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Experimental investigation and performance prediction modeling of a single stage centrifugal pump operating as turbine

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Abstract

In small hydropower systems, Pumps as Turbines (PaTs) represent a cost-effective alternative to conventional turbines as long as the turbine mode performance can be predicted before their installation. A closed-loop test rig for experimental studies on hydraulic pumps and turbines has been built at the Department of Mechanics, Mathematics and Management of the Polytechnic University of Bari, in order to experimentally support the theoretical studies on PaTs. A single stage centrifugal pump has been tested in both direct and reverse mode. Furthermore, a literature survey of PaT models has been conducted and a new model, with a more general applicability, has been proposed.

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Keywords: Pump as Turbine; PaT; Hydropower; Centrifugal Pump; Experimental Study; Performance Prediction Model.

1. Introduction

Among renewable energy sources, Small Hydropower Plants (SHPs) represent a promising solution in electricity production, having no significant environmental impact. Approximately 36% of the current global SHP potential has been developed in 2016. SHPs represent 7% of the total renewable energy capacity and 6.5% of the total hydropower capacity (including pumped storage). Europe has the highest SHP development rate, with nearly 48% of the overall potential already installed [1]. Due to the technological complexity of small size turbines, in SHPs the use of pumps

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operating under reverse mode (the so-called PaTs, i.e., Pumps as Turbines) represents a practical solution, considering their wide range of specific speed numbers and available standard sizes. For this reason, PaTs are often used in water distribution networks and oil pipelines, where they can replace pressure reducing valves [2]; in hydraulic pumped-storage plants, these machines can satisfy the demand when energy peaks occur. Furthermore, also in rural electrification, PaTs can represent an economical solution because can be installed and maintained by local technicians [3].

PaTs can become a cost-effective alternative to traditional turbines as long as their turbine mode performance can be accurately predicted. However, pump manufacturers do not usually offer performance curves of their pumps operating under turbine mode. A large number of theoretical and experimental studies can be found in the literature for the prediction of PaT performance. However, these prediction models show low reliability, accuracy and robustness since they usually have been developed based on a limited number of samples. As result, each model is usually accurate only with respect to the set of considered pumps.

In this framework, a closed-loop test rig for experimental studies on hydraulic pumps and turbines has been built with the aim to experimentally support the theoretical studies in the development of accurate and robust models for prediction of PaT performance. At the same time, the test rig could be an important facility for those pump manufacturers that would like to investigate more deeply their machine operations.

Nomenclature

Acronyms

AC	Alternating Current
BEP	Best Efficiency Point
DC	Direct Current
PaT	Pump as Turbine
SHP	Small Hydropower Plant

Symbols

C_V	flow coefficient of valves [m^3/h]
e_H	relative error in turbine head prediction at BEP [%]
e_Q	relative error in turbine discharge prediction at BEP [%]
h	head conversion factor
H	head [m]
$H_{BEP,P}$	head of the PaT at its BEP under pump mode operation [m]
$H_{BEP,T}$	head of the PaT at its BEP under turbine mode operation [m]
N	rotational speed [rpm]
N_S	specific speed number (Q in [m^3/s], H in [m] and N in [rpm])
$N_{S,P}$	specific speed of the PaT under pump mode operation
$N_{S,T}$	specific speed of the PaT under turbine mode operation
P	power [kW]
q	flow rate conversion factor
Q	flow rate [m^3/h]
$Q_{BEP,P}$	flow rate of the PaT at BEP under pump mode operation [m^3/h]
$Q_{BEP,T}$	flow rate of the PaT at BEP under turbine mode operation [m^3/h]

Greek letters

$\eta_{BEP,P}$	efficiency of the PaT at BEP under pump mode operation [-]
$\eta_{BEP,T}$	efficiency of the PaT at BEP under turbine mode operation [-]

A single stage centrifugal pump has been installed and tested in order to obtain its sets of characteristic curves (under both pump and turbine operating mode). Furthermore, a literature survey of PaTs has been conducted collecting all the considered prediction models. Finally, a new model has been proposed, considering a greater sample constituted by gathering all pumps found during the literature survey, in order to improve the accuracy and the applicability in the prediction of PaT performance.

2. Predictive modeling of PaT performance

As already said in the introduction, different methods for predicting PaT performance have been proposed in the literature; some of them are theoretical and based on the geometry of the machine and some complex phenomena like hydraulic losses [4-6]. These methods are quite comprehensive but they are difficult to be applied in practice because they necessitate very detailed geometric information, which is sometimes available only to the manufacturers. Therefore, many researchers have used different experimental techniques to predict the PaT performance from pump characteristics [7]. In particular, focusing on PaT Best Efficiency Point (BEP), there are mainly two groups of prediction models: authors as Williams [8], Sharma [9], Alatorre [10] and Yang et al. [11] use methods based on the pump efficiency at BEP, whereas others as Derakshan and Nourbakhsh [12], Sing and Nestmann [13], Nautiyal et al. [14], Engeda [15] and Barbarelli et al. [16], use methods based on the pump specific speed number, $N_{s,P}$. Table 1 summarizes these models in chronological order and proves that, nowadays, this topic is still object of research.

Table 1. Some BEP prediction models in literature.

Year	Researcher	Method	$h = H_{BEP,T} / H_{BEP,P}$	$q = Q_{BEP,T} / Q_{BEP,P}$
1957	Stepanoff	BEP	$\frac{1}{\eta_{BEP,P}}$	$\frac{1}{\sqrt{\eta_{BEP,P}}}$
1985	Sharma	BEP	$\frac{1}{\eta_{BEP,P}^{1,2}}$	$\frac{1}{\eta_{BEP,P}^{0,8}}$
1994	Alatorre-Frenk	BEP	$\frac{1}{0,85\eta_{BEP,P}^{5}+0,385}$	$\frac{0,85\eta_{BEP,P}^{5}+0,385}{2\eta_{BEP,P}^{9,5}+0,205}$
2012	Yang	BEP	$\frac{1,2}{\eta_{BEP,P}^{1,1}}$	$\frac{1,2}{\eta_{BEP,P}^{0,55}}$
2008	Derakshan	Ns	$0,0233\left(\frac{N_P Q_{BEP,P}^{0.5}}{(gH_{BEP,P})^{0.75}}\right)+0,6464$	$0,9413\left(\frac{N_P Q_{BEP,P}^{0.5}}{(gH_{BEP,P})^{0.75}}\right)+0,6045$
2009	Sing	Ns	$f(N_{s,P})$	$f(N_{s,P})$
2011	Nautiyal	Ns	$41,667\left(\frac{\eta_{BEP,P}-0,212}{\ln(N_{s,P})}\right)-5,042$	$30,303\left(\frac{\eta_{BEP,P}-0,212}{\ln(N_{s,P})}\right)-3,424$
2016	Engeda	Ns	$\frac{1}{H_{BEP,P}}\left(\frac{\omega D_0}{N_{s,T} D_{s,T} g^{3/4}}\right)^2$	$\frac{D_0^3\omega}{Q_{BEP,P}g^{3/4}D_{s,T}^3 N_{s,T}}$

In order to ensure a more general applicability of the prediction model for PaT performance, this literature survey has been conducted trying to have a greater number of samples to be considered. The result is a sample constituted by 27 pumps operating in a wide range of specific speed, $N_{s,P}$, from 9 to 80. All pumps are listed in table 2, sorted by $N_{s,P}$, with their main characteristic values at BEP, both under pump and turbine operation modes. The KSB Etanorm PaT in the list is the one installed and tested in the test rig of the Department of Mechanics, Mathematics and Management of the Polytechnic University of Bari. This test rig is described in details in the next section. The experimental BEP of the machine has been included in the data set of table 2.

The approach of some authors [11-13, 15-16] is to predict the BEP under turbine operation mode by correlating the prediction factors $h = H_{BEP,T} / H_{BEP,P}$ and $q = Q_{BEP,T} / Q_{BEP,P}$ to the pump specific speed under pump mode operation, $N_{s,P}$.

As known, the specific speed number, N_s, allows one to correlate the parameters of a machine performance (rotational speed, N, discharge, Q, and head, H) with the shape of the impeller. Therefore, N_s is a more useful engineering tool than the pump efficiency at BEP in the prediction of PaT performance. For this reason, in this work, the proposed prediction method is actually based on the $N_{s,P}$. Thanks to the collected experimental data, the specific speed number under turbine mode operation, $N_{s,T}$, and $N_{s,P}$ have been correlated. Successively, the head prediction factor, h, and the empirical $N_{s,T}$ have been correlated. Figure 1 shows these correlations with the interpolating curves, written in Eq. (1) and (2). Considering the data trend, a linear interpolating curve has been used for the correlation between $N_{s,T}$ and $N_{s,P}$, showing an R squared value equal to 0.98. This means a good agreement with experimental data. In the case of correlation between h and $N_{s,T}$, a third order polynomial has been considered, due to the highest R squared value (equal to 0.82), if compared with logarithmic or power trend lines.

Table 2. BEPs of pumps found in literature.

Pump	$N_{s,P}$ [-]	$Q_{BEP,P}$ [m³/h]	$H_{BEP,P}$ [m]	$\eta_{BEP,P}$ [-]	$Q_{BEP,T}$ [m³/h]	$H_{BEP,T}$ [m]	$\eta_{BEP,T}$ [-]	$N_{s,T}$ [-]
Barbarelli 1	9.05	26.6	33.01	0.44	47.09	93.28	0.43	5.52
Barbarelli 2	9.48	27.1	31.41	0.45	50.8	110.8	0.35	5.04
Barbarelli 3	12.8	25.1	20.00	0.55	38.3	43.66	0.51	8.81
Derakhshan 1	14.97	28.8	17.80	0.65	44.86	36.48	0.64	10.9
Barbarelli 4	16.34	19.0	12.00	0.55	34.99	25.5	0.59	12.6
Barbarelli 5	20.23	59.4	19.30	0.65	93.67	38	0.65	15.28
Sing 1	20.98	38.9	14.50	0.77	71.76	25.81	0.725	18.49
Derakhshan 2	23.26	85.3	20.40	0.76	135.9	39.83	0.73	17.77
Sing 2	24.46	95.4	21.50	0.78	180.8	47.37	0.765	18.62
Barbarelli 6	25.47	96.4	19.60	0.73	145.0	33.2	0.73	21.01
KSB Etanorm 200-150-400	26.44	302.5	24.4	0.784	334.5	29.85	0.889	23.87
Barbarelli 7	28.72	35.0	8.50	0.67	55	13.1	0.73	26.03
Barbarelli 8	30.30	87.0	14.52	0.74	131.47	22.4	0.78	26.91
Barbarelli 9	31.21	83.5	12.02	0.72	112.39	17.6	0.76	29.82
Sing 3	35.33	91.4	12.80	0.785	118.8	20.66	0.81	28.12
Sing 4	36.42	55.1	8.38	0.744	84.79	14.43	0.715	30.06
Pugliese 1 [19]	37.67	148.0	39.00	0.787	217.18	72.29	0.83	28.73
Derakhshan 3	39.52	205.9	18.10	0.865	304.5	31.26	0.74	31.89
Sing 5	39.66	237.2	19.80	0.85	320.2	27.81	0.835	35.71
Barbarelli 10	43.48	150.0	12.90	0.76	180	18.8	0.84	35.91
Pugliese 2 [19]	44.76	88.5	44.00	0.765	108.3	57.21	0.86	40.67
Sing 6	45.16	118.8	10.50	0.8	163.56	14.72	0.8	41.13
Derakhshan 4	55.43	385.2	17.50	0.87	439.1	23.45	0.78	47.52
Sing 8	61.26	104.0	6.40	0.72	161.8	9.32	0.743	57.63
Barbarelli 11	64.04	208.5	9.59	0.82	303.59	13.3	0.84	60.46
Barbarelli 12	77.39	77.39	5.32	0.78	157.1	7.82	0.7	64.77
Sing 9	79.21	370.8	10.60	0.84	469.36	14.64	0.755	69.95

The procedure that allows one to predict the turbine operation mode at BEP is summarized as follows: once the specific speed under pump operation mode, $N_{s,P}$, has been calculated, it is possible to evaluate the specific speed under turbine operation mode, $N_{s,T}$, by using Eq. (1):

$$N_{s,T} = 0.9237 N_{s,P} - 2.6588 \tag{1}$$

The head prediction factor, h, can be calculated thanks to $N_{s,T}$ with Eq. (2)

$$h = -0.000023 N_{s,T}^3 + 0.003206 N_{s,T}^2 - 0.145781 N_{s,T} + 3.604636 \tag{2}$$

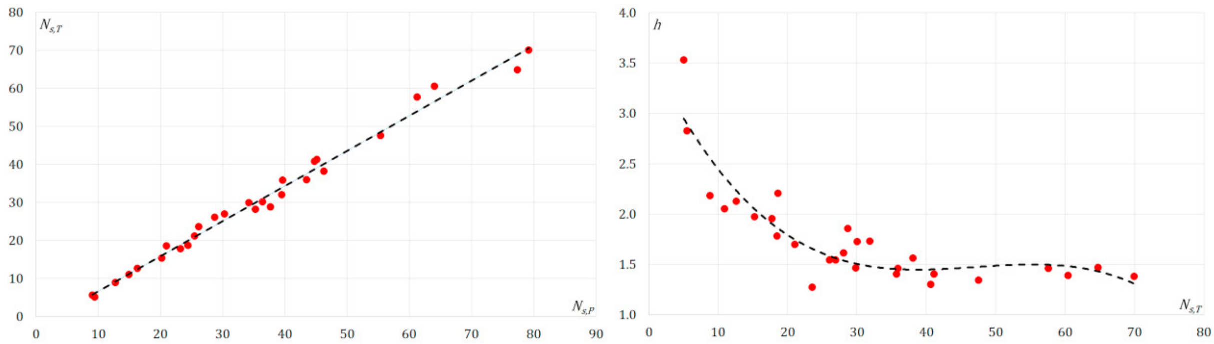

Fig. 1. Specific speed number in turbine mode, $N_{s,T}$, vs. specific speed number in pump mode, $N_{s,P}$ (left) and h factor vs. $N_{s,T}$ (right)

Therefore, the head, $H_{BEP,T}$, and discharge, $Q_{BEP,T}$, at BEP in reverse mode can be calculated by means of the following equations:

$$H_{BEP,T} = h \cdot H_{BEP,P}; \qquad Q_{BEP,T} = \left(\frac{N_{s,T} \cdot H_{BEP,T}^{3/4}}{N}\right)^2 \tag{3}$$

3. The test rig

In order to experimentally support the theoretical studies in the development of accurate and robust models for prediction of PaT performance, a closed-loop test rig for experimental studies on hydraulic pumps and turbines has been built. At the same time, this test rig could be an important facility for those pump manufacturers that would like to investigate more deeply their machine operations.

The test rig, depicted in figure 2, allows one to test machines with a head up to 280 mH_2O, a discharge up to 650 m^3/h, a rotational speed up to 2400 rpm and a power up to 480 kW. Due to the possibility to test pumps either in direct or in reverse mode, two hydraulic circuits can be set by simply acting on manual On/Off valves, as shown in figure 3. In this layout, all the installed measurement devices can be evidenced.

The PaT is installed in the test section, provided by a special platform with a rail system in order to be able to mount hydraulic machines of different sizes. The machine is directly coupled with a DC motor of 480 kW that can work as motor during “pump tests” and generator during “turbine tests”. This is possible because a four quadrant AC/DC converter controls the electric DC motor at constant rotational speed, during pump mode, and at constant torque, during reverse mode.

During “turbine test”, a booster pump (KSB Multitec D with max head 280 m, max discharge 550 m^3/h, max power 516 kW, max rotational speed 2267 rpm and max efficiency 81.2 %), driven by a second electric DC motor (identical to the previous one) and controlled only at constant rotational speed, supplies the hydraulic power for the testing of the PaT. In this case, the motor that drives the booster pump is partially supplied by the power produced by the PaT. The experimental setup for the characterization is constituted of a series of electronic measurement devices: an electromagnetic flow meter (Siemens Sitrans FM Magflow 3100 – accuracy class 0.05%), four-wire RTD PT 100 for water temperature measurements, static (EH Cerabar S) and dynamic (PCB 113B21 – accuracy class 0.05%) pressure transducers down and upstream of both the PaT and the booster pump, a HBM T40B torque meter with integrated angular speed encoder characterized by an accuracy class of 0.05%.

Three electric regulation valves are installed in the piping. Two of them (C_v = 650 m^3/h), installed downstream section of both hydraulic turbomachines, are used to regulate the discharge and the latter, installed on a parallel bypass pipeline, allows one to have a further degree of freedom in the regulation of the head and the discharge.

Furthermore, the test rig is equipped with a surge tank with a capacity of 8 m^3 and an air pressure control system that can increase the absolute pressure up to 11 bar(abs). In the case of cavitation tests, a vacuum pump can reduce the tank absolute pressure down to 0.2 bar(abs); a closed loop cooling system controls the water temperature.

Fig. 2. View of the test rig.

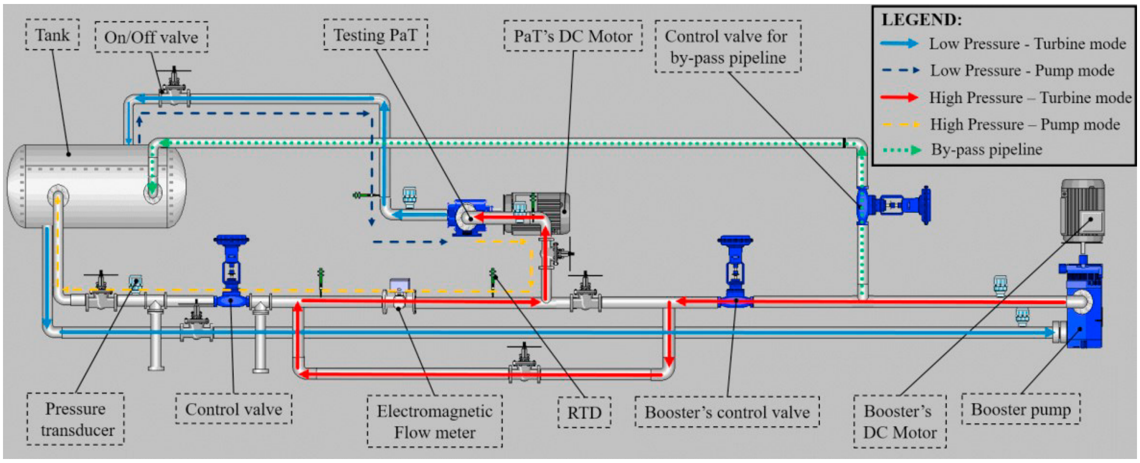

Fig. 3. Pump and Turbine mode layouts.

During tests, a real time LabVIEW software, installed on a PC (inside the control room), allows the user to remotely control the DC motors, the control valves, and perform the data acquisitions. The PC is connected with a National Instruments master/slave hardware configuration: a master NI PXIe-8135 and two slaves NI 9144 chasses with C series I/O modules.

4. Experimental characterization

A KSB Etanorm 200-150-400 PaT has been tested in the test rig. Figures 4 and 5 show the characteristic curves of the PaT under both operation modes (Pump and Turbine) at 1000 rpm. The experimental characterization has been carried out for a wide range of discharges. During pump test, the four-quadrant AC/DC converter automatically maintain constant the rotational speed (0 ÷ 2400 rpm) independently of the discharge Q. During turbine operation mode, the rotational speed is kept constant by varying the electric motor torque, through the same four-quadrant AC/DC converter, at the different booster operation points. Data were collected at 1 s interval for a duration of 30 s, after having reached steady state conditions. In order to prove the repeatability of the measurements, two tests have been carried out in different months. Moreover, the characteristic curves of the PaT, under pump operation mode, has been compared with those provided by the manufacturer catalogue. The experimental efficiency points are provided with their standard deviation bands, whereas the standard deviation of the experimental head and power are within ±3%.

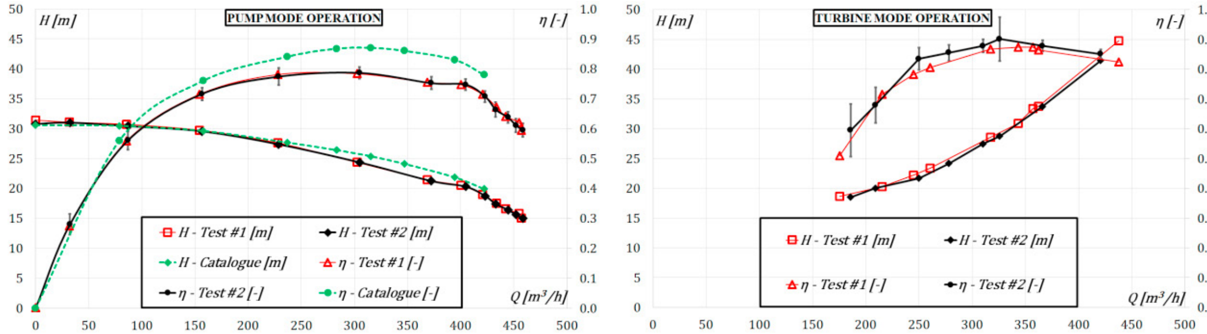

Fig. 4. Experimental and catalogue head, H, and efficiency, η, vs. discharge, Q, under either in pump (left) or turbine (right) operation mode.

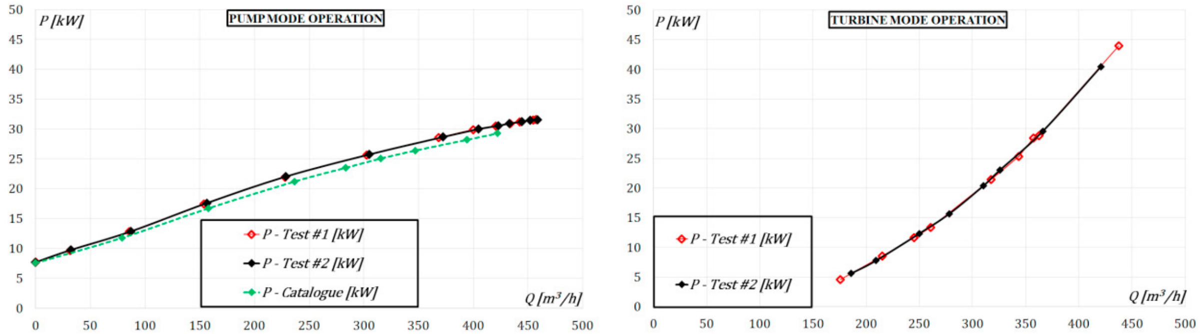

Fig. 5. Experimental and catalogue power, P, vs. discharge, Q, under either in pump (left) or turbine (right) operation mode

5. Results and discussion

The proposed PaT performance prediction model has been applied to a different set of pumps found in other recent works [11, 15, 18-20].The proposed model has been compared with other models in terms of both head and discharge prediction errors. Indeed, figure 6 shows the comparison of relative error in estimating the head, $H_{BEP,T}$, and discharge, $Q_{BEP,T}$, with respect to the experimental data, e_H and e_Q for each pump.

The model shows a good applicability with a wide range of pumps that are different not only in terms of operating BEPs, but also because they are produced by different manufacturers. Indeed, 7 pumps out of 11 are characterized by a head prediction error within ±10%, whilst 8 out of 11 are characterized by a discharge prediction error within ±10%.

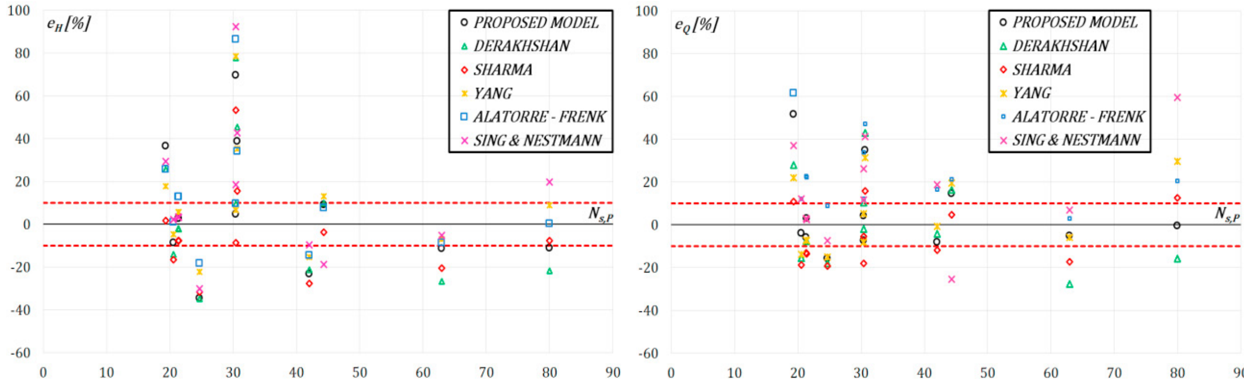

Fig. 6. Comparison of different models for BEP head prediction error, e_H, (left), and discharge prediction error, e_Q, (right) at BEP vs. pump specific speed, $N_{s,P}$.

6. Conclusion

A closed-loop test rig for experimental studies on hydraulic pumps and turbines has been built with the aim to support experimentally the theoretical studies in the development of accurate and robust models for prediction of PaT performance. Initially, a single stage centrifugal pump has been installed and tested in order to obtain its sets of characteristic curves (under both pump and turbine operating mode). Furthermore, a literature survey of PaT models has been conducted. A new model has been proposed, considering a greater sample constituted by gathering all pumps found in other studies, in order to have a better accuracy and a more general applicability in the prediction of PaT performance. It was preferred to base the model on the specific speed number rather than on the efficiency at BEP under pump operating mode, because representing families of different pumps that could work in similar operating conditions. The proposed model has been compared with other models in terms of head and discharge prediction errors by using a new set of pumps. Although data showed a significant dispersion and the sample is not yet big enough, it has presented a greater number of pumps with head and discharge BEP in turbine operation prediction errors within ± 10%. Obviously, further experiments are thus required in order to add a bigger number of pumps in the sample.

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