EFFECT OF IMPELLER TUNING FOR PUMP USED IN TURBINE MODE

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Abstract
This article presents results of experimental verification of construction modifications of radial centrifugal pump in reverse turbine mode. The measured characteristics indicated that a simple adjustment of the impeller by rounding the input edges, along with the reduction in roughness of the flow parts, resulted in an increase in efficiency of 2% and in performance, 1.5% while reducing the flow and gradient. The article also shows the correction of the conversion relationships needed when designing a pump for turbine mode, where it is necessary to take into account the varying efficiency of reversing pump operation. This correction results from real experience and has not yet been described in literature.

Key words: *pump as turbine (PAT), impeller trimming, efficiency.*

INTRODUCTION

In the field of small hydropower plants, hydrodynamic pumps as turbine (PAT) are often used as an alternative to conventional water turbines. This option is of particular interest because of its low investment cost and its applicability, especially for decentralized sources with outputs of up to 100 kW. When building small hydropower plants, the most expensive investment is the price of the turbocharger and hence the effort to find alternatives (*Alatore-Frenk*, *1994*). Given the wide range of pumps on the market, these relatively inexpensive and reliable units are often more advantageous in terms of maintenance than conventional custom-made turbines (*Bláha, Melichar & Mosler 2012*).
However, when selecting pumps for turbine operation, it is necessary to take into account the specificities associated with their reversibility. This concerns in particular the geometry of the flow cross sections, which are usually fixed and as such do not allow for regulation in changing operating conditions. Furthermore, it is necessary to consider that optimal parameters of pump and turbine operation differ (*Nautiyal*, *Varun & Kumar*, *2010*). The design process is based on the hydrotechnical parameters of the site where the machine is to be installed. These parameters are the gradient H (m) and the water flow Q ($m^3 \cdot s^{-1}$). Based on the theory of hydraulic machines, the parameters for selecting the corresponding pump are determined by the conversion, i.e. the water head H_P (m) and the flowrate Q_P ($m^3 \cdot s^{-1}$) (*Güllich, 2014, Munson, Zouny & Okiishi*, *2006*).

$$H_P = \frac{H}{\eta_P{}^x} \qquad [m] \tag{1}$$

$$Q_P = \frac{Q}{\eta_P{}^y} \qquad [m^3 \cdot s^{-1}] \tag{2}$$

where, η_P is the total pump efficiency. The values of the exponents x, y may vary in literature with authors. The most reported values are $x = 2$ and $y = 1$ (*Nautiyal, Varun & Kumar, 2010*).The most appropriate type and size in the pumps manufacturer's catalogue will be then selected according to the parameters H_P a Q_P. However, in some cases, the effectiveness of the machine is often a question. Here, it is assumed that it will not change by reversing the operation. Nevertheless, the efficiency of turbine and pump operations vary, with that of turbine operation being lower. This unfavourable state may be corrected through some design adjustments that lead to increased efficiency. The adjustments mainly concern flow geometry and require a thorough understanding of the hydraulic mechanisms of energy transfer (*Singh & Nestmann*, *2011*). These can be dealt with only by a combination of theory and experimental verification. Some of the modifications are described, for example, in (*Derakhshan & Nourbakhsh, 2008; Polák, 2013; Poláková & Polák, 2016; Raman, Hussein, Palanisamy & Foo, 2013*). This article focuses on experimental verification of impeller adjustments that may be performed on a finished pump and on verification of the above-mentioned conversion relationships, (1) and (2).

MATERIALS AND METHODS

The principle of additional optimization of the pump consists mainly in the reduction of hydraulic losses. There are basically two kinds of these - frictional and local losses. The proposed modifications are such as are easy to perform on the machine and do not further increase the purchase price. Most of the adjustments relate to the impeller and the areas directly connected, i.e. the zones of entry, passage and outlet of the fluid from the impeller (*Singh, 2005; Sedlář, Šoukal & Komárek, 2009*).
When the fluid passes through a pump in turbine mode, the output edges of the impeller blade become input edges. In these spots, the blades are usually ended with sharp edges which are not problematic in pump mode, but cause liquid shock in turbine mode, manifested by an increase in local hydraulic losses and a decrease in efficiency. The magnitude of the hydraulic loss depends on the wake region, i.e. the size of the fluid area affected by the immersed body. Such loss can be reduced by the appropriate shaping of the immersed body, in this case by rounding its edges. Fig. 1 schematically shows the size of wake region during fluid flow by unmodified input profile on the left, modified on the right. Both the edges of the blades and of the rear and front shroud of the impeller on its outer diameter, D_1 were rounded. This modification involves removing the impeller and rounding the edges, e.g. by grinding, which is a very simple process.

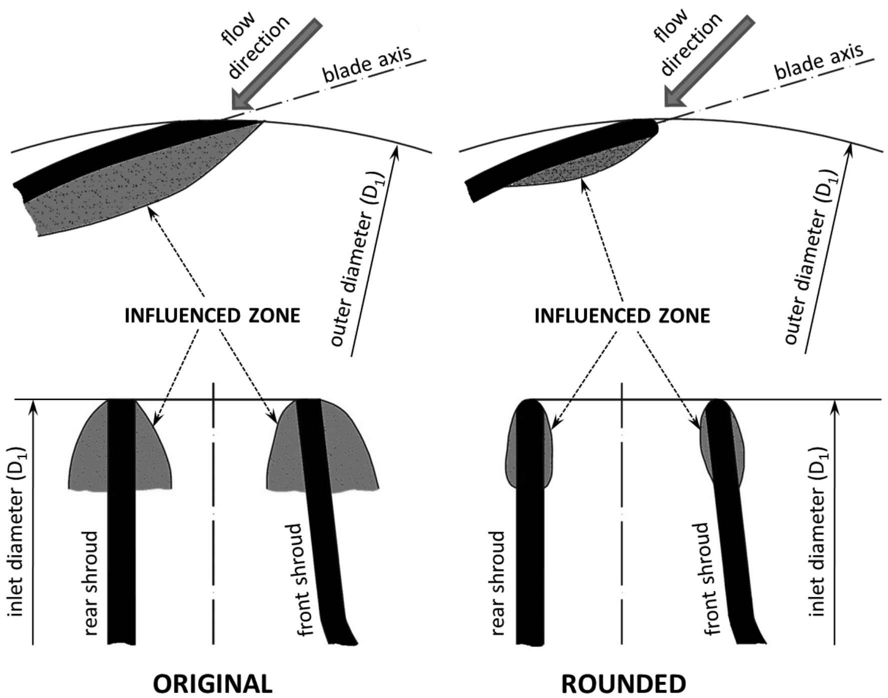

Fig. 1 Influence of rounding of input edges on fluid flow through the impeller

Further reduction of hydraulic losses in turbine operation is achieved by reducing the roughness of the surfaces which are in contact with the flowing fluid. This is achieved through additional machining (grinding, polishing, etc.) and eventually by applying suitable coatings of ceramics, plastic, etc. Such modifications are known for pumps used in special applications, e.g. in the chemical industry (*Güllich, 2003*). However, experience in verifying their effect on turbine pump operation is still lacking in literature.

Experimental verification of modifications

The META Plus 5 pump, manufactured by ISH Pumps Olomouc, was selected for experimental verification of the above mentioned optimization adjustments. From a design point of view, this is a one-stage centrifugal pump with a spiral box and impeller with outer diameter D_1 = 132 mm. The parameters of the original untreated pump at optimal efficiency, at 1450 rpm, are shown in Tab. 1. The diagram of the pump design is shown in Fig. 2.

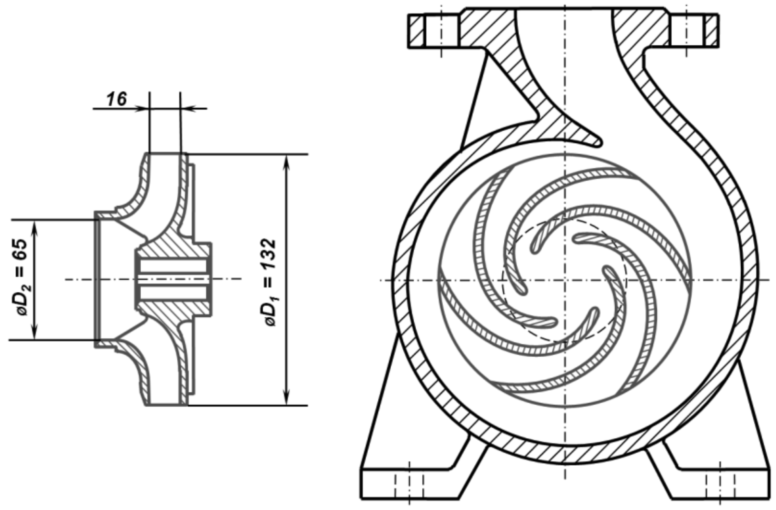

Fig. 2 Diagram of experimental radial centrifugal pump

On the pump above, the characteristics were measured on the hydraulic test circuit in pump and subsequently in turbine modes, on the original, untreated machine. The impeller was then dismantled and adjusted as shown in Fig. 1. The blade edges and both discs on the outer diameter, D_1 were rounded. The adjustment is evident from Fig. 3. In addition, a polymeric coating with a hydrophobic surface was applied to the inner flow parts of the impeller. Its purpose was to reduce the roughness of these surfaces which are wetted by the flowing fluid.

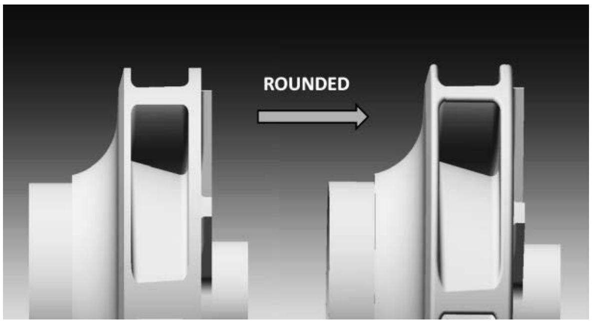

Fig. 3 Detail of the impeller rounding

After the assembly of the modified impeller, the effect of the modifications on operation in turbine mode was verified on the hydraulic circuit. The diagram of the hydraulic test circuit is shown in Fig. 4.

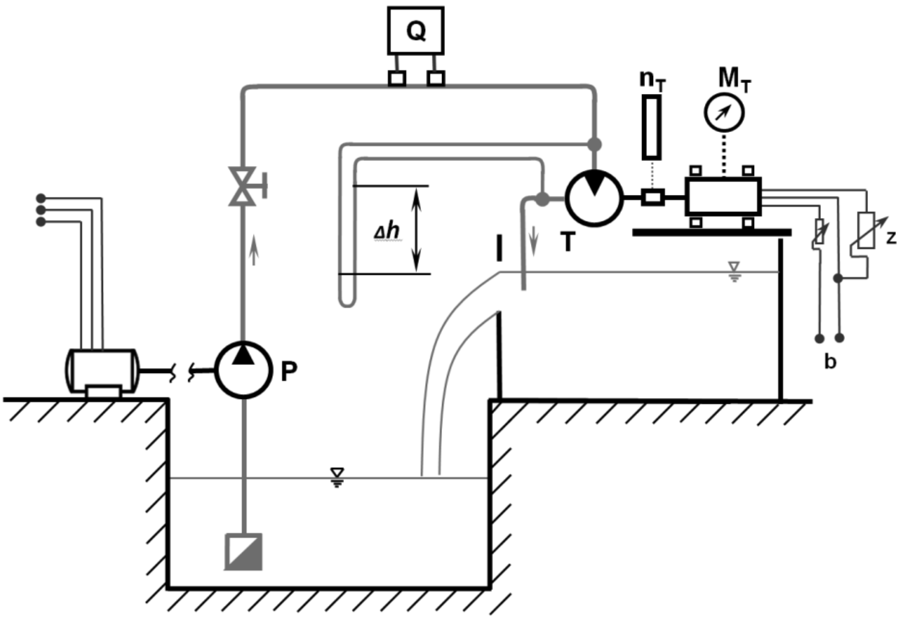

Fig. 4 Hydraulic circuit for testing pump as turbine (PAT)

The test circuit consists of two tanks and a pump (P) which produces hydrotechnical potential for the turbine (T) to be tested. The turbine is connected to a dynamometer (M_T) with continuously adjustable load. The mechanical output of the turbine is given by:

$$P_T = M_T \frac{\pi \cdot n_T}{30} \qquad [W] \tag{3}$$

where, M_T is the torque on the turbine shaft (N·m) and $n_{T,}$ the turbine speed (min^{-1}). The hydrotechnical potential (or hydraulic output delivered to the turbine) is determined by the flow Q and the differential pressure of the mercury manometer, Δh (m):

$$P_w = Q \cdot \rho_w \cdot \left(\frac{\rho_{Hg}}{\rho_w} \cdot \Delta h \cdot g + \frac{v_i^{\,2} - v_o^{\,2}}{g} + g \cdot y \right) \qquad [W] \tag{4}$$

where, ρ_{Hg} is the specific weight of mercury (kg·m^{-3}), ρ_w, the specific weight of water (kg·m^{-3}), v_i, v_o, the velocity of water in the inlet and outlet pipes, respectively (m·s^{-1}) and y, the vertical distance of zones generating pressure (m). The element in brackets of equation (4) expresses the specific energy of the turbine Y_T (J·kg^{-1}) (*ČSN EN ISO 9906, 2013*).
The overall efficiency of the turbine is:

$$\eta_T = \frac{P_T}{P_w} \qquad [-] \tag{5}$$

RESULTS AND DISCUSSION

The results of the experimental verification of pump optimization in turbine operation are in the form of the characteristics shown in Fig 5. The course of the main output quantities is shown here, i.e. the mechanical power P_T, the torque M_T, and the efficiency, η_P, in relation to the unit speeds n_{11}. Unit speeds of hydraulic machines are defined by:

$$n_{11} = \frac{n_T \cdot D_1}{\sqrt{Y_T}} \qquad [\min^{-1}] \tag{6}$$

Furthermore, the characteristic features the course of the gradients and unit flow rates, Q_{11} in relation to the unit speeds, n_{11}. Unit flow rate is defined as:

$$Q_{11} = \frac{Q}{D_1 \cdot \sqrt{Y_T}} \qquad [l \cdot s^{-1}] \tag{7}$$

Black curves correspond to the characteristics of the original untreated pump. Grey curves are the characteristics of the machine modified by the method described above.

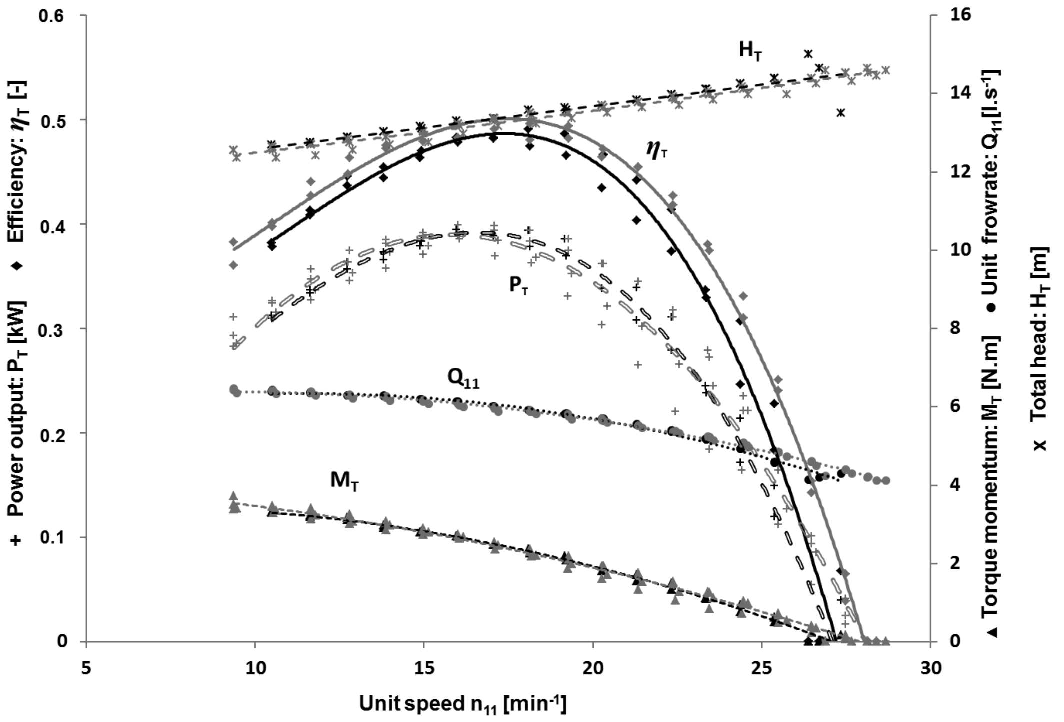

Fig. 5 Characteristics of turbine operation of radial centrifugal pump

Values corresponding to optimal operation at the highest efficiency of the machine (or Best Efficiency Point, BEP) are quantitatively summarized in Tab 1.

Tab. 1 Parameters of optimum pump operation and PAT

BEP		Pump	PAT original	PAT improved
Shaft speed	N [min^{-1}]	1450	1500	1500
Flowrate	Q [$l.s^{-1}$]	3.8	6.0	5.9
Total head	H [m]	5.8	13.4	13.3
Power input/output	P [W]	310	388	394
Total efficiency	η [%]	0.70	0.48	0.50

From the measured characteristics, there is another finding concerning the conversion relations (1) and (2). Recalculation assumes that the machine has the same speed in both pump and turbine modes. Only the direction of rotation of the shaft is the opposite. Maintaining the speed has its justification resulting from the requirement to use the set for electricity production. In this case, it is necessary to respect the frequency of the supply network and hence the generator, or the turbine, speed. However, the use of relations (1) and (2) also assumes unchanging efficiency in both pump modes. This condition may not apply in practice though, especially with smaller pumps, as it has been demonstrated experimentally in this case.

CONCLUSIONS

On the basis of the measured results it can be stated that the radial one-stage pump in turbine operation shows a 22% reduction in overall efficiency compared to the pump operation. Subsequent adjustment of the impeller by rounding the input edges on the outer diameter of the impeller, along with a reduc-

tion in roughness of the inter-blades flow channels resulted in an efficiency gain of 2%. At the same time, flow decreased by 1.7% and the gradient by 0.75%. Mechanical performance improved due to the modification by 1.5%. Overall, such a simple adjustment brings only a partial increase in performance parameters. Its benefits need to be seen as a further addition to the innovative adaptations of the connecting parts of the pump, such as the spiral case.
The measurement results further showed the necessity of correction of the conversion relationships when designing a pump for turbine mode. Based on the quantified parameters of the test pump, the exponents of equations (1) and (2) correspond to the values $x = 2.3$ and $y = 1.3$. It can be stated that the values of the exponents increased in a similar pattern as the efficiency of the pump in the reverse turbine operation decreased. This fact, including the change in the efficiency of the reverse operation of the machine is yet to be described in literature and it is necessary to examine it in further research.

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