

## Selection, control and techno-economic feasibility of Pumps as Turbines in Water Distribution Networks



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### ABSTRACT

Water Distribution Networks (WDNs) are becoming an attractive area of application for small hydropower, which can contribute to the development of distributed energy generation from renewable sources. Indeed, WDNs experience considerable water leakages due to their age and water management authorities often divide the WDNs by inserting Pressure Reducing Valves (PRVs), which waste a potentially recoverable hydraulic head. The replacement of PRVs with Pump as Turbines (PaTs) can be considered as an economically feasible solution to achieve both an effective pressure control and a throttling energy recovery. The selection, installation and control strategy of a PaT in a WDN must consider the variability of the pressure and the flow rate demand. Starting from the evaluation of the available head and the water demand, in this work, we describe a methodology to select from pump catalogues the most suitable PaT and the best control criteria for a specific WDN. Then, knowing the pump geometry, it is possible to predict the characteristic curve of the pump operating as a turbine by using a 1-D performance prediction model. The WDN of a town in the Apulia region (Southern Italy) has been used as a case study. Finally, a techno-economic evaluation has been carried out by considering both economic and environmental benefits.

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### 1. Introduction

In the current world energy scenario, the struggle against the anthropic climate changes and the need of access to clean energy represent two of the seventeen Sustainable Development Goals (SDG) planned by the United Nations in the 2030 development agenda [1]. In order to be achieved, these goals require a strong synergy between technical, economic, political and social strategies. The International Renewable Energy Agency (IRENA) affirmed in its latest report that by 2050 electricity could become the central energy carrier, growing from a 20% share of the final consumption up to an almost 50% share. Considering the e-mobility growth, the primary drivers for this increasing electricity demand will be the world vehicle fleet with over 1 billion electric cars and the

emergence of renewable hydrogen. Overall, renewable energies could supply two-thirds of the final energy consumption [2]. The International Energy Agency (IEA) has recently published an outlook to 2023 about renewable energies, which claims that the share of renewable energies is expected to grow by one-fifth in the next five years reaching the 12.4% in 2023. In these five years, renewables are expected to meet more than 70% of the global electricity generation growth, led by solar photovoltaic (PV) and followed by wind, hydropower and bioenergy [3].

Among clean energy sources, hydropower shows a high level of technological maturity, when considering the large hydropower. Instead, small hydropower still has ample margin for improvement. Indeed, hydropower shows an important contribution in today's electricity mix, meeting 16% of the global electricity demand by 2023, followed by wind (6%), solar PV (4%), and bioenergy (3%). Furthermore, hydropower plays a key role in the grid stabilization being able to quickly respond to rapid load variations, especially now that the shares of variable renewable electricity sources – primarily wind power and solar photovoltaic (PV) – are increasing considerably. Nowadays, small and mini hydro plants are becoming

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## Nomenclature

### Symbols

$C_E$	Hydraulic Energy Harvesting Coefficient [–]
$f$	Frequency [Hz]
$h$	Head factor [–]
$H$	Head [m]
$n$	Rotational speed [rpm]
$n_q$	Specific speed number [rpm, $m^3/h$ , m]
$n_s$	Synchronous speed [rpm]
$p$	Number of magnetic poles [–]
$P$	Power [kW]
$q$	Flow rate factor [–]
$Q$	Flow rate [ $m^3/h$ ]
$T$	Torque [Nm]

### Greek letters

$\eta$	Efficiency [–]
$\rho$	Density [ $kg/m^3$ ]
$\sigma$	Slip factor [–]

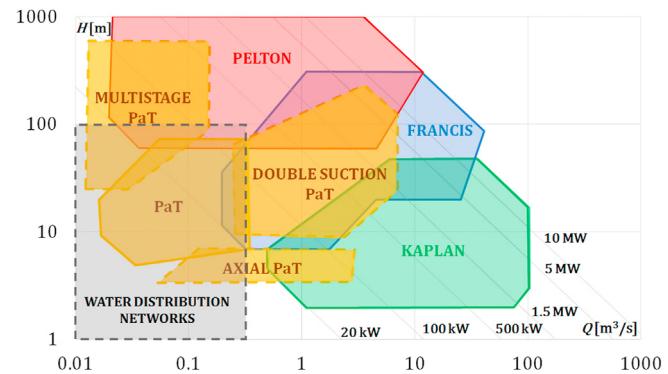
### Abbreviations

B/C	Benefit-Cost ratio
BEP	Best Efficiency Point
DMA	District Metering Area
IRR	Internal Rate of Return
NPV	Net Present Value
PaT	Pump as Turbine
PRV	Pressure Reduction Valve
ROI	Return of Investment
SCADA	Supervisory Control And Data Acquisition
SHP	Small Hydropower
WDN	Water Distribution Network

more and more tempting because of their technical feasibility and cost effectiveness, having a very low impact on the environment [4]. According to the latest World Small Hydropower Development Report by the United Nations [5], the global installed Small Hydropower (SHP) capacity was estimated at 78 GW in 2016, with an increase of approximately the 4% compared to 2013 data. Indeed, approximately 36% of the total global SHP potential has been developed as of 2016. Europe presents the highest SHP development rate, with nearly 48% of the overall potential already installed.

Water Distribution Networks (WDNs) are becoming an attractive area of application for mini hydro plants. Indeed, WDNs experience considerable water leakages due to their age and hence water management authorities often segment the WDNs into District Metering Areas (DMAs) by inserting Pressure Reduction Valves (PRVs). The optimal segmentation of a WDN consists in a multi-objective problem, which is characterized by many parameters (e.g., topology of the network, water demand allocation, pressure constraints, leakages, unsupplied demand). Indeed, different approaches have been used in the literature to construct DMAs, mainly based on multi-objective design strategies in which both technical and economic issues are considered [6–12]. Moreover, techniques based on genetic algorithms have been used to find the optimal DMAs according to specific objective functions. These algorithms are addressed to minimize the network pressure and improve the water quality (for instance, by reducing the water travel time and avoiding chemical and biological issues) [13–17]. An alternative approach to genetic algorithms is to divide the WDN into smaller areas via the method of Geometric Partitioning, which is based on the Recursive Coordinate Bisection (RCB) [18]. The benefits of DMAs have been highlighted by many works, in which these approaches have been applied to case studies [19–22].

Actually, PRVs reduce the network pressure and hence the leakages but at the same time wasting a potentially recoverable hydraulic energy. In recent years, the idea of replacing PRVs with hydraulic turbines has been proving successful, thanks to the activities performed by the scientific community to achieve both a control of the pressure and an energy recovery in WDNs [23–32]. The main challenge is to find a cost-effective small turbine (with power generally lower than 500 kW), suitable in terms of flow rates and available water head for this kind of application (see Fig. 1). PaTs, i.e., conventional pumps used in reverse mode, are playing a leading role in this field, as being an economic alternative to conventional hydraulic turbines, which need to be designed *ad hoc* for the recovery of throttling energy. Applying the concept of “think



**Fig. 1.** Typical operating ranges of hydraulic turbines in Small Hydropower. Adopted from Ref. [34].

*globally, act locally*”, these mini and small hydro power plants could either help towards creating smart grids, or improve the existing ones, for the future electric energy demand.

Despite promising developments in terms of scientific research, currently the PaT share in the hydro turbine market is negligible. This is partially due to the lack of knowledge or interest in the topic by pump manufacturers and hydropower consulting firms, and partially to the several technical challenges yet to be addressed regarding the PaT design and operation. Indeed, the current Technology Readiness Level (TRL, a method for estimating the maturity of technologies developed by NASA) of PaTs is estimated at TRL-4 due to the limited knowledge on the design and operation characteristics of pumps when used as turbines [33]. However, an appropriate approach to this problem must account for the incontrovertible environmental benefit that follows from the conversion into electric energy of the hydraulic energy otherwise dissipated by the valves. Moreover, techno-economic assessments are important to evaluate the economic benefits from PaT energy production. Although the profit margin could be small in some cases, it must be considered that the system exploits hydraulic energy otherwise lost.

Several economic analyses have been performed on the existing WDNs highlighting the advantages of a PaT instead of a conventional turbine for this kind of application. A PaT installation is usually characterized by a payback period of about 2–3 years,

although PaTs are characterized by a lower efficiency than conventional turbines since the last are not normally equipped with variable inlet guide vanes (IGV). A PaT installation can involve an annual income ranging from 25,000 €/year to 50,000 €/year or even more, depending on the desired level of water saving strictly related to the pressure reduction in the WDN [35–38]. Different works propose different methods for economic assessments. For instance, Kramer et al. [39] proposed a new cost classification scheme in order to enable a systematic and generally valid estimation of investment costs for energy recovery power plants. Novara et al. [40] developed a model to allow designers to predict the PaT and generator costs from the nominal flow rate and the available hydraulic head. García et al. [35] proposed a model to determine the most cost-effective solution between PaT and PRV, in terms of Net Present Value (NPV). Chacón et al. [41] used a statistical methodology to select PaTs whose operating point determines the lowest payback period. Moreover, PaTs are employed not only in the hydraulic fields, but also in the process engineering, as described by Stefanizzi et al. [42], Renzi et al. [43] and Chen et al. [44].

The usefulness of PaTs has been confirmed by many case studies published in the technical literature. Muhammetoglu et al. [45] studied a full-scale PaT application into a WDN in Antalya (Turkey) for energy production and pressure reduction; both Rossi et al. [46] and Alberizzi et al. [47] evaluated the potential of PaT installations in the WDN of different towns in the Northern Italy; Balacco et al. [48] proposed to replace an existing PRV with a PaT to supply a charging station for electric vehicles in Casamassima (Apulia region, Southern Italy); Stefanizzi et al. [49] performed a preliminary assessment of a PaT for the same WDN in Casamassima using experimental characteristic curves of a KSB PaT and evaluating different installation layouts. In addition to this work, another preliminary assessment of a PaT installation has been carried out for a larger town (always in Apulia Region, Southern Italy) [50]. Small-scale hydropower systems are also becoming increasingly successful options for hydropower generation in small districts and remote areas, as demonstrated by Balkhair et al. [51], for a case study in Pakistan, or by Arriaga [52], who proposed the installation of a PaT for rural electrification in Lao People's Democratic Republic. As a final example, aware that there are many other works in the literature, Morabito et al. [34] proposed the application of a PaT for micro Pumped Hydro Energy Storage, integrated in a smart grid.

Concerning PaT installations, Lima et al. [53] and Polák [54] underlined that, due to dynamic operations throughout the day in WDNs, a PaT should work under varying operating conditions (head vs. flow rate), i.e., mutable hydropower potential, which does not permit to define a unique operating point for a PaT. Additionally, pump manufacturers do not provide catalogue curves of their pumps running as turbines since this would require the pump to be tested also in turbine mode operation, hence increasing significantly their final cost. This shortage of experimental data has involved a lack of knowledge on PaT performance before their installations. For this reason, a significant effort has been made by several authors in the prediction of PaT performance not only with theoretical approaches, but also with experimental campaigns of pumps tested in reverse mode. Indeed, many experimental activities have been reported by several authors as Barbarelli et al. [55], Derakhshan et al. [56], Nautiyal et al. [57], Pugliese et al. [58], Singh et al. [59], Shi et al. [60], Yang et al. [61], Tan et al. [62], Carravetta et al. [63], Rossi et al. [64], Höller et al. [65] and Cavazzini et al. [66]. The common objective was to develop models, able to predict the Best Efficiency Point (BEP) of the PaT by knowing the BEP of the pump, retrieved from catalogues. The relations between the parameters at both BEPs, are usually given in terms of head,  $h$ , and

flow rate,  $q$ , conversion factors, as stated in Eqs (1) and (2):

$$h = H_{BEP,T}/H_{BEP,P} \quad (1)$$

$$q = Q_{BEP,T}/Q_{BEP,P} \quad (2)$$

Actually, there are mainly two approaches: for instance, Williams [67], Sharma [68], Alatorre [69] and Yang et al. [61] used methods based on the pump efficiency at BEP,  $\eta_{BEP,P}$ , whereas others like Derakhshan et al. [56], Singh et al. [59], Nautiyal et al. [57], Tan et al. [62], Barbarelli et al. [55] used methods based on the pump specific speed number,  $n_{q,P}$ . However, these investigations took place in periods when few experimental data of pumps tested in turbine mode were available in the literature. For this reason, these first studies were useful to create a basis for the next models. Indeed, these prediction models are extremely simple even though seldom they consider off-design conditions (part and over loads). Thanks to the increase number of pumps tested in reverse mode, some other authors have proposed prediction methods based on artificial neural networks [70,71]. Theoretical models for off-design performance prediction are quite comprehensive, but they are difficult to be applied in practice because they necessitate very detailed geometric information, which only manufacturers usually know. For this reason, some researchers, such as Gülich [72], Barbarelli et al. [73], Liu et al. [74], developed theoretical approaches in order to predict PaT performance by taking into account velocity triangles and hydraulic losses models on simplified geometries rather than using statistical and experimental correlations. Venturini et al. [75–77] presented a physics-based model to predict PaT performance curve over the entire range with an optimization procedure to identify model parameters instead of detailed geometrical data.

In the last few years, further investigations have been performed regarding the study of the flow field inside low specific speed PaT runners focusing on the slip phenomenon at their outlets [78,79]. Arani et al. [80] investigated the effect of the volute tongue geometry on PaT performance. For high flow rates, a novel impeller for double suction pumps has been presented and numerically investigated in reverse mode pointing out its higher efficiency in a wide range of flow rates compared to a conventional back-to-back double suction geometry [81].

Under this framework, we propose a methodology to determine the most suitable PaT for a specific WDN, starting from the analysis of the pressure and flow rate patterns. Indeed, starting from these data, it is possible to evaluate the specific speed range required by the site, in order to understand if the operating field is appropriate for PaTs instead of conventional turbines. This methodology is based on the evaluation of the PaT specific speed, hence the specific speed of the pump by means of empirical correlations and its selection from catalogues. Then, once the main geometrical parameters of the pump are known, a 1-D performance prediction model is applied in order to obtain the characteristic curve of the pump operating as turbine.

Initially, the reader is guided through a description of the control methods to be applied to PaT working in off-design conditions (section 2). Indeed, advantages and drawbacks of choosing either a constant or a variable rotational speed control are taken into account with an overview on the coupling of the machines with a three-phase asynchronous motors. In the second part (section 3), the work deals with the description of the proposed methodology for the selection of a PaT, starting from the evaluation of the specific speed required from the installation site. Afterward, the WDN of a town in the Apulia region (Southern Italy), here merely indicated as *Murgia network*, due to confidentiality issues, has been selected as a

case study, in order to apply the methodology and select a PaT useful for throttling energy recovery (section 4). In section 5, the work continues with the evaluation of the energy production by the PaT installation in order to highlight how the PaT application can achieve both pressure control and clean energy production. Finally, a techno-economic analysis has been carried out by considering both the economic and the environmental benefits that the PaT installation can bring to the community.

## 2. Control methods for PaT installed in a WDN

As mentioned in the introduction and shown in Fig. 1, the interest in PaT stems from their capability to operate under operating conditions that conventional hydraulic turbines can barely cover (i.e., flow rates ranging from 10 up to 1800 m<sup>3</sup>/h and head ranging from 10 up to 100 m). However, the installation and the control strategy of a PaT must consider the variability of both pressure and flow rate, which continuously change in a WDN. Usually, thanks to the installation of a PRV in a conventional WDN, the backpressure remains almost constant with only two different values (a higher value during the daytime and a lower value during the night), as the user demand must be guaranteed, whereas the upstream pressure depends on the flow rate pattern. So, the PaT must comply with the same requirements of the PRV. Different control methods can be applied in order to guarantee the off-design conditions of a PaT. As depicted in Fig. 2, in a typical installation scheme the PaT is installed in series downstream of a control valve (PRV #1) and in parallel to a bypass line with its own PRV (PRV #2). Moreover, it is possible to consider a variable rotational speed control on the PaT by means of a frequency inverter.

### 2.1. Throttle and bypass control

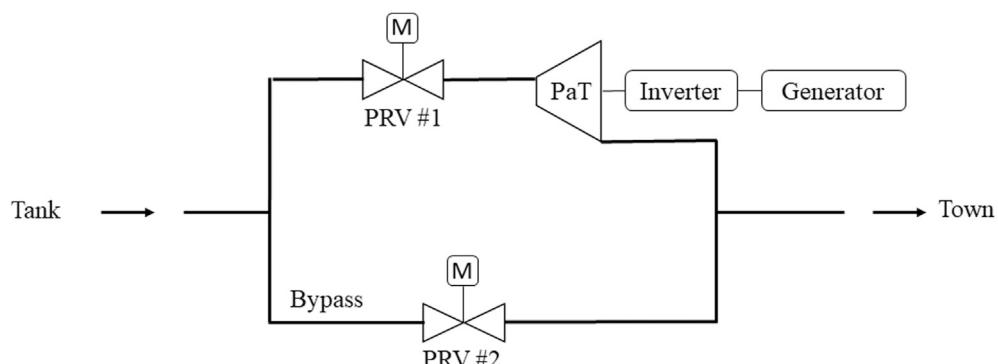
A PaT, being a conventional pump used in reverse mode, is not equipped with adjustable guide vanes to accommodate different flow conditions. For this reason, at constant PaT rotational speed, the control can be performed by means of either a valve, when we want to reduce the flow rate in the WDN, or a bypass, when we need to increase the flow rate in the WDN. The addition of a speed control can help in keeping as high as possible the PaT efficiency and widen the system rangeability. Fig. 3 presents a head,  $H$ , versus volume flow rate,  $Q$ , diagram on which the generic characteristic curve of a PaT at constant rotational speed (black curve) is shown. As expected, in order to allow a greater flow rate to flow through the PaT a higher head is needed. In the same figure, the site characteristics is shown (red curve). In this case, the increase of the flow rate determines a reduction of the available head due to losses. Then, point A represents the operating point when PRV #1 is

completely open (zero head drop). In order to delimit the PaT operating range, two other characteristic curves of the machine need to be taken into account: the runaway ( $T = 0 \text{ Nm}$ ) and the resistance ( $n = 0 \text{ rpm}$ ) curves. The optimal matching occurs when the site characteristic intersects the PaT characteristic at its best efficiency point (here point A). Fig. 3 shows also the characteristic curves of both the PRVs installed in series with the PaT, i.e., PRV #1 (green curve) and PRV #2 (blue curve). Clearly these curves correspond to specific valve closures. For instance, a partial closure of a PRV will determine the increase of its characteristic curve slope.

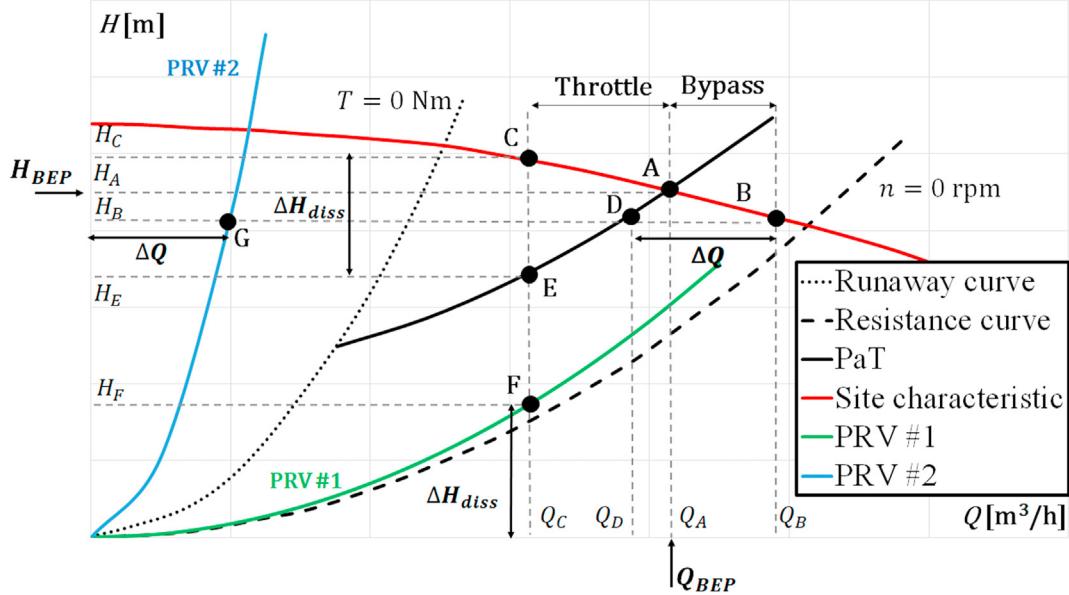
Under the assumptions that the PaT works at constant rotational speed and it is not provided with adjustable guide vanes, its characteristic curve is fixed. This means that we need to act on either PRV #1 or PRV #2 if the WDN requires a flow rate change with respect to  $Q_A$ . When a flow rate lower than that of the design point (e.g.,  $Q_C < Q_A$ ) is required, the control valve PRV #1 should be partially closed. In this case, the flow rate becomes  $Q_C$ , the PaT operating point becomes E, and the PRV #1 operating point becomes F. The head loss of the PRV #1,  $H_F$ , equals the head difference,  $H_C - H_E$ , moving the system operating point in C. In this case the excess head, available on site, is actually dissipated. Following this approach,  $P_F = \rho g H_F Q_C$  is the wasted and no more exploitable power. Whereas, when a flow rate higher than that of the design point (e.g.,  $Q_B > Q_A$ ) is required, the control valve PRV #1 should be fully opened and the valve PRV #2 on the bypass should be partially opened. In this case, the flow rate of the WDN becomes  $Q_B$ , the PaT operating point becomes D, and PRV #2 operating point becomes G. In this case, the head loss in the PRV #2,  $H_G$ , equals the head required by the PaT,  $H_D$ , and the sum of the PaT flow rate,  $Q_D$ , and the bypass flow rate,  $Q_G$ , equals the  $Q_B$ . In this case the power  $P_{bypass} = \rho g H_B (Q_B - Q_D)$  is dissipated. Furthermore, the turbine experiences a power output reduction, due to the reduction of the flow rate, the available head, and the efficiency.

### 2.2. Rotational speed control

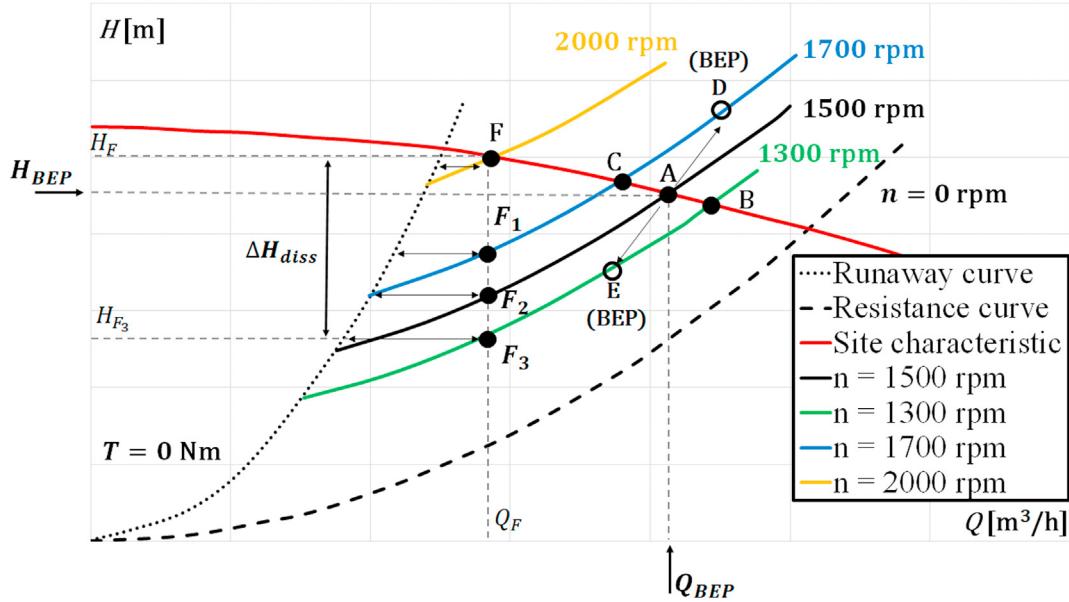
Actually, a PaT can be controlled also by modifying its rotational speed. This type of regulation is advantageous if flow rates and heads of the site are close to the BEP of the machine. Moreover, it is also important to evaluate how close the operating point is with respect to either the runaway or the resistance curve. In many cases these two curves are too close together to allow a speed control over a wide range of flow rates. Fig. 4 presents the characteristic curves of a PaT at different rotational speeds (1300 rpm, 1500 rpm, 1700 rpm and 2000 rpm) with the characteristic of the site (in red). Fig. 4 shows also the runaway ( $T = 0 \text{ Nm}$ ) and the resistance ( $n = 0 \text{ rpm}$ ) curves. The optimal matching occurs when the site



**Fig. 2.** Typical installation scheme of a PaT in a WDN.



**Fig. 3.** Generic PaT characteristic curve at constant rotational speed which intersects the site characteristic curve in the design point, A. Runaway curve ( $T = 0 \text{ Nm}$ ) and resistance curve ( $n = 0 \text{ rpm}$ ) of the PaT. Hydraulic regulation of the machine with PRV #1 (green) for throttle regulation and with PRV #2 (blue) for bypass regulation. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)



**Fig. 4.** Generic PaT characteristic curves at various rotational speeds which intersects the site characteristic curve in the operating points (A, B, C, F). Runaway curve ( $T = 0 \text{ Nm}$ ) and resistance curve ( $n = 0 \text{ rpm}$ ) of the PaT. Rotational speed control to avoid operating condition (point F) close to runaway curve (points  $F_1$ ,  $F_2$ ,  $F_3$ ).

characteristic intersects the PaT characteristic ( $n = 1500 \text{ rpm}$ ) at its Best Efficiency Point (BEP), here again corresponding to point A.

When the required flow rate is lower than the design one (e.g.,  $Q_C < Q_A$ ), it is possible to increase the rotational speed in order to move the operating point from point A to point C. For instance, increasing the rotational speed from 1500 rpm to 1700 rpm allows the new operating point to become point C. With such a control, we don't need to have a head loss in the regulation valve (PRV #1). Furthermore, there is an increase in the available head on the PaT, but also a reduction of the flow rate and the efficiency (the PaT BEP at 1700 rpm moves to point D, hence  $\eta_C < \eta_A = \eta_D$ ) and this generally determines a power reduction. Conversely, when the

required flow rate is greater than the design one (e.g.,  $Q_B > Q_A$ ), the rotational speed can be reduced. For instance, decreasing the rotational speed from 1500 rpm to 1300 rpm allows the new operating point to become point B. With this operation, the PaT BEP at 1300 rpm moves to point E with a slight PaT efficiency reduction ( $\eta_B < \eta_A = \eta_E$ ).

However, if we wanted to guarantee the operating point in F ( $Q_F$  vs.  $H_F$ ), we should significantly increase the rotational speed moving the operating point too close to the runaway curves with a drastic reduction of the efficiency. In order to avoid this circumstance, the rotational speed can be reduced in order to move the operating point F far from the runaway curve (e.g.,  $F_1$  at 1700 rpm,  $F_2$

at 1500 rpm and  $F_3$  at 1300 rpm) but this means that, at the same time, we need to act on the PRV#1.

### 2.3. Coupling with electric motor

Usually PaTs, being commercial pumps sold with their electric motor, are coupled with three-phase asynchronous motors, which are self-starting, reliable and economical. When the PaT is supposed to work at constant speed, it is important to note how its rotational speed,  $n_{PaT}$ , differs from the so-called synchronous speed,  $n_s$ , i.e., the rotational speed of the magnetic field, defined by Eq. (3):

$$n_s = \frac{2f}{p} [rpm] \quad (3)$$

where  $f$  is the frequency of the power supply ( $f = 50$  Hz in Europe) and  $p$  is the number of magnetic poles. Fig. 5 shows a typical torque curve of a three-phase asynchronous motor vs. the motor rotational speed,  $n$ , both in motor and generator mode. Moreover, it is plotted also a generic resistance curve defined by the torque developed by the coupled PaT,  $T_{PaT}$ . The operating point is defined by the intersection between the two curves and it is possible to notice how the PaT works at a rotational speed,  $n_{PaT}$ , greater than the synchronous speed,  $n_s$ . This is due to the slip factor,  $s$ , defined for an asynchronous generator as the difference between operating speed and synchronous speed at the same frequency (equation (4)):

$$s = \frac{n_r - n_s}{n_s} \quad (4)$$

where  $n_s$  is the stator magnetic speed and  $n_r$  is the magnetic field speed associated to the rotor mechanical speed.

The control of the rotational speed is performed by means of a variable frequency drive, which changes the supply frequency in order to match working conditions at different rotational speeds, as shown in Fig. 6.

## 3. Machine selection methodology

Fig. 7 shows schematically the proposed methodology for the selection of a machine to be installed in a WDN. Once the user

knows the daily flow rate and pressure patterns of the installation site, it is possible to define the design operating conditions ( $Q_{site}$  vs.  $H_{site}$ ), hence the BEP of the machine ( $Q_{BEP,T} = Q_{site}$ ,  $H_{BEP,T} = H_{site}$ ). The selection of the design working conditions depends on different aspects, such as the flow rate frequency distributions and the exploitable hydraulic power. Then, the PaT specific speed,  $n_{q,T}$ , can be computed. Starting from these data, it is necessary to predict the BEP that the same machine should provide in pump mode ( $Q_{BEP,P}$  vs.  $H_{BEP,P}$ ) in order to select it from any commercial pump catalogue. This can be done by applying the proposed empirical model.

As depicted in Figs. 8 and 9, the model is based on empirical correlations which basically enable the computation of the BEP in turbine mode from the BEP in pump mode. Conversely, knowing  $n_{q,T}$  it is possible to estimate  $n_{q,P}$  by applying Eq. (5), which reports the empirical correlation of Fig. 8:

$$n_{q,P} = (n_{q,T} + 2.6588)/0.9237 \quad (5)$$

Afterward the head ratio  $h = H_{BEP,T}/H_{BEP,P}$  can be evaluated from Fig. 9. Then, the head and the flow rate in pump mode can be computed by means of the definition of head ratio and specific speed, respectively (Eqs. (6) and (7))

$$H_{BEP,P} = H_{BEP,T}/h \quad (6)$$

$$Q_{BEP,P} = \left( n_{q,P} H_{BEP,P}^{3/4} / n \right)^2 \quad (7)$$

It is worth to notice that the results given by the model depend both on the proposed method and on the set of machines used for calibration. As the calibration set grows, the model can gain generality but may lose accuracy. Moreover, as far as the specific speed range is not too extended, the method can lead to accurate forecasts. For this reason, we have decided to base the selection model on the specific speed number, which is a useful engineering tool because it collects the main performance parameters of the machine. The case study, shown in the following, deals with a specific speed range, which has been widely covered by several PaTs tested in the literature (centrifugal pump type). The sample is large

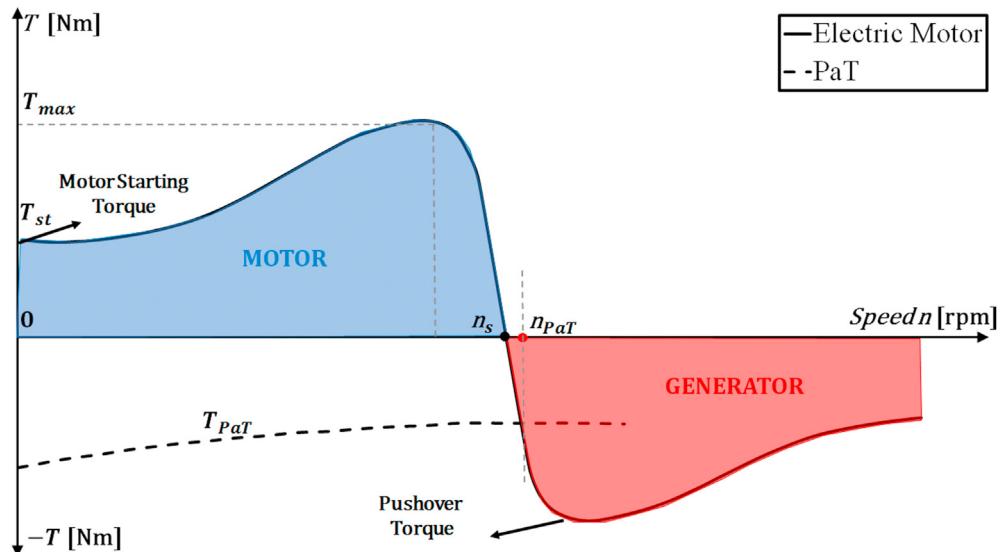
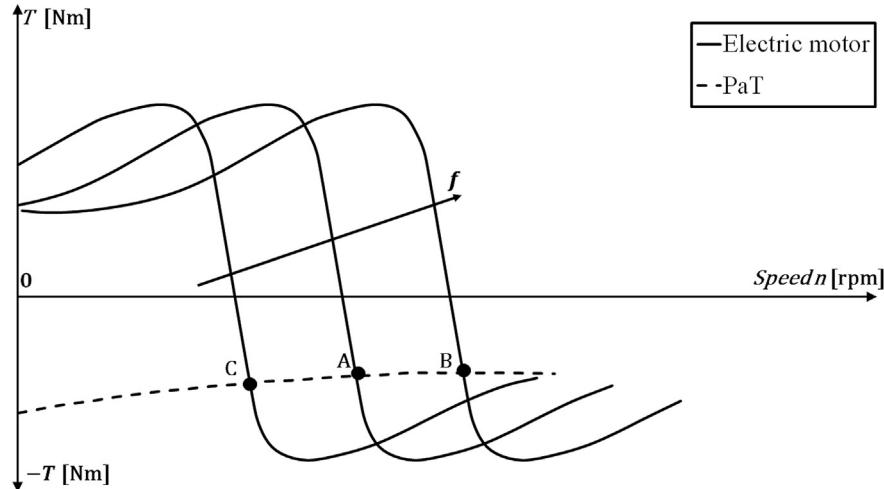


Fig. 5. Typical torque curve of a three-phase asynchronous motor in "motor" and "generator" mode,  $T$ , vs. rotational speed,  $n$ , and PaT torque curve,  $T_{PaT}$ , vs. rotational speed,  $n$ .



**Fig. 6.** Speed regulation of a three-phase asynchronous motor by means of a variable frequency drive.

enough to give us a high coefficient of determination ( $R^2 = 0.98$ ) for the correlation between the specific speed in pump mode,  $n_{q,P}$ , with the one in turbine mode,  $n_{q,T}$  (see Fig. 8). Clearly, in case one wants to move to other types of turbomachines, and therefore to other ranges of  $n_q$ , the methodology should be replicated and new correlation curves should be found which are suitable for the new ranges.

Knowing  $n_{q,P}$ ,  $Q_{BEP,P}$  and  $H_{BEP,P}$ , any pump manufacturer could select the machine from its pump catalogue. At this point, having no experimental characteristic curves of the selected pump operated as turbine, it could be possible to predict its characteristic curve in reverse mode by applying a 1-D performance prediction model. Indeed, this model has been proposed in a previous work [78] for manufacturers by considering detailed geometrical information about the machine (only property of the manufacturers) and taking into account complex phenomena such as hydraulic losses in the volute and in the runner and slip phenomena, which cannot be neglected for PaT due to their reduced number of blades compared to conventional turbines (see Eq (8)).

$$H_{turbine} = \sigma_{turbine} H_{th} + Z_{volute} + Z_{runner} \quad (8)$$

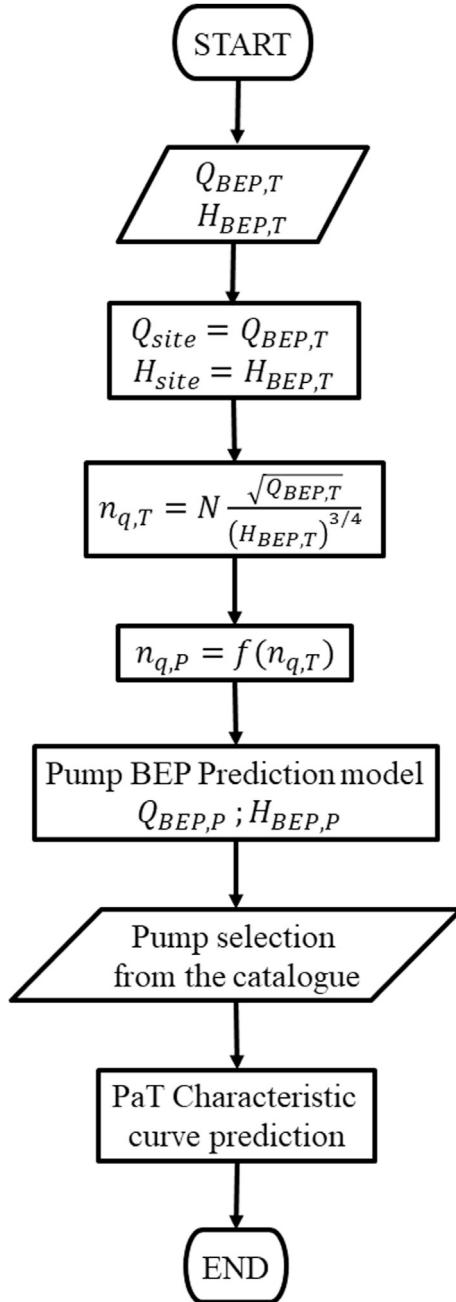
#### 4. Case study

In this framework, an Italian WDN located in the Southern Italy, whose name is omitted due to confidentiality, has been used as case study in order to install a PaT to recover hydraulic energy otherwise wasted by the PRV. The town counts approximately 50,000 inhabitants and it is supplied by a unique reservoir. As shown in Fig. 10, the PRV is installed at the entrance of the town, precisely at the origin of the water distribution network (highlighted in red). The PRV is programmed to guarantee two different pressure levels on the WDN during the day and the night in order to satisfy the daily water demand and to reduce nightly water losses. Indeed, Fig. 11 depicts the flow rate and pressure patterns measured in a typical day upstream and downstream the PRV and recorded every 10 min. During the night (16:10–5:00) the site is characterized by an available pressure drop  $H_{night} = 90$  m, whereas during the daytime (5:10–16:00) the available pressure drop is lower,  $H_{day} = 80$  m allowing a higher pressure level on the WDN. The PaT is supposed to be installed downstream of a first PRV (PRV #1) in order to adapt the pressure level to the desired value when the flow

rate is lower than  $Q_{BEP,T}$ , as depicted in Fig. 12. In order to allow the hydraulic regulation when the required flow rate is higher than  $Q_{BEP,T}$ , another PRV (PRV #2) is installed in parallel to the PaT line. The system is supposed to be installed upstream the main PRV, which is left for safety reason, but that will be left fully open.

The proposed selection procedure has been applied to this case study. Since there are two different flow conditions (one during the night and another during the daytime) with different available heads, the PaT specific speed,  $n_{q,T}$ , must be evaluated in both configurations. In order to evaluate the specific speed, the rotational speed of the machine needs to be known. As mentioned in chapter 2, the rotational speed of the PaT is greater than the speed of the asynchronous generator. Considering an induction motor supplied at 50 Hz with 4 magnetic poles, the synchronous speed is equal to 1500 rpm. At this point, by supposing a slip factor of about 1%, the rotational speed of the PaT should be  $n_{PaT} = 1520$  rpm, which results in  $n_{q,T}(Q_{min}) = 18.77$  and  $n_{q,T}(Q_{max}) = 26.73$ , hence, according to Eq. (5), the pump specific speed should range in between  $n_{q,P}(Q_{min}) = 23.2$  and  $n_{q,P}(Q_{max}) = 31.8$ . This means that the pump specific speeds are within the conventional range of applicability of centrifugal pumps and, in particular, mixed flow centrifugal pumps. In this preliminary study in order to define a unique specific speed, it was decided to select the pump based on the daytime head and the average daytime flow rate ( $Q_{site} = \bar{Q}_{day} = 636.5$  m<sup>3</sup>/h and  $H_{site} = 80$  m). In this way the PaT can work at its BEP when the flow rates are higher. Then, it is possible to evaluate the BEP of the corresponding pump ( $Q_{BEP,P}$  vs.  $H_{BEP,P}$ ) by means of a BEP prediction model, which correlates the BEP in turbine mode to the BEP in pump mode. As a result,  $Q_{BEP,P} = 439.5$  m<sup>3</sup>/h and  $H_{BEP,P} = 48.8$  m. At this point, knowing  $Q_{BEP,P}$  and  $H_{BEP,P}$ , any pump manufacturer could select the proper machine from its own pump catalogue. In this study, thanks to the collaboration with Xylem®, it has been possible to use their pump catalogue in order to obtain available geometric information of the selected machine. In this study, the calculation led to find the Xylem® NSC 150–400/900, as depicted in Fig. 13, where the operating point at the intersection of the two red arrows is close to the pump BEP.

Then, knowing its geometrical information, it is possible to predict the characteristic curve of the pump operating as a turbine by using the 1-D performance prediction model. Fig. 14a and b show respectively the predicted characteristic curves of the PaT in terms of head and efficiency vs. mass flow rate. In Fig. 14a it is also reported the runaway curve, which has been predicted by using the

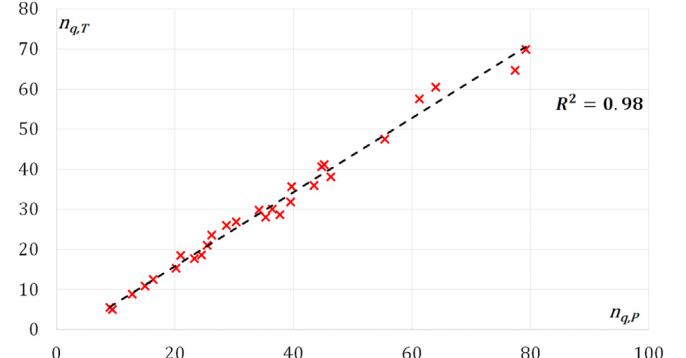


**Fig. 7.** Flow chart of the methodology for the pump selection and PaT curve prediction.

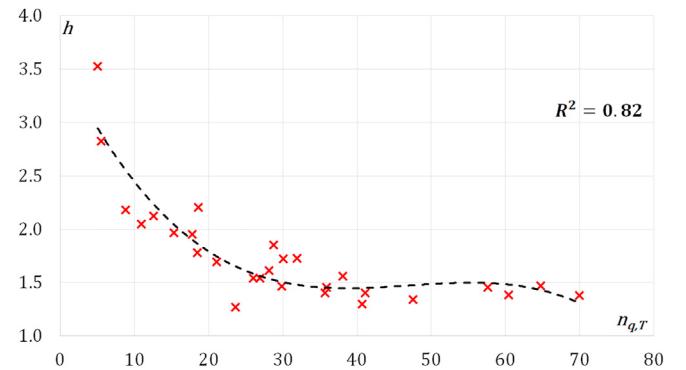
correlations proposed by Gülich [72].

Fig. 15a depicts the predicted characteristic curve of the PaT at 1520 rpm with both the site characteristics during the night (blue) and the daytime (orange). Moreover, the hydraulic regulation methods are highlighted in the same figure: when the flow rate is lower than the design point (this is always the case during the night), the PRV #2 on the by-pass line is closed, whereas the PRV #1 in series with the PaT is used to regulate the head (see Fig. 16a). Otherwise, when the flow rate exceeds the design operating point, the PRV #2 on the by-pass is partially opened and set in control mode in order to reduce the flow rate flowing in the PaT line to the value  $Q_{BEP,T}$ , thus allowing the PaT to operate at its BEP (see Fig. 16b).

In order to reduce the pressure drop on the PRV #1, hence



**Fig. 8.** Specific speed number in turbine mode,  $n_{q,T}$ , vs. specific speed number in pump mode,  $n_{q,P}$ .



**Fig. 9.** Head ratio,  $h$ , vs. specific speed number in turbine mode,  $n_{q,T}$ .

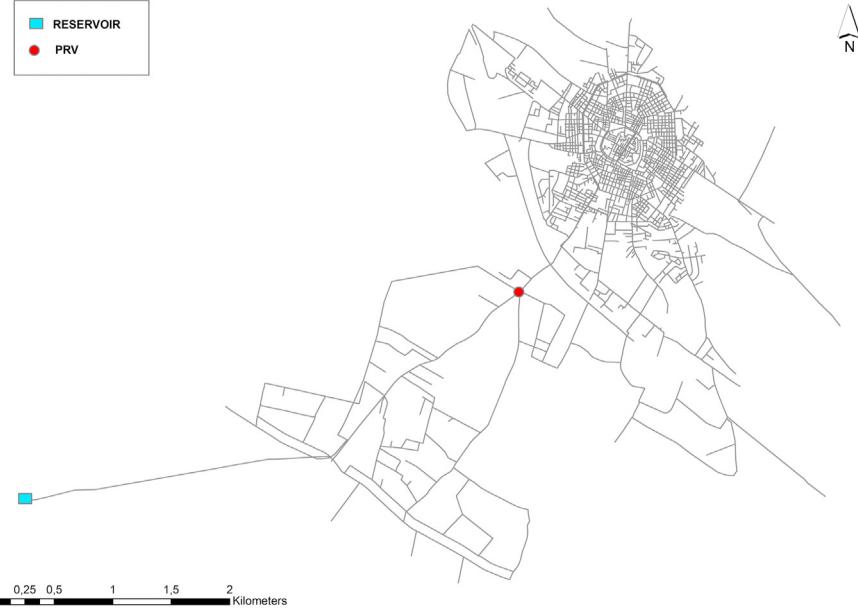
recovering a higher amount of energy, during the night, a case #2 has been proposed by including also the rotational speed control, as depicted in Fig. 15b. In this case, it is possible to reduce the pressure drop in the PRV #1 by increasing the rotational speed from 1520 rpm to 1750 rpm. Although the PaT works at lower efficiency because of the scaling, a considerable amount of energy, otherwise dissipated by the PRV, can be recovered, as it will be shown in the next section. The same strategy can be applied when high flow rates are required by reducing the rotational speed of the machine, in this case, even if the PaT works under off-design conditions, it operates in a range of high efficiency. The latter case has not been considered here and it will be taken into account in further studies.

## 5. Energy production

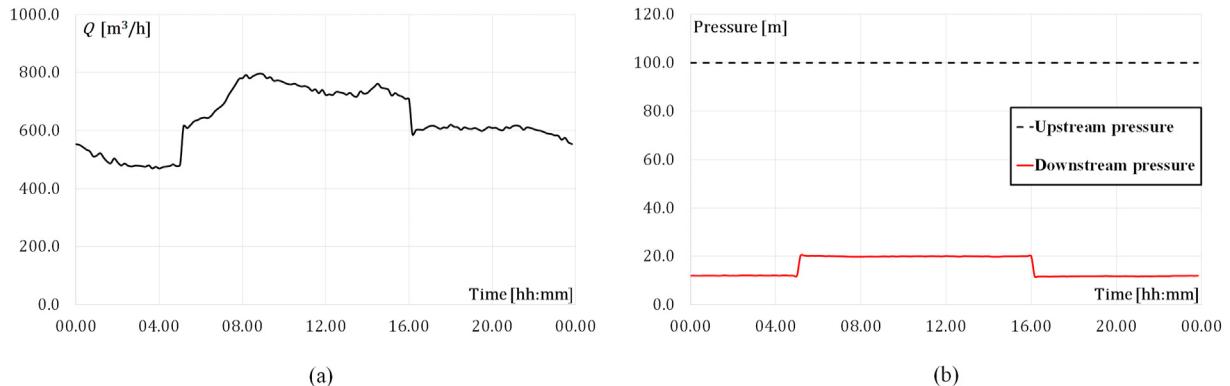
As summarized in Table 1, the site presents a significant daily available energy equal to 3520 kWh/day. A comparison between two scenarios has been carried out in terms of energy exploitation capability. In order to perform this comparison, a Hydraulic Source Harvesting Coefficient has been introduced,  $C_E$ , in order to take into account how much energy can be produced by the PaT over the total hydraulic energy available from the site (Eq. (9)). Table 2 points out the comparison between the two cases: case #1 without any control speed; case #2 with the control speed during the night.

$$C_E = \frac{E_{produced}}{E_{available}} \quad (9)$$

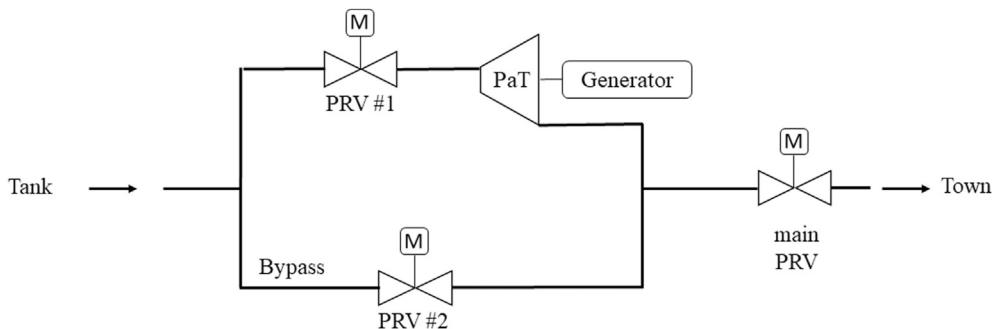
It is possible to notice an identical  $C_E$  during the daytime, because both cases use the same control strategy with the by-pass,



**Fig. 10.** Water Distribution Network of the town considered as case study.



**Fig. 11.** Daily flow rate measured at the origin of the urban distribution (a); pressure patterns measured upstream and downstream the PRV installed at the origin of the urban distribution (b).



**Fig. 12.** Installation scheme of the PaT system.

but a different behaviour during the night, where case #1 shows a  $C_E$  lower than the one of case #2. Indeed, in case #1 the nightly pressure drop is partially throttled by PRV #1 and the corresponding energy is no more exploitable. Using the rotational speed control, such as in case #2, the throttled pressure drop is reduced by moving the PaT characteristic curve towards the nightly site

curve. Obviously, increasing the rotational speed if on one hand it allows the reduction of the pressure drops, on the other hand it forces the PaT to work at higher flow rates with lower efficiencies. Indeed, the overall increase of the total daily energy production of case #2 with respect to case #1 is only equal to 4.5% (from 2241 kWh/day to 2343 kWh/day). Fig. 17 shows the comparison

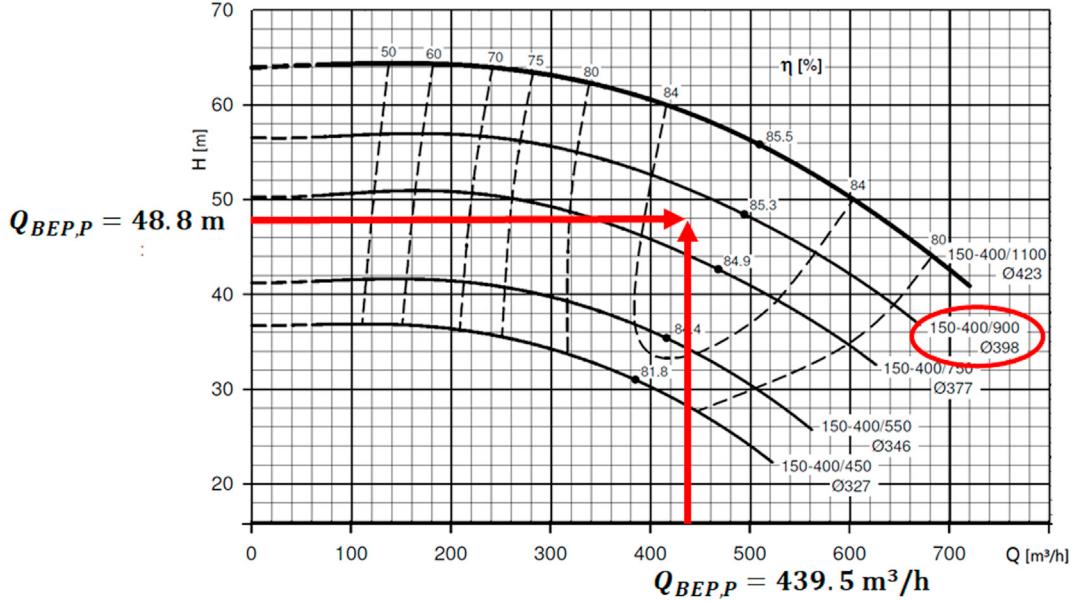


Fig. 13. Pump selection from the manufacturer catalogue ( $Q_{BEP,P} = 439.5 \text{ m}^3/\text{h}$  and  $H_{BEP,P} = 48.8 \text{ m}$ ).

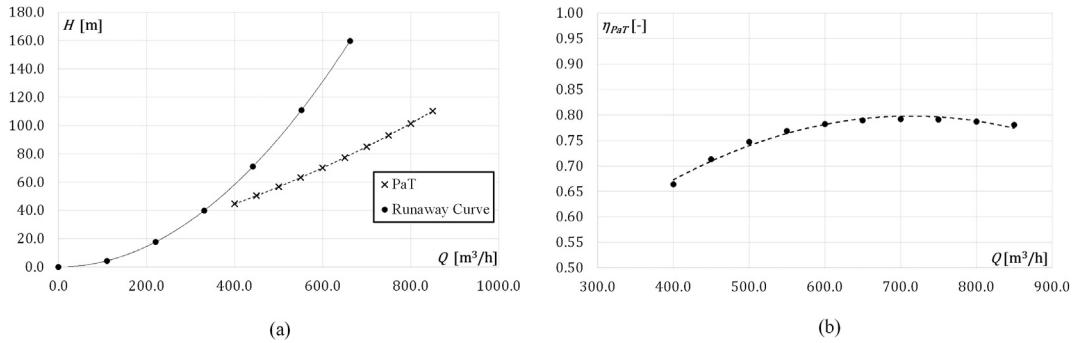


Fig. 14. Predicted characteristic curve of PaT with the runaway curve (a); Predicted PaT efficiency curve vs. flow rate.

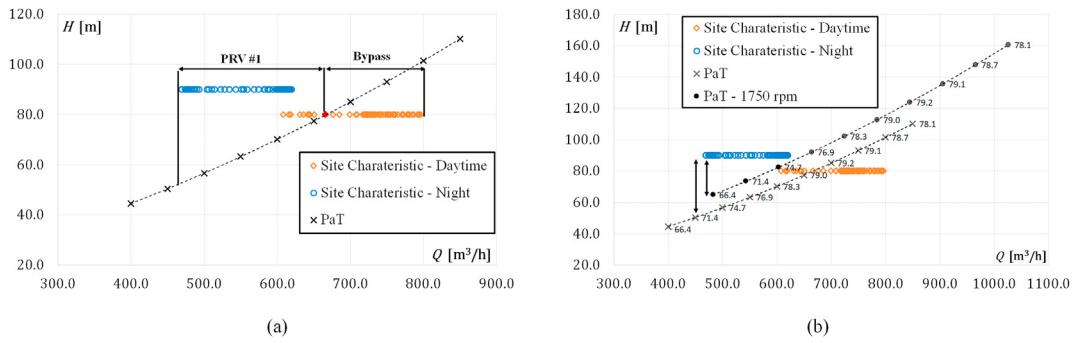


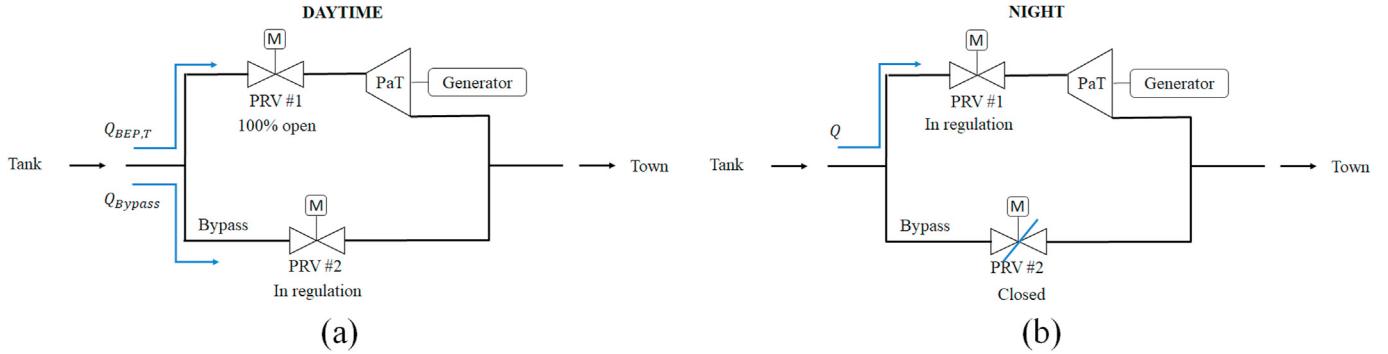
Fig. 15. Case #1: predicted PaT performance curve at constant rotational speed with characteristic curves of the site during the night and the daytime (a); Case #2: rotational speed control of the PaT during the night.

between the two cases in terms of effective power generated by the PaT during the entire day ( $P_{eff} = \eta_{PaT} \rho g H Q$ ).

## 6. Economic analysis

An economic analysis has been carried out in order to evaluate the cost-effectiveness of the use of a PaT in a WDN for hydraulic

energy recovery. In this study, the economic analysis has been carried out considering a useful lifetime equal to 15 years, i.e., the lifetime of a generic mechanical and electrical equipment. Costs were subdivided into capital costs and operation costs: the former sum all the equipment costs necessary to realize the plant including the civil works, whereas the latter consider maintenance and operational costs of every device. In details, maintenance and



**Fig. 16.** Bypass regulation scheme during the daytime (a); throttle regulation scheme during the night (b).

**Table 1**  
Daily Available Energy of the case study.

Night [kWh/day]	Daytime [kWh/day]	Total [kWh/day]
1778	1742	3520

**Table 2**  
Comparison of the two proposed cases in terms of  $C_E$ .

ID CASE	$C_E$ (Night)	$C_E$ (Daytime)	$C_{E,TOT}$ (Entire day)	Energy Production [kWh/day]
CASE #1	0.56	0.72	0.64	2241
CASE #2	0.62	0.72	0.67	2343

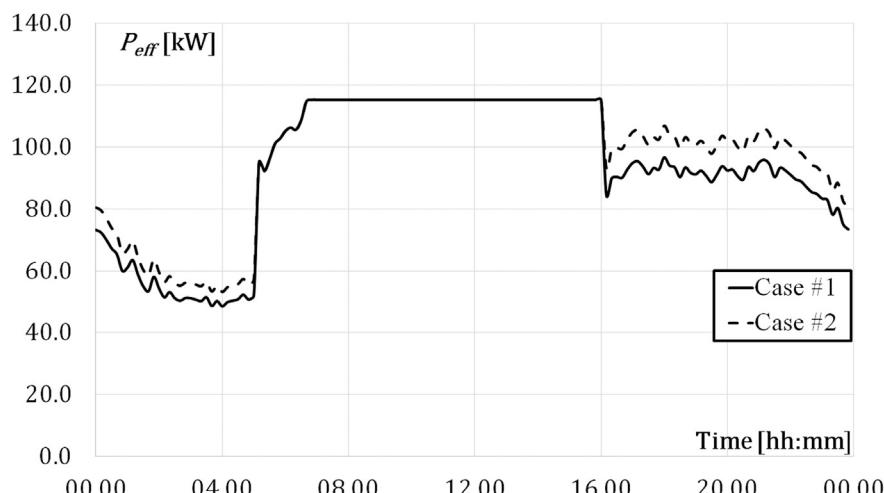
operational costs have been assumed respectively equal to 0.5% and to 2.5% of the capital costs [82]. Table 3 lists all the costs of case #1 provided by pump manufacturer and suppliers, classified by main cost items.

The hydraulic equipment consists in valves, joints, pressure transducers, flow meter, pipes and the by-pass line, whereas the electrical equipment consists in a frequency converter and a PLC based control system. Repositioning costs related to the substitution of the equipment were considered equal to zero since the defined economic life of the system. In order to evaluate the economic effectiveness of this kind of technical solutions, several economic indicators have been considered, such as the Net Present Value (NPV), the Return of Investment (ROI), the Payback Period

and the Internal Rate of Return (IRR).

The annual energy production of the PaT system was estimated at 818,028 kWh/year and the annual revenue for this produced energy was computed as 127,612 €/year, considering a complete feed-in tariff of the produced electricity and a reference sales price fixed by the Italian Energy Manager (GSE) in 2019. Italian Regulatory Authority for Energy, Networks and Environment (ARERA) indicates GSE like the unique authority to whom the energy producers turn to stipulate the agreement that regulates the commercial withdrawal of the electricity fed into the network. Actually, GSE is the responsible of the minimum guaranteed sales price for energy produced by small power generating systems with renewable energy plants with the aim to encourage energy production from renewable sources.

As a result, the NPV, defined as the sum of the actual values of all the costs and revenues associated to the project amounts to 876,797 € by considering a discount rate equal to 5%. The Payback Period is estimated of about 24 months, which is consistent with results of other economic assessments in the literature [34–36]. The Return of Investment is equal to 49%. The Benefit/Cost Ratio (B/C), that compares actual values of benefit and costs amounts to 4.50, whereas the Internal Rate of Return is equal to 19.29% (Table 4). The results of the economic analysis have been reported only for case #1 considering that the addition of the frequency converter (case #2) involves only a marginal improvement of all the economic parameters (e.g., the Payback Period reduced from 1.32 years to 1.29 years), and it makes the entire control system more



**Fig. 17.** Effective power generated by the PaT during the entire day for case #1 and case #2.

**Table 3**  
Costs for the PaT plant.

Items	Cost [€]
PaT	13,000
<b>Hydraulic Equipment</b>	
Valves	112,285
Measurement devices	17,216
Pipes and Fittings	46,794
Labour costs	5,465
<b>Electrical equipment</b>	
SCADA system	6,100
Cabinet and cabling	4,880
Remote control	3,660
Other electrical equipment	1,220
Power electronics	5,878
<b>Civil works</b>	29,150
<b>Engineering</b>	17,247
<b>Total Costs</b>	<b>262,895</b>

complicated, reducing its reliability.

Such an analysis could also gather the environmental benefits that such a technology would entail, i.e., what in economics are commonly called positive externalities. As mentioned in the previous chapter, the water management authority has installed recently a PRV at the origin of the urban water distribution network. This solution has drastically reduced the water leakages with a consequent water saving volume of about 1.57 Mm<sup>3</sup>/year and a full-recovery cost of water of about 1,261,440.00 €/year. The PaT installation could achieve the same results of the PRV in terms of water saving by recovering energy at the same time. Actually, for this analysis, the water saving is already guaranteed by the existing PRV. Moreover, the generation of energy from a clean source would result in a reduction of emissions, which can translate in economic benefits [83], as shown in Table 5. In this case the energy production by the PaT system could involve the reduction of about 327 ton of CO<sub>2</sub>, considering an averaged emission factor of 400 g/kWh obtained by the Italian national data for the last ten years [84].

Considering these environmental benefits and their economic repercussions (in this case only those due to reduction of CO<sub>2</sub> emissions), it is possible to obtain a clear improvement in the economic indices, as shown in the second column of Table 4. In this case the Payback Period has been estimated of about 1.32 years, a very short time compared with the first one; ROI amounts to 73% and B/C results equal to 7, while NPV amounts to 1,554,525 € and, finally, IRR for this study is equal to 31.02%.

**Table 4**  
Economic analysis results.

Economic indicators	Value considering only economic revenues	Value considering environmental benefits
Payback Period	1.93 years	1.32 years
ROI	49%	73%
B/C	4.50	7
NPV	876,797 €	1,554,525 €
IRR	19.29%	31.02%

**Table 5**  
Environmental benefits and revenues of the PaT system.

Item	Environmental benefits	Revenues [€]
Annual energy production by the PaT system [kWh/year]	818,028	127,612 €
Reduction of CO <sub>2</sub> emissions from energy saving [g CO <sub>2</sub> /year]	327,211,044	71,986 €

In addition, the impact of the PaT installation has been assessed in terms of CO<sub>2</sub> reduction against emissions produced by urban circulation of cars. According to the Italian Ministry of Economic Development [84,85], average CO<sub>2</sub> emission by a car in Italy amounts to 115.4 g/km with an average driving distance by a car equal to 11,200 km/year. This can be translated in an average annual CO<sub>2</sub> emission by a car equal to 1.3 ton/year (Table 6). Fig. 18 clearly illustrates the impact in terms of environmental benefit of the proposed solution, that is, the energy produced by the PaT installation could avoid the production of CO<sub>2</sub> emissions comparable to those produced by 255 cars that circulate for a year in an Italian city.

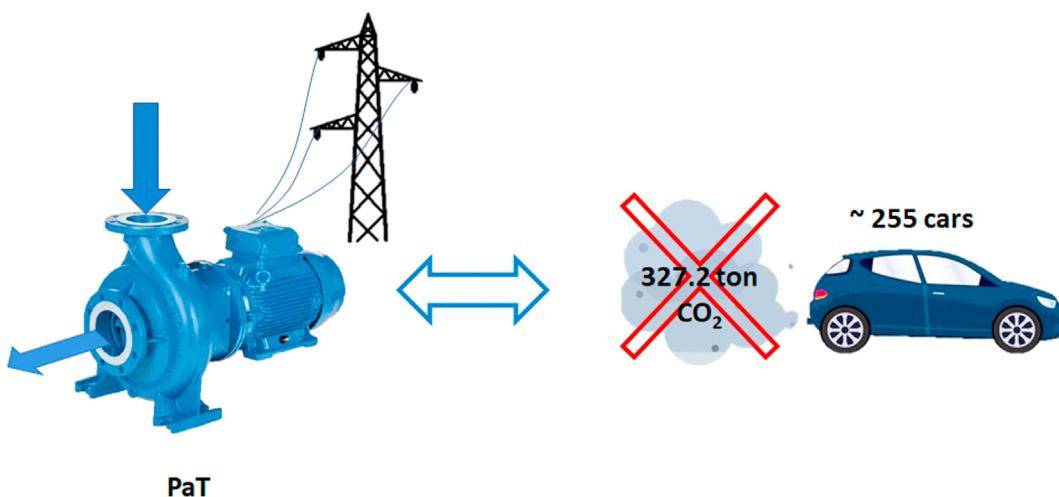
## 7. Conclusions

In this work a methodology to select a PaT and the guidelines to perform a techno-economic analysis for a specific WDN have been proposed. The WDN of a town in the Apulia region (Southern Italy) has been investigated as case study, in order to select a PaT useful for throttling energy recovery. In this proposal, the PaT is supposed to be installed upstream the existing PRV at the origin of the urban distribution network. Starting from the evaluation of the daily pressure and water demand patterns, it is possible to evaluate the specific speed required by the site, in order to understand if the operating field is appropriate for PaTs instead of conventional turbines. Moreover, the methodology aims to allow the user to select an existing machine from a pump catalogue. Then, knowing its geometrical information, it is possible to predict the characteristic curves of the pump operating as a turbine by using a 1-D performance prediction model.

In a preliminary assessment, the average daily flow rate has been chosen as the reference operating condition. This value of flow rate and its corresponding head value have been selected as the turbine BEP operating condition ( $Q_{BEP,T}$  and  $H_{BEP,T}$ ). Then, the corresponding BEP in pump mode ( $Q_{BEP,P}$  and  $H_{BEP,P}$ ) has been calculated by means of a BEP prediction model, which correlates the BEP in pump mode to the BEP in turbine mode. At this point, knowing  $Q_{BEP,P}$  and  $H_{BEP,P}$ , a real machine from a pump catalogue can be selected. In this case, thanks to the collaboration with Xylem®, the Xylem® NSC 150–400/900 pump has been selected from its

**Table 6**  
CO<sub>2</sub> emission by car in Italy.

Item	CO <sub>2</sub> average emission by car in Italy	115.4 kg/(km car)
Average driving distance by car in Italy		11,200 km/(year car)
Annual CO <sub>2</sub> emission by car in Italy		1.3 ton/(year car)



**Fig. 18.** Environmental impact of the PaT installation in terms of reduction of CO<sub>2</sub> emission.

commercial catalogue. Then, knowing its geometrical information, in absence of experimental tests the performance curves of the same machine in reverse mode have been predicted by means of the 1-D performance prediction model (as validated in a previous work).

At this point, two control methods have been evaluated and the Energy Harvesting Coefficient ( $C_E$ ) has been defined in order to compare the two cases in terms of hydraulic energy harvesting and power output. In case #1 the PaT is only hydraulically regulated with a PRV installed in series with the PaT and a by-pass line; in case #2 the rotational speed control is contemplated in order to avoid excessive nightly pressure drops. As results, case #2 implicates marginal energy production increase (4.5%) and for this reason this second case has been finally neglected.

Finally, an economic analysis has been carried out in order to assess the cost-effectiveness of this application. Annual energy production by the PaT system has been estimated at 818,028 kWh/year and the annual revenue for this produced energy is equal to 127,612 €/year, considering a complete feed-in of the produced electricity. Different economic indices have been contemplated in order to assess the cost-effectiveness of the project. As result, considering only the economic revenues, the analysis has shown the Net Present Value (NPV) equal to 876,797 € and the Payback Period of about 24 months. The Return of Investment (ROI) ROI is equal to 49%, while the Benefit/Cost Ratio amounts to 4.50. Finally, the Internal Rate of Return (IRR) is equal to 19.29%. Actually, all these results are based only on the capital costs and the economic revenues. Indeed, the same results become further advantageous if the estimation of environmental benefits, like the reduction of CO<sub>2</sub> emissions and the water saving, shall be included. In this case, the Payback Period has been estimated of about 1.32 years, a very short time compared with the first scenario and with classical turbine application.

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## CRedit authorship contribution statement

**Michele Stefanizzi:** Conceptualization, Methodology, Formal analysis, Investigation, Data curation, Writing - original draft,

Writing - review & editing. **Tommaso Capurso:** Conceptualization, Methodology, Formal analysis, Investigation, Writing - original draft, Writing - review & editing. **Gabriella Balacco:** Conceptualization, Methodology, Investigation, Data curation, Writing - original draft, Writing - review & editing. **Mario Binetti:** Writing - original draft, Methodology, Supervision. **Sergio Mario Camporeale:** Writing - original draft, Methodology, Supervision. **Marco Torresi:** Conceptualization, Methodology, Formal analysis, Investigation, Writing - original draft, Writing - review & editing, Supervision.

## Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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