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# Determination of conversion relations for the use of small hydrodynamic pumps in reverse turbine operation

M. Polák

Czech University of Life Sciences in Prague, Faculty of Engineering, Kamýcká 129, CZ16521, Praha 6, Czech Republic
Correspondence: karel@tf.czu.cz

**Abstract.** In small-scale hydropower, hydrodynamic pumps used in reverse mode are an important economical alternative to conventional water turbines. Efficient utilisation of these devices however requires taking into account all the specifics of the reverse pump operation and optimise the pump design for maximum utilisation of the hydro-technical potential of the deployment site. The article compares existing conversion models, describes initial theoretical assumptions and determines new conversion relations for the design of pumps as turbines (PAT) for the lowest power and specific speed category. The validity of the conversion relations is experimentally verified on a hydraulic test circuit with a radial centrifugal pump tested in both pump and turbine operation modes. The results of the verification of the new conversion relations proved better correspondence to reality within this category of machines than that reported by the previously used conversion models.

**Key words:** pump as turbine (PAT), conversion relations, specific speed, head, flowrate.

## INTRODUCTION

Standard centrifugal pumps may be operated in reverse mode as water turbines. They are often cheaper than specifically designed turbines, especially in case of devices of less than 100 kW (micro-hydro). The first pump turbine had been set at a remote farm in the Yorkshire Dales of the North England in 1930. This scheme has been working for a five year testing time, after which its reliability was confirmed before being transferred to other countries. (Williams, 1994) Especially in developing countries, small and micro hydropower plants are very effective source for electricity generation with energy pay-back time less than other conventional electricity generation systems. Using pump as turbine (PAT) is an attractive, significant and cost-effective alternative. Pump manufacturers do not normally provide the characteristic curves of their pumps working as turbines. (Naeimi, et al., 2017) However, in order to use a pump in a micro-hydro scheme, the turbine performance must be found either by testing or by calculation. Several methods have been suggested for predicting the turbine performance based on the data for pump performance at best efficiency, but they produce a wide range of results. (Williams, 1994; Bláha et al., 2011) In this paper, nine such methods are compared using an analysis of the effects of poor turbine prediction on the operation of a pump as turbine at a typical micro hydro site. None of these methods gives an accurate prediction, especially for the smallest pumps. This was the cause for searching such conversion relations that would better reflect reality, especially for machines with the lowest power and specific speed. Data from experimental measurements of the performance characteristics of ten radial centrifugal pumps were used for the purpose of determining the conversion relations. Three of them were verified at the author's workplace (No. 1, 2 a 3 – see Table 1 a Table 2), others were taken from (Singh, 2005; Singh, 2011).

## MATERIALS AND METHODS

### Determination of pump performance parameters in turbine operation

The methodology for the conversion of the parameters was verified on radial single stage centrifugal pumps in the range of power $P = 0.1 \div 9$ kW and specific speed $N_{qT} = 15 \div 70$ rpm. A typical representative in this category is a META series pump, manufactured in the Czech Republic by ISH PUMPS Olomouc, a.s. which was verified at the author's workplace. The pump's scheme, including its parameters as provided by the manufacturer is shown in Fig. 1.

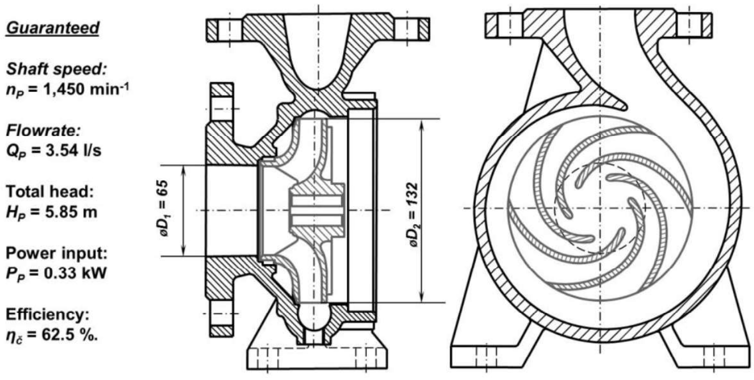


**Figure 1.** Centrifugal pump for experimental tests and original pump parameters (Source: Polák, 2017).

Verification tests were conducted on a hydraulic circuit in the Fluid Mechanics laboratory at the Faculty of Engineering, Czech University of Life Sciences Prague. The circuit diagram is shown in Fig. 2. Characteristics of the pump and its modifications in the turbine mode were measured in the first stage. Subsequently the effect of the modifications in the pump operation was verified.

The testing circuit consisted of a set of two reservoirs with pipes and control and measuring elements. With this setting the tested pump (T-P) was measured in turbine mode – by closing valve V2 while regulating valve V1, the water flows in the direction of dashed arrows, while the feeding pump (FP) creates the hydro-technical potential for the turbine. After rearranging the valves V1 and V2, the machine (T-P) was tested in the pumping operation on the same circuit – by closing the valve V1 and controlling valve V2 the water flows in the direction of grey arrows. The dynamometer (M) with continuous revolutions control allows operation in motor and braking mode. The dynamometer has cylindrical stator. It is placed on a bed allowing a slight rotation, which enables a measurement of the reaction torque at load. Dynamometer is a DC machine connected to continuous resistance load control unit. The water flow was measured using an ultrasonic flowmeter Siemens SITRANS FUP1010. The turbine shaft speed was measured using an infrared sensor TESTO 465 (Polák, 2017, ČSN EN ISO 9906).

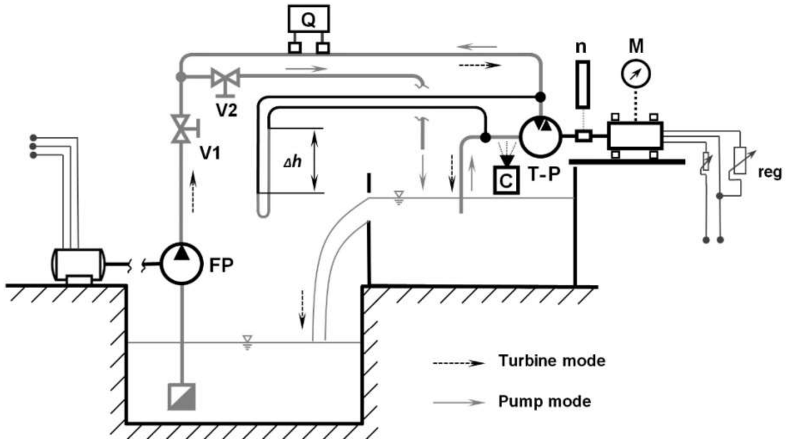


**Figure 2.** Hydraulic circuit scheme for testing turbines/pumps. Q – flowmeter, FP – feed pump (for turbine mode), T–P tested pump/turbine, V1, V2 – control valves, M – dynamometer, n – revolution counter, reg – load/drive control of the turbine/pump (Source: Polák, 2017).

The characteristics of the pump and consequently the turbine operation were experimentally determined on the test circuit for all ten monitored pumps. Then the parameters corresponding to optimum operation at best efficiency point (BEP) were determined from the characteristics, i.e. flow, total head, efficiency and shaft speed. An overview of the performance parameters of each machine in pump mode is given in Table 1. An overview of the performance parameters of the same machines in turbine mode is in Table 2.

**Table 1.** Overview of performance parameters in pump mode (Source: author; Singh, 2005)

| **Pump No.**<br>Impeller diameter: $D_1$, mm | **1**<br>96 | **2**<br>132 | **3**<br>132 | **4**<br>258 | **5**<br>206 | **6**<br>174 | **7**<br>264 | **8**<br>200 | **9**<br>139 | **10**<br>224 |
|---|---|---|---|---|---|---|---|---|---|---|
| Total head: $H_P$, m | 2.27 | 5.7 | 6.5 | 21.5 | 15.8 | 8.38 | 19.8 | 10.5 | 5.6 | 10.6 |
| Flowrate: $Q_P$, L s$^{-1}$ | 0.76 | 3.8 | 3.8 | 26.5 | 25.4 | 15.3 | 65.9 | 33 | 13.5 | 103 |
| Shaft speed: $N_P$, rpm | 1,450 | 1,450 | 1,450 | 1,500 | 1,500 | 1,450 | 1,450 | 1,450 | 1,450 | 1,450 |
| Efficiency: $\eta_P$, % | 0.44 | 0.7 | 0.73 | 0.78 | 0.79 | 0.74 | 0.85 | 0.80 | 0.76 | 0.84 |
| Specific speed: $N_{qP}$, rpm | 22 | 24 | 22 | 25 | 35 | 36 | 40 | 45 | 46 | 79 |

**Table 2.** Overview of performance parameters in turbine mode (Source: author; Singh, 2005)

| **Pump No.**<br>Impeller diameter: $D_1$, mm | **1**<br>96 | **2**<br>132 | **3**<br>132 | **4**<br>258 | **5**<br>206 | **6**<br>174 | **7**<br>264 | **8**<br>200 | **9**<br>139 | **10**<br>224 |
|---|---|---|---|---|---|---|---|---|---|---|
| Total head: $H_T$, m | 14.6 | 13.5 | 13.8 | 17.1 | 15.5 | 13.4 | 16 | 13.7 | 10.6 | 10 |
| Flowrate: $Q_T$, L s$^{-1}$ | 3.0 | 6.0 | 5.5 | 30.1 | 28.6 | 22.7 | 67.5 | 43.9 | 19.7 | 108 |
| Shaft speed: $N_T$, rpm | 1,450 | 1,450 | 1,450 | 900 | 1,300 | 1,400 | 1,100 | 1,400 | 1,600 | 1,200 |
| Efficiency: $\eta_T$, % | 0.2 | 0.48 | 0.55 | 0.77 | 0.81 | 0.72 | 0.84 | 0.8 | 0.76 | 0.76 |
| Specific speed: $N_{qT}$, rpm | 15 | 16 | 15 | 19 | 28 | 30 | 36 | 41 | 38 | 70 |
| Power output: $P_T$, W | 86 | 381 | 410 | 3,863 | 3,523 | 2,134 | 8,847 | 4.691 | 1,557 | 7,992 |

**Verification of the conformity of existing conversion relations**

On the basis of the experimentally obtained data, the validity of the existing conversion relations was verified according to various authors (Frosina et al., 2017; Derakhshan & Nourbakhsh, 2008; Nautiyal et al., 2010). An overview of existing conversion relations is in Table 3.

**Table 3.** Conversion relations according to various authors

| Author | Source, Year | Head ratio $H_T/H_P$ | Flowrate ratio $Q_T/Q_P$ | Remarks |
|---|---|---|---|---|
| Stepanoff | (Stepanoff, 1957) | $\frac{1}{\eta_P}$ | $\frac{1}{\sqrt{\eta_P}}$ | Accurate for Ns = 40÷60 |
| Childs | (Childs, 1962) | $\frac{1}{\eta_P}$ | $\frac{1}{\eta_P}$ | - |
| Hancock | (Hancock, 1963) | $\frac{1}{\eta_T}$ | $\frac{1}{\eta_T}$ | - |
| Grover | (Grover, 1980) | $2.693 - 0.0229N_{sT}$ | $2.379 - 0.0264N_{sT}$ | Applied for Ns = 10÷50 |
| Hergt | (Lewinsky, 1987) | $1.3 - \frac{6}{N_{qT} - 3}$ | $1.3 - \frac{1.6}{N_{qT} - 5}$ | - |
| Sharma | (Sharma, 1985) | $\frac{1}{\eta_P{}^{1.2}}$ | $\frac{1}{\eta_P{}^{0.8}}$ | Accurate for Ns = 40÷60 |
| Schmiedl | (Schmiedl, 1988) | $-1.4 + \frac{2.5}{\eta_P}$ | $-1.5 + \frac{2.4}{\eta_P{}^2}$ | - |
| Alatorre-Frenk | (Alatorre-Frenk, 1994) | $\frac{1}{0.85\eta_P{}^5 + 0.385}$ | $\frac{0.85\eta_P{}^5 + 0.385}{2\eta_P{}^{9,5} + 0.205}$ | - |
| Güllich-volute | (Güllich, 2008) | $\frac{2.4}{\eta_P{}^2} - 1.5$ | $\frac{2.5}{\eta_P} - 1.4$ | - |

The parameters of the pump mode from Table 1 were used in order to verify the validity of the above mentioned conversion relations. Conversion results, i.e. flows and gradients for turbine mode, were then compared with the real values recorded in the laboratory (Table 1). Fig. 3 and Fig. 4 shows an example of a comparison of the results for two pumps of the same specific speeds, but different sizes and power. The figures show differences between conversion results and reality for pump No. 3 ($D_I$ = 132 mm, $N_{qT}$ = 22 rpm, $P_T$ = 410 W) and pump No. 4 ($D_I$ = 258 mm, $N_{qT}$ = 25 rpm, $P_T$ = 3.9 kW). Results of the calculations according to all nine authors listed in Table 3 are lined up on the horizontal axis. Fig. 3 show the relative differences of the turbine head $\Delta H_T$ calculated according to:

$$\Delta H_T = 100 \cdot \frac{H_{T*} - H_T}{H_T} \; [\%] \qquad (1)$$

where $H_T$ is the real turbine head measured on the test circuit, and $H_{T*}$ is the turbine head calculated according to the respective author.

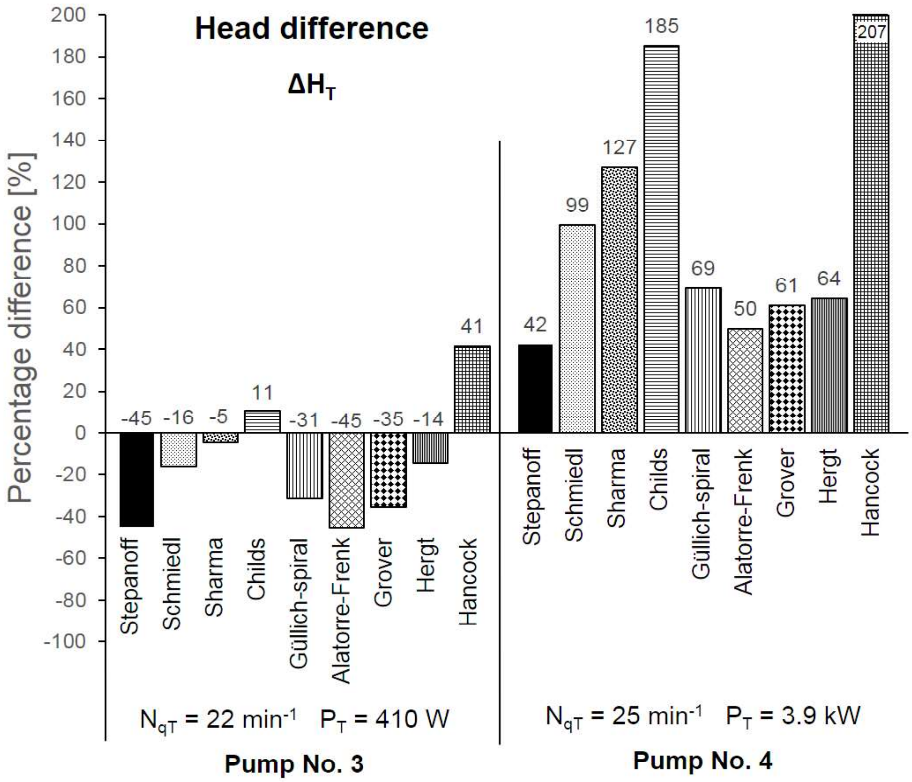


**Figure 3.** Comparison of turbine head calculation results according to various authors.

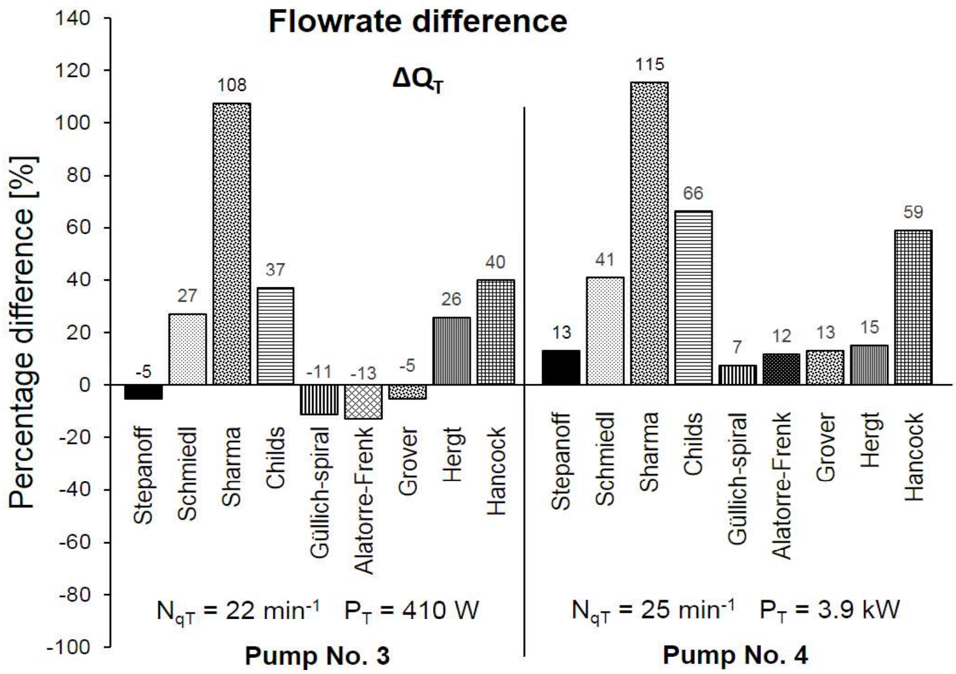


**Figure 4.** Comparison of turbine flowrate calculation results according to various authors.

Fig. 4 show the relative differences of the turbine flowrate $\Delta Q_T$ according to:

$$\Delta Q_T = 100 \cdot \frac{Q_{T*} - Q_T}{Q_T} \; [\%] \qquad (2)$$

where $Q_T$ is the real turbine flowrate recorded on the test circuit and $Q_{T*}$ is the value of the turbine flowrate calculated according to the respective author.

It is apparent from comparison of the graphs in Fig. 3 and Fig. 4 that the best results (the smallest differences) are presented by the conversion model according to Stepanoff. The following Fig. 5 shows the comparison of the differences of the conversion for all tested pumps only according to Stepanoff. On the x-axis, the pumps are arranged according to the revolutions $N_{qT}$ (horizontal x-axis numbers) and power $P$ (vertical x-axis numbers). On the y-axis, the relative values of the deviation of the real values from the calculated flowrate $\Delta Q_T$, as well as from the $\Delta H_T$) are presented (horizontal numbers above the x-axis.

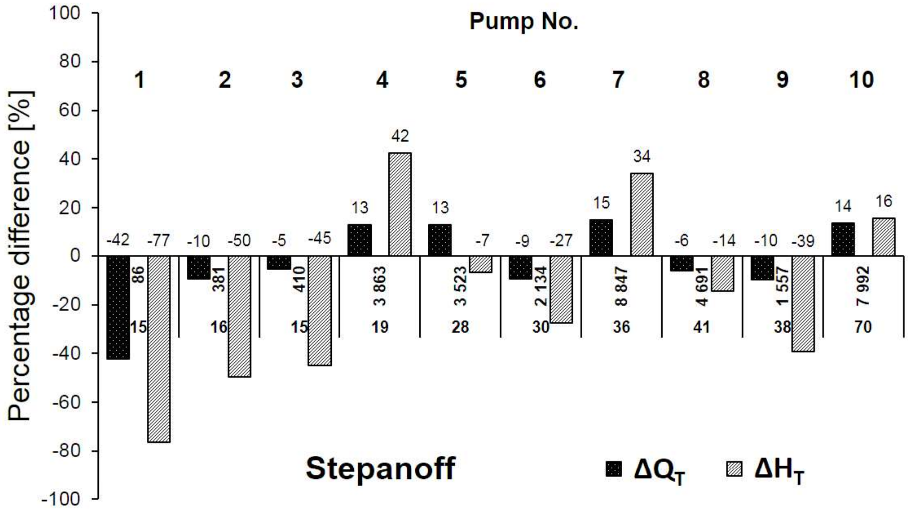


**Figure 5.** Comparison of conversion calculations according to Stepanoff.

**Determination of the conversion relations for the lowest power category**

The conversion of parameters according to Stepanoff is based solely on the efficiency of the pump. It assumes that the efficiency is the same both in the pump and turbine mode. However, this does not apply generally. The efficiency is lower in the turbine mode particularly for machines with the lowest specific speed and power. In these cases, it is necessary to take into account the reduction in efficiency and include it in the conversions. Furthermore, it shows that the coefficients in the conversion relations for small pumps cannot be considered as constant. This implies the necessity of coefficients variability along with the definition of the area of their use. These circumstances led to the search for such conversion relations that would meet these assumptions.

It is apparent from the comparison in Fig. 5 that the greatest difference between the calculated values and the reality is for the five pumps with the lowest power where the differences exceeded 40% (especially in the turbine head). It concerned specifically pumps No. 1, 2, 3, 6 and 9. These pumps were focused on with the aim to find such conversion relations, which would better reflect the reality.

The test results from the laboratory tests were used to determine more appropriate conversion relations. A prerequisite for their derivation was the hypothesis that the efficiency of small pumps in the turbine mode changes in a similar relation as the specific speed does. This relation can be expressed by equation (3):

$$\Delta\eta = \Delta N_q \tag{3}$$

where:

$$\Delta\eta = \frac{\eta_P - \eta_T}{\eta_P} \tag{4}$$

and

$$\Delta N_q = \frac{N_{qP} - N_{qT}}{N_{qP}} \tag{5}$$

where $\eta_P$, $\eta_T$ is pump or turbine efficiency, and $N_{qP}$, $N_{qT}$ is specific speed by the flowrate in pump or turbine mode. Specific speed is defined by a generally known relation (Munson 2006; Melichar et at. 1998):

$$N_q = N \cdot \frac{Q^{\frac{1}{2}}}{H^{\frac{3}{4}}} \tag{6}$$

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 $H_T$ is 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 (7):

$$Q_P = Q_T \cdot \eta_P{}^x \tag{7}$$

and respective pump head $H_P$:

$$H_P = H_T \cdot \eta_P{}^y \tag{8}$$

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. Conversion rates for turbine mode depending on the original pump parameters can be expressed from formulas (7) and (8) as $Q_T = Q_p/\eta_p{}^x$ and $H_T = H_p/\eta_p{}^x$.

## RESULTS AND DISCUSSION

An overview of the results comparison of the conversion model according to equations (3) to (8) proposed by the author of the article is given in Fig. 6. The calculation differences of the five smallest pumps No. 1, 2, 3, 6 and 9 are presented here, for which the calculation was designed. The results showed that, compared to Stepanoff's model (difference above 40%), the authors' conversion model achieved significantly better conformity with the reality (differences up to 5% or 15%). Higher difference occurred only at turbine head calculation for pump No. 6. The probable cause of this is generally higher sensitivity of the turbine head calculation to the equation constants used. It is also similar for Stepanoff's model (see $\Delta H_T$ on Fig. 5). Another reason can be also exceeding the interval defining the optimum use of the conversion relations. This implies the need for correction of the conversion model, which is the subject of further research.

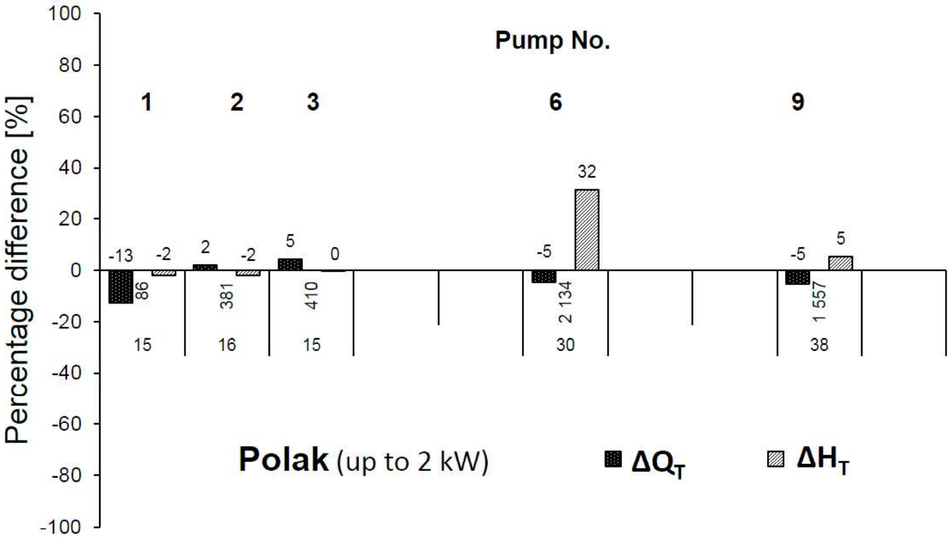


**Figure 6.** Comparison of conversion calculations according to author

## CONCLUSIONS

The conversion relations for the radial centrifugal pump parameters for turbine mode were verified in this study. The results of experimental verification of ten radial single stage centrifugal pumps in the range of specific speed $N_{qT} = 15 \div 70$ rpm and power $P = 0.1 \div 9$ kW were used for this purpose. By comparing the conversion results according to nine different authors, it was found that Stepanoff's model was the most conforming. However, it failed for pumps with the smallest power and specific speed, where deviations exceeded 40%. The probable cause is the changing efficiency of the pump in turbine mode, which is not considered by Stepanoff's model. The author of this study took into account changing efficiency and included it in his own conversion relations. With these new relations, the results for pumps with the smallest power differ from reality by less than 5 or 15%. The conversion was subsequently verified on the remaining five larger pumps. It appeared that the difference between the calculation results and the reality increased with increasing power and specific speed. Therefore, the author's proposed conversion model is suitable for 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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