

# Experimental characterization of two Pumps As Turbines for hydropower generation



Francesco Pugliese <sup>a,\*</sup>, Francesco De Paola <sup>a</sup>, Nicola Fontana <sup>b</sup>, Maurizio Giugni <sup>a</sup>, Gustavo Marini <sup>b</sup>

<sup>a</sup> Department of Civil, Architectural and Environmental Engineering, University of Naples "Federico II", Via Claudio 21, 80125, Naples, Italy

<sup>b</sup> Department of Engineering, University of Sannio, Piazza Roma 21, 82100, Benevento, Italy

## ARTICLE INFO

### Article history:

Received 2 February 2016  
Received in revised form  
8 June 2016  
Accepted 23 June 2016  
Available online 6 July 2016

### Keywords:

Pump As Turbine  
Hydropower  
Experimental data  
Water Distribution Network  
Centrifugal pump  
Best Efficiency Point

## ABSTRACT

In recent years, the use of turbines or Pumps operating As Turbines (PATs) has been proven to be a sustainable alternative for managing Water Distribution Networks (WDNs), by coupling pressure control and leakage reduction with hydropower generation.

Pumps running in reverse mode can be an effective alternative to using turbines for energy production in WDNs. Many commercial models are readily available on the market and a number of economic and technical advantages for installation, operation and maintenance can be found. Theoretical and experimental criteria for predicting pump performance in turbine mode and for the optimal installation of a PAT in WDNs can be found in the literature. Nevertheless, the prediction of PAT characteristic curves is still an unresolved issue, because of the lack of information provided by manufacturers and the few laboratory campaigns that focus on the topic.

For this purpose, the laboratory results in the present study aim to assess the performance of pumps operating in reverse mode. Two centrifugal pumps were investigated: a centrifugal horizontal single-stage pump and a vertical multi-stage pump. Experiments were compared with theoretical models available in the literature, in order to assess their reliability in predicting PAT performance when data are lacking.

© 2016 Elsevier Ltd. All rights reserved.

## 1. Introduction

Hydropower generation is a relatively recent topic in the field of Best Management Practices of Water Distribution Networks. It defines a relevant category in the operative approach of leakage control, focused on the recovery of the excess pressure [1], by generating electric power [2].

Pressure Reducing Valves (PRVs) are often used in WDNs to prevent the downstream hydraulic grade from exceeding a set value. Nevertheless, the excess head can be exploited for hydropower generation by using turbines and/or Pumps As Turbines (PATs) [3]. PATs are pumps running in reverse mode, by inverting

flow direction and using the electric motor as a generator [4]. The possibility of using pumps operating in turbine mode has been widely accepted since the third decade of XX Century [5,6] but only in recent years it was pointed out the benefit of their use in WDNs [7–12]. Jain and Patel [13] provided a comprehensive review of the state-of-the-art of PATs, summarizing the main researches carried out.

PATs generally exhibit worst performance against reaction micro-turbines but, at the same time, investment and maintenance costs are largely lower [14]. PATs also show the benefit arising from the wide set of pump models commercially available, with easier installation, maintenance activities and availability of spare parts [15]. Various researchers indicated the single-stage centrifugal pumps, operating in the range of low to medium head, as the most convenient, from both technical and economic standpoint [16–19].

On the other hand, the range of flow rates over which a single PAT unit can operate is much smaller than in a conventional turbine. This issue, critical in WDNs, can be overtaken by coupling a by-pass line to the main power generation line where the PAT is

**Abbreviation:** PAT, Pump As Turbine; WDN, Water Distribution Network; PRV, Pressure Reducing Valve; BEP, Best Efficiency Point; CFD, Computational Fluid Dynamics; PLC, Programmable Logic Controller; SCADA, System Control And Data Acquisition.

\* Corresponding author.

E-mail address: [francesco.pugliese2@unina.it](mailto:francesco.pugliese2@unina.it) (F. Pugliese).

installed for energy production. On both lines, a PRV is installed, for pressure and flow regulation [10,14]. At the same time, higher operative adaptability can be obtained by installing a frequency converter to regulate the rotational speed at varying inflow discharge [9].

From a technical viewpoint, the main issue of selecting PATs is the lack of information, as performance curves are rarely made available from manufacturers. In order to obtain this information, some authors developed experimental and theoretical models to assess PAT performances. Almost all these models were introduced as function of the Best Efficiency Point (BEP), which represents the operating condition with maximum efficiency.

By using a semi-empirical one-dimensional approach, some authors [20–25] developed models to predict discharge and head drop ratios in turbine and pump mode, as a function of the efficiency  $\eta_p$  of the machine in direct mode. At a generic operating point, the efficiency  $\eta$  can be calculated, for pumps and turbines (or PATs), respectively as:

$$\eta_p = \gamma Q H_p / P_p \quad (1)$$

$$\eta_t = P_t / (\gamma Q H_t) \quad (2)$$

with  $\gamma$  specific gravity,  $Q$  flow discharge,  $H$  hydraulic head,  $P$  used or generated power. The subscripts  $p$  and  $t$  refer to the pump mode and turbine mode, respectively.

Hancock [26] correlated discharge and head drop ratios with the efficiency in turbine mode  $\eta_t$ . Schmiedl [27] correlated discharge and head drop ratios to the hydraulic efficiency of the machine  $\eta_{hp}$ , which was calculated as follows:

$$\eta_{hp} = \sqrt{\eta_p^{0.5} \eta_t^{0.5}} \quad (3)$$

By means of experiments on 35 pumps, Williams [28] showed that the Sharma's method [23] best fits test results with deviations from experimental data of around  $\pm 20\%$  [29]. Nautiyal et al. [19] correlated them with both efficiency  $\eta_p$  and specific speed  $N_{sp}$  in direct operation, expressed as:

$$N_{sp} = N Q^{\frac{1}{2}} / H^{\frac{4}{3}} \quad (4)$$

or

$$N_{sp} = N P^{\frac{1}{2}} / H^{\frac{5}{4}} \quad (5)$$

with  $N$  rotational speed (rps). In case of multi-stage pump,  $H$  in Eqs. (4) and (5) is the head of a single-stage.

Saini and Ahmad [30] introduced a chart to calculate parameters at BEP in turbine mode as a function of the specific speed in pump mode and correction factors for head drop and discharge.

Grover [31], Hergt [32] and Joshi [33] calculated head and discharge ratios at BEP as a function of specific speed in reverse mode  $N_{st}$ .

Gantar [34] carried out empirical tests on propeller pumps running in reverse mode, with different impeller blade set angles and identified a larger region with a higher efficiency in turbine mode. Based on experimental data, Amelio et al. [35] derived the characteristic curves for centrifugal pumps operating in reverse with  $N_{sp}$  between 9 and 65. They showed that the efficiency of the investigated machines in pump mode was higher than in reverse mode for  $N_{sp}$  lower than 16 and higher than 56, whereas a higher efficiency in turbine mode was obtained for  $N_{sp}$  between 16 and 56.

Fernandez et al. [17] carried out experiments on centrifugal pumps both in direct and reverse mode, with a speed ranging

between 21 and 41.5 rps. They compared the related BEP points and derived performance curves.

Derakhshan and Nourbakhsh [36,37] tested 4 centrifugal pumps in turbine mode, with  $N_{sp}$  lower than 60. They introduced an innovative approach to predict the BEP of a PAT as a function of the pump characteristics. Relationships predicting head drop and generated power ratios in turbine mode, for  $N_{st}$  up to 150, were proposed:

$$H_t = 1.0283 \left( \frac{Q_t}{Q_{tb}} \right)^2 - 0.5468 \left( \frac{Q_t}{Q_{tb}} \right) + 0.5314 \quad (6)$$

$$\begin{aligned} P_t = & -0.3092 \left( \frac{Q_t}{Q_{tb}} \right)^3 + 2.1472 \left( \frac{Q_t}{Q_{tb}} \right)^2 - 0.8865 \left( \frac{Q_t}{Q_{tb}} \right) \\ & + 0.0452 \end{aligned} \quad (7)$$

in which the subscript  $b$  refers to the conditions at the BEP. Eqs. (6) and (7) were obtained for flow rate numbers:

$$\phi = Q / (ND^3) \quad (8)$$

lower than 0.40, where  $D$  is the impeller diameter in m.

The most relevant relationships are summarized in Table 1, in terms of discharge and head ratios.

Singh and Nestmann [38] introduced an optimization routine to predict performance and select radial flow centrifugal pumps in turbine mode. The relationships were validated experimentally for very low values of rotational speed.

Nautiyal et al. [14] carried out an empirical investigation on centrifugal pumps, pointing out that machines operating in turbine mode were able to work at higher head and flow rates, whereas the BEP of the PAT was lower than in direct mode. They introduced a selection criteria to reduce the difference between theoretical methods and experimental results as a function of mechanical factors, pumped liquid and system characteristics.

Alternative approaches to predict PAT performance are based on the development of CFD models, which are able to overtake the analytical difficulties of solving the equation arising from the fluid-dynamic field [14,18,25,37,39,40].

In any case, CFD models require validation through experimental data, especially when re-designing internal components of pumps to improve the efficiency in reverse mode [41]. As well known, the performance of a pump in turbine mode can be improved by optimizing impeller geometry [42–48].

Despite the models available in literature, predicting PAT curves is still an unresolved issue, because of the lack of information provided by manufacturers and the few laboratory tests that focus on the topic. In addition, the available models have been derived from experiments on horizontal single-stage pumps, although different types of pumps are available on the market and commonly used (e.g. vertical axis pumps, multi-stage pumps).

Consequently, the results of a laboratory experiments were analyzed and discussed in the paper, so as to overcome some lack of knowledge on the performance of pumps operating in reverse mode. Experiments aimed at assessing: a) the validity of the model proposed in Refs. [36,37] for flow rate numbers up to 1.3; and b) the reliability of such model for a vertical multi-stage pump, since the cited relationships were derived from experiments on horizontal centrifugal pumps. Data currently available from the literature are derived from experiments carried out on horizontal single-stage pumps, consequently it is still an open question whether they can be used also for different type pumps.

Obtained experimental data also allowed to validate the existing

**Table 1**

Models for calculating discharge and head ratios.

Model	$Q_{tb}/Q_{pb}$	$H_{tb}/H_{pb}$
Nautiyal et al. [19]	$30.303[(\eta_p - 0.212)/\ln(N_{sp})] - 3.424$	$41.667[(\eta_p - 0.212)/\ln(N_{sp})] - 5.042$
Stepanoff [20]	$\eta_p^{-1/2}$	$\eta_p^{-1}$
Childs [21]	$\eta_p^{-1}$	$\eta_p^{-1}$
Sharma [23]	$\eta_p^{-0.8}$	$\eta_p^{-1.2}$
Alatorre - Frenk & Thomas [24]	$(0.85\eta_p^5 + 0.385)/(2\eta_p^{9.5} + 0.205)$	$1/(0.85\eta_p^5 + 0.385)$
Yang et al. [25]	$1.2/\eta_p^{0.55}$	$1.2/\eta_p^{1.1}$
Hancock [26]	$\eta_t^{-1}$	$\eta_t^{-1}$
Schmidl [27]	$-1.5 + 2.4/\eta_{hp}^2$	$-1.4 + 2.5/\eta_{hp}$
Grover [31]	$2.379 - 0.0264N_{st}$	$2.693 - 0.0229N_{st}$
Hergt [32]	$1.3 - 1.6/(N_{st} - 5)$	$1.3 - 6/(N_{st} - 3)$
Derakhshan & Nourbakhsh [36]	$f(N_{sp})$	$f(N_{sp})$

relationships predicting the BEP of the machine in reverse mode from data available in normal mode.

An inverter was also used to obtain the largest possible number of operating conditions, so as to vary in a wide range the rotational speed of the impeller. Data were presented in terms of dimensionless numbers for generalization. According to the approach proposed in Ref. [36], analytic relationships predicting PATs' performance curves were derived with respect to BEP conditions. Analysis of uncertainty was also developed to predict the reliability of the proposed formulas.

## 2. The experimental set-up

Experiments were carried out at the Hydraulic Laboratory in the Department of Civil, Architectural and Environmental Engineering, University of Naples Federico II, using a four loops laboratory WDN (Fig. 1).

The network was made of cast iron and steel with diameter DN 150 and supplied by an air chamber, which guaranteed an inlet maximum pressure of around 70 m and a flow discharge of up to 50 l/s. Motorized gate valves, needle valves and PRVs were installed for flow control and regulation. Pressure transducers and flow meters were also deployed within the system to characterize flow at the most significant nodes of the network [49].

A needle valve was installed at the network inlet to regulate pressure, whereas butterfly valves were installed at the outlets to regulate outflow. Valves were remotely controlled by electric actuators, connected to PLCs using a wired connection with Ethernet/IP protocol. The system was remotely controlled through an in-house SCADA system.

Two PATs were installed at Nodes S.S. and M.S. At Node S.S. (Fig. 2) a horizontal single-stage electric centrifugal pump was installed (model LOWARA FHE 80-200/220).

At Node M.S. (Fig. 3) a vertical multi-stage centrifugal pump was installed (model LOWARA 92SV2GH150T).

The pump characteristics were summarized in Table 2, whereas in Fig. 4a,b the performance curves of the two investigated pumps were given.

An electromagnetic flow meter (with accuracy 0.20%) and a pressure transducer (pressure range 0–10 bar and accuracy 0.25%) upstream and downstream of the PAT were also installed.

The PATs were connected to an electric frequency converter (Input Ratings: 380–415 V, 44 A, 48–63 Hz; Output Ratings: 0–415 V, 47 A, 0–300 Hz; Power Rating: 5.5–110 kW; accuracy 1.00%), for electric regulation [50]. It allowed the regulation of PATs rotational speed  $N$  and the measurement of generated power  $P$ .

Experiments were carried out for a wide range of inflow discharge  $Q$ . During each run, the discharge was set (almost) constant and the PAT rotational speed  $N$  was varied through the inverter. Discharge  $Q$ , head drop  $H$  and generated power  $P$  were

measured, whereas efficiency  $\eta_t$  at any operating condition was calculated through Eq. (2).

The rotational speed was varied between a minimum of 5 rps and the maximum rotational speed compatible with the actual discharge through the PAT. An increase of 0.5 rps was considered during each run. The discharge running the PAT was varied by opening the node outlets. An automatic control was set to maintain a constant discharge at increasing  $N$ , by varying the opening degree of the needle valve at the network inlet. In any case, due to the need of a dead band in the control algorithm, a small variation (in the order of 0.1–0.2 l/s) can be observed during the experiments. Data were collected at 1 s interval and the duration of recording was always greater than 1 min. Steady state conditions were achieved before recording and outliers were discarded. In any case, time averaged values were used for further analysis, so as to reduce sampling random error.

The uncertainty analysis associated to the experimental measurements was performed according to Abernethy et al.'s method [51]. The measurement error was calculated as the sum of a fixed error and a random error. For flow rate  $Q$  and power  $P$ , the fixed error was estimated as a function of the accuracy of the electromagnetic flow meter and the frequency converter, respectively. The error propagation method was applied for evaluating the fixed error of both head drop  $H$  (as a function of the pressure transducers' accuracy) and efficiency  $\eta$ .

The random error was estimated by considering a 95% confidence level of the Student's t-distribution with  $t_{95} = 2.00$ , since the degrees of freedom are always greater than 30 [52].

The uncertainty of flow, head, power and efficiency thus resulted  $\pm 0.26\%$ ,  $\pm 2.09\%$ ,  $\pm 2.93\%$  and  $\pm 3.61\%$ , respectively.

## 3. Results and discussion

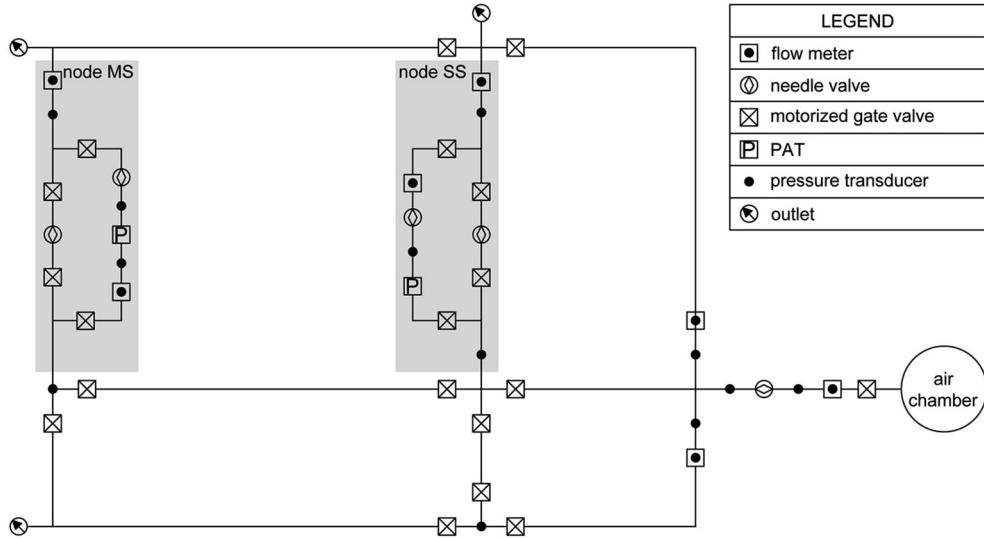
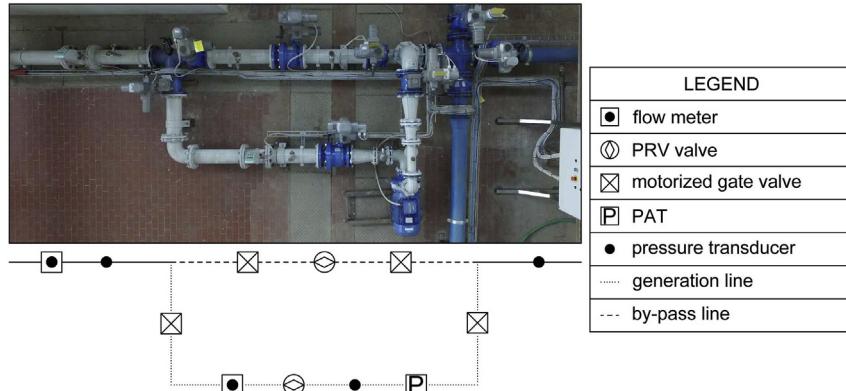
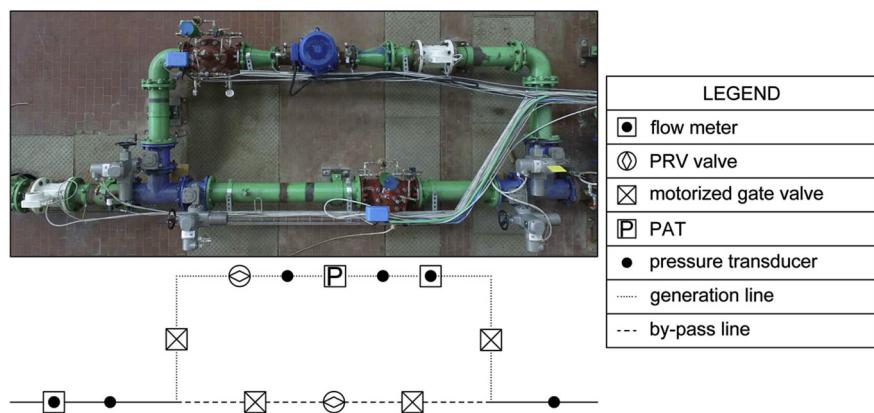
The results obtained from the experiments were summarized in what follows for both S.S. and M.S. PATs. For S.S. PAT, flow discharge was varied from 10 l/s to 45 l/s, and  $N$  from 5 rps to 36 rps, for a total of 2671 experiments. To compare data corresponding to quite different operating conditions, results were given in dimensionless form, derived from affinity law [53], through flow rate number  $\varphi$ , head number  $\psi$  and power number  $\pi$ , which are constant for similar machines operating with the same efficiency  $\eta$ :

$$\psi = gH / (N^2 D^2) \quad (9)$$

$$\pi = P / (\rho N^3 D^5) \quad (10)$$

in which  $g$  is the acceleration of gravity and  $\rho$  is the fluid density.

As result, the flow rate number  $\varphi$  varied between 0.08 and 1.33, head number  $\psi$  between 3.37 and 413.59, power number  $\pi$

**Fig. 1.** Layout of laboratory network.**Fig. 2.** Detail of equipment and transducers at Node S.S. (Plan and schematic).**Fig. 3.** Detail of equipment and transducers at Node M.S. (Plan and schematic).

between 0.01 and 67.34. Since the generated power was measured at the inverter output and not directly at the motor, the power number was calculated by taking into account the inverter consumption, estimated in around 0.04 kW.

For the vertical multi-stage pump, flow discharge was varied

between 9 l/s and 28 l/s, and  $N$  from 5 rps to 36 rps, for a total of 1647 experimental data. Flow rate number  $\varphi$  varied between 0.10 and 1.73, head number  $\psi$  between 5.91 and 1000.88, and power number  $\pi$  between 0.01 and 230.94. As for horizontal single-stage PAT, the power number was calculated by adding the generated

**Table 2**

Characteristics of the investigated PATs in normal mode.

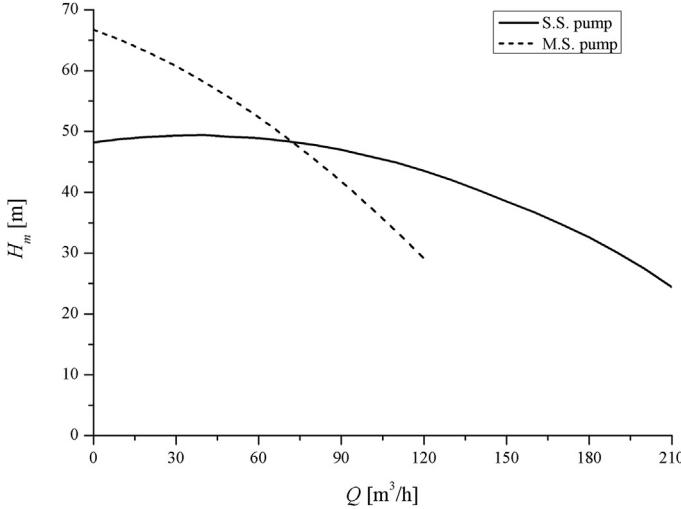
PAT	S.S.	M.S.
No. poles	2	2
No. impellers	1	2
Impeller diameter $D$ [m]	0.189	0.146
Rotational Speed $N$ [rps]	48.33	48.33
Motor power $P$ [kW]	22.0	15.0
Efficiency $\eta_p$ at BEP [%]	78.7	76.5
Discharge $Q$ at BEP [ $\text{m}^3/\text{h}$ ]	148.0	88.5
Head $H_m$ at BEP [m]	39.0	44.0
Specific speed $N_{sp}$ [m, $\text{m}^3/\text{s}$ ]	37.75	44.76

**Table 3**

Operation at BEP in reverse mode for S.S. and M.S. PATs.

PAT	$\varphi$	$\psi$	$\pi$	$\eta_t$
S.S.	0.18	8.50	0.96	0.61
M.S.	0.20	11.27	1.63	0.72

$$\frac{P_t}{P_{tb}} = 4.000 \cdot 10^{-3} \left( \frac{Q_t}{Q_{tb}} \right)^3 + 1.386 \left( \frac{Q_t}{Q_{tb}} \right)^2 - 0.390 \left( \frac{Q_t}{Q_{tb}} \right) \quad (11)$$

**Fig. 4.** Pump characteristic curves in direct mode (S.S. and M.S. pumps).

power measured by the inverter by 0.04 kW, in order to account for its consumption.

The dimensionless variables identified at the BEP when running in reverse mode were shown in Table 3 for both pumps.

Since the specific speed  $N_{sp}$  for S.S. and M.S. PAT is 37.75 and 44.76 respectively, they both fall within the range of the Derakhshan and Nourbakhsh's model. The flow rate number  $\varphi$  was varied in a much wider range instead. Consequently, experiments allowed to assess whether Eqs. (6) and (7) reliably predict head number and power number also for flow rate numbers greater than 0.40.

### 3.1. Single-stage PAT

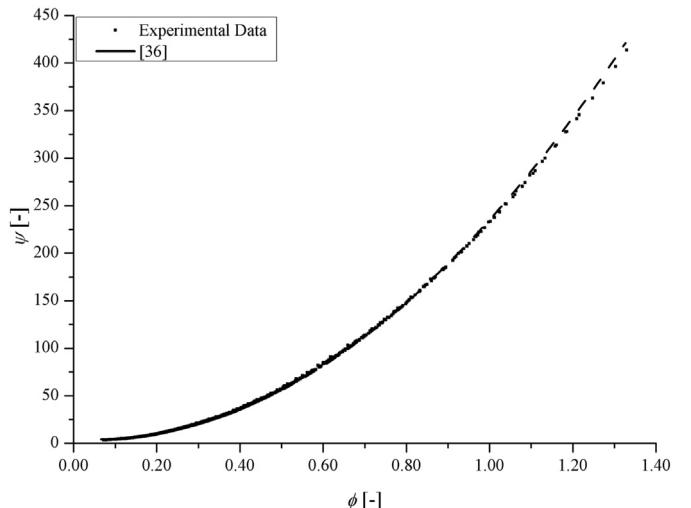
For S.S. PAT, head number showed a good agreement between experiments and Eq. (6), with deviations in the order of a few percent within all the investigated range of flow rate numbers (Fig. 5).

Power number showed instead a good agreement with Eq. (7) only for  $\varphi < 0.40$ , whereas significant deviation was found for higher values (Fig. 6). Data exhibited a monotonically increasing trend at increasing  $\varphi$ , whereas Eq. (7) showed a maximum for  $\varphi \approx 0.81$  and a decreasing trend for higher flow rate numbers.

Consequently, Eq. (7) is not able to predict power numbers for  $\varphi > 0.40$ , unlike Eq. (6), which is effective also for flow rate number outside the investigated range. A new curve was inferred for the power number, also reliable for a wider range of flow rate numbers. The structure of a third order polynomial was maintained for the relationship, and coefficients were calculated by minimizing the differences between experiments and theoretical model. It resulted the following relationship:

which has a range of validity wider than Eq. (7), up to flow rate numbers of around 1.30. Eq. (11) was also plotted in Fig. 6, showing the good agreement with experimental data within the whole range of investigated flow rate numbers.

Finally, a significant reduction in the efficiency at BEP when operating in reverse mode was found. Experiments showed that the BEP was achieved for  $\varphi \approx 0.18$  with a maximum efficiency of around 61%. Such a value is 18% lower than the efficiency at BEP

**Fig. 5.** Experimental data and Derakhshan and Nourbakhsh's model for  $\psi(\varphi)$  (S.S. PAT).

when operating in normal mode. The result is quite different from ones obtained by Amelio et al. [35], who measured values at BEP in reverse mode greater than in direct mode for  $N_{sp}$  between 16 and 56.

According to Eq. (2), analytic relationship for efficiency was also proposed. When using Eq. (7), efficiency can be calculated as:

$$\frac{\eta_t}{\eta_{tb}} = \frac{-0.3092 \left( \frac{Q_t}{Q_{tb}} \right)^3 + 2.1472 \left( \frac{Q_t}{Q_{tb}} \right)^2 - 0.8865 \left( \frac{Q_t}{Q_{tb}} \right) + 0.0452}{1.0283 \left( \frac{Q_t}{Q_{tb}} \right)^3 - 0.5468 \left( \frac{Q_t}{Q_{tb}} \right)^2 + 0.5314 \left( \frac{Q_t}{Q_{tb}} \right)} \quad (12)$$

whereas the following relationship is obtained, when the power curve of Eq. (11) is used:

$$\frac{\eta_t}{\eta_{tb}} = \frac{4 \cdot 10^{-3} \left( \frac{Q_t}{Q_{tb}} \right)^3 + 1.386 \left( \frac{Q_t}{Q_{tb}} \right)^2 - 0.390 \left( \frac{Q_t}{Q_{tb}} \right)}{1.0283 \left( \frac{Q_t}{Q_{tb}} \right)^3 - 0.5468 \left( \frac{Q_t}{Q_{tb}} \right)^2 + 0.5314 \left( \frac{Q_t}{Q_{tb}} \right)} \quad (13)$$

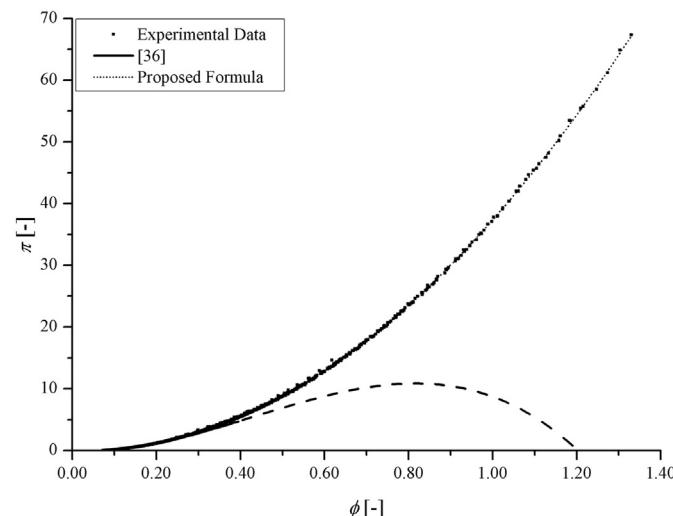
Both Eqs. (12) and (13) were plotted in Fig. 7 for comparison with experiments. As for power curve, measured data show a better agreement using Eq. (13) within all the investigated range, whereas Eq. (12) returns reliable values only for  $\phi < 0.40$ .

In Figs. 5–7 a dashed line was used for  $\phi > 0.40$ , to better identify the range of investigated values by Derakhshan and Nourbakhsh [36].

### 3.2. Multi-stage PAT

Head number, power number and efficiency for the multi-stage PAT were plotted in Figs. 8–10 as function of flow rate number. Data showed again high correlation, with a very low scattering, for both head and power numbers. The efficiency at BEP when running in turbine mode was found around 72.0%, for a flow rate number  $\phi \approx 0.20$  (Fig. 10). Such value is slightly lower (around 5%) in reverse mode with respect to the direct one. The result is in good agreement with previous researches, which showed similar efficiency in turbine mode and in pump mode.

In Figs. 8 and 9, Eqs. (6) and (7) were compared with



**Fig. 6.** Experimental data, Derakhshan and Nourbakhsh's model and proposed formula for  $\pi(\phi)$  (S.S. PAT).

experimental data. Newly derived Eq. (11) was also plotted, as a function of flow rate number.

As for S.S. PAT, Eq. (7) was able to predict the power curve only for  $\phi < 0.40$ , whereas very different behavior was observed for higher values (Fig. 9). Also in this case, Eq. (7) returned wrong values when the flow rate number was outside the investigated range. Same considerations apply for efficiency curve (Fig. 10). Eq. (12) is reliable only for lower values of  $\phi$ , whereas the proposed Eq. (13) shows a good agreement within the whole investigated range. In Figs. 8–10 a dashed line was used for  $\phi > 0.40$ , in accordance with approach applied for S.S. PAT.

Eqs. (6) and (11) returned a fairly good agreement with experiments, with maximum differences around 20% for head and 30% for power. In both cases, predicted values were lower than measured data, thus resulting in underestimated head and power numbers.

### 3.3. Prediction of discharge and head ratios

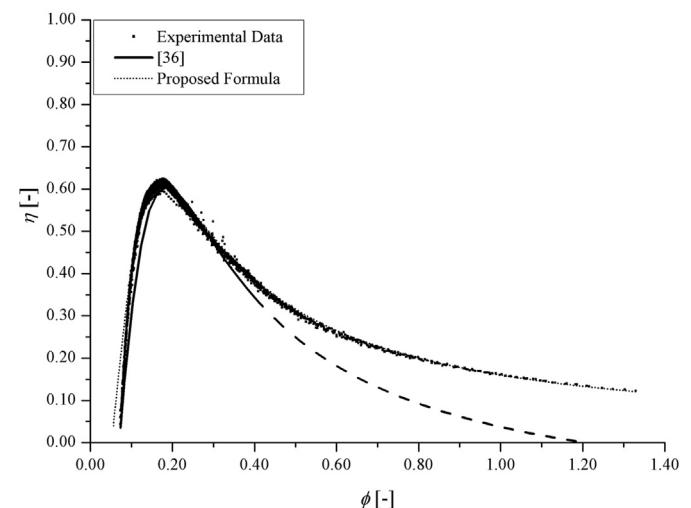
Finally, analysis was carried out to assess the reliability of relationships given in Table 1 to predict discharge and head ratios when operating in normal and reverse mode. Data for direct operation were obtained from manufacturers' datasheets, and from experiments for reverse operation. A rotational speed of 48.33 rps was considered for analysis. Details of machine operation at BEP in direct and reverse mode were given in Table 4.

Measured values were compared with theoretical models summarized in Table 1. Results were given in Tables 5 and 6 for the horizontal single-stage and vertical multi-stage PATs, respectively.

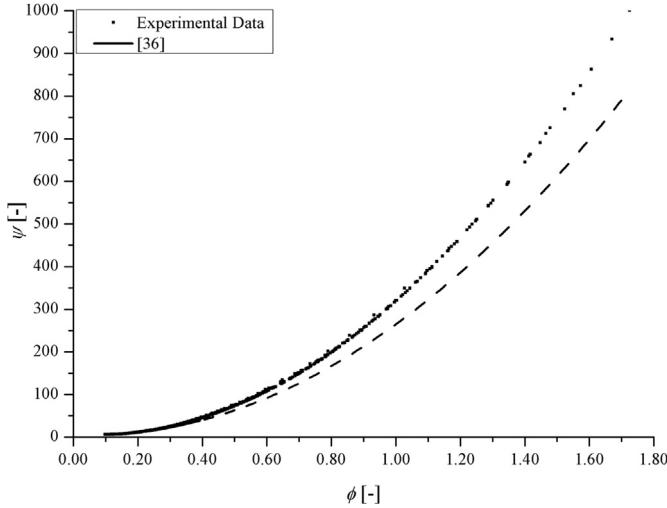
Although data showed a significant dispersion, for almost all models differences are within 20–30%, as pointed out by Nautiyal et al. [19] and Yang et al. [25]. The model with the best agreement is the Hancock model [26], with differences in the order of 10% for head and discharge ratios for both pumps.

## 4. Conclusions

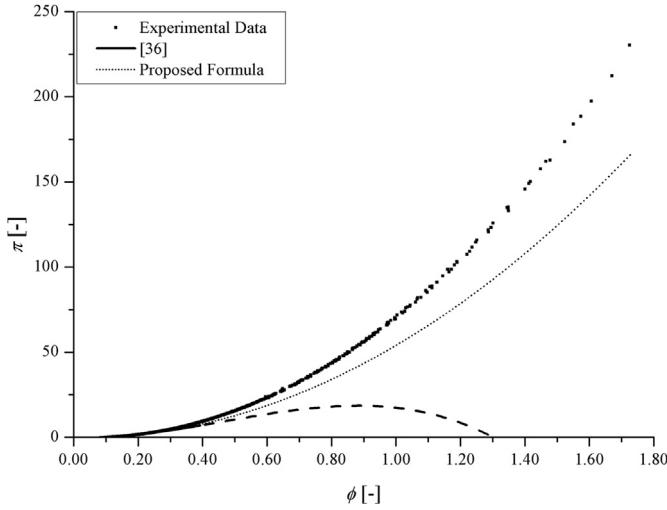
The paper discusses laboratory experiments on two centrifugal pumps operating in reverse mode. For the horizontal single-stage pump, results showed that the model of Derakhshan and Nourbakhsh is reliable also outside the investigated range (flow rate numbers lower than 0.40) for head number, whereas it fails to



**Fig. 7.** Experimental data, Derakhshan and Nourbakhsh's model and proposed formula for  $\eta(\phi)$  (S.S. PAT).



**Fig. 8.** Experimental data and Derakhshan and Nourbakhsh's model for  $\psi(\phi)$  (M.S. PAT).



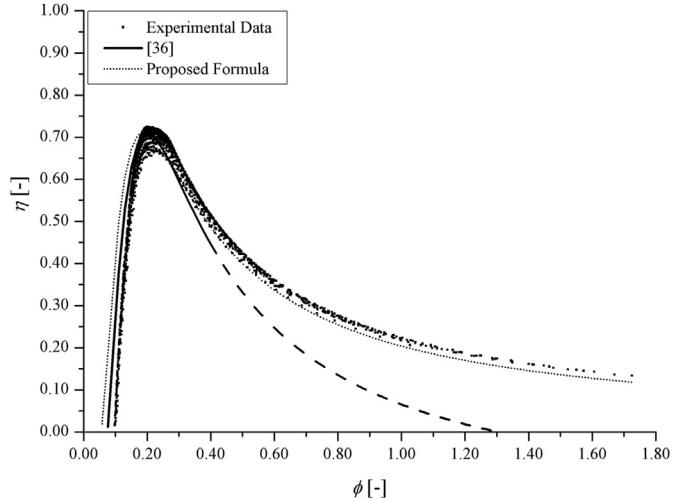
**Fig. 9.** Experimental data, Derakhshan and Nourbakhsh's model and proposed formula for  $\pi(\phi)$  (M.S. PAT).

predict generated power for greater flow rate numbers. Consequently, a different equation was proposed, which predicts reliable power numbers within the whole investigated range.

Furthermore the comparison between experimental data and theoretical mono-dimensional approaches pointed out how, for the considered machines, the BEP in reverse mode can be predicted, in the first approximation, as a function of BEP in direct mode, through several analytic models in literature, with differences generally lower than 25–30%.

Results were extended to a vertical two-stage centrifugal pumps, showing again the validity of the power curve equation provided by Derakhshan and Nourbakhsh only for flow rate numbers lower than 0.40, while both the head curve and the proposed equation for the horizontal pump under-estimated experiments by around 20–30%.

Further experiments are thus required, in order to test different models of vertical single and multi-stage pumps and assessing whether different relationships are required or differences fall within the uncertainties of experimental runs.



**Fig. 10.** Experimental data, Derakhshan and Nourbakhsh's model and proposed formula for  $\eta(\phi)$  (M.S. PAT).

**Table 4**  
Pump characteristics and operation at BEP for 48.33 rps in reverse mode.

PAT	$Q_{bt}$ [l/s]	$H_{bt}$ [m]	$P_{bt}$ [kW]	$N_{st}$ [m, m <sup>3</sup> /s]	$\eta_{hp}$ [-]	$Q_{bt}/Q_{bp}$ [-]	$H_{bt}/H_{bp}$ [-]
S.S.	60.33	72.29	26.03	28.73	0.83	1.47	1.86
M.S.	30.09	57.21	12.18	40.67	0.86	1.22	1.30

**Table 5**  
Discharge and Head Ratios calculated according to literature models (S.S. PAT).

Model	$Q_{bt}/Q_{bp}$	$H_{bt}/H_{bp}$	$\Delta(Q_{bt}/Q_{bp})$	$\Delta(H_{bt}/H_{bp})$
Nautiyal et al. [19]	1.37	1.56	-6.31%	-16.25%
Stepanoff [20]	1.13	1.27	-23.18%	-31.62%
Childs [21]	1.27	1.27	-13.41%	-31.62%
Sharma [23]	1.21	1.33	-17.46%	-28.27%
Alatorre - Frenk & Thomas [24]	1.56	1.56	6.52%	-16.13%
Yang et al. [25]	1.37	1.56	-6.71%	-15.96%
Hancock [26]	1.63	1.63	11.22%	-12.17%
Schmidl [27]	1.96	1.60	33.31%	-13.90%
Grover [31]	1.62	2.04	10.44%	9.51%
Hergt [32]	1.23	1.07	-16.00%	42.59%
Derakhshan & Nourbakhsh [36]	1.39	1.54	-5.08%	-16.98%

**Table 6**  
Discharge and Head Ratios calculated according to literature models (M.S. PAT).

Model	$Q_{bt}/Q_{bp}$	$H_{bt}/H_{bp}$	$\Delta(Q_{bt}/Q_{bp})$	$\Delta(H_{bt}/H_{bp})$
Nautiyal et al. [19]	0.98	1.02	-19.59%	-21.59%
Stepanoff [20]	1.14	1.31	-6.60%	0.53%
Childs [21]	1.31	1.31	6.78%	0.53%
Sharma [23]	1.24	1.38	1.21%	6.07%
Alatorre - Frenk & Thomas [24]	1.68	1.65	37.15%	26.56%
Yang et al. [25]	1.39	1.61	13.59%	23.92%
Hancock [26]	1.39	1.39	13.27%	6.64%
Schmidl [27]	1.73	1.50	41.42%	15.42%
Grover [31]	1.31	1.76	6.63%	35.48%
Hergt [32]	1.26	1.14	2.53%	-12.27%
Derakhshan & Nourbakhsh [36]	1.52	1.74	24.58%	33.76%

## Acknowledgments

This work was supported by the EU PON/FESR “Ricerca e Competitività” 2007–2013 under projects PON01\_01596 “WaterGRID”

and PON04a2\_F “BE&SAVE – AQUASYSTEM – SIGLUD”.

## References

- [1] F. De Paola, M. Giugni, E. Galdiero, A jazz based approach for optimal setting of pressure reducing valves in water distribution networks, *Eng. Opt.* 48 (5) (2015) 1–13, <http://dx.doi.org/10.1080/0305215X.2015.1042476>.
- [2] M. Giugni, N. Fontana, D. Portolano, Energy saving policy in water distribution networks, in: International Conference on Renewable Energies and Power Quality (ICREPQ 2009), *Renewable Energy & Power Quality Journal (RE&PQJ)* in Valencia, Spain, 2009.
- [3] M. Giugni, N. Fontana, A. Ranucci, Optimal location of PRVs and turbines in water distribution systems, *J. Water Res. Plann. Manage.* 140 (9) (2009), [http://dx.doi.org/10.1061/\(ASCE\)WR.1943-5452.0000418](http://dx.doi.org/10.1061/(ASCE)WR.1943-5452.0000418), 06014004.
- [4] J.M. Chapallaz, P. Eichenberger, G. Fischer, *Manual on Pumps Used as Turbines* vol. 11, Vieweg, Ed. Gate-Gtz, Braunschweig, 1992, ISBN 3-528-02069-5.
- [5] D. Thoma, C.P. Kittredge, Centrifugal pumps operated under abnormal conditions, *Power* 73 (1931) 881–884.
- [6] A. Tamm, A. Braten, B. Stoffel, G. Ludwig, Analysis of a standard pump in reverse operation using CFD, in: PD-05, 20<sup>th</sup> IAHR-symposium, 2000. Charlotte, North Carolina, USA.
- [7] H. Ramos, A. Borga, Pumps as turbines: an unconventional solution to energy production, *Urban Water J.* 1 (3) (1999) 261–263, [http://dx.doi.org/10.1016/S1462-0758\(00\)00016-9](http://dx.doi.org/10.1016/S1462-0758(00)00016-9).
- [8] M. Arriaga, Pumps as turbine – a pico-hydro alternative in Lao People's democratic republic, *Renew. Energy* 35 (2010) 1109–1115, <http://dx.doi.org/10.1016/j.renene.2009.08.022>.
- [9] A. Caravetta, G. Del Giudice, O. Fecarotta, H. Ramos, Energy production in water distribution networks: a PAT design strategy, *Water Res. Manag.* 26 (2012) 3947–3959, <http://dx.doi.org/10.1007/s11269-012-0114-1>.
- [10] N. Fontana, M. Giugni, D. Portolano, Losses reduction and energy production in water distribution networks, *J. Water Res. Plan-ASCE* 138 (3) (2012) 237–244, [http://dx.doi.org/10.1061/\(ASCE\)WR.1943-5452.0000179](http://dx.doi.org/10.1061/(ASCE)WR.1943-5452.0000179).
- [11] N. Fontana, M. Giugni, D. Portolano, Closure to “Losses reduction and energy production in water distribution networks”, *J. Water Res. Plan-ASCE* 140 (2) (2014) 271–273, [http://dx.doi.org/10.1061/\(ASCE\)WR.1943-5452.0000380](http://dx.doi.org/10.1061/(ASCE)WR.1943-5452.0000380).
- [12] M. De Marchis, C.M. Fontanazza, G. Freni, A. Messineo, B. Milici, E. Napoli, V. Notaro, V. Puleo, A. Scopa, Energy recovery in water distribution networks. Implementation of pumps as turbines in a dynamic numerical model, *Procedia Eng.* 70 (2014) 439–448, <http://dx.doi.org/10.1016/j.proeng.2014.02.049>.
- [13] S.V. Jain, R.N. Patel, Investigations on pump running in turbine mode: a review of the state-of-the-art, *Renew. Sust. Energy Rev.* 30 (2014) 841–868, <http://dx.doi.org/10.1016/j.rser.2013.11.030>.
- [14] H. Nautiyal, V. Goel, A. Kumar, Reverse running pumps analytical, experimental and computational study: a review, *Renew. Sust. Energy Rev.* 14 (2010) 2059–2067, <http://dx.doi.org/10.1016/j.rser.2010.04.006>.
- [15] K.H. Motwani, S.V. Jain, R.N. Patel, Cost analysis of pump as turbine for pico hydropower plants – a case study, *Procedia Eng.* 51 (2013) 721–726, <http://dx.doi.org/10.1016/j.proeng.2013.01.103>.
- [16] R. Lueneburg, R.M. Nelson, in: Lobanoff, Ross (Eds.), *Hydraulic Power Recovery Turbines, Centrifugal Pumps: Design and Applications*, 1985, pp. 246–282. Texas, USA.
- [17] J. Fernandez, E. Blanco, J. Parrondo, M.T. Stickland, T.J. Scanlon, Performance of a centrifugal pump running in inverse mode, in: Proceedings of the Institution of Mechanical Engineers, Part a: Journal of Power and Energy, 218 (4), 2004, pp. 265–271, <http://dx.doi.org/10.1243/0957650041200632>.
- [18] J. Fernandez, R. Barrio, E. Blanco, J. Parrondo, A. Marcos, Numerical investigation of a centrifugal pump running in reverse mode, in: Proceedings of the Institution of Mechanical Engineers, Part a: Journal of Power and Energy, 224 (3), 2010, pp. 373–381, <http://dx.doi.org/10.1243/09576509JPE757>.
- [19] H. Nautiyal, V. Varun, A. Kumar, S. Yadav, Experimental investigation of centrifugal pump working as turbine for small hydropower systems, *Energy Sci. Tech.* 1 (1) (2011) 79–86, <http://dx.doi.org/10.3968/g1293>.
- [20] A.J. Stepanoff, *Centrifugal and Axial Flow Pumps*, second ed., John Wiley & Sons, Inc, New York, USA, 1957, p. 276.
- [21] S.M. Childs, Convert pumps to turbine and recover HP, in: *Hydrocarbon Processing and Petroleum Rejner*, 41 (10), 1962, pp. 173–174.
- [22] B.M. McClaskey, J.A. Lundquist, *Hydraulic power recovery turbines*, ASME Conf. 76 (65) (1976).
- [23] K.R. Sharma, *Small Hydroelectric Project-use of Centrifugal Pumps as Turbines*, Kirloskar Electric Co, Bangalore, India, 1985.
- [24] C. Alatorre-Frenk, T.H. Thomas, The pumps as turbines approach to small hydropower, in: *World Congress on Renewable Energy*, 1990 (Reading).
- [25] S.S. Yang, S. Derakhshan, F.Y. Kong, Theoretical, numerical and experimental prediction of pump as turbine, *Renew. Energy* 48 (2012) 507–513, <http://dx.doi.org/10.1016/j.renene.2012.06.002>.
- [26] J.W. Hancock, Centrifugal pump or water turbine, *Pipe Line News* (1963) 25–27.
- [27] E. Schmiedl, *Serien-Kreiselpumpen im Turbinenbetrieb, Pumpentagung* Karlsruhe, 1988. A6.
- [28] A. Williams, The turbine performance of centrifugal pumps: a comparison of prediction methods, *Proc. Inst. Mech. Eng. A. J. Power Energy* 208 (1) (1994) 59–66.
- [29] A. Williams, *Pumps As Turbines Users Guide*, International Technology Publications, London, England, 1995.
- [30] R.P. Saini, N. Ahmad, Selection procedure of centrifugal pump to use as turbine for small hydro power stations, in: *Proceedings of National Seminar on Small HydroPower Project-civsem'93*, 1993, pp. 64–71. Coimbatore, India.
- [31] K.M. Grover, Conversion of Pumps to Turbines, GSA Inter Corp, Katonah, New York, 1980.
- [32] P. Hergt, The influence of the volute casing on the position of the best efficiency point, in: Proc. 11<sup>th</sup> IAHR Sympo, Amsterdam, Holland 3, 1982, p. 69.
- [33] S. Joshi, A.G.L. Holloway, L. Chang, Selecting a high specific speed pump for low head hydro-electric power generation, in: *Canadian Conference on Electrical and Computer Engineering (CCECE)*, 2005, pp. 603–606.
- [34] M. Gantar, Propeller pump running as turbines, in: *Conference on Hydraulic Machinery*, Ljubljana, Slovenia, 1988, pp. 237–248.
- [35] M. Amelio, S. Barbarelli, N.M. Scornaienchi, *Caratterizzazione al banco prova di pompe centrifughe utilizzate come turbine*, in: *Proceedings of 55<sup>th</sup> National Congress ATI*, Bari – Potenza, Italy, 2000 (in Italian).
- [36] S. Derakhshan, A. Nourbakhsh, Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds, *Exp. Therm. Fluid. Sci.* 32 (3) (2008) 800–807, <http://dx.doi.org/10.1016/j.expthermflusci.2007.10.004>.
- [37] S. Derakhshan, A. Nourbakhsh, Theoretical, numerical and experimental investigation of centrifugal pumps in reverse operation, *Exp. Therm. Fluid Sci.* 32 (8) (2008) 1620–1627, <http://dx.doi.org/10.1016/j.expthermflusci.2008.05.004>.
- [38] P. Singh, F. Nestmann, An optimization routine on a prediction and selection model for the turbine operation of centrifugal pumps, *Exp. Therm. Fluid Sci.* 34 (2) (2010) 152–164, <http://dx.doi.org/10.1016/j.expthermflusci.2009.10.004>.
- [39] S. Rawal, J. Kshirsagar, Numerical simulation on a pump operating in a turbine mode, in: *Proceedings of the 23<sup>rd</sup> International Pump Users Symposium*, Houston, Texas, USA, 2007, pp. 21–27.
- [40] O. Fecarotta, A. Caravetta, H. Ramos, CFD and comparisons for a pump as turbine: mesh reliability and performance concerns, *Intern. J. Ener. Environ.* 2 (1) (2011) 39–48.
- [41] A. Bozorgi, E. Javidpour, A. Riasi, A. Nourbakhsh, Numerical and experimental study of using axial pump as turbine in pico hydropower plants, *Renew. Energy* 53 (2013) 258–264, <http://dx.doi.org/10.1016/j.renene.2012.11.016>.
- [42] D. Cohrs, *Investigations of a multi-stage pump operated as a turbine, Untersuchungen an einer mehrstufigen rückwärtslaufenden kreiselpumpe im turbinenbetrieb*, verlag und bildarchiv, German, 1977, pp. 8–41. W.H. Faragallah.
- [43] V.S. Lobanoff, R.R. Ross, *Centrifugal Pumps: Design and Applications*, second ed., Gulf publishing company, Texas, 1992.
- [44] M. Suarda, N. Suarnadwipa, W.B. Adnyana, Experimental work on the modification of impeller tips of a centrifugal pump as a turbine, in: *Proceedings of the 2nd Joint International Conference on Sustainable Energy and Environment (SEE 2006)*, Bangkok, Thailand, 2006.
- [45] S. Derakhshan, A. Nourbakhsh, B. Mohammadi, Efficiency improvement of centrifugal reverse pumps, *J. Fluid Eng-T ASME* 131 (2009) 21103–21107, <http://dx.doi.org/10.1115/1.3059700>.
- [46] P. Singh, F. Nestmann, Internal hydraulic analysis of impeller rounding in centrifugal pumps as turbines, *Exp. Therm. Fluid Sci.* 35 (1) (2011) 121–134, <http://dx.doi.org/10.1016/j.expthermflusci.2010.08.013>.
- [47] S.S. Yang, C. Wang, K. Chen, X. Yuan, Research on blade thickness influencing pump as turbine, *Adv. Mech. Eng.* (2014) 1–8, <http://dx.doi.org/10.1155/2014/190530>.
- [48] S.V. Jain, A. Swarnkar, K. Motwani, R.N. Patel, Effects of impeller diameter and rotational speed on performance of pump running in turbine mode, *Energy Convers. Manage.* 89 (2015) 808–824, <http://dx.doi.org/10.1016/j.enconman.2014.10.036>.
- [49] N. Fontana, M. Giugni, L. Gielmo, G. Marini, Real time control of a prototype for pressure regulation and energy production in water distribution networks, *J. Water Res. Plan-ASCE* 142 (7) (2016), [http://dx.doi.org/10.1061/\(ASCE\)WR.1943-5452.0000651.04016015](http://dx.doi.org/10.1061/(ASCE)WR.1943-5452.0000651.04016015).
- [50] A. Dannier, A. Del Pizzo, M. Giugni, N. Fontana, G. Marini, D. Proto, Efficiency evaluation of a micro-generation system for energy recovery in water distribution networks, in: 5th International Conference on Clean Electrical Power. Renewable Energy Resources Impact (ICCEP15), Taormina, Italy, 2015, pp. 689–694, <http://dx.doi.org/10.1109/ICCEP.2015.7177566>.
- [51] R.B. Abernethy, B.D. Powell, D.L. Colbert, D.G. Sanders, J.W. Thompson Jr., *Handbook Uncertainty in Gas Turbine Measurements*, USAF AEDC-TR, 1973, pp. 73–75. IS&N: 87664–483-3.
- [52] R.J. Moffat, Contributions to the theory of single-sample uncertainty analysis, *J. Fluid Eng-ASME* 104 (1982) 250–260.
- [53] A.J. Stepanoff, *Centrifugal and Axial Flow Pumps: Theory, Design and Application*, John Wiley & Sons, Inc, New York, USA, 1948.