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Application of conversion model for designing hydrodynamic pumps in turbine mode

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Abstract. The use of the smallest water resources has been coming again to the centre of interest in recent years. A water engine – turbine, is the heart of these power plants. This is usually the highest expense for the investor, in terms of cost. The effort is therefore to seek investment less demanding alternatives. One of them is the use of hydrodynamic pumps in reverse turbine operation. This paper provides a methodology for conversion of parameters of the smallest power pumps (micro hydro sites) to turbine operation. The conversion model is based on the results of experimental research at the author's workplace and is suitable for pumps with low specific speeds and outputs. The pump design process for turbine mode is complemented by a practical example for a specific deployment site. This example also serves to verify the accuracy of the conversion model.

Key words: pump as turbine (PAT), conversion model, specific speed, efficiency.

INTRODUCTION

The design of a hydrodynamic pump as turbine (PAT) is based on good knowledge of flow ratios in the reverse operation of the machine. However, compared to a conventional turbine, there are some limitations such as the impossibility of control by means of adjustable guide vanes. But these limitations are counterbalanced by the main advantage of these applications, and that is the price of a machine, that is often much lower than that of a conventional turbine. When designing a PAT, it is necessary to take into account all the specific characteristics and choose a pump from commercial manufacturer's offer that meets all the requirements of the investor. The knowledge of pumps conversion relations for turbine mode is a prerequisite for correct design. A whole range of conversion models was created according to various authors in connection with this issue, for example (Stepanoff, 1957; Childs, 1962; Hancock, 1963; Grover, 1980; Sharma, 1985; Lewinsky, 1987; Schmiedl, 1988; Alatorre-Frenk, 1994; Williams, 1994; Derakhshan & Nourbakhsh, 2008; Güllich, 2008; Nautiyal et al., 2010; Carravetta et al., 2017; Frosina et al., 2017 and Naeimi et al., 2017). These authors, however, proceeded mainly from the experience with large power plants. But small machines have specifics during operation, which complicate the simple transfer of conversion relations for this category or the results achieved by them may differ considerably from reality. One reason is that the efficiency of a pump converted to turbine mode may change. Another reason is the assumption of constant conversion relations across a wide range of specific speeds and pump powers. The author of the article presents his own conversion model here, based on the results of experimental verification of small pumps turbine operation.

MATERIALS AND METHODS

Determining the conversion relations

The starting point for determining the conversion relations was the assumption that the efficiency of the pump changes in the same relation as the specific speed does when changing to a turbine mode. Among others, this results from the requirement to maintain the same circumferential speed of the impeller, or the same speed in both machine modes. This is due to the use of synchronous motors, which can be operated in a generator mode, and the frequency of the electric power grid needs to be maintained for the electricity produced (Bláha et al., 2011). Specific speed by the flowrate is defined by a generally known relation (Melichar et al., 1998; Munson et al., 2006):

$$N_q = N \cdot \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} \tag{1}$$

where N is shaft speed, Q pump or turbine flowrate and H pump or turbine head. Based on an analysis of the results of the experimental verification, the conversion relations were expressed. The turbine flowrate Q_T and the head H_T are determined from the known hydrotechnical potential of the deployment site. The pump flow rate Q_P is determined from these values using the conversion relation:

$$Q_P = Q_T \cdot {\eta_P}^x \tag{2}$$

and respective pump head H_P:

$$H_P = H_T \cdot {\eta_P}^y \tag{3}$$

where x and y are constants. Their values are in range of $x = 1 \div 1.6$ and $y = 2 \div 2.6$ and they depend on specific speed and size of the pump. The values H_P and Q_P are then used to select a specific pump from the manufacturer's catalogue using the *H-Q* characteristics (Polák, 2018).

Example of use of conversion model

Use of the conversion model mentioned above is demonstrated on an example of a particular installation. The principle of the PAT design is to use the multiple approximation method several times until the results of the next step match the results of the previous step.

Example specification: Select a suitable pump from the manufacturer's catalogue for use in turbine mode at a location where the hydrotechnical potential given by flow rate $Q_T = 6.0$ L s^{-1} and net gradient $H_T = 13.5$ m is available.

In the first step of the conversion, it is necessary to determine the specific speed in turbine mode according to Eq. (1). This is based on the requirement for the same shaft speed, i.e. the expected turbine shaft speed will be $N_P = N_T = 1{,}450$ rpm.

$$N_{qT} = N_T \cdot \frac{Q_T^{\frac{1}{2}}}{H_T^{\frac{3}{4}}} = 1{,}450 \cdot \frac{0.006^{\frac{1}{2}}}{13.5^{\frac{3}{4}}} = 16\, rpm \quad (4)$$

The calculated specific speed value is used to predict the efficiency of the pump using the Erhart's diagram in Fig. 1. The diagram represents achievable efficiency of pumps in relation to specific speed (horizontal axis) and flow (vertical axis), i.e pump size. Thick dashed lines represent the values of the example and their intersection indicates the expected efficiency of the pump; here η_{P1} = 56%.

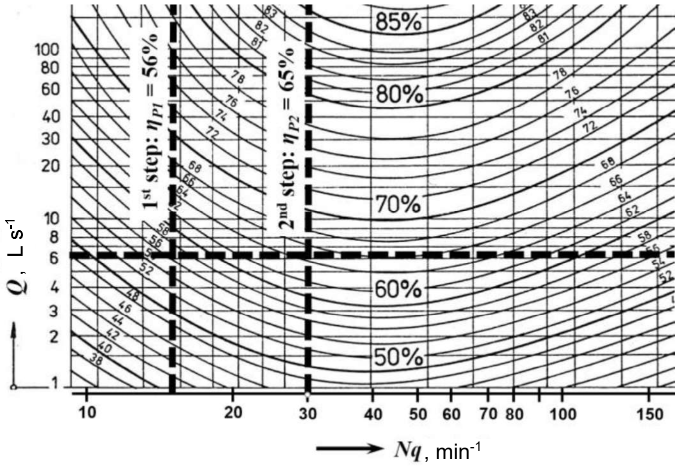

Figure 1. Erhart's diagram (Melichar et al., 1998).

The next step is to calculate the flowrate and pump head according to Eqs (2) and (3) where x and y are constants. Their values are in range of $x = 1 \div 1.6$ and $y = 2 \div 2.6$. For the calculation, their mean value from the interval will be used, i.e. $x = 1.3$ and $y = 2.3$. The pump flowrate will be according to Eq. (2):

$$Q_{P1} = Q_T \cdot \eta_{P1}{}^{x} = 6 \cdot 0.56^{1.3} = 2.82\ L\,s^{-1} \quad (5)$$

The pump head will be according to Eq. (3):

$$H_{P1} = H_T \cdot \eta_{P1}{}^{y} = 13.5 \cdot 0.56^{2.3} = 3.56\ m \quad (6)$$

Calculation of the pump speed with the values Q_{P1} and H_{T1} will follow:

$$N_{qP1} = N_T \cdot \frac{Q_{P1}^{\frac{1}{2}}}{H_{P1}^{\frac{3}{4}}} = 1{,}450 \cdot \frac{0.00282^{\frac{1}{2}}}{3.56^{\frac{3}{4}}} = 30\, rpm \quad (7)$$

Following is the second step of the multiple approximation method, when according to the Erhart's diagram the corrected expected pump efficiency is determined using the N_{qP1} value from Eq (7): η_{P2} = 65%. After substituting into Eqs (2) and (3), the corrected pump flowrate Q_{P2} is given:

$$Q_{P2} = Q_{P1} \cdot \eta_{P2}{}^{x} = 6 \cdot 0.65^{1.3} = 3.43\ L\,s^{-1} \quad (8)$$

and pump head H_{P2} is given:

$$H_{P2} = H_{P1} \cdot \eta_{P2}{}^{y} = 3.36 \cdot 0.65^{2.3} = 5.01\ m \qquad (9)$$

By substituting Q_{P2} and H_{P2} values into Eq. (1), newly corrected value of specific speed N_{qP2} = 25 rpm is obtained. The results of the flowrate or the head calculations from Eqs (5) and (6) or (8) and (9) do not match, which means further correction by repeating the calculations in the next step. However, the corresponding values of Q_{P1} and Q_{P2} or H_{P1} and H_{P2} are already close. This means that these values are already close to real values, so there is more detailed idea of the expected pump parameters. It can be therefore used to select a pump from the manufacturer's catalogue. Assuming $N_P = N_T$ = 1,450 rpm, a pump of the appropriate size can be selected using Q_{P2} = 3.43 L s^{-1} and H_{P2} = 5.01 m – see Fig. 2. This represents a section of *H-Q* field diagram for radial centrifugal pumps from the manufacturer's catalogue where the intersection of dashed lines in the dark grey field indicates the specific pump type and size.

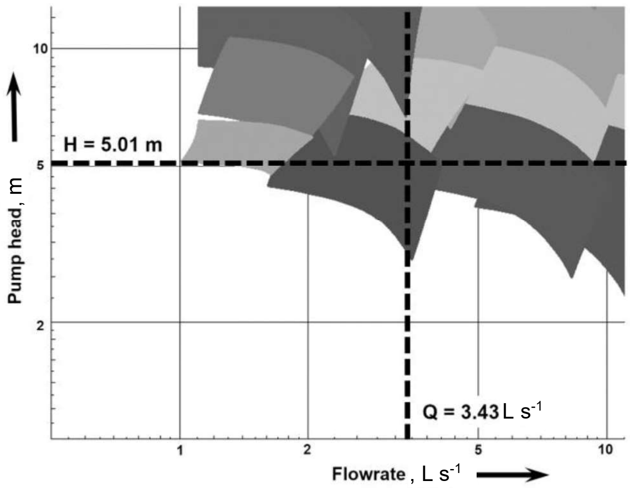

Figure 2. *H-Q* field diagram from the pumps manufacturer's catalogue (ISH Pumps, 2014).

For thus determined pump, the performance characteristics and efficiency can be found in the catalogue. In this case, the efficiency of the pump is η_{P3} = 70%. After substituting this value into Eqs (2) and (3), it can be proceed with the third step in the calculation from which the results are obtained: Q_{P3} = 3.77 L s^{-1} and H_{P3} = 5.94 m. These will be used for calculating specific speed: N_{qP3} = 23 rpm. In the fourth step, the flowrate and head obtained according to Eqs (2) and (3) are Q_{P4} = 3.77 L s^{-1} and H_{P4} = 5.94 m. The results of the fourth step are identical to the results of the third step, so the calculation can be terminated.

Based on the results of the conversion model for the above-mentioned hydrotechnical potential Q_T = 6.0 L s^{-1} and H_T = 13.5 m, the most suitable is a pump with optimum operating parameters: total head H_{P4} = 5.94 m, flowrate Q_{P4} = 3.77 L s^{-1} and shaft speed N_P = 1,450 rpm.

In addition to the above-mentioned basic proposal, an estimation of expected efficiency in the turbine mode can be made:

$$\eta_T = \eta_{P3}\left(1 - \frac{N_{qP3} - N_{qT}}{N_{qP3}}\right) = 0.7\left(1 - \frac{23 - 16}{16}\right) = 0.49 \qquad (10)$$

RESULTS AND DISCUSSION

The validity of the conversion model was verified by comparing the calculated results of the performance parameters with experimentally obtained values measured on the same pump. Verification tests were conducted according to (ČSN EN ISO 9906) on a hydraulic circuit in the Fluid Mechanics laboratory at the Faculty of Engineering, Czech University of Life Sciences Prague (Polák, 2017). Table 1 summarizes the results of the conversion model, the measured performance parameters and their relative difference determined according to:

$$\text{Difference} = 100 \cdot \frac{Model - Reality}{Reality}\ \% \qquad (11)$$

Table 1. Comparison of conversion model with reality

Impeller diameter: D_1, mm	Model	Reality	Difference
Total head: H_T, m	5.94	5.7	+ 4%
Flowrate: Q_T, L s^{-1}	3.77	3.8	- 1%
Shaft speed: N_T, rpm	1450	1450	0
Efficiency: η_T, %	49	48	+ 2%
Specific speed: N_{qT}, rpm	23	24	- 4%

Comparing the results of the conversion model with reality showed very close matching (differences up to 5%) and therefore the author's methodology can be recommended for wider use. Proposed conversion model is suitable for small pumps with a power output of up to 2 kW and a specific speeds up to 40 rpm for which it was primarily designed. The question of determining exact value of conversion factors *x* and *y* for different speeds and outputs remains open for further research.

CONCLUSIONS

Various authors addressed the issue of pump performance parameters conversion for turbine mode in different ways. But their solutions are mostly based on experience with large energy units. However, a simple transfer of these conversions to small devices does not give satisfactory results. The author of the article presents his own methodology which is based on the results of the experimental verification of turbine operation of small pumps. The conversion is performed by multiple approximation method until the results of the consecutive steps match. Besides the conversion of the basic power parameters, an estimate of the expected pump efficiency in turbine mode can also be provided. The method is completed by an illustrative example, which also serves to compare its results with reality. This comparison proved a very good match with reality. The proposed procedure can therefore be recommended for wider use in the area of small pumps with power output of up to 2 kW and specific speed up to 40 rpm for which it was primarily designed.

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