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THE INFLUENCE OF THE IMPELLER CONSTRUCTION ON THE PERFORMANCE OF ONE CHANNEL PUMP

1.Introduction

One channel pumps dedicated for contaminated fluid transportation. Its advantages is as a large width of the passage which is the result of one blade application.

The main purpose of this work is to analyse flow structures in the three types of single bade pumps equipped with three different impellers. In such analysed cases the volutes were identical.

2. The object of research

The object of the research is a sewage pump where three different types of impellers were investigated (fig. 1). The main operating and geometrical parameters of the analysed one channel pumps are listed in table 1.

Table 1. Operating and geometrical parameters of the analysed one channel pumps.

Name	Symbol	Impeller No. 1	Impeller No. 2	Impeller No. 3
Performance parameters of the pump				
Design flow rate	Q, m^3/h	123	129	122
Design head	H, m	10.8	11.1	11.5
Rotating speed	n, $1/min$	1450	1450	1450
Specific speed	n_q	45	45	43
Geometrical parameters of the impeller				
Outlet diameter	d_2, mm	243	240	246
Inlet diameter	d_1, mm	105	100	100
Inlet width	b_1, mm	106	82	70
Outlet width	b_2, mm	85	82	82
Blade wrap angle	α, °	330	320	425
Outlet blade angle	β_2, °	14	18	26

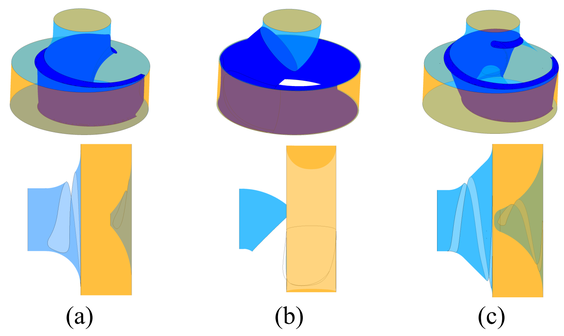

Fig. 1. Solid model of one channel pump impeller: (a) impeller No. 1, (b) impeller No. 2, (c) impeller No. 3.

The numerical simulation of the pumping process in one channel-pump passages was conducted by means of ANSYS CFX software (fig. 2).

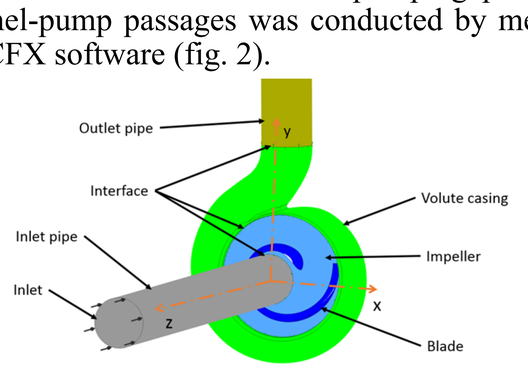

Fig. 2. Model dedicated for numerical simulations.

3. Results and discussion

In general, obtained H–Q curves have similar steepness (fig. 3). The differences in the head between each impeller are not regular in function of discharge. In the design point, the difference between the achieved outcomes equals approximately 3–5 % and the maximum value – 10% – was noticed at lower flow rates. Impeller No. 2 is characterized by specific shape of the meridional cross–section and blade (it is very wide), which has influence on the slope of the H–Q curves.

Efficiency curve obtained for impeller No. 2 has aseparate form. It is located higher for flow rates in the range (0.3–1.0)Q_{nom} reference to other impellers and maximum values of efficiency were noted for flow rates in the range (0.7–1.5) Q_{nom}. The reduction of hydraulic losses is the main reason of the achieved results.

The power consumption curves of the analysed pumps have similar steepness. A slight difference could be observed on the curves form caused by alteration in the head and losses characteristics.

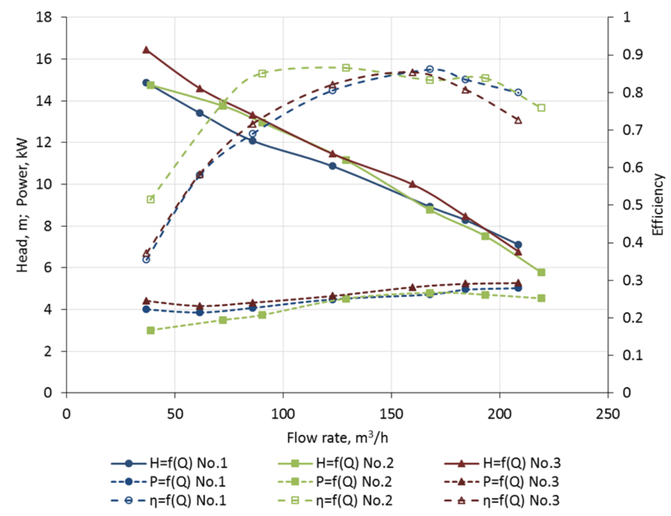

Fig. 3. The comparison of numerical simulation results for different impellers.

The peak–to–peak height values of the pulsation of head at Q_{nom}. are: 39 m (32%), 2 m (18%) and 4 m (30%), for pump with impellers: No. 1, No. 2 and No. 3, and the average values are: 12.2 m, 11.5 m and 13 m respectively (fig. 4). The highest values were obtained to the position in which the trailing edge of the blade is located 120° to the tongue of the casing. The smallest value was obtained to the position when the trailing of the blade edge is passing the tongue of the volute.

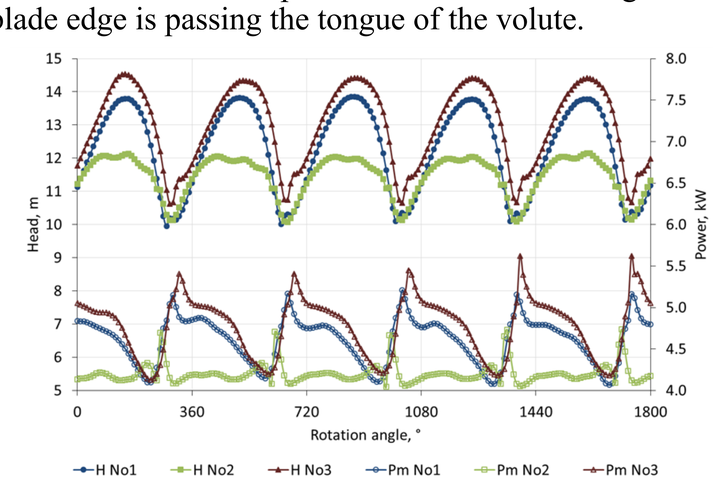

Fig. 4. Fluctuation of the head and power refers to blade position

When power consumption trend is taken into consideration, the peak–to–peak height values are about: 1.1 kW (23%), 0.6 kW (15%) and 1.3 kW (27%) for pump with impellers: No. 1, No. 2 and No. 3.The average values are: 4.6 kW, 4.2 kW and 4.8 kW respectively. Power curves have different shape in comparison with head curves. Rapid increase of power consumption is observed when the trailing edge is passing the tongue. This does not correspond to the high peak of the head. After that, slow decrease of power for impellers: No.1 and No. 3 and sudden drop for No.2 is observed. In general power consumption curve analysed in function of rotation angle is more uniform for impeller No. 2. The reason can be explained by smaller hydraulic losses in the impeller and volute casing.

The pressure distribution depends on the position of the impeller. The comparison of the absolute pressure fields indicates that they are similar for both cases: No. 1 and No. 2 (fig. 5). The small number of blades (one blade) cause not uniform pressure distribution. The differences in pressure for pump with impeller No. 2 and the other two units are quite large. The highest divergences were achieved in suction side of the blades and the smallest in the volute casing. The outlet diffusers were characterized by the identical pressure distribution.

Uneven pressure distribution around the blade and in the volute casing encourages to make the analysis of the radial forces in the pump. Calculated radial hydrodynamic loads for all impellers, by means of equation:

$$(F = (f_x^2 + f_y^2)^{1/2}),$$

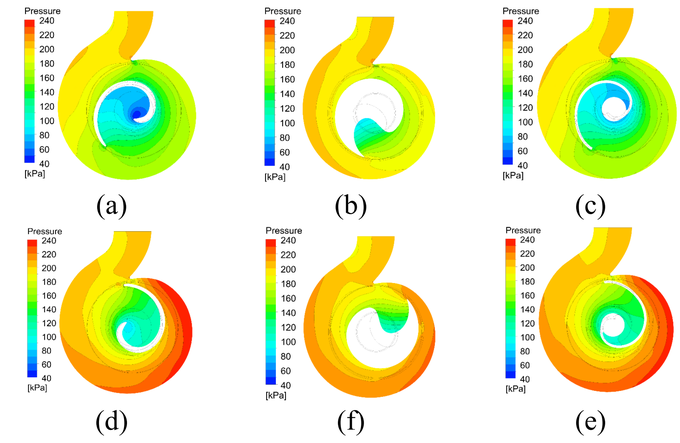

Fig. 5. Pressure distribution in considered pumps in the Q_{nom}: (a) impeller No. 1 (max. head), (b) impeller No. 2 (max. head), (c) impeller No. 3 (max. head); (d) impeller No. 1 (min. head), (f) impeller No. 2 (min. head), (e) impeller No. 3 (min. head).

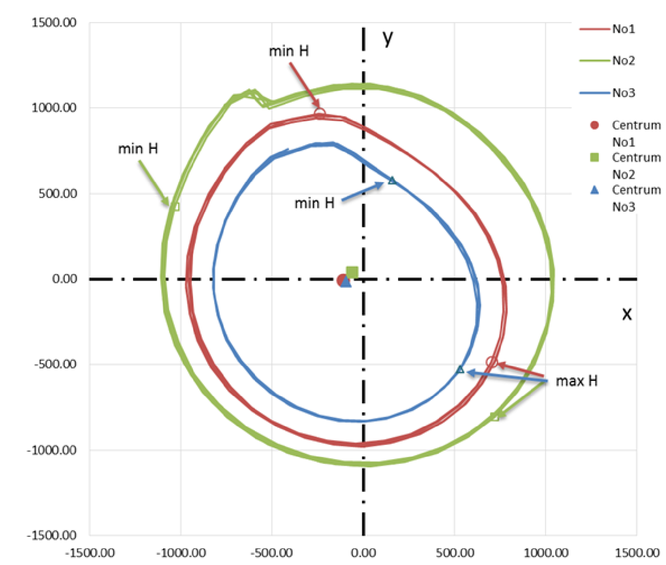

Fig. 6. Calculated radial hydraulic force, [N].

Radial forces are imbalanced, because the divergences from the centre are approximately 100 N on axis X and 20 N on axis Y for pump with impeller No. 1 and No. 3 (fig. 6). When analysing impeller No. 2 it is 70 N and 40 N respectively. The smallest hydraulic radial force was obtained in the course of model No. 3 examination. The possible reason of this situation is the biggest wrap angle of the blade. Considering point of operation with maximum head, the hydraulic forces are directed in the same way for all impellers. It confirms that the position of the trailing edge determines the maximum head. The points of minimum head operation correspond to the vectors of the hydraulic force characterized by different direction for each impeller.

Conclusions

In general, obtained H–Q curves have similar steepness. The differences between the achieved head curves obtained for each impeller are not regular. In the design point it is approximately 3 –5 %.

Efficiency curves of impellers No.1 and No. 3 are similar, but the curve obtained for impeller No. 2 has a separate, more advantageous form.

The differences between the power consumption curves close to the best point equal less than 4% and this value is rising away from the Q_{nom}.

Unsteady simulation shows the pulsation of head and power at Q_{nom}. The highest values were obtained to the position in which the trailing edge of the blade is located 120° to the tongue of the volute and the smallest for the position of the impeller when the trailing edge of the blade is passing the tongue of the spiral casing. Power pulsation curves have different shape in comparison with head pulsation curves. In general, power consumption curve analysed in function of rotation angle is more uniform for impeller No. 2.

The absolute pressure distributions at the mid–span of the impellers and volute cross-sections indicates that inner pressure distributions are similar for cases with impellers: No. 1 and No. 2. The application of one blade causes irregular pressure distribution. The differences in the achieved outcomes between pump with impeller No. 2 and the other two are quite large.

Calculated radial hydrodynamic force in a function of the position of the impeller confirms its unbalance, the centres of hydrodynamic forces are shifted mainly along X axis and could cause additional vibrations.