to [11], [12] upon solution of problems of hydrodynamic analysis: standard $k-\varepsilon$, $k-\varepsilon$ "*FlowVision*" and *SST*. As could be mentioned on the basis of predictions (Fig. 3b), irrespective of the error estimation of the model (by root mean square or by arithmetic mean deviation), the models $k-\varepsilon$ "*FlowVision*" and SST demonstrated high agreement between experiments and predictions. The error of SST model was $\delta(h_{hl})_{rms} \approx 1.0 \div 2.1\%$ in the range of $n_{el} = 76 \div 183$ thousand computation cells, $k-\varepsilon$ "*FlowVision*" – $\delta(h_{hl})_{rms} \approx 2.4 \div 3.4\%$. Taking this into account for computation analysis based on Du50 pipe and verification of the developed procedure of consideration for wettability properties during numerical simulation in *FlowVision*, the following settings were recommended: basic roughness $\varDelta_{eq.neutr.} = 96$ μm, SST model of turbulence, computation grid 15 (see Table 2) with 183 thousand cells with 1 level of adaptation covering 2 layers of main grid.

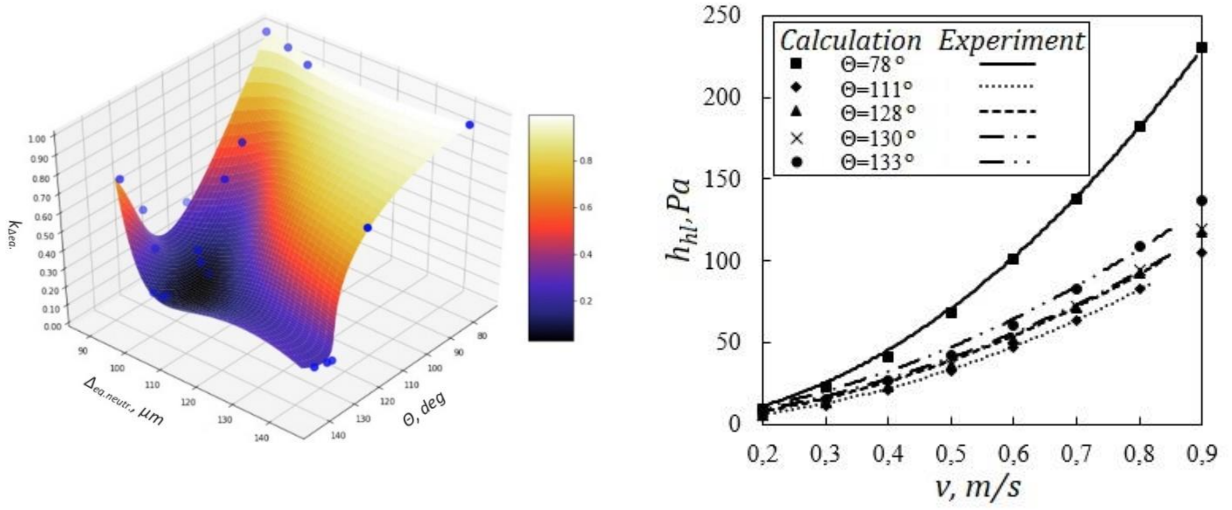
After approximation by Bezier surface of the (4,4) order applied to experimental data of Du25, Du50, Du65, and Du80 pipes with wetting angles $\Theta = 75 \div 140°$ of flow passage surfaces and roughness of $\varDelta_{eq.neutr.} = 88 \div 145$ μm, the following equation was derived $k_{\varDelta_{eq.}} = f(\varDelta_{eq.neutr.}, \Theta)$ (Fig. 4).

Aiming at control verification, the following was performed:

– using $k_{\varDelta_{eq.}} = f(\varDelta_{eq.neutr.}, \Theta)$, the $k_{\varDelta_{eq.}}$ parameters were predicted for Du50 pipe with $\varDelta_{eq.neutr.} = 96$ and wetting angles $\Theta = 78°$, $\Theta = 111°$, $\Theta = 128°$, $\Theta = 130°$, $\Theta = 133°$;

– $\varDelta_{eq.hydroph.}$ were determined for Wall BC of computation model;

– numerical simulation was carried out using FlowVision software and qualitative evaluation of agreement between predicted and experimental trends.



a) b)

Fig. 4 Verification of the developed procedure of consideration for wetting properties in computation model: a) visual presentation of the function $k_{\Delta_{eq.}} = f(\Delta_{eq.neutr.}, \Theta)$; b) $h_{pl} = f(v)$ plots of Du50 pipe with $\Theta = 78° \div 133°$

As can be seen in Fig. 4, the numerical simulations are characterized by qualitative agreement with experimental data, which allows to apply the proposed procedure of consideration for wettability of flow passage surfaces during numerical simulation using *FlowVision* software for more complicated problems.

B. Computation studies of M29 pump-as-turbine with modified wettability of flow passage surface

This section of computation analysis was aimed at evaluation of efficiency variation of M29 pump-as-turbine in turbine mode upon variation of wettability of flow passage surface. For numerical simulation in *FlowVision* the computational 3D model was developed (Fig. 5), its parameters and BC are summarized in Table 3.

Data were monitored using the following variables: $H = \frac{p_{in}\text{-}p_{out}}{\rho g}$ was the pressure drop at inlet and outlet of hydraulic unit in turbine mode (turbine mode), where p_{out}, p_{in} were the full pressures averaged by surfaces with BC No. 1 and 7, respectively; $\eta = \frac{M\omega}{\rho g Q H}$ was the efficiency, $\omega = 157$ s-1 was the angular speed, M was the torque determined by the surface with BC No.2 , $Q = k_Q Q_{nom}$ was the working mode in terms of flow rate; Q_{nom} was the nominal flow rate of hydraulic unit in turbine mode, $k_Q = 0.4 \div 1.6$ was the consumption coefficient allowing to preset operation mode of the hydraulic unit. Predictions were terminated in the case of error of head $\varepsilon_H \leq 10^{-8}$.

The main elements for modification were the blade system (BS) and the volute chamber (VC). Modification of disk cavities and slot seals was not analyzed in this work.

TABLE III. PARAMETERS OF M29 PUMP-AS-TURBINE COMPUTATIONAL MODEL

Model of turbulence	Subregions	BC (see Fig. 5)	
		No.	Description
SST	Impeller (Imp): Local VC; RPM: ω	1	Flow passage surface: BS BC: Wall Roughness: $\Delta_{eq.neutr.M29}, \Delta_{eq.hydroph.M29}$ Wetting angle: $\Theta_{neutr.M29}, \Theta_{hydroph.M29}$
		2	Flow passage surface: Driven and driving disks, seal band of Imp BC: Wall Roughness: $\Delta_{eq.neutr.M29}$ Wetting angle: $\Theta_{neutr.M29}$
		3	BC: Coupled (sliding surfaces)
	Casing: Global VC; RPM: 0	4	Flow passage surface: Inlet of hydraulic unit (in pump mode) BC: Inlet Inlet pressure: 100 kPa

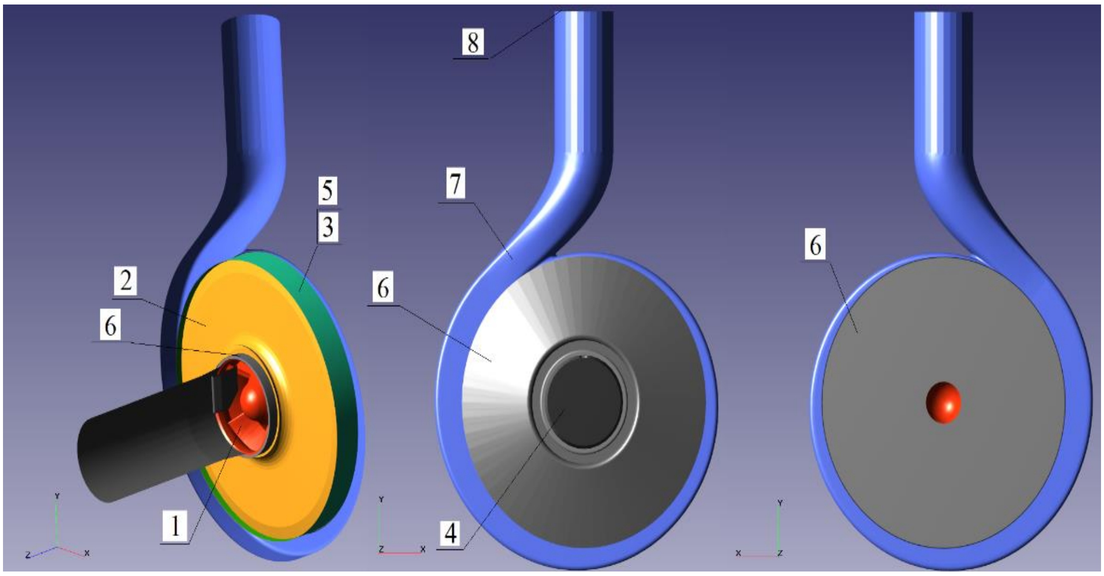
		5	BC: Coupled (sliding surfaces)
		6	Flow passage surface: Cavities of driven and driving disk, seal rings, supply BC: Wall Roughness: $\Delta_{eq.neutr.M29}$ Wetting angle: $\Theta_{neutr.M29}$
		7	Flow passage surface: VC BC: Wall Roughness: $\Delta_{eq.neutr.M29}, \Delta_{eq.hydroph.M29}$ Wetting angle: $\Theta_{neutr.M29}, \Theta_{hydroph.M29}$
		8	Flow passage surface: Outlet of hydraulic unit (in pump mode) BC: Outlet Mass flowrate: $v_M = -\rho Q/F_{out}$

$\Delta_{eq.neutr.}$ was determined on the basis of agreement between predictions in *FlowVision* and experimental studies of M29 pump-as-turbine, carried out at CHV "SIGMA" test bench (Lutin, Czech Republic) and in VUT laboratory (Brno. Czech Republic). The wetting angle $\Theta_{neutr.M29}$ of respective neutral surface of flow passage of the hydraulic unit was determined on the basis of $f(\Delta_{eq.neutr.M29}, \Theta_{neutr.M29})\big|_{k_{\Delta_{eq.}}=1}$. $\Theta_{hydroph.M29}$ and $\Delta_{eq.hydroph.M29}$ were determined by Eqs. (7) and (8) after searching for extreme value of $k_{\Delta_{eq.}} = f(\Delta_{eq.neutr.}, \Theta)$.

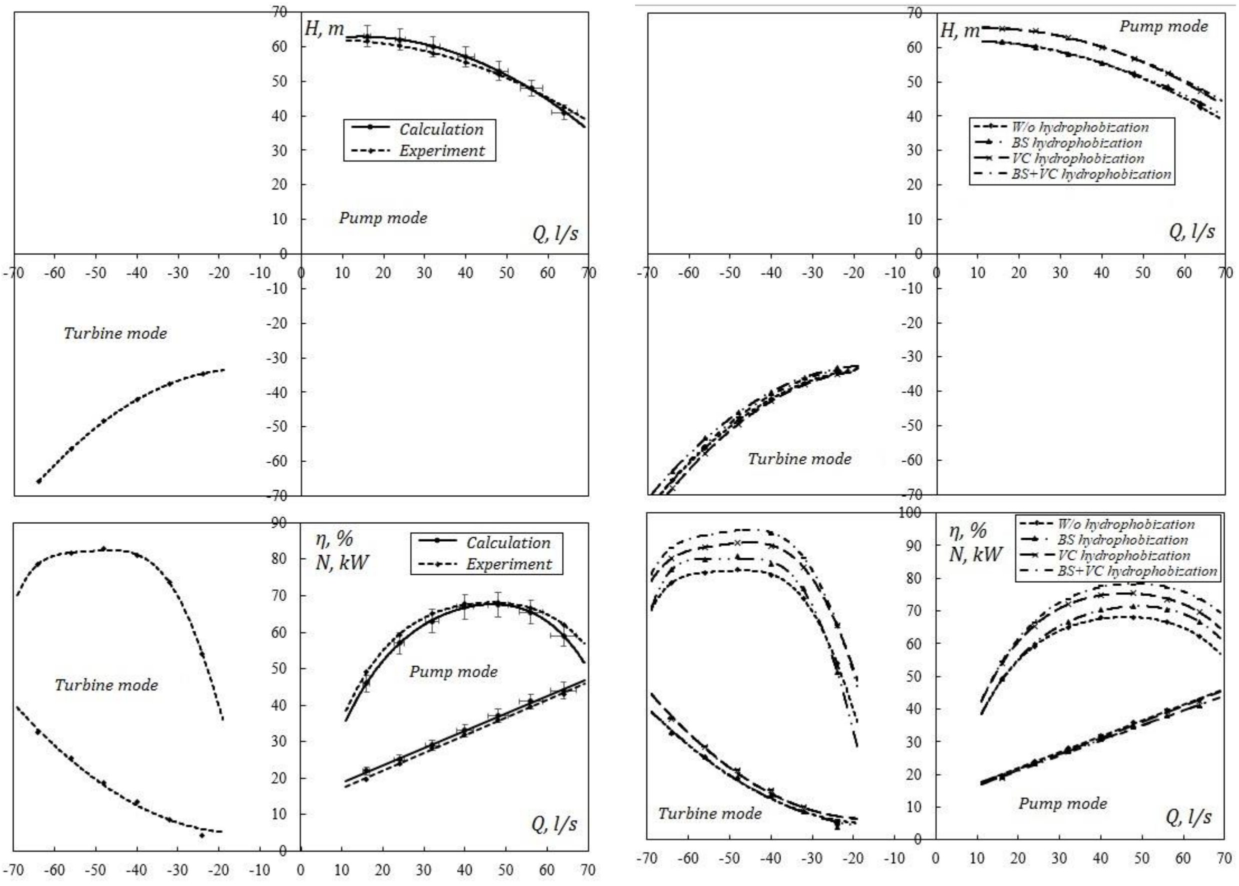
$$f(\Delta_{eq.neutr.M29}, \Theta_{hydroph.M29})\big|_{k_{\Delta_{eq.M29}} \to min} \qquad (7)$$

$$\Delta_{eq.hydroph.M29} = k_{\Delta_{eq.M29}} \Delta_{eq.neutr.M29} \qquad (8)$$

ON the basis of numerical simulation (Table 4) in *FlowVision*, the error of efficiency, head, and capacity of M29 pump-as-turbine was not higher than 5% in the feed range of $Q = 25 \div 40$ l/s in comparison with experimental data. Maximum increase in efficiency $\Delta\eta$ of the pump-as-turbine, achieved according to prediction at complex hydrophobization of BS and VC, was $\Delta\eta = 10.34\,\%$ in pump mode and $\Delta\eta = 12.19\,\%$ in turbine mode. It should be taken into account that the efficiency increase is determined mainly by scaling effect. Thus, the M29 pump-as-turbine impeller diameter of only $D_2 = 0.4$ m is comparatively small, and the influence of microrelief and surface wettability is significant. It is quite expectable that in a turbine of megawatt class $\Delta\eta$ would be lower.



a)



b) c)

Fig. 5 Predictions on the basis of M29 pump-as-turbine with modified wettability of functional surfaces: a) computation model; b) experimental

IV. CONCLUSION

1. The procedure of consideration for wetting properties was developed during solving $3D$ problems of hydrodynamic analysis and the equation was derived: $k_{\Delta_{eq.}} = f(\Delta_{eq.neutr.}, \Theta)$, which allowed to predefine BC upon numerical simulation of flows in passage elements of hydraulic equipment with modified wettability, including hydraulic units and water conduits.

2. After verification of the developed procedure based on Du50 pipe using *FlowVision* software, it was established that the results of numerical simulation qualitatively agreed with experimental data, thus allowing to use the proposed procedure of consideration for wettability of flow passage surfaces during numerical simulation.

3. The influence of wettability of single elements of flow passage of M29 pump-as-turbine on its energy characteristics was evaluated in pump and turbine modes. According to the evaluations, the maximum effect was observed upon integrated hydrophobization of blade system and volute chamber of M29 equaling to $\Delta\eta = 10.34\%$ and $\Delta\eta = 12.19\%$ in pump and turbine modes, respectively.

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