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Study of characteristic curves of rotodynamic pumps working as turbines

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Abstract

Centrifugal pumps working in reverse mode can be a good alternative for conventional water turbines. Prediction of the pumps as turbine characteristic curve is still complicated due to the lack of information provided by the producers and rare research focusing on this subject. In some cases of pumping systems it is important to estimate the operating point in turbine mode basing on pump test data. In this paper the pump characteristics in turbine mode for ten pumps with a specific speed between 10–261 were compared and analyzed. The formulas used to predict the best performance point and characteristics of pump as turbine was reviewed. Own formulas for calculating optimum points of pump in the turbine mode was proposed.

Keywords: Pump as turbine (PAT); Rotodynamic pump; Hydraulic power; Recovery turbine

Nomenclature

BEP	–	best efficiency point
D	–	impeller diameter, m
g	–	gravitational acceleration, m/s^2
H	–	head, m
N_s	–	pump specific speed
n	–	rotational speed, rpm
n_{gt}	–	turbine specific speed
Q	–	flow rate, m^3/h

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Greek symbols

α_p	–	pump dimensionless specific speed
β_t	–	turbine dimensionless specific speed
γ, α_t, β_p	–	dimensionless parameters from [2] equation 8,9,10 respectively
η	–	hydraulic efficiency
$\varkappa$	–	relationship between pump efficiency at BEP point and pump specific speed acc. [3]
φ	–	dimensionless head
ψ	–	dimensionless flow

Subscripts

h	–	hydraulic efficiency
p	–	refers to pump
t	–	refers to pump in turbine mode

1 Intruoduction

The idea of using pumps as turbines is very attractive and can be a good alternative to standard solutions applying water turbines. Rotodynamic pumps have relatively simple construction what provides long failure-free operation and accessible maintenance. They are readily available due to the multitude of manufacturers and pumps suppliers. Economically it has been found that a pump operating as a turbine in the range of 1 to 500 kW allows for the return on investment costs over two years or less, which is considerably shorter than in a conventional turbine [1].

Choosing the proper pump as a turbine (PAT) requires knowledge of the pump characteristics working in turbine (reverse) mode. It is a demanding task because these characteristic curves are not usually supplied with the pump or did not made available by the producers. In addition the prediction methods of reverse mode are often focusing on the best efficiency point (BEP) and gives many different results. Manufacturers of such equipment often do not have turbine test benches focusing on determining the typical flow characteristics of the pump. Many researchers, including Stepanoff [4], Childs [16], Sharma [6], Wong [19], Williams [9], Alatorre-Frenk [5], show the equations used to predict pump performance as a turbine. The purpose of these formulas is to calculate the BEP of a pump turbine mode using the characteristics of the pumps provided by the manufacturer.

2 Prediction method overview

This section provides a brief description of several prediction methods available in the literature. These formulas will be used to compare estimate accuracy of turbine mode characteristics.

Childs [16] claimed that the best turbine and pump efficiency for the same machine are practically equal. Equations of the head, H, and the flow rate, Q, are reported by:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_p}\,, \tag{1}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p}\,. \tag{2}$$

Stephanoff [4] has proposed a method that uses pump efficiency. Gives the ratios of turbine and pump head and flow to the hydraulic efficiency of the pump. He states that the flow of a pump working as a turbine at the same speed would have the following values:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_{hp}}\,, \tag{3}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_{hp}^2}\,. \tag{4}$$

Because hydraulic efficiency is not usually known, the following simplification has been introduced:

$$\eta_{hp} = \sqrt{\eta_p}\,. \tag{5}$$

Taking into account this simplification we get

$$\frac{Q_t}{Q_p} = \frac{1}{\sqrt{\eta_p}}\,, \tag{6}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p}\,. \tag{7}$$

Sharma [6] developed a prediction method that also uses coefficients depending on the efficiency of the pump:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_p^{0.8}}\,, \tag{8}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p^{1.2}}\,. \tag{9}$$

Alatorre-Frenk [5] for calculating head, flow and efficiency in reverse mode, used efficiency, head and flow in pump mode. Formulas are as follows:

$$\frac{Q_t}{Q_p} = \frac{0.85\eta_p^5 + 0.385}{2\eta_p^{9.5} + 0.205}\,, \tag{10}$$

$$\frac{H_t}{H_p} = \frac{1}{0.85\eta_p^5 + 0.385} , \tag{11}$$

$$\eta_t = \eta_p - 0.03 . \tag{12}$$

Schmiedl [14] presented forecasting method uses the estimated value of hydraulic efficiency. In this method the efficiency of the hydraulic system is calculated from the following relation:

$$\eta_{hp} = \sqrt{\eta_p^{0.5}\eta_t^{0.5}} . \tag{13}$$

The equations for the BEP in Schmiedl method are as follows:

$$\frac{Q_t}{Q_p} = -1.5 + \frac{2.4}{\eta_{hp}^2} , \tag{14}$$

$$\frac{H_t}{H_p} = -1.4 + \frac{2.5}{\eta_{hp}} . \tag{15}$$

Grover [17] has presented the following equations (but they are) limited in their application to specific speeds in the range of $10 < N_s < 50$:

$$\frac{Q_t}{Q_p} = 2.379 - 0.0264 n_{qt} , \tag{16}$$

$$\frac{H_t}{H_p} = 2.693 - 0.0229 n_{qt} . \tag{17}$$

Hergt [18] calculates the head and flow in reverse mode using the specific speed PAT in BEP:

$$\frac{Q_t}{Q_p} = 1.3 - \frac{1.6}{n_{qt} - 5} , \tag{18}$$

$$\frac{H_t}{H_p} = 1.3 - \frac{6}{n_{qt} - 3} . \tag{19}$$

Derakhaskan and Nourbakhsh [2] formulated a PAT of the BEP prediction method using pump BEP point based on experimental research of four pumps. Their recommend to predictions pumps with the specific speed $N_s < 60$:

$$\begin{aligned} \gamma &= 0.0233\alpha_p + 0.6464 , \\ \alpha_t &= 0.9413\alpha_p - 0.6045 , \\ \beta_t &= 0.849\beta_p - 1.2376 . \end{aligned} \tag{20}$$

Nautiyal and his team [3], who relied on their own research and on other researcher's data, performed an analysis to estimate the relationship between pump efficiency at optimal point and specific speed. They received the following relationship:

$$\varkappa = \frac{\eta_p - 0.212}{\ln N_s} . \tag{21}$$

Using $\varkappa$, they determined the relationship between the flow and head in pump and turbine mode:

$$\frac{Q_t}{Q_p} = 30.303\varkappa - 3.424 , \tag{22}$$

$$\frac{H_t}{H_p} = 41.667\varkappa - 5.042 . \tag{23}$$

3 Proposed formula

Using the collected data for BEPs in pump and turbine mode, a graph of the dimensionless turbine flow relative to the pump flow and the dimensionless head in turbine mode relative to the head in the pump mode was drawn. It is worth noting that the points of dimensional flow are linear – approximation points are presented in Fig. 1. The equation for this straight line can be represented as follows:

$$\varphi_t = 1.99^{0.5}\varphi_p + 2^{-6.48} . \tag{24}$$

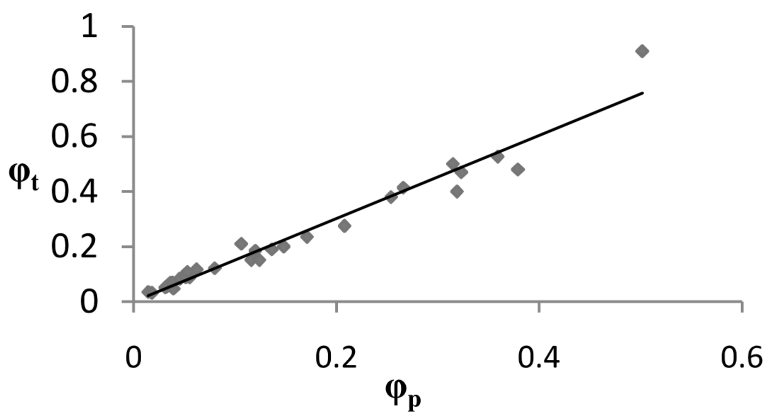

Figure 1. Dimensionless flow at BEP in pump and turbine mode φ – see Eq. (27).

Approximation points presented in Fig. 2 is a linear function also proposed for the prediction head coefficient

$$\psi_t = 4.74^{0.5}\psi_p - 5.72^{0.5}\,. \tag{25}$$

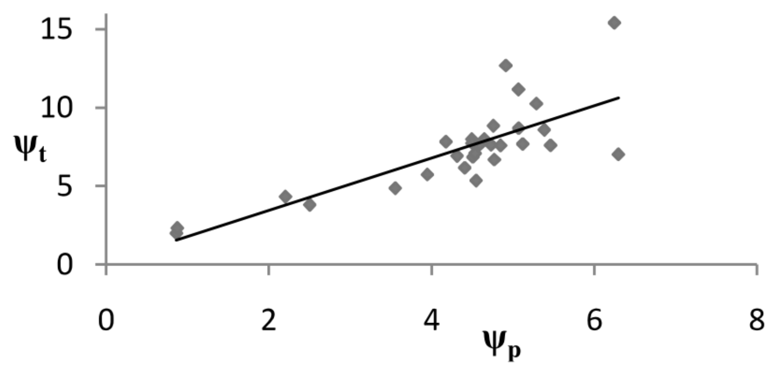

Figure 2. Dimensionless head at BEP in pump and turbine mode, ψ – see Eq. (28).

The following equation gives satisfying results for $N_s < 90$, which works better for higher specific speeds:

$$\psi = 1.5^{0.5}\psi_p + 1.34^{0.5}\,. \tag{26}$$

4 Diagram

Using the research [7,8] conducted to prepare complete characteristic of rotodynamic pump. The results are used to draw diagrams for machine working in pump mode and reverse mode. The results are shown dimensionless parameters:

- dimensionless flow coefficient

$$\psi = \frac{Q}{nD^3}\,, \tag{27}$$

- dimensionless head coefficient

$$\varphi = \frac{gH}{n^2D^2}\,. \tag{28}$$

Characteristic curves of pump and turbine mode for different specific speed from 10 to 261 are presented in Figs. 3 and 4. This diagrams and experiment data was applied to comparison formulas to PAT's characteristic curves by means of proposed formula.

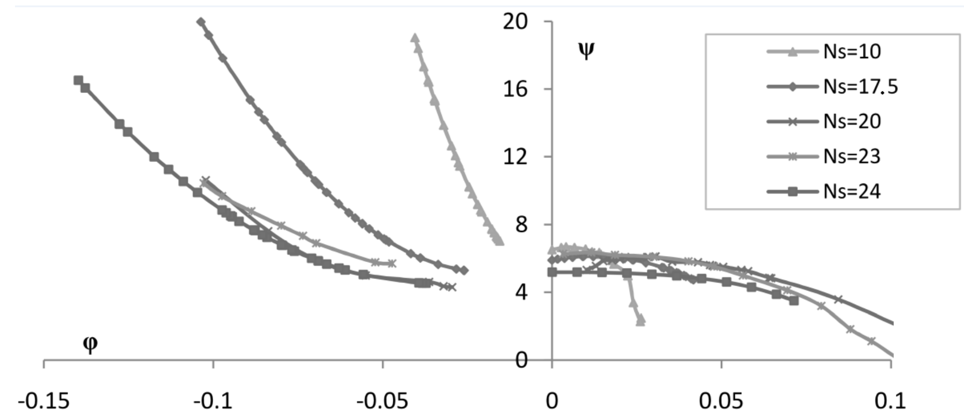

Figure 3. Dimensionless head curves for $N_s = 10$–24.

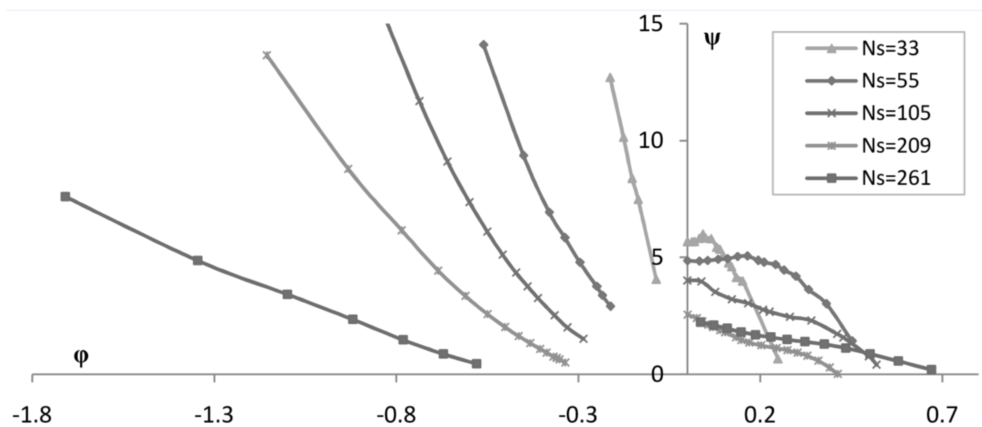

Figure 4. Dimensionless head curves for $N_s = 33$–261.

5 Comparison prediction method

A comparison of the turbine best performance prediction methods is presented in Tab. 1. It shows the percentage estimating mistake of the flow coefficient, φ_t, and the head coefficient, ψ_t, obtained from the predictions in relation to the experimental data. Graphic illustrations of errors are shown in Figs. 5 and 6. By looking at the results of comparisons, it can be stated that there is no explicit method of forecasting the operating point on the whole range of pumps with different speed ratings. The Childs method [16] has given satisfactory results in predicting the flow value.

Table 1. Comparison of prediction method accuracy.

Pump		Experimental		Childs		Stepanoff		Sharma		Grover		Derakhshan		Prop. formula	
N_s	Source	φ_t	ψ_t	φ_t[%]	ψ_t[%]	φ_t[%]	ψ_t[%]	φ_t[%]	$\psi =_t$[%]	φ_t[%]	ψ_t[%]	φ_t[%]	ψ_t[%]	φ_t[%]	ψ_t[%]
9	[10]	0.069	30.91	0.7	-16.4	-27.4	-16.4	-11.7	-4.7	12.1	8.2	-47.2	-7.1	-9.8	-13.1
10	[7]	0.035	15.42	-15.9	-17.3	-41.1	-17.3	-27.0	-4.6	-12.9	-0.2	-86.0	-14.5	-9.6	-27.3
15.1	[11]	0.054	7.03	-19.2	20.5	-30.3	20.5	-23.8	27.9	18.9	110.2	-13.1	77.7	5.4	60.9
17.5	[7]	0.063	8.59	-18.2	-6.9	-32.9	-6.9	-24.5	-0.8	5.5	43.7	-66.4	20.9	-4.6	8.6
19.3	[12]	0.047	7.70	19.0	-5.3	-0.3	-5.3	10.9	-1.7	56.2	49.8	27.9	25.7	41.8	13.8
19.9	[8]	0.084	7.61	-26.4	-3.0	-36.7	-3.0	-30.7	-3.1	1.0	60.7	-16.4	34.8	-9.8	24.9
23.1	[8]	0.088	8.70	20.6	12.1	-13.1	12.1	5.8	-27.8	10.9	26.2	-3.3	5.5	1.2	-0.6
24	[7]	0.088	7.67	-19.3	-17.7	-31.2	-17.7	-24.3	-12.3	2.4	28.4	-6.4	7.3	-4.4	-0.7
24.5	[13]	0.117	11.17	-32.1	-41.8	-40.0	-41.8	-35.4	-38.8	-8.6	-3.2	-51.3	-19.1	-16.0	-22.6
30.4	[14]	0.122	6.86	-13.4	-13.6	-24.5	-13.6	-18.1	-8.7	3.7	31.2	95.9	9.6	2.0	8.2
36.4	[13]	0.185	8.00	-12.8	-21.9	-24.8	-21.9	-17.8	-17.1	-8.1	8.1	-11.6	-9.0	-2.5	-3.4
39.7	[13]	0.200	6.70	-12.9	-16.2	-19.7	-16.2	-15.7	-13.4	-1.4	27.1	-23.2	7.7	10.2	19.3
42	[12]	0.190	7.84	-7.4	-31.3	-18.5	-31.3	-12.0	-27.8	-8.8	-7.8	-4.3	-21.3	7.2	-14.6
46.4	[13]	0.275	7.60	-0.5	-16.0	-13.2	-16.0	-5.8	-11.3	-12.7	4.4	6.0	-9.4	10.8	8.0
55	[8]	0.380	6.93	-10.9	-17.0	-22.8	-17.0	-15.9	-12.1	-39.8	-12.1	-24.3	-19.9	-2.8	1.0
61.3	[13]	0.414	5.75	-10.8	-4.6	-24.3	-4.6	-16.4	1.9	–	–	–	–	-6.6	7.9
64	[11]	0.527	5.37	25.6	56.1	-7.5	56.1	11.1	76.4	–	–	–	–	-1.7	40.0
79.1	[13]	0.480	4.88	-6.0	-13.3	-13.9	-13.3	-9.2	-10.2	–	–	–	–	13.8	9.2
94.4	[13]	0.400	3.82	-3.9	-21.1	-12.5	-21.1	-7.4	-18.1	–	–	–	–	15.4	10.4
105	[8]	0.470	4.34	4.2	-23.1	-15.3	-23.1	-4.1	-16.4	–	–	–	–	-0.6	-11.2
209	[8]	0.500	2.01	-3.1	-34.0	-21.9	-34.0	-11.1	-28.1	–	–	–	–	-8.9	10.1
261	[8]	0.910	2.34	-18.9	-100.0	-33.1	-45.2	-24.9	-40.8	–	–	–	–	-20.9	-4.9

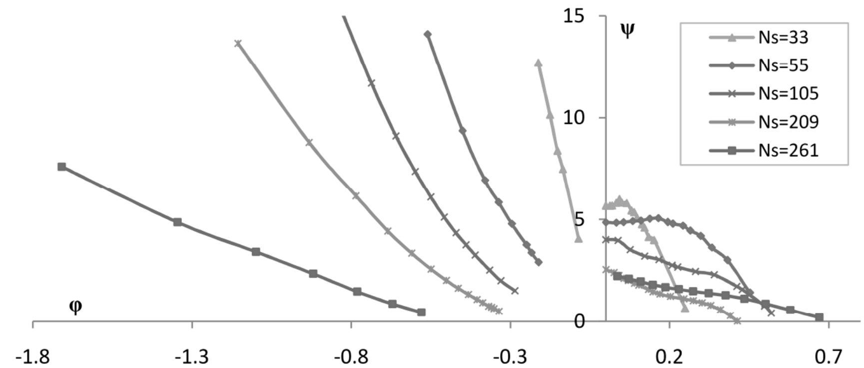

Figure 5. Variation between used prediction method of the flow coefficient for different calculation methods.

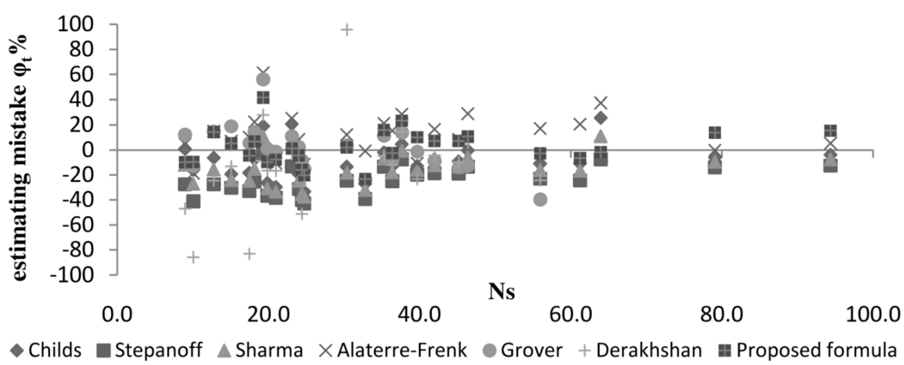

Figure 6. Variation between used prediction method of the head coefficient for different calculation methods.

For several pumps, predictions calculated with formulas are significantly higher than for others. For pump with $N_s = 15.1$, the disproportion for lift height ψ_t is from 20% to 110%. Similarly, the situation looks for pump with $N_s = 64$ for which the predictive difference is from 39% to 99% above the test outcome. For pump with $N_s = 19.3$, the flow coefficient, φ_t, for the formulas proposed by Alaterre-Frenk [5], Grover [7], Derakhshan [2] and given in this work has been considerably inflated.

It is difficult to determine which of the formulas works best for the specific range because two closely related pump identities, e.g., $N_s = 24$, and 24.5, have very different deviations from the same prediction formula. Comparison of the average estimation mistake magnitude for all comparable pumps are in [15], selected are presented in Tab. 2. This table provides a quick view of which of the formulas generates the smallest variation between turbine mode parameters. The proposed formulas give satisfactory results for the entire range of pumps being compared. The Childs and Grover methods provide a relatively good result only for the flow coefficient. The BEP gives the image of work under fixed conditions. As known, the operating conditions may vary depending on the pressure or flow available. Another important issue is the possibility of estimating values in terms other than denominated working conditions.

Table 2. Comparison of accuracy of the prediction methods.

Prediction method	Average estimation of mistake magnitude	
	φ_t[%]	ψ_t[%]
Childs	14.0	22.8
Stepanoff	23.9	20.9
Sharma	17.2	17.9
Alaterre-Frenk	17.0	16.8
Grover	13.5	25.7
Derakhshan	26.7	15.1
Proposed formula	10.8	13.8

The best solution is to have a complete flow characteristic of pump. In paper [5], a method has been proposed for converting the entire turbine characteristics to the substitution of the best turbine mode. The use of the proposed formula for calculation BEP and turbine characteristic for three cases curves is presented in Fig. 7.

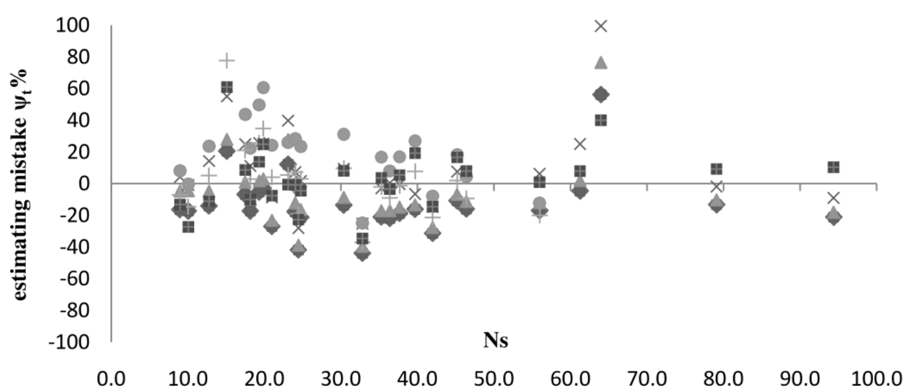

Figure 7. Dimensional flow characteristics of pump reverse mode for the measured data and the predictions Derakhaskan and Nourbakhsh. Formula for the BEP prediction from this paper.

6 Conclusions

Various methods of predicting the best turbine mode are discussed. Their own formulas have also been proposed, which can be used to predict the operating point in reverse mode. Presented formulas, they have been checked to what extent they agree with the measurement data. In comparison, thirty pumps were considered for which the specific speed difference was in the range of $N_s = 9$–261.

The comparison shows that the available formulas are characterized by large discrepancies. Also, it is difficult to pinpoint the range of specific speed for which the formula works best, because for similar specific speed the formulas give significant differences. Other parameters, such as efficiency, rotational velocity, or flow channel geometry, would have to be considered in order to indicate a particular formula. For the whole range of pumps compared, the smallest average estimation mistake was obtained based on the formulas proposed in this article. This indicates the possibility of using them in case of selection of the turbine pump in a satisfactory part of the cases. However, it should be borne in mind that the formula may be subject to a large error for specific pumps. If the application requires thorough knowledge of the course of the characteristics and knowledge of the flow parameters of the machine, it would be necessary to carry out experimental research.

A good solution to this problem of pump selection and technology development would be to identify suitable for generating energy from rotodynamic pumps available on the market by defining performance characteristics for the pump series and establishing the general relationship between parameters in pump and turbine modes. Pump manufacturers should be interested in this area as it would allow them to expand the market for their products.

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