New test-rig for micro hydropower turbomachines

João Delgado[1,2,3], João P. Ferreira[2] Dídia I.C. Covas[2], François Avellan[3]

[2] CERIS, Instituto Superior Técnico, Universidade de Lisboa, Av. Rovisco Pais 1, 1049-001 Lisboa, Portugal

[3] Laboratory for Hydraulic Machines, École Polytechnique Fédérale de Lausanne, Avenue de Cour 33 Bis CH-1007 Lausanne, Switzerland

[1]*joao.borga.delgado@tecnico.ulisboa.pt, joao.delgado@epfl.ch*

ABSTRACT

Energy recovery in water supply systems has been pointed as an interesting opportunity. The available powers for recovery are usually small and, therefore, there is the need to develop cost-effective solutions. Pumps running as turbines are pointed as a prospective solution for this purpose. However, there are still some challenges to address, namely the prediction of turbine mode performance. For this purpose, a new test-rig was assembled to test different pumps in turbine mode and operational constraints. This paper shows the details of the developed experimental facility, data acquisition and control system. Preliminary experimental results are presented for the efficiency hillchart, as well as the influence of backpressure in the pressure fluctuations downstream the pump. The development of this test-rig aims at collecting reliable data for developing formulations to accurately predict the pump as turbine performance.

Keywords: Pumps running as turbines, experimental facility, efficiency measurement

1 INTRODUCTION

Pressurised water supply systems (WSS) are infrastructures with significant potential for recovering energy [1], as they supply water many times with excessive pressure. This excessive energy is usually dissipated in [2]: (i) pressure reducing valves; (ii) control valves at the inlet of storage tanks; (iii) break pressure tanks. Powers available for recovery are usually below 250 kW [2]. Additionally, the operation of the energy recovery units needs to comply with the safe water supply to the population [3]. To sum up, there is an interesting engineering challenge for developing technical solutions for efficiently recovering energy, while controlling pressure in WSS.

Pumps running as turbines (PATs) are pointed out as a cost-effective solution for this purpose, as they have a low initial cost, are available for a wide range of head/discharge and have easy maintenance and installation [4]. However, some challenges still need to be addressed. Firstly, PATs have a low efficiency under fluctuating discharge, which is a characteristic of WSS due to the population variable water consumption [5]. Secondly, the difficulty of accurately predict the full characteristics in turbine mode [6], which are of the utmost importance for the design and economic analysis of PAT hydropower recovery schemes.

The strong efficiency decrease in off-design operation is caused by the fixed geometry and absence of inlet flow control [7]. Among several solutions [8], [9], variable speed operation [10] is proposed to reduce the limited efficiency under variable discharge. The PAT turbine mode performance characteristics are not provided by pump suppliers. Singh and Nestmann [11] proposed a consolidated model based on the “specific speed-specific diameter” plots [12] with very good results to estimate the best efficiency point (BEP) in turbine mode. Derakhshan and Nourbakhsh

[13] presented empirical equations to describe the entire domain of operation of PATs based on a regression analysis of an extensive set of data experimentally collected. Despite this formulation seems to be accepted by other authors [14], it must be used with care, as the formulation outputs strongly depend on the conditions of the experimental tests performed and centrifugal pumps used.

A new experimental facility for testing pumps running as turbines was developed at the Laboratory of Hydraulics and Environment from Instituto Superior Técnico, Universidade de Lisboa. This new facility aims at the collection of reliable experimental data to develop formulations to describe the variable speed performance of PATs. Additionally, the experimental set-up allows investigating other subjects, such as the operating conditions outside normal operation range (e.g., sudden load rejection) and influence of cavitation in the PAT performance.

This paper aims, firstly, at the description of the test-rig design, measurement and control equipment, as well as the description of the working philosophy of the experimental tests performed. Secondly, it presents preliminary experimental results, showing the test-rig scope of application. Finally, results are discussed and perspectives for future work are outlined.

2 EXPERIMENTAL SET-UP

2.1 Overview

Experimental methods allow the accurate determination of hydraulic turbomachines performance. Standardised documents such as the IEC60193 [15] outline recommendations regarding measurement processes and test-rig characteristics to measure hydraulic (e.g., discharge, pressure, water level) and mechanical (torque, speed) parameters. Several experimental can be carried out to determine the turbomachines performance, such as the: (i) Efficiency hillchart characterization; (ii) pressure fluctuations; (iii) extended operation range (i.e., four-quadrant operation).

2.2 Test-rig

A new experimental facility (Figure 1) was developed at the Laboratory of Hydraulics and Environment at Instituto Superior Técnico, Universidade de Lisboa. The test-rig is composed of a closed-loop pressurised steel pipe system, with diameters ranging from 50 mm to 200 mm. The facility allows the flow circulation in two directions, for testing both pump and turbine modes. The test-rig was developed following the recommendations of the international standard IEC60193 [15]. The facility is supplied with water from a 40 m^3 storage tank.

(a) (b)

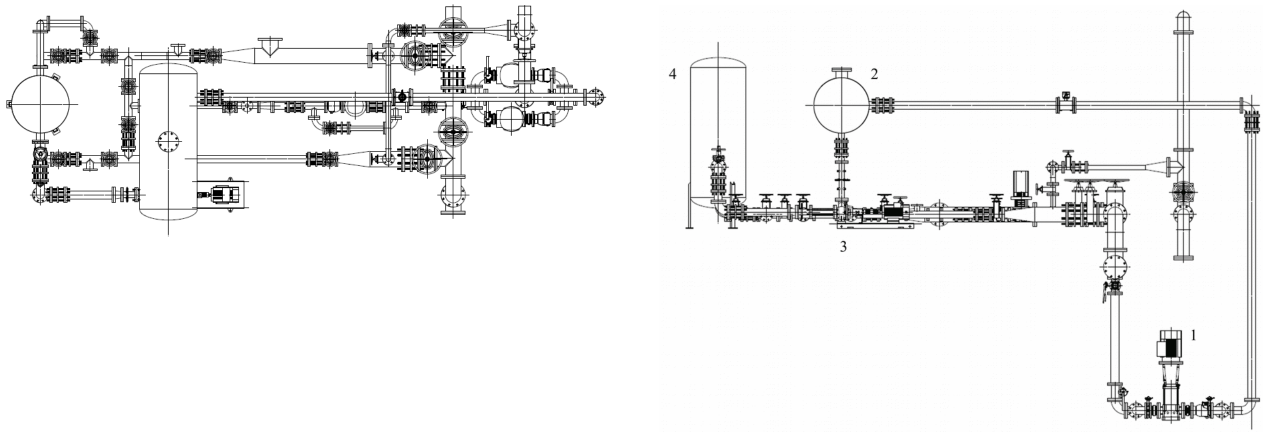

Figure 1. Experimental facility layout: (a) plan view; (b) side-view. Legend: 1. Supply pumps; 2. Upstream pressure vessel; 3. Test platform; 4. Downstream pressure vessel

The facility has two recirculating multistage centrifugal pumps (denominated onwards as supply pumps) connected in parallel to supply the circuit with hydraulic power. The pumps, with a rated power of 2x15 kW, deliver a maximum discharge and pressure of 2x85m^3/h and 8 bar, respectively. The supply pumps are equipped with variable speed drives to control the supplied hydraulic power.

Downstream the pressurised pumps there is a 1 m^3 pressure vessel, with the objective of stabilising the flow and to provide a uniform velocity profile at the inlet of the tested turbomachine. The pressure vessel is equipped with two valves for supplying and releasing pressurised air. The test platform, installed downstream this pressure vessel, allows the integration of different types and sizes of turbomachines. It is equipped with a 15 kW regenerative variable speed drive, which allows the turbomachine speed control between 300 min^{-1} and 3000 min^{-1}. The energy produced is directly injected into the power grid of the laboratory and used to partially supply the recirculating pumps.

Downstream the test platform there is a pressure vessel, which allows creating different setting levels using compressed air. Additionally, this vessel allows collecting air bubbles generated by the backpressure valve or turbomachine cavitation. The vessel discharges to three pipelines which are connected to the supply pumps. The pipelines can be used together or separately, depending on the head loss required in the experimental facility for a given experimental test.

2.3 Measurement equipment, data acquisition and control

The test-rig is equipped with state-of-the-art instrumentation for measuring hydraulic, mechanic and electrical parameters. The measurement equipment allows the measurement of the turbomachine efficiency with a systematic relative uncertainty [15] of 0.6%. The temperature θ is measured using a Novus PT100 temperature sensor. The discharge Q is acquired with an ABB electromagnetic flowmeter. Two differential pressures are measured using Emerson differential pressure transducers. One between the high-pressure section and low-pressure section Δp_1 (for characterising the head) and another between the low-pressure section and the turbomachine reference section Δp_2 (for characterising the setting level). Additionally, there are two WIKA gauge pressure sensors to measure upstream and downstream wall-pressure fluctuations, p_1 and p_2 respectively. Torque T is retrieved using a HBM torque transducer assembled on the turbomachine shaft and the rotating speed n is acquired using and IFM inductive sensor. Active and reactive supplied electric power, P_{sup} and Q_{sup} respectively, are acquired using a Lovato digital power analyser. Static manometers and process indicators are installed for visual indication to the test platform user. Table 1 presents the main characteristics of the measurement equipment.

Measurements are acquired using a data acquisition hardware (NI cDAQ9178) from National Instruments™, which is connected to a personal computer. Ten simultaneous measurements are carried out at a rate of 5 kHz. The data acquisition hardware is equipped for reading analogue signals in voltage (NI 9205) and current (NI 9203), and to produce voltage analogue outputs (NI 9269) for controlling the three variable speed drives. The data acquisition and control software of the test-rig was developed based on the LabVIEW™ 2014 Queued Message Handler architecture [16]. This architecture allowed to develop applications where multiple tasks occur in parallel. Therefore, it allows to simultaneously control the three variable speed drives, acquire, log, process and visualise data in real-time, without heavy computational resources.

Table 1. Measurement equipment characteristics

Variable	Output signal	Range	Precision
Q	4 mA – 20 mA	0 $m^3 \cdot h^{-1}$ – 160 $m^3 \cdot h^{-1}$	± 0.4%
Δp_1	4 mA – 20 mA	0.0 – 7.5 bar	± 0.035%
Δp_2	4 mA – 20 mA	-2.1 – 0.2 bar	± 0.035%
T	2 mA – 18 mA	-100 N·m – 100 N·m	± 0.3%
n	4 mA – 20 mA	0 Hz – 50 Hz	± 0.2%
θ	0 V – 10 V	0 ºC – 100 ºC	± 0.3%
P_{sup}	4 mA – 20 mA	-15 kW – 15 kW	± 0.5%
Q_{sup}	4 mA – 20 mA	-15 kVAr – 15 kVAr	± 2.0%
p_1	4 mA – 20 mA	0.0 bar – 6.0 bar	± 1.0%
p_2	4 mA – 20 mA	0.0 bar – 2.5 bar	± 1.0%

3 PRELIMINARY EXPERIMENTAL TESTS

The preliminary experimental tests were performed in a single-stage centrifugal pump with a unit specific speed, given by Eq. , n_q = 67.4. The pump has the following characteristics in pumping mode: rated discharge Q_R = 30.0 $l \cdot s^{-1}$; rated head H_R = 14.7 m; rated speed = 2920 min^{-1}; and a rated shaft power P_R = 5.5 kW. The pump has a high-pressure section diameter D_1 = 127 mm and a low-pressure section diameter D_{I} = 80 mm.

$$n_q = N_R \frac{Q_R^{\frac{1}{2}}}{H_R^{\frac{3}{4}}}$$

Where N_R = rated speed [min^{-1}]; Q_R = rated discharge [$m^3 \cdot s^{-1}$]; H_R = rated head [m].

The preliminary experimental tests consisted of the determination of the characteristic curves for pump and turbine mode and the measurement of downstream pressure fluctuations in turbine mode for different backpressure conditions. The efficiency in pump mode and turbine mode, η^P and η^T respectively, are obtained based on Eq.

$$\eta^P = \frac{\rho g Q H}{2\pi n T} ; \eta^T = \frac{2\pi n T}{\rho g Q H}$$

where ρ = water density [$kg \cdot m^{-3}$]; g = gravity acceleration [$m \cdot s^{-2}$]; Q = discharge [$m^3 \cdot s^{-1}$]; H = turbomachine head [m], given by Eq. [15]; n = rotating speed [s^{-1}]; T = shaft torque [N·m].

$$H = \frac{\Delta p_1}{\rho g} + \frac{C_1^2 - C_2^2}{2g}$$

Where C_1 and C_2 = flow velocity [m·s^{-1}], respectively at the high-pressure and low-pressure measurement section.

The characteristic curves in pump mode were obtained for constant speeds. At pump mode, the speeds were 2920 min^{-1}, 2400 min^{-1} and 1500 min^{-1}. At turbine mode, the speeds were 3000 min^{-1}, 2400 min^{-1} and 1500 min^{-1}. The constant speed of the pump was achieved by controlling the variable speed drive frequency. The head loss required for determining the entire characteristic curve in pump mode is produced by a control valve installed in the pipeline between the downstream pressure vessel and the by-pass of the supply pumps. The head at the inlet of the PAT was obtained by varying the frequency of the supply pumps. The downstream pressure fluctuation was measured for the maximum efficiency point of the PAT at 3000 min^{-1}. The fluctuations were measured for two backpressure values, which were controlled by setting the pressure of the downstream vessel with compressed air.

Figure 2 and Figure 3 present the Q-H curve and Q-η curves in pump and turbine modes, respectively. The discharge is considered positive for turbine mode and negative for pump mode [15]. The maximum efficiency for 3000 min^{-1} could not be measured, as the maximum discharge of the test-rig is 37.5 l·s^{-1}. These figures show that the pump operates at higher discharges and heads in turbine mode. Maximum efficiencies in pump and turbine () modes are 73.6% and 70.8%, respectively, both for 1500 min^{-1}. In both operating modes, the maximum efficiency for each speed is obtained at lower discharges. This is particularly relevant for recovering energy in WSS, as the PAT speed can be regulated, aiming to operate at the best efficiency. Additionally, it allows the recovery of energy for smaller discharges, where the operation in synchronous speed (3000min^{-1}) would not allow (e.g., for 20 l/s the η = 7.9% for 3000 min^{-1} and η = 70.4 % for 1500 min^{-1}).

(a) (b)

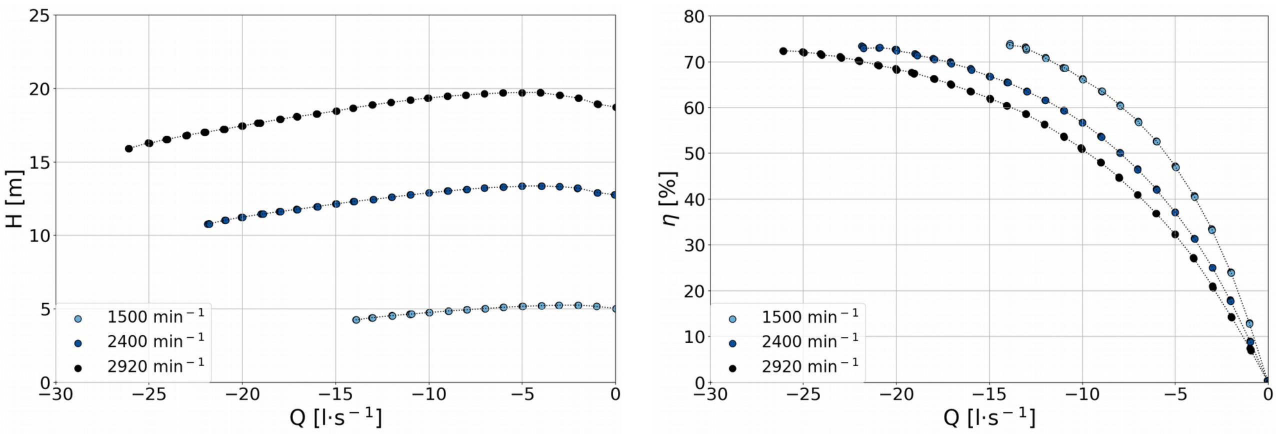

Figure 2. Pump mode performance: (a) Q-H curve; (b) Q-η curve

(a) (b)

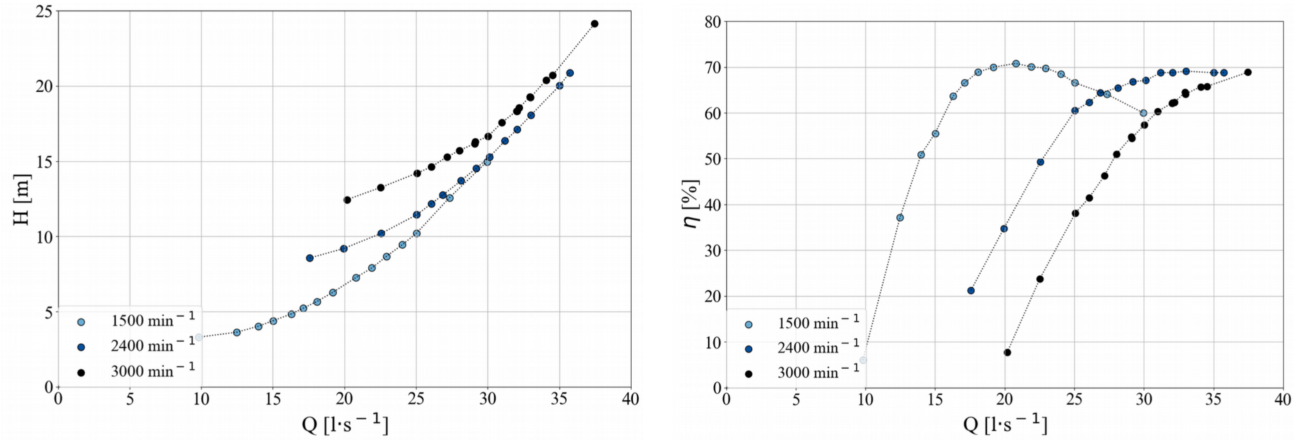

Figure 3. Turbine mode performance: (a) Q-H curve; (b) Q-η curve

Table 2. BEP for each rotational speed in pump and turbine mode

Mode of operation	N [min^{-1}]	Q [l·s^{-1}]	H [m]	η_{max} [%]
Pump	1500	13.9	4.2	73.6
	2400	21.8	10.8	73.1
	2920	26.6	15.8	73.4
Turbine	1500	20.8	7.3	70.8
	2400	32.1	17.1	69.1
	3000	37.5	24.1	68.9

Figure 4 presents the temporal variation of downstream static pressure coefficient C_P, given by Eq. , for different local cavitation coefficients χ_{nD} = 7.5 and χ_{nD} = 12.5, given by Eq. . These coefficients correspond, respectively, to an average backpressure $p_{\bar{2}}$ = 1.24 bar and $p_{\bar{2}}$ = 2.03 bar. Measurements were performed for the BEP at 3000 min^{-1} ().

$$C_P = \frac{p_2 - p_{\bar{2}}}{\rho g H}$$

$$\chi_{nD} = \frac{p_{\bar{2}} - p_v}{\rho n^2 D_{\bar{1}}^2}$$

Where p_2 = pressure at the low-pressure section [Pa]; $p_{\bar{2}}$ = average pressure at the low-pressure section [Pa]; p_v = vapour pressure [Pa]; $D_{\bar{1}}$ = outlet diameter of the impeller [m].

Figure 4 shows that the backpressure has a significant effect in the pressure fluctuations downstream the PAT. Table 3 presents the maximum and minimum values of static pressure coefficient and corresponding pressure variations, respectively to average pressure.

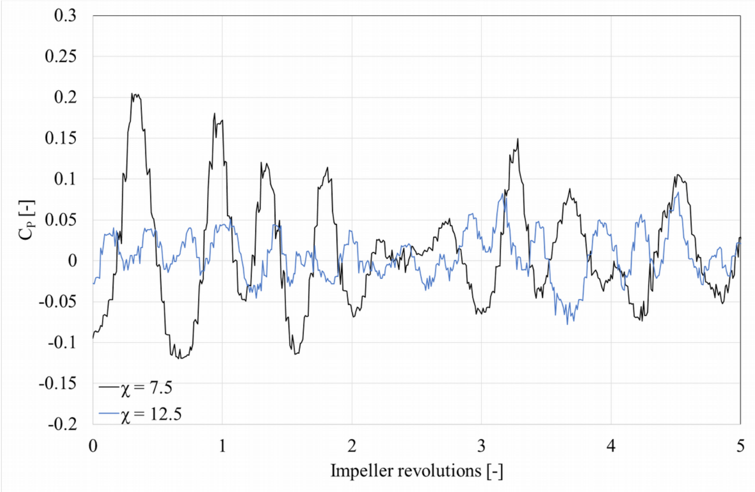

Figure 4. Temporal variation of static pressure coefficient fluctuation

Table 3. Maximum and minimum static pressure coefficient variation and corresponding pressure

χ_{nD} [-]	C_P [-]		$p_2 - p_{\bar{2}}$ [bar]	
	max	min	max	min
7.5	4.71	2.00	0.205	0.084
12.5	-2.75	-1.74	-0.125	-0.078

4 FINAL REMARKS AND PERSPECTIVES

This paper presents the technical details of a new test-rig for micro hydropower turbomachines up to 15 kW. The test-rig was designed according to the IEC 60193 recommendations [15]. The measurement equipment allows the determination of the turbomachine efficiency with a maximum systematic relative uncertainty of 0.6%. The data acquisition and control software allows to operate the facility and to observe the real-time the hydraulic, mechanic and electrical properties of the tested turbomachine, as the data processing of the ten measurements is performed in real-time.

This paper shows preliminary data collected on a single-stage centrifugal pump. Data obtained shows that the efficiencies in both operating modes are of the same order of magnitude. Also, it showed that variable speed operation is particularly relevant for recovering energy in WSS, as one can set the PAT to operate at maximum efficiency under fluctuating discharge. It also showed that the backpressure conditions strongly influence the pressure fluctuations downstream the PAT. This is relevant, as the integration of PATs in WSS must respect the safe and stable operation of WSS.

Additional research is being carried out at the moment. Several PATs with different unit specific speeds are being tested. Apart from the experimental tests shown, research is being developed on the determination of the extended operating range (i.e., four-quadrants operation), cavitation mapping in the PAT operating domain, measurement of pressure and torque fluctuations for different operating points and determination of runaway conditions.

Acknowledgements

The authors acknowledge Portuguese national funding agency for science, research and technology (FCT) and the Laboratory for Hydraulic Machines for funding João Delgado PhD grant, under the joint doctoral programme H2Doc (PD/00206/2012) between Instituto Superior Técnico (IST) and

École Polytechnique Fédérale de Lausanne (EPFL). The authors gratefully acknowledge European Commission's LIFE Programme for funding LIFE Smart Water Supply System project (LIFE14 ENV/PT/000508) for partially funding the test-rig.

References

[1] A. McNabola, P. Coughlan, and A. P. Williams, "Energy recovery in the water industry: an assessment of the potential of micro-hydropower," *Water Environ. J.*, vol. 28, no. 2, pp. 294–304, Jun. 2014.

[2] A. McNabola, P. Coughlan, L. Corcoran, C. Power, A. Prysor Williams, I. Harris, J. Gallagher, and D. Styles, "Energy recovery in the water industry using micro-hydropower: an opportunity to improve sustainability," *Water Policy*, vol. 16, no. 1, p. 168, Feb. 2014.

[3] ESHA - European Small Hydropower Association, "Energy recovery in existing infrastructures with small hydropower plants," Retrieved from http://www.esha.be/fileadmin/esha_files/ documents/SHAPES/Multipurpose%20schemes %20brochure %20SHAPES.pdf, 2010.

[4] A. A. Williams, N. P. A. Smith, C. Bird, and M. Howard, "Pumps as Turbines and Induction Motors as Generators for Energy Recovery in Water Supply Systems," *Water Environ. J.*, vol. 12, no. 3, pp. 175–178, Jun. 1998.

[5] D. Loureiro, A. Mamade, M. Cabral, C. Amado, and D. Covas, "A Comprehensive Approach for Spatial and Temporal Water Demand Profiling to Improve Management in Network Areas," *Water Resour. Manag.*, vol. 30, no. 10, pp. 3443–3457, Aug. 2016.

[6] S. V. Jain and R. N. Patel, "Investigations on pump running in turbine mode: A review of the state-of-the-art," *Renew. Sustain. Energy Rev.*, vol. 30, pp. 841–868, Feb. 2014.

[7] D. R. Giosio, A. D. Henderson, J. M. Walker, P. A. Brandner, J. E. Sargison, and P. Gautam, "Design and performance evaluation of a pump-as-turbine micro-hydro test facility with incorporated inlet flow control," *Renew. Energy*, vol. 78, pp. 1–6, 2015.

[8] B. Orchard and S. Klos, "Pumps as turbines for water industry," *World Pumps*, vol. 2009, no. 8, pp. 22–23, Aug. 2009.

[9] A. Carravetta, G. Del Giudice, O. Fecarotta, and H. M. Ramos, "Energy Production in Water Distribution Networks: A PAT Design Strategy," *Water Resour. Manag.*, vol. 26, no. 13, pp. 3947–3959, Oct. 2012.

[10] A. Carravetta, G. del Giudice, O. Fecarotta, and H. Ramos, "PAT Design Strategy for Energy Recovery in Water Distribution Networks by Electrical Regulation," *Energies*, vol. 6, no. 1, pp. 411–424, Jan. 2013.

[11] P. Singh and F. Nestmann, "A Consolidated Model for the Turbine Operation of Centrifugal Pumps," *J. Eng. Gas Turbines Power*, vol. 133, no. 6, p. 63002, 2011.

[12] O. Cordier, "Aehnlichkeitsbedingungen für Strömungsmaschinen," *Brennst.-Waerme-Kraft*, vol. 5, no. 10, pp. 337–340, 1953.

[13] S. Derakhshan and A. Nourbakhsh, "Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds," *Exp. Therm. Fluid Sci.*, vol. 32, no. 3, pp. 800–807, Jan. 2008.

[14] F. Pugliese, F. De Paola, N. Fontana, M. Giugni, and G. Marini, "Experimental characterization of two Pumps As Turbines for hydropower generation," *Renew. Energy*, vol. 99, pp. 180–187, Dec. 2016.

[15] International Electrotechnical Commission, "Hydraulic turbines, storage pumps and pump-turbines - Model acceptance tests," *Int. Stand. IEC 60193, 2nd ed*, 1999.

[16] National Instruments, "Queued Message Handler Template documentation," 2017. [Online]. Available: http://www.ni.com/tutorial/53391/en/. [Accessed: 29-Jun-2017].