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HYDROPOWER ENERGY RECOVERY SYSTEMS: DEVELOPMENT OF DESIGN METHODOLOGIES FOR PUMPS-AS-TURBINES IN WATER NETWORKS

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SUMMARY

Given the urgency of tackling the challenges offered by climate change and natural resource depletion, most nations around the world have committed towards reducing their impact on the ecosystem. At the same time, the water industry is among the top energy intensive sectors among industrialised countries since large amounts of energy are required to pump and treat water for human consumption. As such, it is a priority to increase the sustainability of the industry by reducing its energy demand and associated pollutant emissions.

One of the possible ways to achieve this goal is to implement on a vast scale hydropower energy recovery from pressurized pipelines, in order to generate clean renewable energy at sites where currently an excess of pressure is being dissipated by means of a throttling valve. While hydropower is a mature technology with a long history of practical applications, large portions of its potential on small sites (with installed power lower than 100 kW) and on existing pipelines is currently untapped. The main reasons behind it are the unsuitability of certain types of turbines to operate in-line within a conduit and the large cost of miniaturising conventional custom-made hydro turbines down to small sizes.

A type of hydro converter which has the potential to meet the technical challenges of in-pipe small scale hydropower at a reduced cost are Pumps As Turbines (PAT), consisting of regular water pumps utilized in reverse as turbines. Despite this, their adoption worldwide as energy recovery devices has been sporadic and the technology has remained extremely marginal as for the number of applications. An investigation into the reasons behind this has led to the identification of three key reasons: the lack of interest from pump manufacturers towards commercializing their products as turbines, the lack of robust PAT design guidelines and a general lack of awareness about PATs from a wide portion of the potentially interested audience of water and energy professionals, academics and practitioners.

While it is reasonable to assume that pump manufacturers may become interested in PATs in a close future once the market for this type of device will no longer be marginal with respect to their core business, the present PhD dissertation focused on developing a comprehensive and robust PAT selection strategy under the format of a software application, able to assist designers during the selection of the most appropriate turbine for any given site. At the same time, in order to tackle the mentioned lack of awareness and demonstrate the potential of the technology to a wide

audience, the same software was used to design a laboratory test rig and two full scale pilot plants later built and commissioned in Ireland and Wales.

In order to build a reliable PAT selection tool several technical and economic aspects of the technology were at first analysed separately by gathering information on hundreds of PATs from both academic and industrial sources, and assembling the largest data set on the topic compiled up-to-date. Such an amount of information has allowed the development of specific mathematical models capable of predicting the PAT performances and cost over a wide range of application which were subsequently built into the selection software.

As a second step, a laboratory experimental campaign has been carried out to test a small-scale PAT prototype over a wide set of operating conditions and connected to electric generators of different type. The main findings from this experiment have informed the construction of a full-scale PAT test rig located within the premises of Trinity College Dublin. This experimental setup has been designed to re-create fluid conditions that can realistically be found in a water supply network, and the previously developed PAT selection software has been utilized to choose the most appropriate turbine. The experiment led to the desired outcomes since it confirmed the validity of the technology and informed the design of the planned demonstration sites. Moreover, the PAT tested in the lab proved 29% more efficient than what predicted by the software which in turn based its calculations on the pump efficiency declared by the manufacturer which was found to be underestimated.

As a last step, two pilot plants were designed and built featuring each a PAT offering a maximum power output close to 4 kW. One of the two installations consisted of a run-off-river scheme in northern Wales providing energy to a remote historical farmhouse, while the second one was located on the supply line of a water treatment works in southern Ireland. The commissioning of the demonstration sites confirmed the validity of the PAT selection software which had been used to choose the specific turbine for each location, with both units performing according to the expectations. Additionally, both installations showed an attractive return on investment (8.5 and 5 years respectively) and a competitive cost of generated electricity (0.148 and 0.079 €/kWh) which would have not been possible with the adoption of a conventional and more expensive hydro turbine. Together, the two demo sites are expected to avoid the emission of 21.5 tons of CO_{2eq} in the atmosphere per year.

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Last but not least, I am immensely thankful towards my family who has always supported me.

LIST OF ABBREVIATIONS

AC: Alternating Current

BEP: Best Efficiency Point

BPT: Break Pressure Tank

CFD: Computational Fluid-Dynamics

CO₂: carbon dioxide

DC: Direct Current

DN: Nominal Diameter

ELC: Electronic Load Control

ER: Electric Regulation

EROEI: Energy Returned Over Energy Invested

FIT: Feed-In Tariff

GHG: Greenhouse Gas

GIS: Geographic Information System

GUI: Graphic User Interface

HDPE: High-Density Polyethylene

HOF: Hands-Off-Flow

HPM: Hydrostatic Pressure Machine

HPW: Hydrostatic Pressure Wheel

HR: Hydraulic Regulation

IE: International Efficiency standards according to IEC 60034-30-1

LCOE: Levelized Cost Of Electricity (€/kWh)

MHP: Micro-Hydropower

MPPT: Maximum Power Point Tracker

NFGWS: National Federation of Group Water Schemes

NPV: Net Present Value

PAT: Pump As Turbine

PM: Permanent Magnet

PRV: Pressure Reducing Valve

PV: Photovoltaic

RMSE: Root Mean Square Error

SDM: Staudruckmaschine

SHP: Small-Hydropower

SME: Small-Medium Enterprises

SSE: Sum of Squares Error

TPES: Total Primary Energy Supply

UN: United Nations

VAT (Value Added Tax)

VOS: Variable Operating Strategy

VSD: Variable Speed Drive

WEN: Water-Energy Nexus

WSS: Water Supply System

WWTP: Waste Water Treatment Plant

NOMENCLATURE

1D: one-dimensional

2D: two-dimensional

3D: three-dimensional

C: cost (€)

CS: cost per unit power (€/kW)

D: impeller diameter (mm or m)

f: frequency (Hz)

g: gravitational acceleration, equal to $9.81 \text{ (m/s}^2\text{)}$

H: net head (m)

h: non-dimensional head parameter (-)

I: current (A)

k_n : speed constant of the generator (RPM/V)

N: rotational speed (RPM)

n: rotational speed (RPS)

$N_{s,p}$: specific speed according to discharge (-)

N_s : specific speed according to discharge (-)

P: power output (W or kW)

p: pressure (Pa or bar)

pp: number of pole pairs (-)

Q: nominal flow rate (m^3/s or l/s)

q: non-dimensional flow parameter (-)

R: resistance (Ohm)

R^2 : determination coefficient (-)

s: slip factor (-)

T: Torque (Nm)

U: flow mean velocity (m/s)

V: voltage (V)

z: geometric height (m)

GREEK SYMBOLS

γ : dimensionless parameter (-)

η : efficiency (-)

η_p : plant overall efficiency (-)

ρ : fluid density (kg/m^3)

Φ : non-dimensional flow rate (-)

Ω : non-dimensional specific speed (-)

ω : rotational speed (rad/s)

SUBSCRIPTS AND SUPERSCRIPTS

av: available

eq: equivalent

gen: generator

h: head

i: internal

max: maximum

p: pump

q: flow

rel: relative

s: specific

t: turbine

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1. INTRODUCTION

Behind the fundamental process of supplying communities and businesses with clean water as well as collecting and treating wastewater, lies enormous technical, scientific, logistic and organizational challenges which are often unknown to the general public.

Water utilities are among the largest electricity users in industrialized countries, and therefore are also responsible for a significant quota of CO₂ emissions. Population growth, ageing infrastructures, more stringent quality standards, as well as other factors have caused a constant increase in the energy demand from the water sector up to the present days. Utilities from around the world are struggling to reduce their operating cost, contain water leakage and provide a reliable service while at the same time maintaining user's bills as low as possible.

Among the possible solutions to address the problem is the partial recovery of kinetic energy by means of hydraulic turbines from sites within water infrastructures where it is currently dissipated. This approach could partially offset the overall amount of electricity required for the whole process. However, the available potential for energy recovery is often too small in order to conveniently install a conventional hydraulic turbine as an energy recovery device. Conventional turbines are generally not economically viable below power outputs of 20-50 kW and the majority of sites within water infrastructure falls below this range. Therefore, development of new types of hydro turbines characterized by lower purchase price and reliable performances may enable the exploitation of large portions of this untapped hydro potential.

An effective and low-cost technology for Micro Hydropower (MHP) hereby investigated are Pumps as Turbines (PAT), consisting of standard water pumps running in reverse mode with the connected induction motor acting as a generator

The present PhD research has been funded by the Dŵr Uisce project, which is a five-year multidisciplinary programme aimed at increasing the sustainability of the water sectors of Ireland (Republic of) and Wales (UK).

1.1. RESEARCH QUESTIONS

Which are the major bottlenecks to hydraulic energy recovery schemes from water networks?

Strategies for energy recovery from the water industry must be formulated in order to make hydropower more attractive even after the general ceasing or gradual decrease of Feed-in-Tariffs (FIT) incentives for renewable generation in Europe.

Which are the main barriers to the adoption of Pump as Turbines (PAT) as energy recovery devices?

Why aren't Pumps as Turbines more used on the market, either as energy recovery devices or as micro hydro generators? It's been almost a whole century since the first team of technicians determined that pumps can operate fairly efficiently as turbines when the flow direction through them is reversed. Still, their market presence is risible around the world.

Albeit having a few shortcomings (e.g. lack of in-built flow rate regulation, low part-load efficiency and difficult availability of performance charts in turbine mode from manufacturers) most authors agree in attributing to PATs very compelling advantages in terms of lower cost, maintenance simplicity and availability. Despite this, the up-to-date adoption of PATs is marginal in two sectors for which they would be ideally suitable:

- micro and small hydro, with some exception in developing countries where likely it was the only hydro converter locally manufactured and available;
- energy recovery from water networks or as replacement in throttling valves common in many industrial sectors.

1.2. RESEARCH TARGET

How shall the above concerns be addressed?

A balanced response should be made up of several aspects:

- the formulation of sound and comprehensive design guidelines for PATs as energy recovery devices, including a measure of the involved project uncertainties;
- a successful laboratory campaign in order to test the concept and validate the model being built;

- the design and construction of two full-size demonstration sites.

1.3. THE DŴR UISCE PROJECT

My Ph.D. candidate position was part of the multidisciplinary Dŵr Uisce (“Distributing our Water Resources: Utilizing Integrated, Smart & low-Carbon Energy”) research programme involving Trinity College Dublin and Bangor University. The Dŵr Uisce programme has been developed as a continuation of the previous HydroBPT project and was funded under Interreg Ireland-Wales Programme 2014-2020. Its main target is to improve the long-term sustainability of water resources in Ireland and Wales from economic, environmental and social perspectives. The main steps to accomplish such targets are:

- benchmark assessment of the water sector in Wales and Ireland;
- development of new technological platforms for electricity and heat recovery from water supply systems including prototyping, business plan assessment and installation of pilot schemes;
- development of a cross-border smart specialization cluster in the field of water-energy challenges;
- investigation on climate change impacts and resilience of water industry in Ireland and Wales.

2. RESEARCH CONTRIBUTIONS

2.1. RESEARCH MODEL

My main task as Ph.D. student at Trinity College under guidance of Dr. Aonghus McNabola consisted of developing a multicriteria and multidisciplinary decision support system enabling the design of a Micro-Hydropower (MHP) plant powered by Pumps as Turbines (PATs) for water networks. The structure of the design algorithm developed is shown in Figure 2-1.

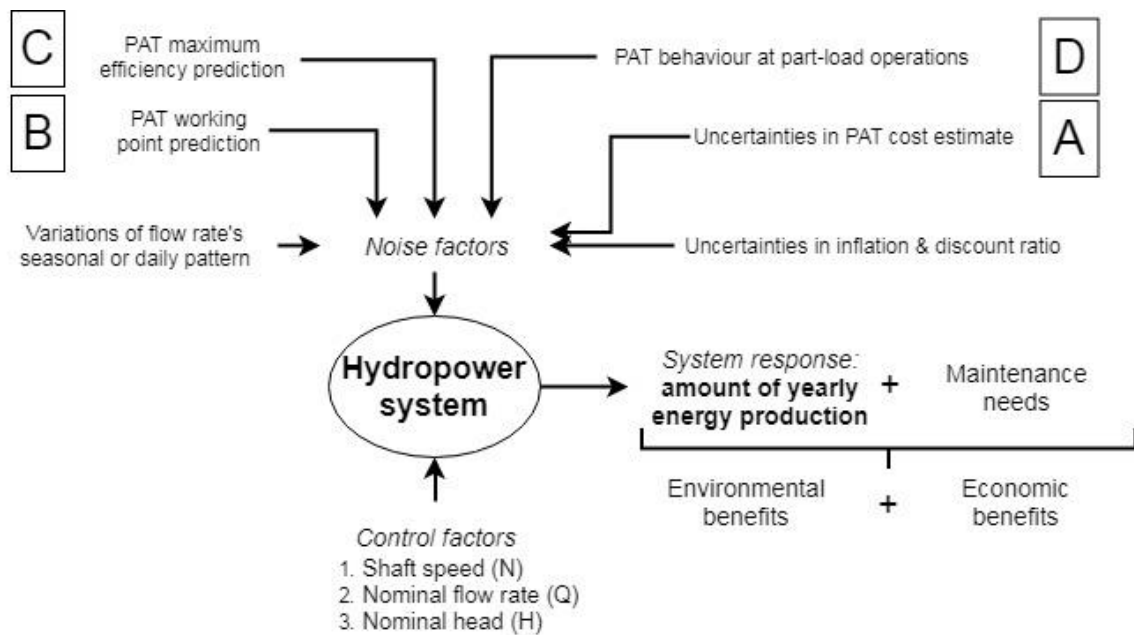


FIGURE 2-1. STRUCTURE OF THE PLANNED DESIGN ALGORITHM.

Within this framework, novel contributions arose from:

- A. determination of a robust PAT cost model and comparison with purchase price of other hydraulic turbines;
- B. evaluation and benchmarking of the existing mathematical correlations able to predict the PAT working point from its pump characteristics;
- C. creation of a model for the prediction of PAT maximum efficiency for any given nominal design conditions;
- D. development of a set of equations to extrapolate the complete characteristic curve of a PAT from its Best Efficiency Point (BEP).

All these contributions shall be merged together in a software able to produce a robust design of a PAT for any given potential site given the specific set of resources and constraints. This work, supported and validated by laboratory tests, shall represent the basis for designing effectively two full-scale Micro-Hydropower demonstration plants in Ireland and Wales thus fulfilling the targets of Dŵr Uisce project. In Figure 2-2 are graphically shown the research aims and the tools adopted for the purpose.

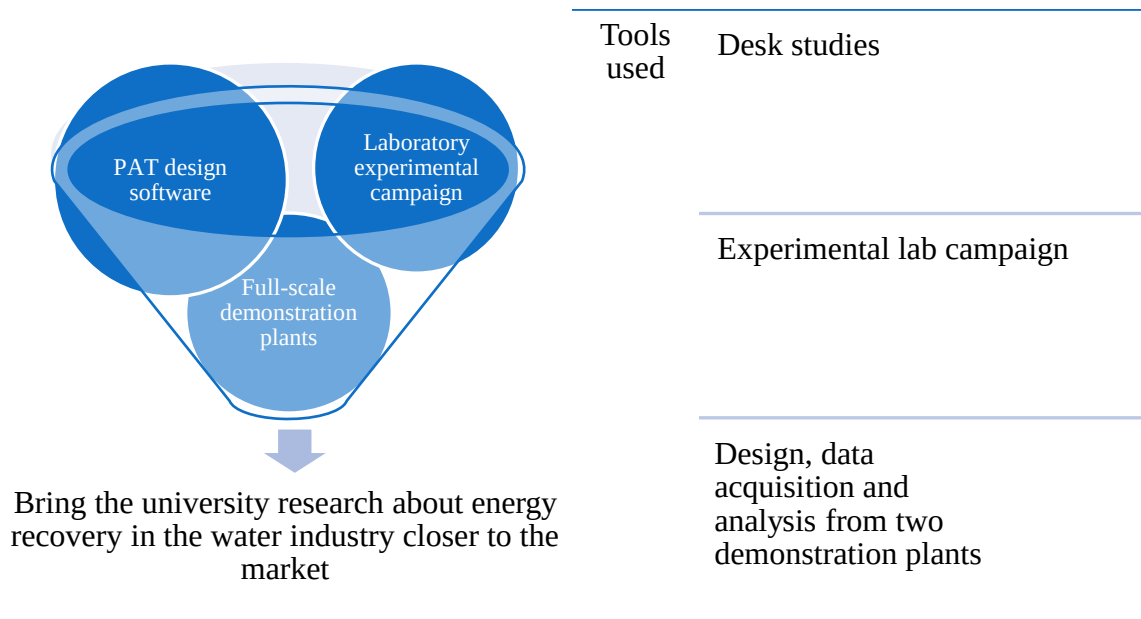


FIGURE 2-2. SCHEME OF RESEARCH TARGETS AND TOOLS.

The expected practical outputs from the research are:

- showcasing the concept of PAT utilization and energy recovery from water networks to a wide audience thanks to the laboratory setup for PAT testing and the demonstration sites;
- generating a set of updated guidelines for PAT sizing in the form of a software in order to foster future applications of the technology;
- disseminating the results and insights gathered through publications in high-impact scientific journals and by attending relevant scientific conferences;
- delivering a series of feasibility studies regarding PAT energy recovery to a series of Small-Medium Enterprises (SMEs).

2.2. METHODOLOGY

The planned work involved different phases and skills:

1. execution of desk studies based on the available literature in order to develop the key knowledge to build the PAT design algorithm as described in Figure 2-1.
2. laboratory testing of different PATs in order to acquire a deeper knowledge about pumps running in reverse mode and validating the software outputs;
3. identification of the location, design, construction supervision and performance analysis from two demonstration plants to further validate and improve the developed methodology;

3. LITERATURE REVIEW

Water and energy are the two fundamental commodities at the basis of our everyday lives, our wellness and economies. Their availability, quality and economic value play an essential role for the status and the development of households, communities, agriculture and industries.

3.1. SUSTAINABILITY OF ENERGY SECTOR

The world energy sector is heavily dependent on fossil fuels, which supply most of the heat and electricity needed for human activities. A significant indicator of this is the Total Primary Energy Supplied (TPES), indicating the overall energy production, storage and net imports. In Figure 3-1 (a) the share of the yearly world TPES is shown divided by fuel, while Figure 3-1 (b) displays the share by fuel of electricity production in 2014 (International Energy Agency, 2016).

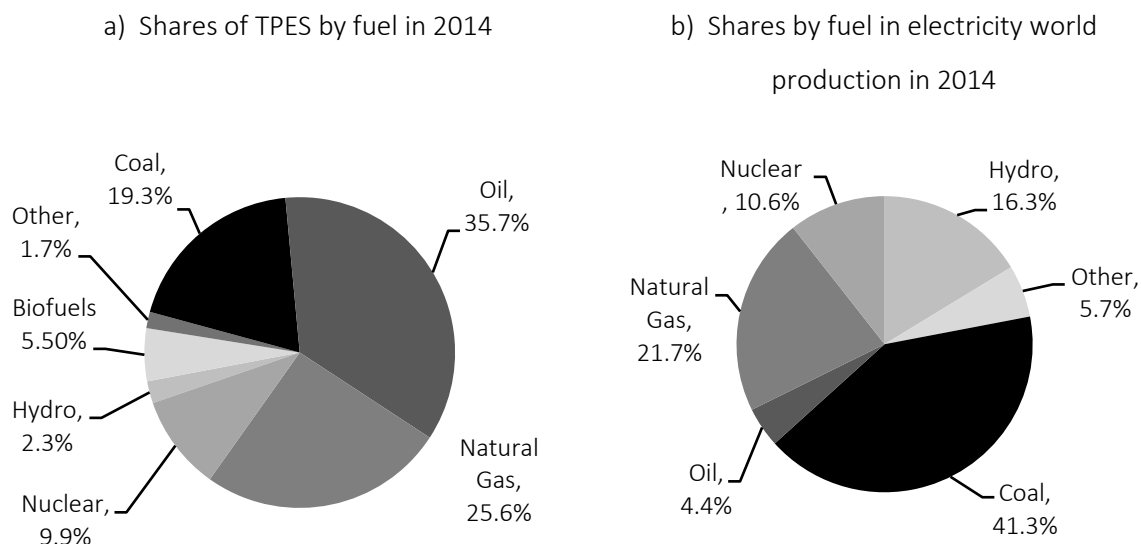


FIGURE 3-1. WORLD TPES AND ELECTRICITY PRODUCTION BY SOURCE IN 2014 SHOWING THE RELEVANCE OF HYDROPOWER AMONG OTHER RENEWABLE ENERGY TECHNOLOGIES (INTERNATIONAL ENERGY AGENCY, 2016).

Despite the evident supremacy of fossil fuels in the global energy mix seen in Figure 3-1, several drivers exist which compel the transition towards more efficient and renewable power sources.

Firstly, the physical limit of natural resources is undermining the long-term availability of fossil fuels (coal, oil, natural gas) and fissile elements (thorium and uranium) which constitute the large majority of present energy feedstock. Despite that, the technical availability of such energy sources may stretch for as long as another century at current levels of consumption, as seen in Figure 3-2, their economically viable availability is questionable. In fact, the progressive depletion of

conventional and easy-to-reach energy resources implies a shift towards unconventional resources (e.g. shale gas, tar sands) which require additional technology and efforts to be extracted and are more energy intensive (Murphy & Hall, 2011).

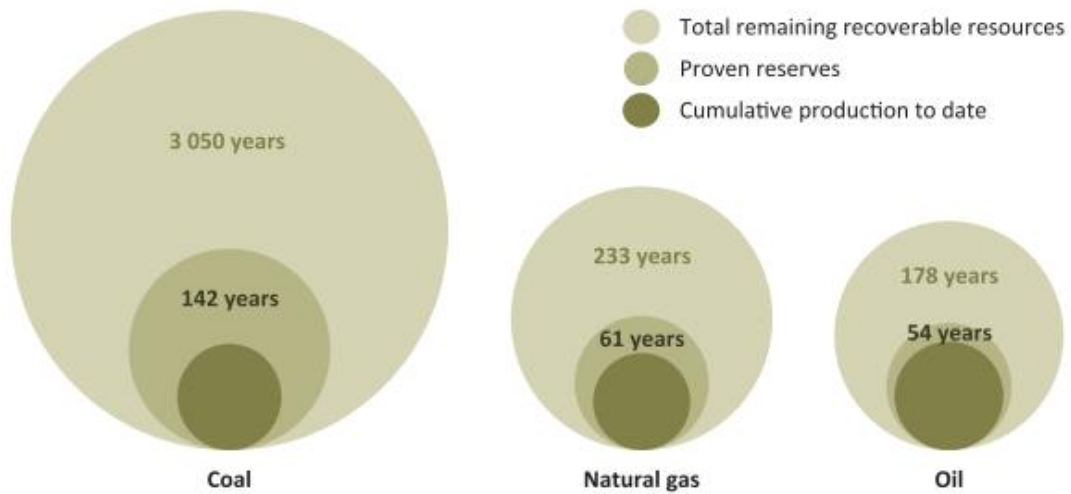


FIGURE 3-2. FOSSIL ENERGY RESOURCES BY TYPE, SHOWING THE FORECASTED AVAILABILITY OF COAL, NATURAL GAS AND OIL BASED ON PROVEN RESERVES AND YEARLY CONSUMPTION RELATIVE TO THE YEAR 2013 (INTERNATIONAL ENERGY AGENCY, 2016).

A significant parameter to assess the extraction difficulty of resources is the EROEI, defined as the ratio between the energy content of an energy raw material (defined as “energy returned”) and the energy necessary to extract, produce or refine it (“energy invested”). Since the early phases of fossil fuel extraction their EROEI value has been constantly dropping and approaching unit, corresponding to the energy payback level at which the energy needed to extract a resource equals the energy embedded in the same. The evolution over time of crude oil EROEI extracted in the US can be seen in Figure 3-3 (Cleveland, 2005).

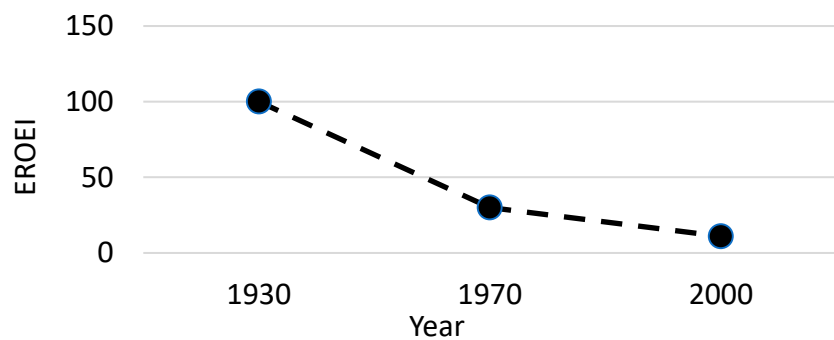


FIGURE 3-3. HISTORICAL SERIES OF CRUDE OIL EXTRACTED EROEI IN THE LOWER 48 US STATES, SHOWING A MARKED DECREASE BETWEEN THE YEARS 1930 AND 2000 (CLEVELAND, 2005).

Besides, emissions of fossil CO₂ are deemed by a large part of the scientific community as the main greenhouse gas responsible for the climate change phenomenon. This is expected to have particular negative impacts on low-income countries and the most vulnerable social groups in developed countries (Haines, et al., 2006).

Starting from the Kyoto Protocol which was adopted in 1997 and entered into force in 2005, during the last decades a number of international treaties has been negotiated by representatives of world governments with the aim of mitigating and reducing CO₂ emissions into the atmosphere. The declared intent of the 2016 Paris Agreement was to hold the increase of the Earth average temperature “well below 2 °C” with respect to the pre-industrial era following the recommendations of the Intergovernmental Panel on Climate Change (Rogelj, et al., 2016). At the same time, the European Union has implemented a series of unilateral strategies to progressively reduce its carbon emissions among which the 2030 climate and energy framework and the European plan on climate change which aims at reaching carbon neutrality in the EU by 2050 (European Commission, 2018).

In order to foster the transition towards a cleaner and fossil-free energy mix, most industrialized countries have implemented during last decades a system of Feed-In Tariffs (FIT) at which electricity produced from renewable sources shall be paid when sold to the national grid. As of 2016-2017 most European countries have suspended such FIT schemes (Council of European Energy Regulators, 2018).

3.2. WATER SECTOR

The UN Water agency estimates the number of people that by 2025 will be living in countries characterized by absolute water scarcity as 1.8 billion, while globally two thirds of the entire world population is expected to cope with water stress conditions by the same date (UN Water, 2015). The effects of climate change are believed to induce an increased occurrence of extreme meteorological events such as droughts or floods, therefore actions to decrease vulnerability of water resources need to be undertaken (Ramos, et al., 2010a). Threats posed by desertification of fertile lands and uneven distribution of water resources in the long term may contribute to controversies and conflicts between neighbouring regions or states (Gleick, 2014).

According to a recent report, over one billion jobs worldwide are heavily dependent on water, corresponding to sectors such as agriculture, fishing, mining, power generation, food and textiles.

Besides, another billion jobs in sectors like construction, manufacturing (e.g. industry of paper, wood, metals, rubber and plastics transformation) and recreation are considered to be moderately water-dependent. In total, almost 80% of the global workforce relies on economic, healthy and accessible water resources (UN Water, 2016). Despite accounting for a share of global water use lower than 1% (Gössling, 2002), the hospitality/tourism sector often exists in areas with limited freshwater resources and is responsible for both direct water use (e.g. kitchens, toilets, showers, laundry service, swimming pools, garden irrigation) and indirect water use related mainly to the need for food, energy and construction materials (Gössling, et al., 2012).

Significant amounts of energy are required to collect, purify, transport and treat water. Water utilities have been facing an increased energy intensity in the last decades due to increased demand, increasing contamination of resources and the issuing of more stringent regulations on water quality requirements (Pabi, 2013).

3.3. WEN: POSSIBLE SYNERGIES

Energy and water are strongly interconnected in terms of resource uses, and such links are referred to as the Water-Energy Nexus (WEN). The WEN definition is also sometimes broadened to include Food production and consumption, acquiring the name of Water-Energy-Food Nexus. The extent of such links reveals itself on multiple scales and is a matter of growing concern given the predicted worldwide increase of energy, water and food demand expected as higher than +50% by 2050 (IRENA, 2015).

From the one side, the world energy sector is heavily dependent on water for energy generation, cooling, raw material extraction and bioenergy production. At the same time, water industries are amongst the largest energy consumers in order to carry out their tasks, which include pumping, drainage, treatment and purification, desalinization and water distribution (IRENA, 2015; Pabi, 2013; Cooley & Wilkinson, 2012). On the other side, it is evident how the food industry requires consistent inputs in the form of water and energy to perform its basic stages like irrigation, fertilization, harvesting, food processing and storage. Strategies and technologies which illustrate a global management of such three fundamental resources can be traced on different scales. On a large regional scale, examples include the construction and operation of large multiple-use water reservoirs able to satisfy water supply for human consumption and agriculture needs and to provide a source of clean renewable energy to the surrounding communities as well. A recent implementation of such a concept is the Portuguese Alqueva multipurpose project consisting in

two artificial reservoirs having combined capacity of 4,256hm³ feeding several hydropower stations for a total installed capacity of 530MW and supplying water for both human needs and irrigation to most of the Alentejo region (EDIA, 2017). Another example is the creation of hybrid Water Supply Systems (WSS) which rely on a multiple arrays of different energy generation units and integrate pumped water energy storage in potable water transport and storage infrastructures (Vieira & Ramos, 2009). Another way in which the nexus between energy and water becomes explicit is the similarity in respective consumption patterns: the moments of the day with highest electricity demand correspond generally to the aggregated peak water demand from residential and industrial consumers.

The need for an integrated resource management approach has been stated by many researchers as a way to cope with the predicted rise in demand and resource scarcity (Scott A., et al., 2011). The decision making complexities involved shall be overcome through the elaboration of policies and the inclusion of both public and private decision makers (Glassman, et al., 2011). An example of how application of different policies may affect simultaneously the future energy and water sectors can be traced in a recent research in which authors analysed the consequence in terms of water demand evolution against different future energy production scenarios which may result from alternative policies (Carrillo & Frei, 2009). The results of the analysis for Spain showed that by 2030 the projected water demand of the energy sector is supposed to be 25% higher than the 2005 baseline, and the increase of renewable energy shares may have contrasting outcomes on this figure. For instance, an increase in biofuels production for the automotive sector would increase the stress on water resources while a rise in the installed wind power capacity is expected to reduce the water demand of the energy sector (Carrillo & Frei, 2009).

Besides the cited predicted rise in resource demand expected within the next decades, another element expected to add complexity in the system is the influence that global warming is expected to have in all human activities and natural resources. The predicted modifications in precipitation and temperature distribution may affect deeply the water, energy and food sectors in unpredictable ways and produce relevant cost increases (Cooley & Wilkinson, 2012). Thus, tending towards a higher resilience level should be a key target of any integrated policy view.

Climate change has been reported to have a tangible effect on water utilities, causing them to rely on large scale strategic water transfers by pipeline or by road during periods of draught and conversely to cope with sudden storms requiring additional drainage capacity (Arrell, 2014).

3.4. SUSTAINABILITY OF WATER SUPPLY THROUGH MICRO-HYDROPOWER

Production of renewable energy from water utilities is traditionally restricted to the treatment of wastewater treatment sludge, either through incineration or through anaerobic digestion and biogas production (Zakkour, et al., 2002). However, energy recovery through the application of Micro-Hydropower (MHP) schemes within water supply infrastructures is considered a valid opportunity for reducing the energy dependence of water utilities and the consequent CO₂ emissions, as well as enhancing security of supplies and balancing the increasing energy costs (Wallace, 1996; Gaius-obaseki, 2010; Zakkour, et al., 2002). Ultimately, advantages of such an approach are of different types:

- Environmental: because of the reduction in grid energy consumption and the associated pollutant emissions such as CO₂, NO_x, SO_x and particulate matter. As an example of this, a group of researchers performed a life cycle analysis computing the environmental burdens associated to three hydro energy recovery installations in the water industry having capacity of 15, 90 and 140 kW across Wales and Ireland. As all three schemes were grid-connected, their environmental impacts were compared with those from a natural gas combined-cycle power plant which constitute the backbone of power generation capacity in many European countries. The results of the analysis demonstrated that over the lifespan of the equipment the micro-hydro installations on average reduced the GHG emissions by over 99% as well as significantly lowering the impacts in terms of fossil fuel depletion, acidification potential and human toxicity potential (Gallagher, et al., 2015a);
- Economic: since ownership of Micro-Hydropower (MHP) generation assets along their networks would constitute a revenue source for water utilities. MHP schemes have a long design lifetime, between 25 and 40 years, and thus can lead to long-term cost savings (Zakkour, et al., 2002; Rentricity, 2014);
- Social: because on the one side the generated clean renewable energy could be utilized locally benefitting small communities, and on the other side the increased incomes faced by water companies may translate into a reduction of bills paid by water end-users.

Another advantage arising from MHP installations within WSS is the absence of traditional market and regulatory barriers to new hydro developments, i.e. limited need for building authorizations,

no water concession fees, no interference with ecosystem, no visual intrusion (Zakkour, et al., 2002).

The concept of hydropower technologies applied to water transmission lines where large flow rates and potentials exist is not a new one, but significant challenges lie in the miniaturization of the normal physical scale of turbines to enable exploitation of the small potentials that occur in most distribution networks shown as “dissipation nodes” in Figure 3-4. As reviewed by many authors, several opportunities exist for power generation within water networks, as described in the following sub-chapters.

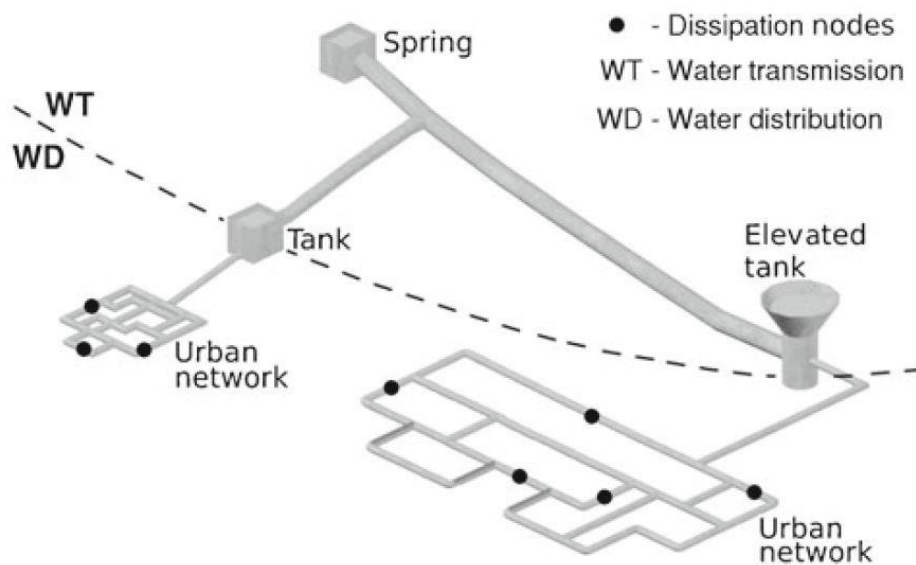


FIGURE 3-4. WATER SUPPLY SYSTEM WITH HIGHLIGHTED PRESSURE DISSIPATION NODES ALONG THE DISTRIBUTION NETWORK (CARRAVETTA, ET AL., 2012).

A 1989 resource assessment study carried out by Salford University estimated the potential for MHP development within water industry in England, Northern Ireland and Wales consisted of 17MW of exploitable capacity (Salford Civil Engineering, 1989), while another research study estimated at 64MW the recoverable energy potential in Western German water systems (Apfelbacher & Etzold, 1988). However, such mentioned figures contain evident underestimates of the actual potential available nowadays and are based on technical and economic data which is not up-to-date.

A research team from Turin Polytechnic School established a methodology to assess the hydropower potential of WSS from Piedmont region in Italy. The results show an intrinsic potential

in the interval between 21 and 31 MW, while the potential which is economically viable lay between 14 and 26 MW (Soffia, et al., 2010).

More recently, a study was launched to identify the potential for energy recovery in the water industries of Ireland and Wales related to elevated service reservoirs, Pressure Reducing Valves (PRV) sites or Waste Water Treatment Plants (WWTP) outfalls (Gallagher, et al., 2015b). After investigating the existing technical and financial constraints the authors proposed a four-step methodology for the selection of feasible MHP sites as can be seen in Figure 3-5. Out of the total analysed 238 sites across the two investigated regions, the authors shortlisted 80 most suitable sites and evaluated the technical and economic challenges for implementing MHP projects in such locations (Gallagher, et al., 2015b).

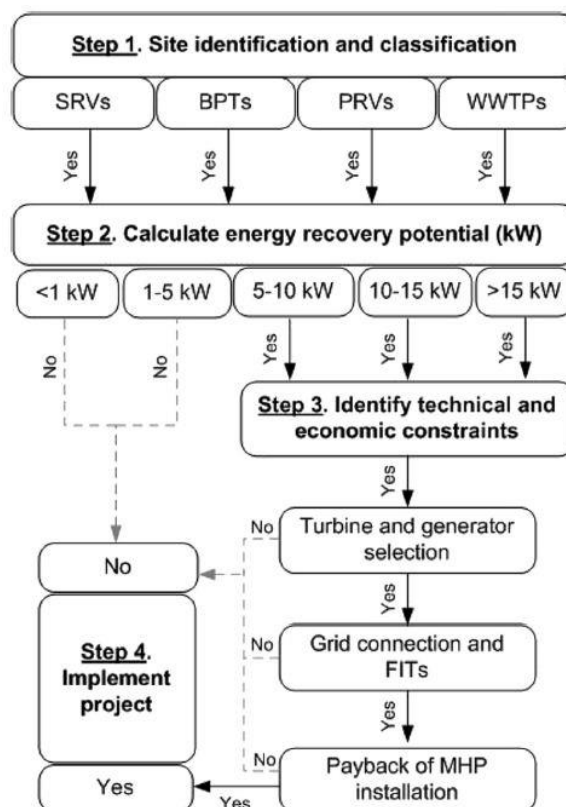


FIGURE 3-5. PROPOSED FOUR-STEP METHODOLOGY TO ASSESS THE VIABILITY OF MHP PROJECTS IN WATER INDUSTRY (GALLAGHER, ET AL., 2015B).

3.4.1. OVERPRESSURES IN GRAVITY-FED WATER CONVEYANCE PIPES

While many WSS use underground water resources, others are supplied by gravity from elevated reservoirs or constant water streams. Under such circumstances, it is common to introduce Break

Pressure Tanks (BPT) at given intervals of the conveyance system in order to avoid excessive pressures to be built within pipes (McNabola, et al., 2011). Such concrete-made tanks can potentially be replaced by hydraulic turbines designed to dissipate the excess pressure and bring it down to the level required by downstream treatment processes and at the same time generate electricity as displayed schematically in Figure 3-6 (Wallace, 1996). Recent research by McNabola et al. (2011) assessed the economic feasibility of turbine installation for 7 BPT units existing in Ireland and estimated a payback period lower than 10 years for schemes having installed power higher than 5 kW.

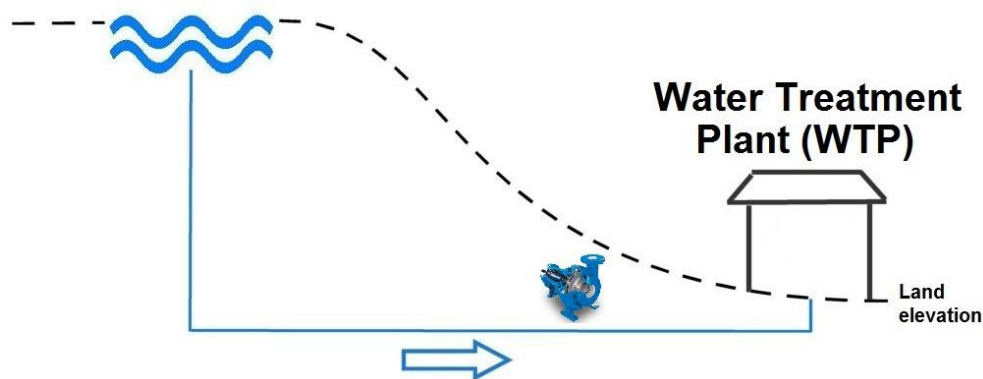


FIGURE 3-6. TURBINE PLACEMENT AT THE INLET OF A WATER TREATMENT PLANT (WTP) FED BY AN ELEVATED RESERVOIR.

Another approach to the same theme is the one proposed by US-based Company Lucid Energy which developed a hydraulic hydrokinetic vertical-axis turbine to be inserted directly inside the main water conveyance pipes and is capable of dissipating the excess energy generated by gravity (Lucid Energy, 2017).

3.4.2. MICRO-HYDROPOWER FOR PRESSURE CONTROL AS A PRV REPLACEMENT

Despite the fact that water utilities consume large amounts of energy to pump and pressurize water, situations commonly occur for which a determined pipe branch in the distribution line holds an excessive pressure with respect to the desired level. Such situations commonly arise when a single elevated water tank is implemented to pressurize the whole network for reliability reasons and at the same time the population served is spread over a non-geographically uniform territory as shown in Figure 3-7.

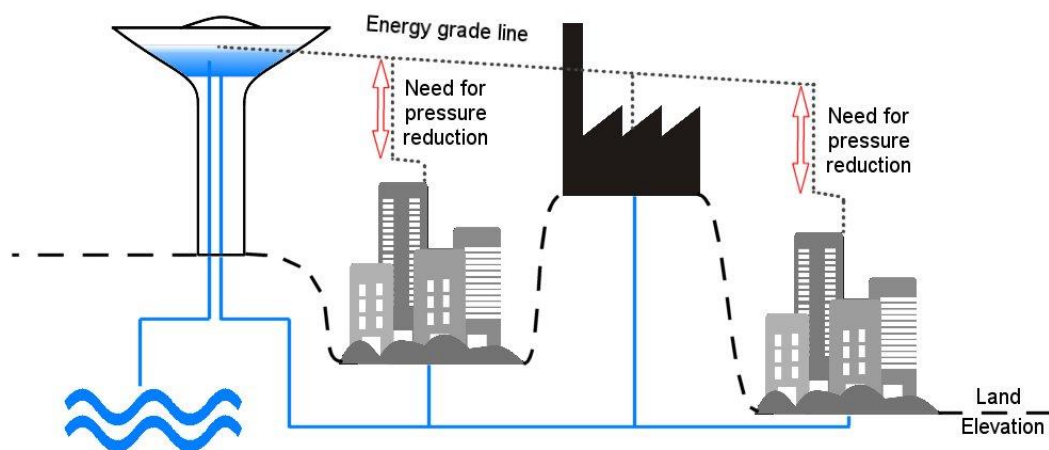


FIGURE 3-7. SCHEME OF A WATER NETWORK HAVING PIPE BRANCHES HOLDING AN EXCESSIVE WATER PRESSURE.

While it is important to maintain a sufficient pressure in the pipes to avoid water contamination, systematic overpressures do not only constitute an obvious waste of the energy for pumping processes but also can increase significantly the system leakage losses. In fact, water leakages occurring through pipe fissure or defective junctions constitute a waste of resources, an increase in maintenance needs, and produce economic losses. Besides, reducing pressures also means reducing the frequency of pipe bursts and lowering the pressure-related water consumption – e.g. garden watering (Mistry, 2011). A common solution to avoid such overpressure is the application at chosen nodal points of Pressure Reducing Valves (PRVs) able to lower pressure down to the desired level (Mistry, 2011; Ramos, et al., 2004). Ordinary PRVs have small dimensions and are designed to operate in line with the water pipe according to different control modes: constant pressure drop, adjustable or fixed downstream pressure level set (Ramos, et al., 2004).

The flow rate and pressure drop through any given valve can vary largely along a typical day, and is subject of sudden seasonal increases or decreases in water consumption patterns due to human activities (Corcoran, et al., 2012). Pumps as Turbines (PATs) have been pointed out by many studies as a technology fit for PRV substitution or alongside installation in order to reduce excess pressure and generate electricity alongside (Ramos, et al., 2005; Ramos, et al., 2010a; Giugni, et al., 2009; Gaius-obaseki, 2010).

In her 2014 Master Thesis, Tânia Calado modelled on EPANET two water distribution networks located in the Portuguese municipality of Loures in order to assess the viability of substituting PRVs

of current use with PATs acting as pressure control equipment. Results show that the use of PATs can reduce the water pressure without compromising the effectiveness of the network and provide a similar effect as PRVs. The investment payback time for an installed capacity of about 4kW has been found to vary from 8 up to more than 30 years according to the hydraulic conditions, magnitude of FIT at which energy is sold and the discount rate (Calado, 2014).

Another research study focused on the analysis of a water network in the Naples region of Italy, in order to identify the optimal locations for installation of PATs. Results show that the choice of optimal PAT nodal position for the maximization of energy yield and the minimization of water losses due to pressure-driven leakage doesn't lead to the same results. Additionally, the installation in the cited network of multiple PAT units having nominal power of 9 and 20kW was analysed in financial terms, showing a payback period between 2.5 and 3 years in all cases (Fontana, et al., 2011). However, authors recommended to consider such figures merely as representing an order of magnitude since several cost factors were neglected (e.g. grid connection tariff and installation of control devices) and the cost of civil works was estimated as a mere 30% of PAT cost (Fontana, et al., 2011).

De Marchis et al. (2014) performed a preliminary analysis assessing the viability of PAT use as a pressure control device applied to the water distribution network of Palermo city, Italy. The studied network was designed to supply around 35,000 inhabitants and deliver a maximum of 400 l/capita per day with pipe diameters ranging from 110 to 400 mm. A network model was built and validated with recorded data, and four different nodal points were chosen and evaluated as preferred locations for PAT installation. The authors proposed a cost figure of 2,000 €/kW for hydromechanical equipment purchase, including PAT, motor/generator unit and related electrical components. Besides, the extents of civil works were estimated as 30% of the cost of hydromechanical equipment as proposed by a previous study (Fontana, et al., 2011). The size of the four evaluated PATs ranges from 1.9 to 8.6 kW and the estimated project payback varies between 2 and 6 years (De Marchis, et al., 2014). However, such values reflected the author's choice of neglecting many cost factors such as design, planning, grid connection, installation and authorization costs which may have a relevant influence on final cost.

The same research group has subsequently evaluated a second network in Italy to identify the most convenient locations for the installation of PATs as pressure reducing devices within it (De Marchis, et al., 2016). The payback time was found to vary between 2 and 12 years, but the results

showed that private storage tanks, widely used by water end users in the Mediterranean basin, may jeopardize the water demand at nodes and therefore the energy production from a PAT with respect to the results of numerical simulations (De Marchis, et al., 2016).

Fecarotta et al. (2014) proposed a method for the selection of the most appropriate PATs as replacements of PRVs in which economic benefits derived both from energy production and from leakage rate reduction. The methodology, which was based on the Variable Operating Strategy described later in Chapter 3.6.7, was then applied to a hypothetical network according to six scenarios featuring a different number of turbines and different prices for water and electricity. Results demonstrated that water savings had a higher impact than revenues generated from electricity sales, on the plant Net Present Value (Fecarotta, et al., 2015).

The US based Company Rentricity is specialized in design and construction of MHP schemes harnessing the excess pressure in pipes of water distribution networks. Following on from a 2006 feasibility study, in 2014 a PAT in series with a control valve was installed in a bypass to a PRV station existing in the network owned by the city of Barre, Vermont. The main challenges of the project were the tuning of the close-coupled control system between the PAT and related valves in order to maintain the prescribed downstream pressure at any flow rate and in presence of hydraulic transients. The nominal PAT power was 12kW, corresponding to a 77m head drop, flow rate of 25l/s and efficiency of 64%. The typical yearly energy yield corresponded to around 90MWh and the total project cost corresponded to 175,000\$. Consequently, the unit cost of Barre PAT scheme resulted equal to 10,800€ per installed kW (Rentricity, 2014). Such a cost figure is remarkably high with respect to similar projects, which is probably due to the case-specific need for civil works and the sophisticated flow and pressure control equipment implemented. The project has been considered viable thanks to a Vermont state Clean Energy Fund which covered the 57% of total expenditure (Rentricity, 2014).

In her PhD thesis, Tracey Lydon presented a case study of PAT application within the Dublin water distribution network as a pressure reducing device running in parallel to an existing PRV located under Black Horse Bridge. Two alternative scenarios were analysed: installation of a single 31kW PAT unit or of two PATs in parallel for a total capacity of 41.7kW. The unit cost of installation was estimated as 3,090 and 4,165€/kW respectively, with a payback time of 4.7 and 6.7 years (Lydon, 2016).

Zeropex AS was a Norwegian turbine manufacturer commercializing the Difgen pressure control system consisting of a lobe turbine, generator, variable drive and electric cabinet. The system was purposely designed for inline installation as PRV replacement for oil/gas industry and later applied to the water sector, providing control of pressure drop and flow rate while avoiding surges and waterhammer phenomenon (ABB, 2015). Difgen systems were available in a variety of sizes capable of processing from 7 to 300l/s with a maximum operating pressure of 10 bar (Zeropex, 2012). Two pilot plants were built as replacement of pressure reducing valves at the inlet of water treatment works operated by Welsh Water in 2013 and 2014. However, these schemes have been decommissioned after a few years (Welsh Water, personal communication, 2018) due to a series of operational issues:

- low efficiency (one unit could only yield 8-12 kW instead of the 30 kW expected);
- maintenance problems, low life of bearings and couplings;
- excessive noise and vibrations induced in the pipeline.

The above issues can be related to the specific technology of the Zeropex Difgen units (rotary lobe PAT) and have not been reported in other centrifugal PAT installations.

Researchers from Bolzano, Italy performed a lab testing campaign on a PAT followed by a feasibility study regarding the deployment of a 19.2 kW centrifugal unit as energy recovery device within an existing aqueduct (Rossi, et al., 2016). The daily energy production was computed as 338 kWh with a shaft speed of 2,900 RPM. However, the authors of the study in order to identify the key parameters of the PAT relied on applying the affinity laws later shown in Equations (6)-(8) on the tested prototype. This choice resulted in the identification of a PAT whose impeller size does not match with the standard offer of normalized pumps and would therefore require a special manufacturing process.

A paper from the National Research Council of Canada focused on the viability of micro hydro turbines as pressure control devices in water networks. Based on pressure and flow values of a hypothetical water main, the authors proposed a probabilistic analysis using Monte Carlo simulations to assess the turbine performances using daily flow profiles as a random variable in order to achieve a more robust design. Results showed that the water demand pattern had a significant impact on project payback time (Colombo & Kleiner, 2011).

3.4.3. HYDROPOWER POTENTIAL IN IRRIGATION NETWORKS

Irrigation networks are a common infrastructure in countries where a significant deficit between precipitation and evapotranspiration during dry months occurs with high frequency. Such networks can be either made up of open irrigation channels or pressurized pipe systems and may incorporate opportunities for low-head hydropower with significant environmental and economic benefits associated (Estrada Tarragó, 2015).

A research group from Italy in 2015 proposed a method to analyse irrigation networks and assess their energy recovery potential from data obtained in computerized Geographical Information System (GIS) databases. The methodology developed was applied to the irrigation networks of the Italian Piedmont region and resulted in an estimated potential for MHP development between a conservative value of 3.5MW up to 9MW (Butera & Balestra, 2015).

3.4.4. HYDROPOWER POTENTIAL IN CORRESPONDENCE OF STORAGE RESERVOIRS

Elevated water storage facilities such as the one shown in Figure 3-9 are a common solution to provide an adequate pressure to the distribution network and at the same time ensure the system reliability enabling water supply by means of gravity action without the need for electricity (Twort, et al., 2000). Such storage tanks are kept at atmospheric conditions, but are fed with pressurized water. Overpressures are usually dissipated in inlet valves, whose replacement with a hydraulic turbine could produce significant amounts of energy as schematically displayed in Figure 3-8 (Novara, 2016).

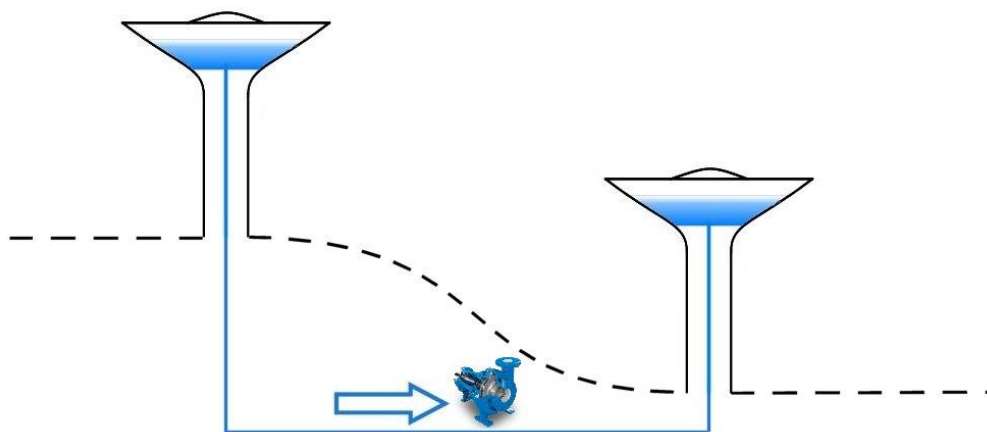


FIGURE 3-8. SCHEMATIC DRAWING OF ENERGY RECOVERY POTENTIAL BETWEEN TWO STORAGE RESERVOIRS AT DIFFERENT HEIGHTS.

A significant example of energy recovery from water entering a service reservoir is the design of a hydropower plant at Beiriz water storage reservoir located in Northern Portugal. The reservoir is divided into three tanks with an overall capacity of 10,000 m³ having maximum height equal to 3m. Installation of a Crossflow turbine as replacement of the previously existing intake valve was investigated, with results showing that up to 52MWh/year of electricity could be generated (Cenor-Projetos de Engenharia, 2011; Samora, 2016b). The flow duration curve relative to inflow measured at Beiriz reservoir is shown in Figure 3-10, with the values of minimum and maximum flow rate achieved by the chosen turbine highlighted.



FIGURE 3-9. ELEVATED CONCRETE-MADE WATER TOWER AT SCURZOLENCO (IT).

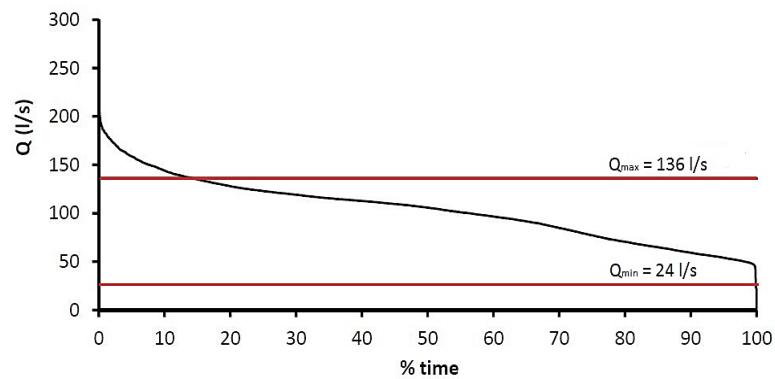


FIGURE 3-10. AVERAGE FLOW DURATION CURVE AT INLET OF BEIRIZ RESERVOIR (PT) AND LIMITS OF OPERATIONS OF CHOSEN TURBINE (CENOR-PROJETOS DE ENGENHARIA, 2011).

3.4.5. HYDROPOWER POTENTIAL AT WASTEWATER TREATMENT INFRASTRUCTURE

Many infrastructures for wastewater treatment potentially offer opportunities for MHP installations since they generally offer low heads counterbalanced by large and easily predictable flow rates as schematically shown in Figure 3-11. Besides, implementation of MHP in correspondence of water treatment plants could reduce the investment cost and avoid typical hydropower shortcomings such as visual impact, sediment accumulation and interference with ecosystems (Power, et al., 2016). Additionally, a secondary added benefit brought by hydro installations is the increased oxygenation in the water stream (Zakkour, et al., 2002).

Waste Water Treatment Plant (WWTP)

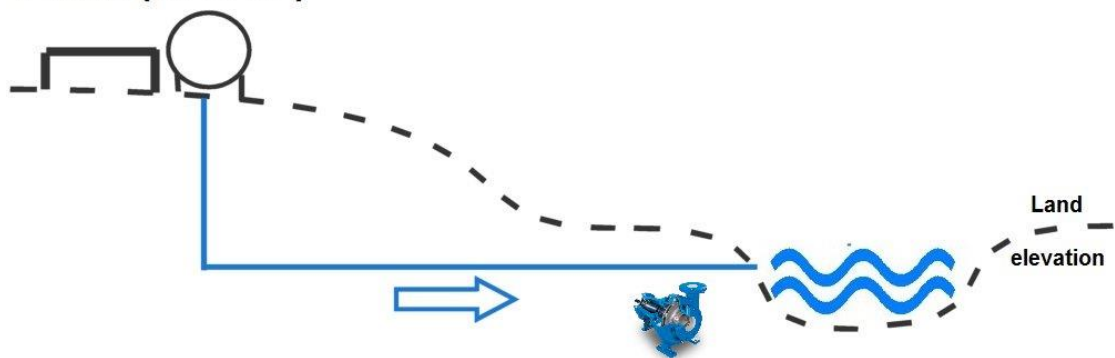


FIGURE 3-11. SCHEMATIC DRAWING OF ENERGY RECOVERY POTENTIAL AT THE OUTFALL OF A WASTE WATER TREATMENT PLANT.

A study was carried out to establish a procedure to assess the potential for hydropower energy recovery from wastewater systems, either upstream or downstream of Waste Water Treatment Plants (WWTP). Inputs to the algorithm were Geographic Information System (GIS) data and inflows from a number of WWTPs, together with local technical and economic constraints. Such methodology was then applied to the case-study of Switzerland, covering around 8 million inhabitants over an area of 41,000 km². Results indicated that a cumulative potential of 9.3 GWh/year is economically viable, corresponding to 19 WWTP sites out of which six are already equipped with hydro generation schemes (Bousquet, et al., 2017). Extrapolating such data to the whole Europe, totalling 740 million of inhabitants as for 2010 (UN Dept. of Economics and Social Affairs, 2015), the rough figure of potential hydraulic energy recovery from wastewater process only could correspond to 860 GWh/year.

In order to accommodate the large flow rates and low head differences typical of many WWTP sites, unconventional low-head energy converters have been considered. An example of this is the 75kW Archimedean screw installed in 2015 at the outfall of Csepel Island treatment plant shown in Figure 3-12. Commissioned in 2010, the plant is a large complex designed to treat 50% of effluents produced in the Budapest metropolitan area (Budapest Waterworks, 2017).



FIGURE 3-12. THE 75KW ARCHIMEDEAN SCREW CONVERTER AT CSEPEL ISLAND WWTP, HUNGARY.

3.5. MICRO-HYDROPOWER

3.5.1. KEY CONCEPTS OF HYDROPOWER

The term “hydropower” refers to a vast series of technologies allowing the generation of electricity from the potential, kinetic or piezometric energy held by a mass of water. The first applications of hydropower concepts date back to the 19th century and since then a large number of hydro power plants have been built, giving a fundamental contribution to the processes of electrification and industrialization. With respect to other renewable power sources, energy from water is reputed to offer high cost-effectiveness, long lifetime and high operational flexibility (International Energy Agency, 2010). Besides, the greenhouse gases (GHG) emissions rates of electricity produced

through hydropower are among the lowest across all technologies and vary between 2 g of CO_{2,eq} per kWh up to 180 when the construction of a new artificial reservoir in a tropical climate is involved (Steinhurst, et al., 2012). Not only are hydropower technologies able to operate as both base and peak load, but also play a fundamental role in grid stability thanks to pumped storage water plants which are able to store excess electricity and supply it when best needed. Most hydro development plans to-date have been based on construction of large artificial reservoirs able to respond to many needs (electricity, drinking water, irrigation) but which bring about significant environmental and social impacts such as modifying hydrological regimes, worsening water quality, creating a barrier to sediment transport and fish migration, displacing communities, damaging cultural heritage and inundating fertile lands (Paish, 2002; Kumar, et al., 2011).

The impact of hydropower towards climate change mitigation has been generally evaluated as positive since being a vastly used renewable energy source it offsets a large portion of Greenhouse Gas (GHG) emissions from fossil fuel-based generation technologies. Besides, the creation of artificial multipurpose water reservoirs is deemed to increase the resilience against changes in the water availability patterns (Berga, 2016). In fact, even though the impact of climate change on global or continent-wide precipitation patterns has been evaluated as negligible it is predicted that there will be significant locally determined variations in water resource availability associated with a general increase in the frequency of extreme events - draughts or floods - which may affect the existing and future hydropower installations (Kumar, et al., 2011). Potential for further development and climate change mitigation is highlighted in the opportunities that exist in multipurpose hydropower (e.g. turbines in water supply pipe networks or in existing dams without a turbine). Challenges facing the technology and its applications include the lack of low-cost alternatives to conventional large machines. A lack of reliable information on the cost of micro hydropower in the literature is also highlighted as a barrier to future development (Kumar, et al., 2011).

Some researchers have suggested that in the context of climate change the development of new Small-Hydropower installations (SHP) without a proper planning may reduce the amount of water available for other purposes (e.g. domestic use and irrigation) and increase the vulnerability of the ecosystem as depicted in Figure 3-13. In order to avoid such impacts the authors recommend shifting the focus from mere power generation towards an integrated management of water resources as previously described in Chapter 3.3 (Kelly-Richards, et al., 2017).

Overall, after 2014 a declining trend in the commissioning of new hydro capacity based on large reservoir projects is expected as shown in Figure 3-14 due to environmental concerns, reduced availability of suitable sites and reduced economical attractiveness of such programs (International Energy Agency, 2015).

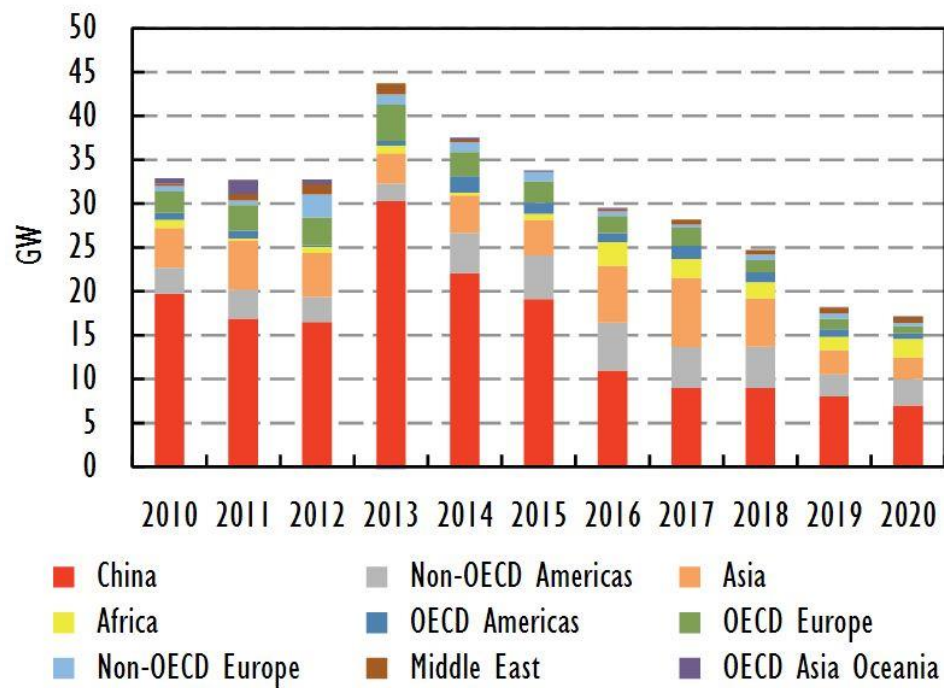


FIGURE 3-14. FORECASTED HYDROPOWER ANNUAL CAPACITY ADDITIONS BY REGION (INTERNATIONAL ENERGY AGENCY, 2015).

3.5.2. WHY MICRO?

Technologies for hydropower on a small scale tend to be clean and cost-effective and offer excellent prospects for new renewable energy installations in Europe as well as offering a high potential for rural electrification in less developed countries (Paish, 2002).

With respect to large hydro projects, small and in particular Micro-hydropower (MHP) offer shorter gestation periods, lower capital requirements and simpler design and commissioning. At the same time, since they do not rely on the building of large reservoirs, they have a lower land use and reduced environmental impact. Moreover, they encourage participation of the local communities and commonly employ local manpower (Thilak, 2010). Despite the unavailability of specific data on Micro hydro, having installed power less than 100kW, the global Small hydro installed capacity has

been calculated as 110GW in 2015 and is expected to experience a 2.5% yearly growth up to 2024 (Global Market Insights, 2016).

The economic impact on small communities in Wales of Micro-Hydropower (MHP) schemes has been compared to that of other renewable energy sources in a report for the Hydropower Stakeholder Group (Bere, et al., 2015). Even if the cumulative contribution of MHP schemes towards Wales's total energy demand fell below 1% as of 2015, such plants proved significant in terms of income and new employment opportunities generated at the full benefit of local communities. The researchers found that for every 1MW of MHP capacity installed 10 full-time jobs were created (defined as person-years of employment) which compares positively to the performance of other renewable energy sources – e.g. solar PV on average only resulted in 3.3 jobs (Bere, et al., 2015). The main challenges towards a more widespread MHP adoption in Wales were identified by the authors as the need of upgrading power transmission lines and of coping with the decline of Feed-In-Tariffs (Bere, et al., 2015). As explained in a newspaper article based on the previously cited report, the small scale of MHP schemes and the concept of community ownership of new installations are powerful antidotes to investment costs otherwise inaccessible to individuals and to the public opposition that large renewable energy projects are often faced with (Meadway, 2015).

3.5.3. *TURBOMACHINERY GENERALITIES*

Hydraulic energy converters are specifically designed to transform energy of a fluid stream into rotating mechanical energy at the shaft. The power output of a hydraulic turbine is defined as:

$$P = \eta g \rho H Q \quad (1)$$

where:

P: power output (W)

η : machine efficiency (-)

g: gravitational acceleration, equal to 9.81 (m/s²)

ρ : fluid density (kg/m³)

H: net head (m)

Q: nominal flow rate (m³/s)

Available hydraulic potential is usually expressed as head H having measure unit of meters of equivalent water column height (m). The complete formulation for determining H is shown in Equation (2), and accounts for fluid pressure, altitude and velocity.

$$H = \frac{p}{\rho g} + z + \frac{U^2}{2g} \quad (2)$$

where:

p : pressure (Pa)

ρ : fluid density (kg/m^3)

g : gravitational acceleration, equal to $9.81 \text{ (m/s}^2\text{)}$

z : geometric height (m)

U : flow mean velocity (m/s)

In turbomachinery performance analysis affinity laws are widely used to express relationship existing between power and other interrelated parameters. The concept of specific speed (N_s) is used to distinguish families of geometrically similar machines, defined as follows:

- according to discharge value, for pumps or turbines:

$$N_s = N \frac{\sqrt{Q}}{H^{3/4}} \quad (3)$$

An alternative formulation of the same concept is the so-called “non-dimensional specific speed” Ω defined as:

$$\Omega = \omega \frac{\sqrt[2]{Q}}{(gH)^{0.75}} \quad (4)$$

- according to power, for turbines:

$$N_{s,p} = N \frac{\sqrt{P}}{H^{5/4}} \quad (5)$$

where:

N : rotational speed (RPM)

H : available head (m)

Q: flow rate (m³/s)
 ω : rotational speed (rad/s)
g: gravitational acceleration (m/s²)
P: power at shaft (kW)

When referring to specific speed in the rest of the manuscript in the absence of a specific subscript “p” the parameter N_s will refer to the formulation of specific speed according to the discharge.

For a family of geometrically similar pumps or turbines (i.e. having the same specific speed) it is possible to relate between them the main characteristic parameters through the following mathematical expressions:

$$\frac{Q'}{Q} = \frac{N'}{N} \left(\frac{D'}{D} \right)^3 \quad (6)$$

$$\frac{H'}{H} = \left(\frac{N'}{N} \right)^2 \left(\frac{D'}{D} \right)^2 \quad (7)$$

$$\frac{P'}{P} = \left(\frac{N'}{N} \right)^3 \left(\frac{D'}{D} \right)^5 \quad (8)$$

According to the presented affinity law the efficiency at the BEP of any similar machine assumes a constant value η independent from variations in rotational speed and impeller diameter, which isn't fully representing real PAT behaviour. In fact, affinity laws disregard the effects of parameters which do not directly depend on velocity (e.g. viscosity) and therefore may lead to inaccurate results in term of power prediction (Simpson & Marchi, 2013). However, despite the existence of alternative methods developed to predict behaviour of similar PATs (Fecarotta, et al., 2016), affinity laws are widely considered as a simple yet quite reliable method to be used in turbomachinery design.

3.5.4. HYDROPOWER CLASSIFICATION BY SIZE AND TYPE

Hydropower schemes are usually sorted into categories according to nominal capacity and plant type as presented in Table 3-1 and Table 3-2 respectively.

TABLE 3-1. CATEGORIES OF HYDROPOWER PLANTS BY TYPE.

Impoundment hydro	Associated with a reservoir
Run-of-river hydro	Without storage capacity, diverting a portion of flowing water from a river
Pumped storage hydro	Pumping or turbinng water from two reservoirs at different heights
In-stream hydro	Utilizing directly flowing water in channels or pipes
Multipurpose hydro	A scheme apt for energy generation as a complementary benefit within an engineering project pursuing a different primary scope

It must be noticed however that the boundaries between categories of Table 3-1 are often blurry, since a single plant may fall under multiple classifications (International Energy Agency, 2015).

TABLE 3-2. CATEGORIES OF HYDROPOWER PLANTS BY SIZE AND RELATIVE UTILIZATION OF THE GENERATED POWER (IRENA, 2012).

Denomination	Capacity range	Power utilization	
		Connected to a grid	Stand-alone
Large-hydro	$P > 100\text{MW}$	Always	/
Medium-hydro	$20\text{MW} < P < 100\text{MW}$	Mostly	/
Small-hydro	$1\text{MW} < P < 20\text{MW}$	Usually	Rarely
Mini-hydro	$100\text{kW} < P < 1\text{MW}$	Possibly	Rarely
Micro-hydro	$5\text{kW} < P < 100\text{kW}$	Possibly	Possibly
Pico-hydro	$P < 5\text{kW}$	Possibly	Usually

3.5.5. HYDRO CONVERTERS REVIEW

Apart from the conventional bespoke hydropower turbines several other devices have been proposed or implemented to extract energy from a water stream, as can be seen in Figure 3-15 where different technologies are classified according to their operating principles.

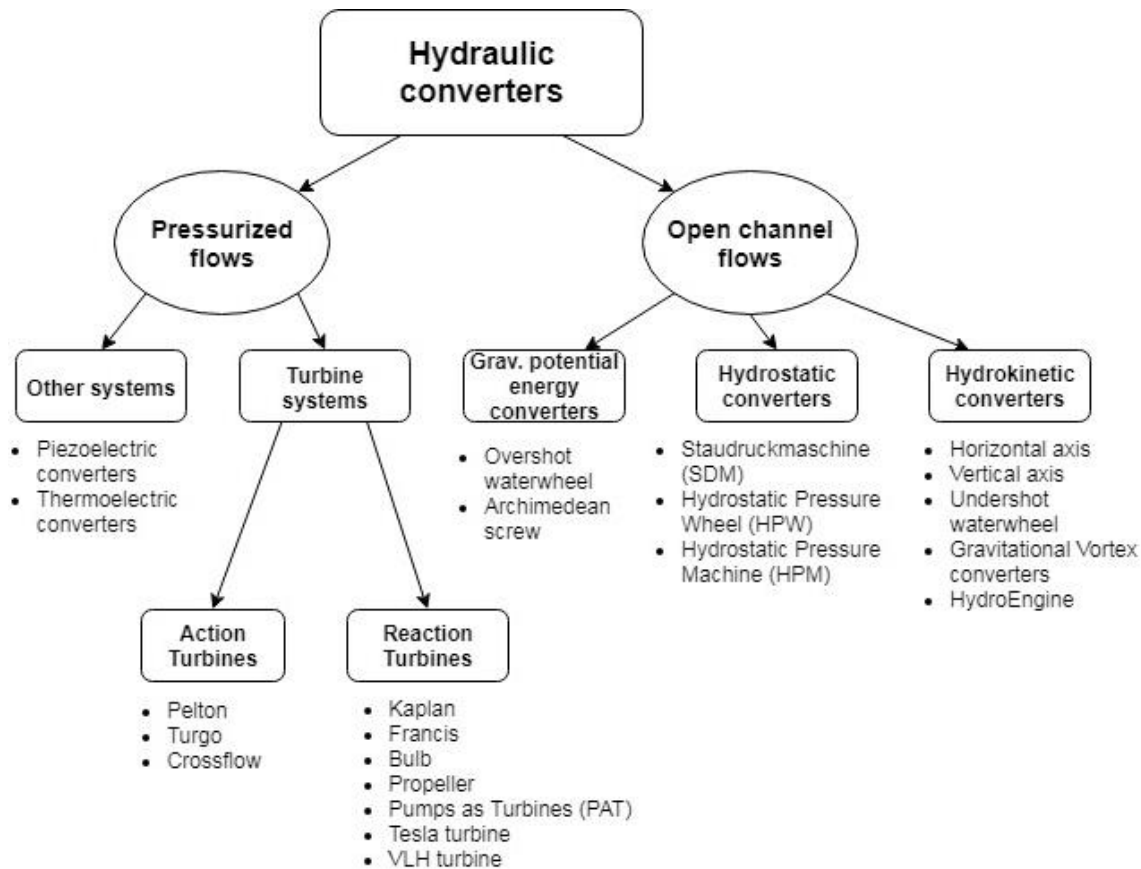


FIGURE 3-15. TECHNOLOGY BRANCHES OF HYDROPOWER ENERGY CONVERTERS SUITED FOR EITHER PRESSURIZED OR OPEN CHANNEL FLOWS.

3.5.5.1. Pressurized flows

- Turbine systems
 - Action turbines
 - Pelton - First patented in 1880, the Pelton wheel consists of a circular arrangement of double-spoon shaped buckets mounted along a disc. It is one of the most common turbine types for high heads, and is manufactured on a scale from Pico and Micro power output to Megawatt-size units.

- Turgo - Originally conceived as a modification of a Pelton wheel design, the Turgo wheel is a simpler turbine type able to work under medium head and flow parameters.
- Crossflow – Derives its name from the fact that by design incoming water crosses twice the blades when entering and exiting the impeller. Despite being slightly less efficient than other turbine types it is inexpensive, highly flexible and can cope with debris in the water flow without getting damaged.
- Reaction turbines
 - Kaplan – Widely used turbines for low heads schemes, they resemble a regular propeller but have in-built active control surfaces (blades pitch and distributor geometry).
 - Francis – Among the most common and iconic turbine type widely used at utility-size plants, made up of a spiral distributor and a fully submerged impeller.
 - Propeller – Similar to a Kaplan turbine but without moving parts. Within the framework of the European Project HYLOW started in 2008 a new concept of tubular propeller turbine was developed specifically to be installed within existing pipes with minimal modifications. Two versions of the turbine have been developed, having four and five-bladed propeller, and the experimental campaign allowed to assess the efficiency of the converter as equal to 60% under design low-head conditions (Samora, et al., 2016a). The turbine was designed as a modular unit to be installed in a pipe bend and the possibility of multiple runners in series configuration is possible in order to produce a larger head drop.
 - Tesla - Invented by Nikola Tesla in the early 20th century but still lacking practical application examples, Tesla turbines are machines of simple design which rely on fluid viscosity to produce mechanical power (Ho-Yan, 2011). Despite being considered as potentially suitable for application in pico hydro generation, their global efficiency has been experimentally proved to lag well behind that of conventional water turbines.

- Pumps as Turbines (PAT) – Nonconventional hydraulic energy converters consisting of water pumps running in reverse mode. Extensively described in Chapter 3.6.
- Rotary lobe PAT – A specific sub category of PATs based on rotary lobe sludge booster pumps which includes the Difgen turbogenerator described in Chapter 3.4.2.
- VLH turbine - Developed by MJ2 Technologies, the VLH turbine consists of a propeller connected with a variable speed generator and assembled in a movable housing protected by a sediment protective screen (Bozhinova, et al., 2013). Such units are designed in open-air concrete water channels with head differences between 1.4 and 3.2 m, flow rate of 10-30m³/s and nominal power between 100 and 500kW (MJ2 Technologies, n.d.).

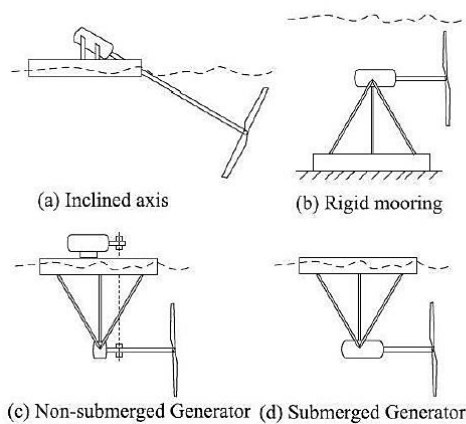
3.5.5.2. Open channels flows

- Gravitational potential energy converters
 - Overshot waterwheel - Waterwheels are among the oldest sources of mechanical power used, and were a relatively common technology before disappearing in the 1950s (Bozhinova, et al., 2013). Overshot waterwheels are composed of a series of buckets arranged circularly and are fed with falling water from above the level of wheel axis. Among other types of waterwheels, they are the most efficient - up to 90% for a large range of flow rates - although they require a head difference of 2.5-10m to be effective (Müller, 2013)
 - Archimedean screw - Despite being one of the oldest hydraulics machines known to mankind, the first studies regarding its utilization as a turbine rather than as a pump date back to recent years (Müller & Senior, 2009). It consists of a helical solid profile inserted in an inclined rotating drum. In literature such converters are shown to have nominal power in the range 3-300kW and maximum achievable efficiency of around 80% (Bozhinova, et al., 2013).
- Hydrostatic converters – Compact waterwheel design for application in open channels with reduced head differences able to act as a weir to maintain a level difference between two sides

of a channel. The initial prototype was named Staudruckmaschine (SDM), while the second-generation designs developed along the period 2007-2013 under EU's HYLOW project from universities of Southampton and Darmstadt were named Hydrostatic Pressure Wheel (HPW) and Hydrostatic Pressure Machine (HPM).

- Hydrokinetic converters:
 - Horizontal and vertical axis converters - As reviewed by (Khan & al., 2009), a large number of hydrokinetic converters have been studied or deployed in order to produce energy from either inland water bodies (e.g. channels and rivers) or offshore resources like waves, marine currents or tides. Most of proposed designs consist in vertical or horizontal axis units, of which an overview is given in Figure 3-16 (Lago, et al., 2010).

a) Horizontal axis hydrokinetic converters



b) Vertical axis hydrokinetic converters

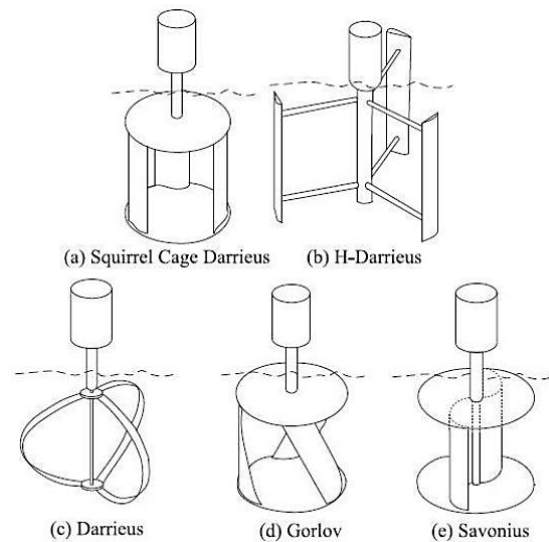
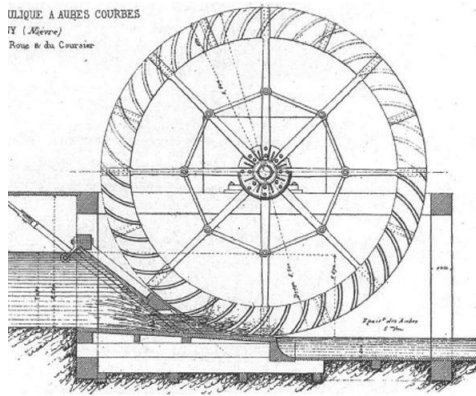


FIGURE 3-16. VISUAL REPRESENTATION OF MOST COMMON TYPES OF HYDROKINETIC WATER ENERGY CONVERTERS: A) HORIZONTAL AXIS CONVERTERS; B) VERTICAL AXIS CONVERTERS (KHAN & AL., 2009).

- Undershot waterwheel - When a waterwheel is fed with water at the level of its wheel axis or below it is called breastshot and undershot respectively (Müller, 2013). The most common wheel designs in this field are the Poncelet and the Zuppinger wheel, typically built with large diameters (3-6m) and able to process large flow rates with a minimum head difference of 0.5m. Typical values of peak efficiency are equal to 65% for the Poncelet wheel and 75% for Zuppinger units (Bozhinova, et al., 2013). The representative power output from such units is up to 16kW per meter width (Müller &

Wolter, 2004), and sketches of a Poncelet and a Zuppinger waterwheel are shown in Figure 3-17.

a) Poncelet waterwheel (Poncelet, 1827)



b) Zuppinger waterwheel (Müller, 1899)

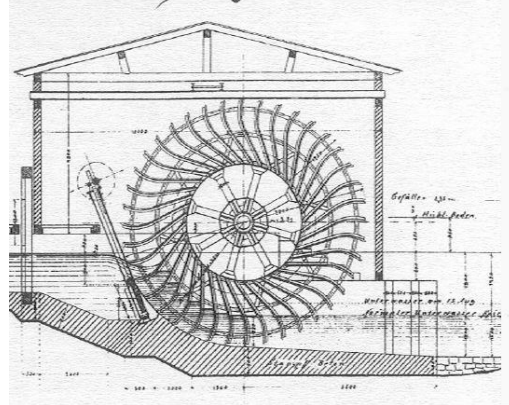


FIGURE 3-17. VISUAL REPRESENTATION OF PONCELET AND ZUPPINGER WATERWHEEL DESIGNS.

A series of lab model tests and desk studies developed at Queen's University Belfast suggest that a Breast wheel with improved geometries could reach a top efficiency of 87.3% and compare positively with other hydro turbines for low heads both in terms of ecological impact and installation cost. In particular, a unit equipped with synchronous belt drive cascade and generator speed drive could have installation cost per installed power of 4,600-5,100€/kW instead of 10,300-14,700€/kW for similar-sized standard Kaplan turbines (Müller & Wolter, 2004).

- HydroEngine TM – Commercialized by Natel Energy as of 2019, it consists of a low-head hydraulic converter based on multiple rotating blades fixed on a belt (USCID, 2009).
- Gravitational Vortex converters – It consists of a round volute chamber in which water forms a vortex which drives the blades of a turbine located at the drain of the chamber.

3.5.6. COST COMPARISON OF MICRO-HYDROPOWER SCHEMES

Generally speaking, the cost per installed capacity of a MHP scheme is highly site-specific and dependent on a series of factors like location, amount of needed civil works, flow and head variability (Kirk, 1999; Ogayar & Vidal, 2009). The range of typical turnkey investment cost for Mini

and Micro-Hydropower have been previously estimated as displayed in Table 3-3 (European Commission, 2002).

TABLE 3-3. TYPICAL TURNKEY INVESTMENT COST FOR SMALL AND MICRO HYDROPOWER (EUROPEAN COMMISSION, 2002).

	Installed capacity	Electricity production cost	
		0.05 – 0.15 €/kWh	0.02 – 0.08 €/kWh
Small hydro	1 – 10 MW	600 – 2,000 €/kW	/
	500 – 1000 kW	1,300 – 4,500 €/kW	1,000 – 3,000 €/kW
	100 – 500 kW	1,500 – 6,000 €/kW	900 – 3,500 €/kW
Micro hydro	< 100 kW	1,500 – 6,000 €/kW	900 – 3,500 €/kW

Further information about the MHP unit investment cost available in literature show ranges between 3,700 and 7,400€/kW (Corcoran, et al., 2013) and 3,000 – 6,000€/kW (Gaius-obaseki, 2010). Another recent research assessed the investment cost for a 100kW MHP scheme as varying between 1,000 and 1,400 €/kW (McNabola, et al., 2011).

A US-based microturbines installer suggested in recent years an estimation of 5,300€/kW for schemes with nominal output around 30kW and 2,600€/kW for larger capacities (Colombo & Kleiner, 2011).

Other authors instead suggest higher values of unit cost in the range 10,300-14,700€/kW for small low-head Kaplan or Crossflow turbines (Müller & Wolter, 2004), while the 12kW PAT scheme described in Chapter 3.4.2 needed a unit investment cost of 10,800€/kW to be built (Rentricity, 2014). Also, a recent study suggests that replacement of Pressure Reducing Valves with MHP schemes could present unit investment cost higher than 12,000€ per installed kW (Gallagher, et al., 2015b). With such values the investment return time could easily extend beyond 10 years which is the limit beyond which projects are commonly classified as unviable (McNabola, et al., 2014b). The overall investment cost for the dual PAT installation at the Beliche reservoir in southern Portugal described in Chapter 3.6.3 was reported as 75,000 €. Dividing this value by the 11 kW of installed capacity the resulting unitary cost is equal to 6,818 €/kW (Livramento, 2013).

A further study which analysed the cost of MHP schemes comparing plants adopting Pumps as Turbines technology with others powered by conventional water turbines found that for nominal

powers lower than 200-300kW the former require a lower investment cost with respect to the latter (Ramos & Ramos, 2010b). A chart comparing the unit investment cost of hydropower plants adopting PATs with plants adopting standard hydro turbines is shown in Figure 3-18.

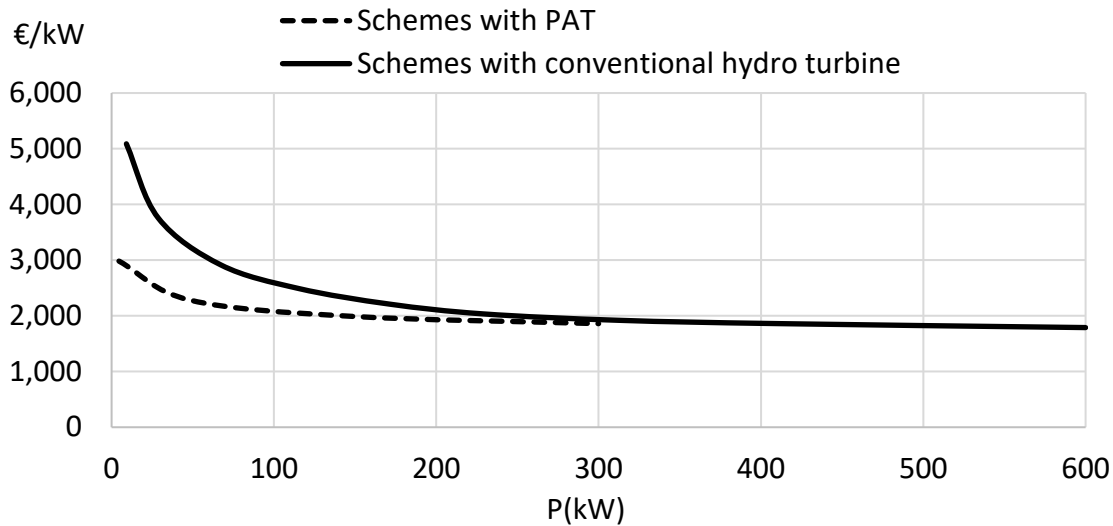


FIGURE 3-18. COMPARISON OF COST PER INSTALLED POWER FOR MHP SCHEMES ADOPTING DIFFERENT GENERATING TECHNOLOGIES (RAMOS & RAMOS, 2010B).

3.5.7. COST COMPARISON OF ELECTRO-MECHANICAL EQUIPMENT FOR MICRO-HYDROPOWER

The term “electro-mechanical equipment” or “generating set” refers to the assembly constituted by a turbine, the connected generator and the related electric devices. In typical MHP applications the cost of the electro-mechanical component can account for a portion of total project budget of between 30 and 40% (Ogayar & Vidal, 2009; Jain & Patel, 2014). But in hydropower schemes having nominal capacity in the range of a few kW the turbine/generator assembly may account for up to 70% of the total project cost (Jain & Patel, 2014; Gallagher, et al., 2015b). As a result, it is important to rely on models capable of predicting such relevant investment costs.

The Welsh-based Centre for Alternative Technology suggested the specific cost of a 5kW Pelton turbine professionally installed could correspond to 5,700€/kW (Centre for Alternative Technology, 2017), while due to economies of scale the total cost of a 2kW scheme is typically equal to the 67% of the cost of a 5kW one. Another report focused on the Welsh MHP sector suggested a unit cost between 5,000 and 8,000 GBP/kW (Bere, et al., 2015).

A chart displaying the cost envelopes of the generating set alone using data from hydro projects in the UK is shown in Figure 3-19. The unit cost of hydromechanical equipment for low head

applications is generally higher than the one for high head schemes, since as a general rule the latter will employ a generating set characterized by higher speed and reduced physical dimensions (Kirk, 1999). Figure 3-19 also displays the cost per installed kW of the whole generating set regarding Pelton, Francis and Kaplan turbines installed in Mini and Micro hydro schemes across Europe (Ogayar & Vidal, 2009).

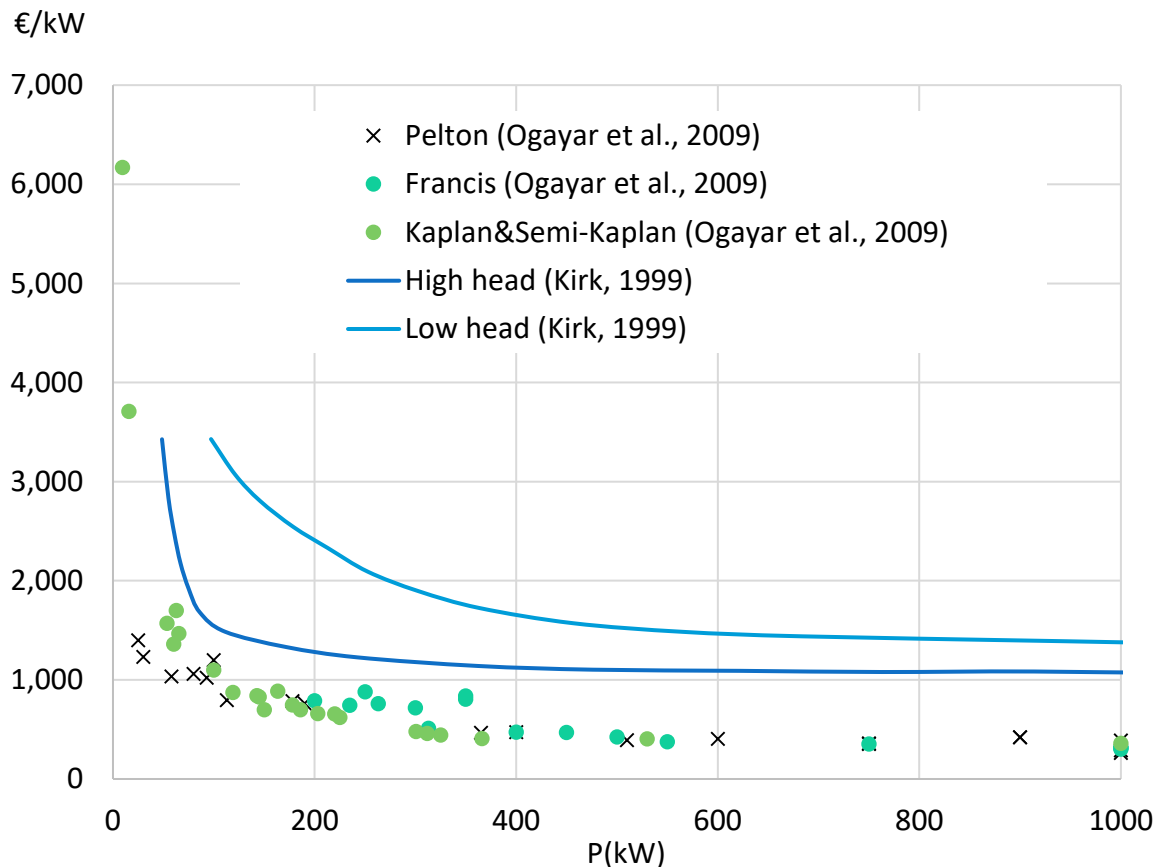


FIGURE 3-19. COST OF ELECTRO-MECHANICAL EQUIPMENT FOR SMALL AND MICRO HYDRO.

The gap between values of unit cost presented by Ogayar et al. (2009) and those predicted by Kirk as visible in Figure 3-19 can be explained by the fact the cost envelope dates back to ten years earlier with respect to the timeframe in which the data about actual cost of electro-mechanical equipment was gathered.

Within the same research, (Ogayar & Vidal, 2009) introduced a cost model for the prediction of purchase cost of electro-mechanical equipment in Mini and Micro hydropower based on parameters of nominal head (m) and installed power (kW). The corresponding equations are shown in Table 3-4.

TABLE 3-4. COST DETERMINATION OF ELECTRO-MECHANICAL EQUIPMENT FOR SMALL AND MICRO HYDRO
(OGAYAR & VIDAL, 2009).

Turbine type	Unit cost (€/kW)
Pelton	$17,693 \cdot P^{-0.364473} \cdot H^{-0.281735}$
Francis	$25,698 \cdot P^{-0.560135} \cdot H^{-0.127243}$
Kaplan	$19,498 \cdot P^{-0.583380} \cdot H^{-0.113901}$
Semi-Kaplan	$33,236 \cdot P^{-0.583380} \cdot H^{-0.113901}$

A research conducted more recently in Bangor University (UK) and Trinity College Dublin (IE) updated the cost equations for Pelton, Francis and Kaplan turbines of Table 3-4 using the same data as (Ogayar & Vidal, 2009) but accounting for inflation changes between 2008 and 2014 and excluding installations having nominal capacity higher than 300kW (Gallagher, et al., 2015b). Additionally, a similar equation based on P and H parameters was proposed to predict the specific cost of a pressure control turbine for use in water pipe networks named by the authors as “PC Turbine”. All four resulting equations are shown in Table 3-5 together with their respective ranges of capacities.

TABLE 3-5. UPDATED COST DETERMINATION OF ELECTRO-MECHANICAL EQUIPMENT FOR SMALL AND MICRO HYDRO (GALLAGHER, ET AL., 2015B).

Turbine type	Unit cost (€/kW)	Capacity range (kW)
PC turbine	$37,792 \cdot P^{0.531} \cdot H^{-0.119}$	11-30
Pelton	$7,316 \cdot P^{0.698} \cdot H^{-0.114}$	25-300
Francis	$150 \cdot P^{1.309} \cdot H^{0.028}$	80-300
Kaplan	$22.155 \cdot P^{0.440} \cdot H^{-0.152}$	9-300

Besides, within the same research authors used the equations of Table 3-5 to evaluate the acquisition cost of generating sets for MHP applications in selected water networks of Ireland and

Wales. The obtained results relative to energy recovery from elevated service reservoirs, PRV sites or WWTP outfalls are visible in terms of average, minimum and maximum specific cost in the chart of Figure 3-20 (Gallagher, et al., 2015b).

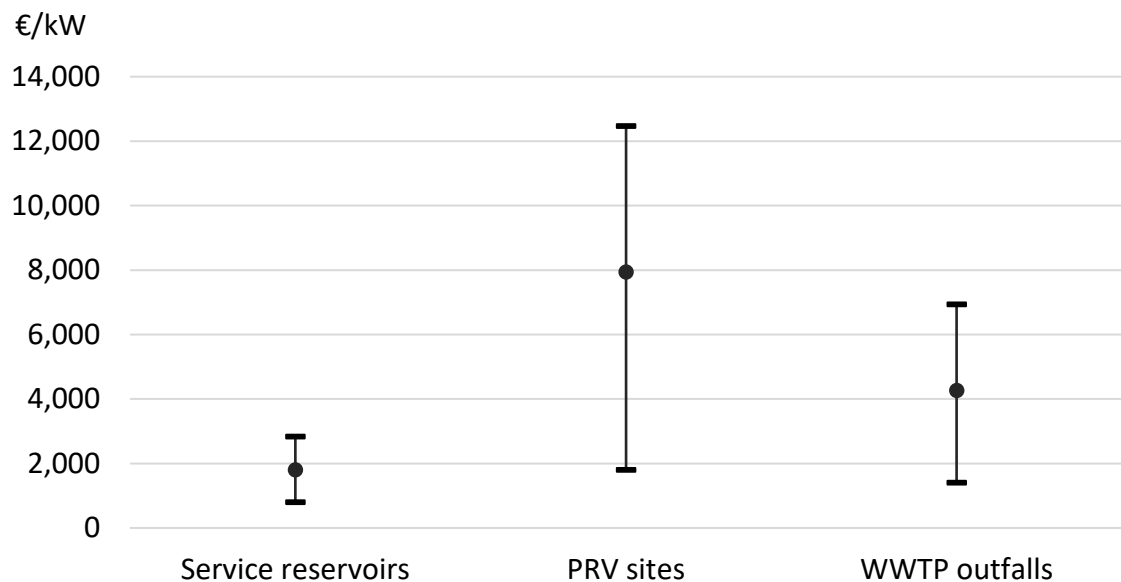


FIGURE 3-20. COST OF MHP GENERATING SET IN DIFFERENT APPLICATIONS WITHIN WATER NETWORKS: SERVICE RESERVOIRS, PRV SITES AND WWTP OUTFALLS (GALLAGHER, ET AL., 2015B).

3.6. PUMPS AS TURBINES

Pumps as Turbines (PATs) are a type of unconventional reaction turbine having behaviour similar to that of Francis runners. Their function is to convert kinetic and pressure energy of the fluid into mechanical energy of the runner. They are commonly commercialized as composite pump and motor/generator units, coupled by a fixed shaft to an asynchronous induction type motor unit (Agarwal, 2012; Williams, 1996). Unlike other conventional machines which require manufacture according to the client's specifications, pumps are a very common piece of equipment widely available in different sizes and functionality anywhere around the globe. When used as turbines, the rotor moves in the opposite direction as during pump operations and the motor unit may generate electrical power. With respect to a conventional Francis turbine, a PAT meant to work at identical duty point is characterized by a runner of larger dimensions having blades of opposite curvature as it is designed to avoid instabilities during flow deceleration when used in pump mode (Alatorre-Frenk, 1994).

The oldest recorded PAT installation started operations in Orchard Mesa, US in 1926 and consisted of a pump running as turbine powering a second pump used to supply water for irrigation purposes (Shafer, 1983). However, the first scientific study evaluating the possibility of running pumps in reverse mode with acceptable efficiencies dates back to the 1930s (Thoma & Kittredge, 1931). During the following decades an increasing interest has been raised on gaining an in-depth view of the physics behind pump behaviour in turbine mode (Singh, 2005).

An important reference parameter of every turbo machine is the Best Efficiency Point (BEP), defined as the coordinates in terms of flow rate and head drop between pump inlet and outlet which lead to the highest machine efficiency. The BEP of a pump differs significantly from the BEP of the same machine operated as a PAT, with the latter one being able to process a higher flow rate and produce a greater pressure drop respect to the first.

At first PATs application has been limited to a small range of niche cases namely pumped water hydro plants and energy recovery from excess pressure in chemical plants (Singh, 2005). In more recent times, PATs have been implemented and studied as efficient hydro converters for low-head hydropower and in-pipe energy recovery in WSS. Since the main limiting factor to the development of small scale hydropower is the high investment cost of the hydromechanical equipment, PATs are a recommendable solution to promote small hydro development (Agarwal, 2012; Motwani, et al., 2013). PATs although more site-dependent present lower cost of electricity per household with respect to solar PV arrays in the context of rural electrification in developing countries (Maher, et al., 2003).

The main advantages of PATs over conventional hydro converters can be summed up as (Alatorre-Frenk, 1994; Chapallaz, et al., 1992; Garay, 1990; Williams, 1996):

- compact dimensions;
- resistance to abrasion (e.g. by silt embedded in stream water) and corrosion;
- ability to process two-phase flows and work in explosive-risk environment;
- they are mass manufactured and readily available with short delivery time;
- highly standardized, easy maintenance and prompt availability of spare parts;
- low investment cost;
- proven results and effectiveness.

Despite this, the main shortcomings of PAT application are:

- high uncertainties of PAT performance prediction from pump mode characteristic curves;
- lack of information about performance charts of a large number of PATs;
- rigid geometry, causing low part-load efficiency and impossibility of flow modulation.

Additionally, when selecting the turbine and generator type to empower a MHP scheme, key choice parameters must include: (a) degree of standardization and simplicity, and (b) an evaluation of an appropriate technology (Kirk, 1999). Such factors tend to favour the application of reverse running pumps, and balance the mentioned PAT drawbacks (Orchard & Klos, 2009).

The operation of a pump/turbine unit can be described in quadrants in order to represent the various combinations of key parameters Torque (T), Head (H) and rotational speed (n) as seen in Figure 3-21. As can be noticed from the cited chart the window of operation of PATs is limited between the occurrence of zero rotational speed $N=0$, situation known as “locked rotor”, and the situation of zero torque $T=0$ which is commonly referred to as “runaway”.

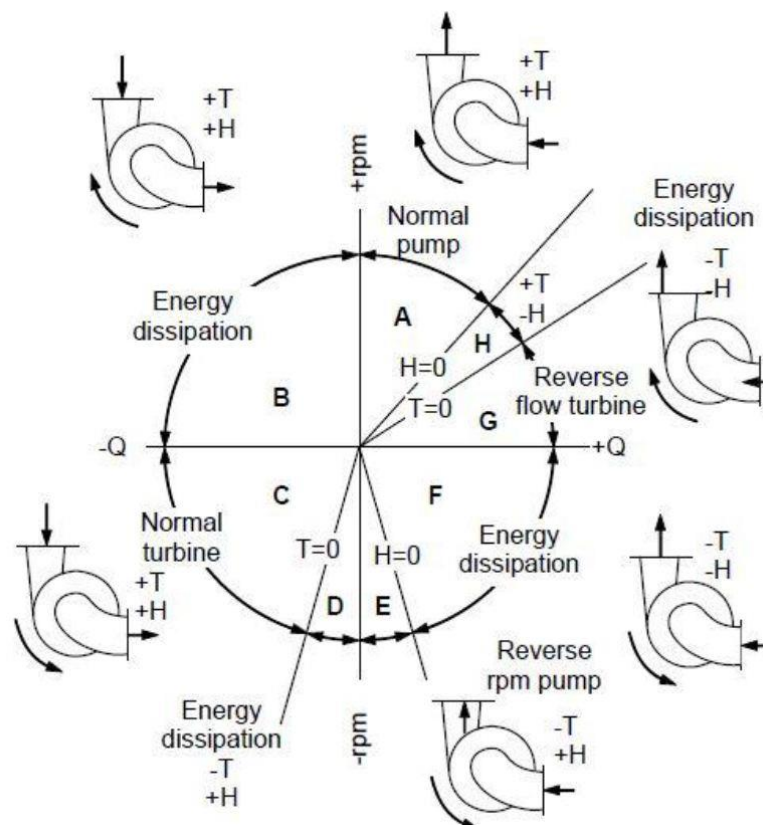


FIGURE 3-21. FOUR QUADRANTS PUMP OPERATIONS CHART (BAUMGARTEN & GUDER, 2005).

3.6.1. PAT TYPES AND APPLICATION RANGE

Generally speaking, pumps may be divided into horizontal and vertical units: while the former often feature a spiral volute case, the latter are normally designed with fixed-blades diffuser (Alatorre-Frenk, 1994). A more comprehensive prospect of pump types and their suitability for being used as turbines is shown in Figure 3-22 as proposed by (Williams, 1995).

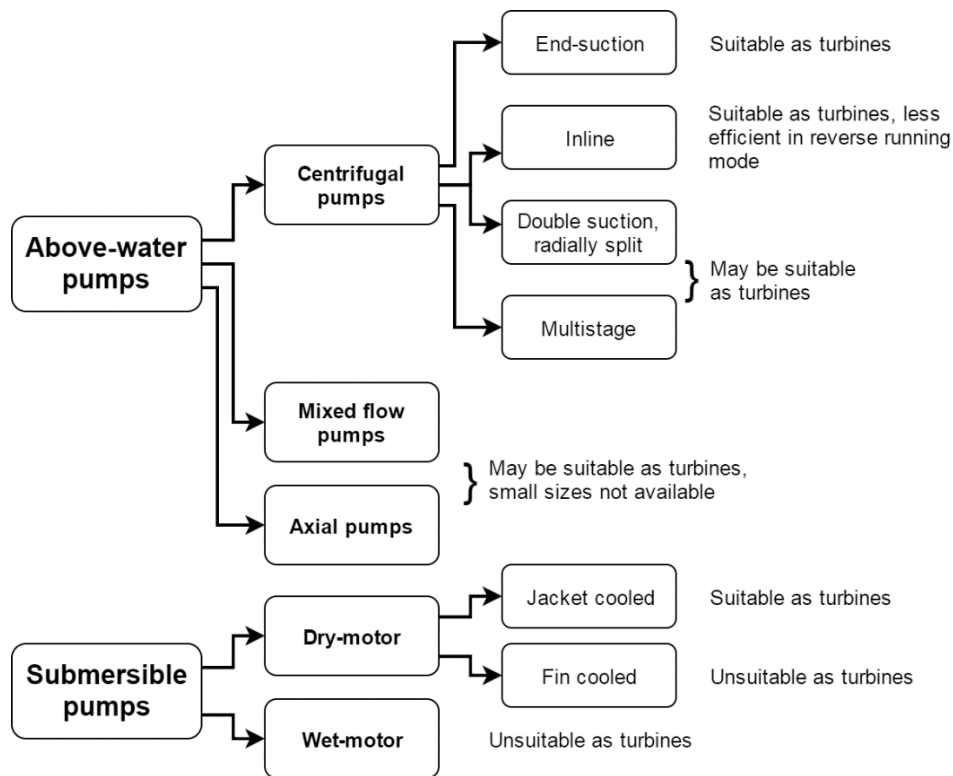


FIGURE 3-22. PUMP TYPES AND THEIR SUITABILITY AS TURBINES, ADAPTED FROM (WILLIAMS, 1995).

Among the existing designs of hydraulic pumps/PATs, “centrifugal” or “radial” units seen in Figure 3-23 (a) are the most used worldwide in a wide variety of application fields (Grundfos industry, 2004). The name is derived from the radial path followed by the fluid in the rotor: from the centre to the periphery when running as a pump and in the opposite direction when flow is reversed (Ramos, 2000). To achieve a higher head drop across the machine, more impellers can be assembled in series to create a multistage unit. Conversely, a double flow radially split pump/PAT design involves a single radial open rotor fed by two symmetric inlets and enable processing a higher flow rate with respect to a standard radial unit (Grundfos industry, 2004). A second type of pump/PAT design is the axial one as shown in Figure 3-23 (c), in which the fluid interacts with a propeller following a trajectory parallel to the pump axis. Such units are particularly suitable to processing high flow rates with low head difference (Ramos, 2000).

Finally, mixed flow pumps/PATs shown in Figure 3-23 (b) stand in between the applicability range of radial and axial units and have an impeller shaped in a similar way as a Francis turbine (Ramos, 2000).

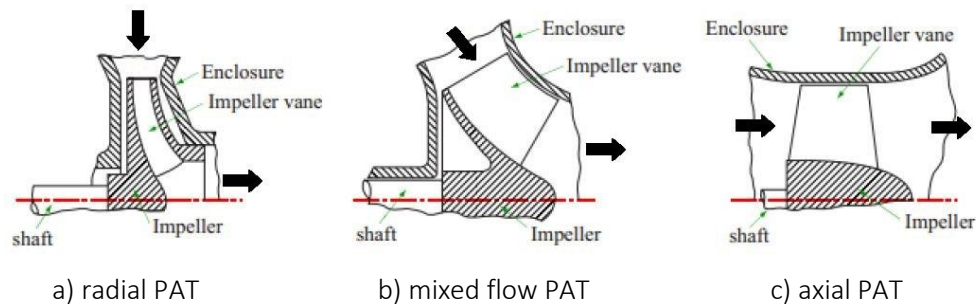


FIGURE 3-23. TYPES OF PAT IMPELLERS AND RELATIVE FLOW DIRECTION: A) RADIAL PAT; B) MIXED FLOW PAT; C) AXIAL PAT (ENGINEERING SCIENCE DATA UNIT, 2014).

Another special pump/PAT design is that of submersible units, which can possibly be fitted inside a pipe connected to draft tube exploiting small head differences in flowing rivers.

A PAT application chart as proposed by (Chapallaz, et al., 1992) is shown in Figure 3-24, showing indicative ranges for radial, multistage, double flow, mixed flow and axial PATs. Instead, two additional application graphs suggested respectively by Andritz and KSB are displayed in Figure 3-25 and Figure 3-26.

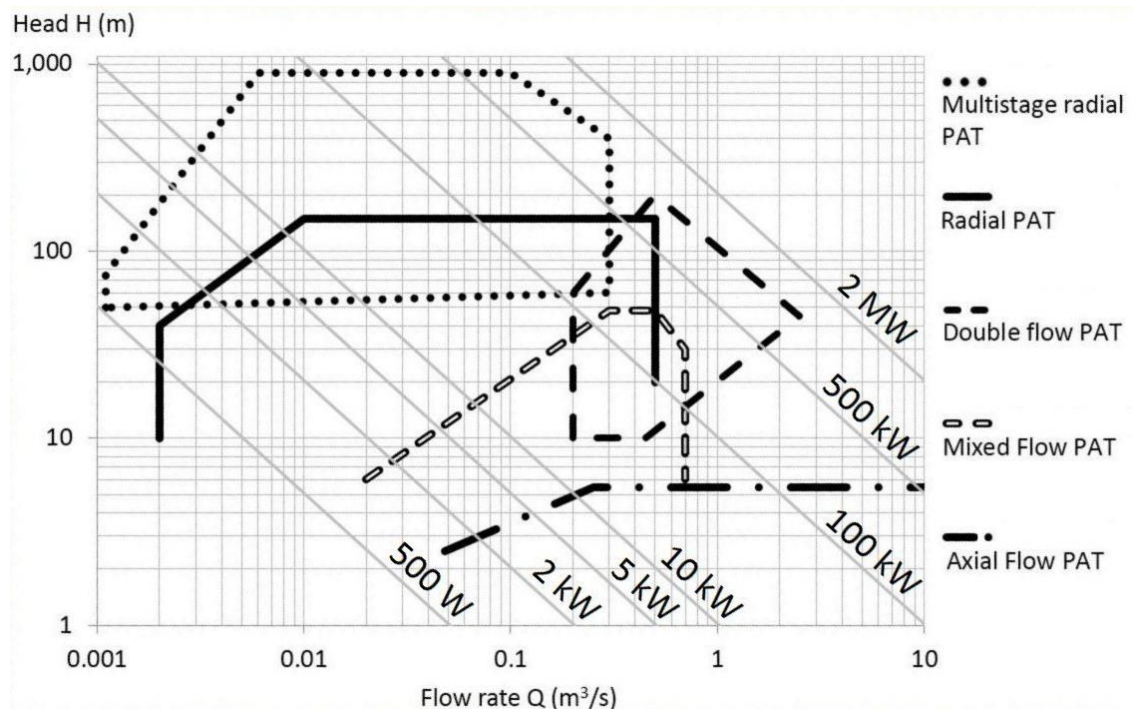


FIGURE 3-24. PAT APPLICATION CHART (CHAPALLAZ, ET AL., 1992).

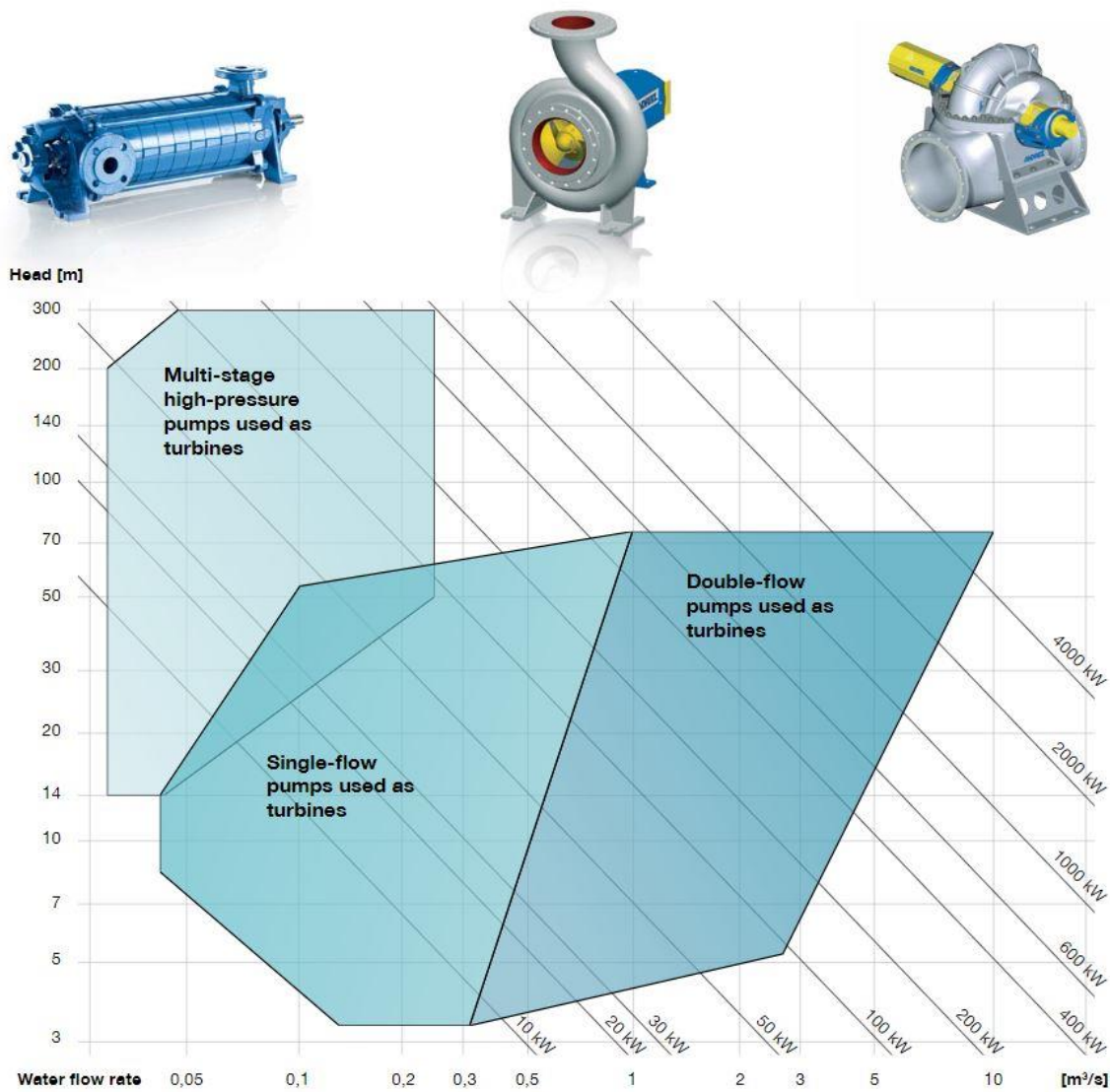


FIGURE 3-25. APPLICATION RANGE OF DIFFERENT TYPES OF PATS UP TO 4 MW (ANDRITZ, 2018).

In terms of power, limits in PAT technology applicability is given by off-the-shelf availability which normally extends up to nominal powers of 100-250kW (Alatorre-Frenk, 1994). However, larger units up to 2MW which require dedicated manufacturing may also be employed as turbines (Garay, 1990).

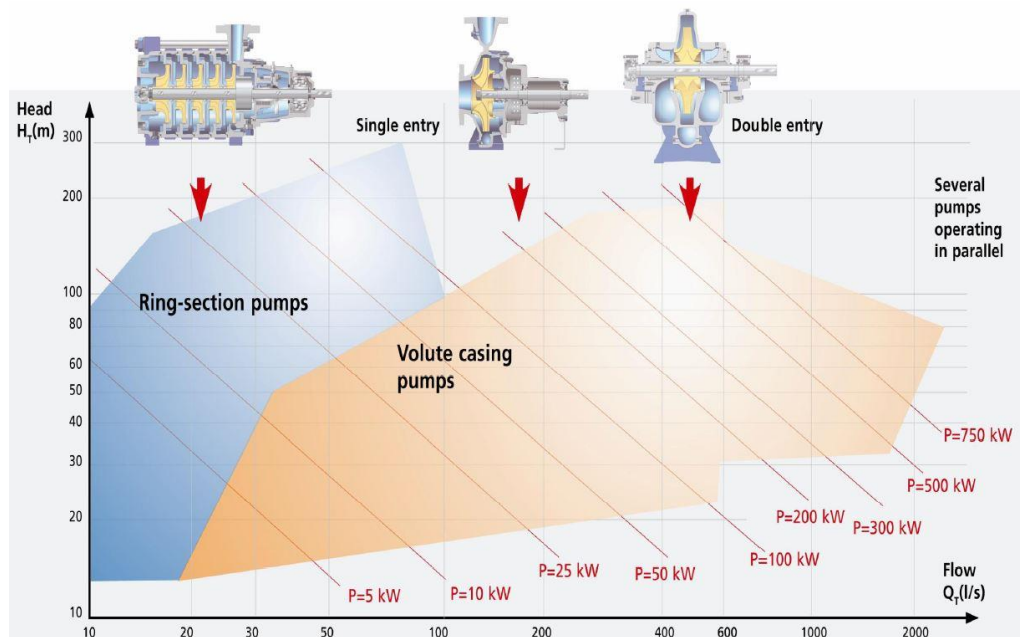


FIGURE 3-26. INDICATIVE APPLICATION RANGE OF DIFFERENT TYPES OF PATS UP TO 1 MW (KSB, 2011).

Instead, in terms of specific speed, PATs applicability has both a lower and an upper limit. For extremely low values of specific speed the efficiency reachable by reaction turbines (including Francis and PATs) falls below that of impulse turbines (e.g. Pelton) due to the friction developed by water passing through narrow flow passages (Alatorre-Frenk, 1994). As for the upper boundary, high specific speeds indicate the adoption of axial PATs which present several limitations over standard radial units: higher acquisition cost, lack of experimental data on reverse mode running available, lower efficiency and increased threats posed by cavitation and runaway phenomena (Alatorre-Frenk, 1994). However, more recent research found axial-flow PATs a cost effective solution for low-head hydropower especially when fitted with adjustable guide vanes (Qian, et al., 2016).

3.6.2. CHOICE OF GENERATOR FOR PAT SCHEMES

If connection to a stable energy source is available, typically a power grid, the preferred choice for generator unit is that of a standard asynchronous induction unit which tend to be less expensive and easier to maintain than synchronous generators (Henderson, 1996). Another advantage is that pumps are commonly commercialized as combined “mono-bloc” units which include the pump body connected to an induction motor/generator unit and fixed to a rigid support frame. Also, induction generators allow for inexpensive frequency regulation through a digital inverter (Williams, 1995).

The effective speed of induction generators is a function of the frequency “f” of the current exciting the stator coils, the number of magnetic pole pairs “pp” and the slip factor “s” through the following equation:

$$N(\text{RPM}) = \frac{60 \cdot f(\text{Hz})}{pp} (1 + s) \quad (9)$$

The slip factor is typically in the range of 2-5%, while the grid frequency in Europe corresponds to 50Hz, and most commercially available pump units come with motors having pp=1, 2 or 3 (Williams, 1995).

Standard 3-phase single speed induction motor/generators are divided into International Efficiency classes “IE” ranging from IE1 to IE4, where the latter corresponds to the highest efficiency standard as shown in Figure 3-27 (ABB, 2014).

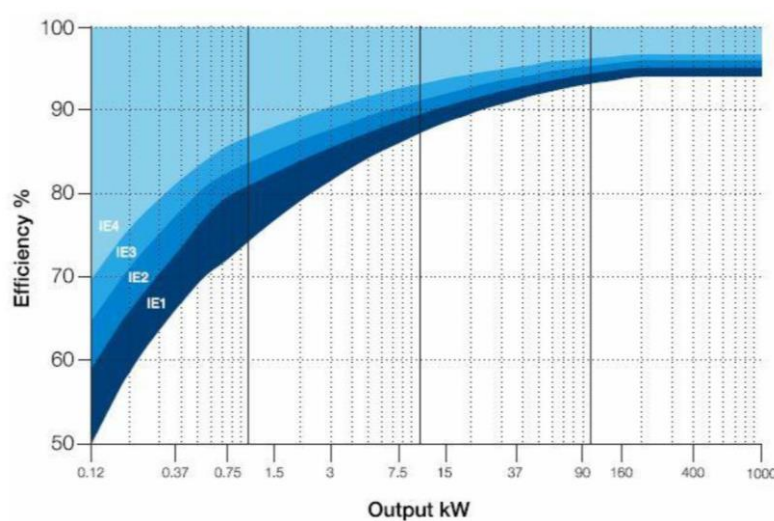


FIGURE 3-27. PERFORMANCE LIMITS OF INDUCTION MOTORS INTERNATIONAL EFFICIENCY CLASSES PER NOMINAL POWER OUTPUT (ABB, 2014).

The efficiency of induction motors running as generators is usually comparable to their rated efficiency as motors, although it has been reported that units with low motor efficiency may offer a significantly lower generator efficiency (Deprez, et al., 2006). A 2% lower efficiency of units running as generators compared to the one when running as motors has also been reported (Williams, 1995).

Finally, when selecting a combined PAT-generator assembly for a determined hydropower scheme caution must be taken in verifying that the maximum power output does not exceed the 80% of rated motor power in order to avoid overloading the unit (Williams, 1995).

In cases when an off-grid installation is required three choices of generator are typically available (Fernandes, et al., 2019):

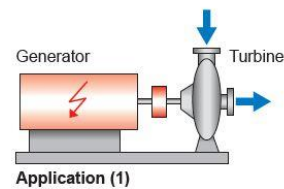
- synchronous generator / alternator whose excitation field is provided by a coil energized by means of a set of brushes;
- permanent magnet synchronous generator / alternator;
- SEIG (Self-Excited Induction Generator) consisting of a regular induction motor/generator connected to a bank of capacitors.

3.6.3. PATs APPLICATION LITERATURE

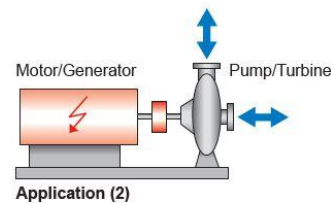
Pumps as turbines have been utilized in a large variety of applications under different configurations summarized in Figure 3-28.

The tested possibilities for application are diverse:

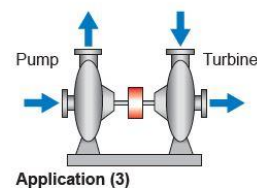
- as a pump which is directly coupled to the generator (1)



- as a combination pump / turbine with motor / generator (2)



- a pump, directly coupled with a further pump as turbine (3)



- or as a pump with a motor and coupled pump as turbine (4)

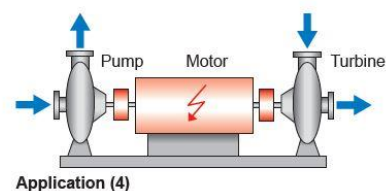


FIGURE 3-28. DIFFERENT CONFIGURATION OF PATs (HÄNY AG, 2006).

Several examples of existing PAT-based power stations are described in the following sub-chapters.

3.6.3.1. Hydropower plants for direct energy generation

The MVE Vyšni Lhoty power station located in Czech Republic is powered by three KSB Etanorm PATs running in parallel and housed in a powerhouse located at the foot of a dam. Such a configuration was selected in order to achieve a high operational flexibility, and the details of the installed turbines are listed in Table 3-6 (KSB, 2011)

TABLE 3-6. DESIGN PARAMETERS FOR PATS CHOSEN FOR THE VYŠNI LHOTY POWER PLANT (KSB, 2011).

PAT 1	Type	Centrifugal
	Manufacturer	KSB
	Model	Etanorm G 150-400
	Nominal power output	90 kW
PAT 2	Type	Centrifugal
	Manufacturer	KSB
	Model	Etanorm RG 200-400
	Nominal power output	110 kW
PAT 3	Type	Centrifugal
	Manufacturer	KSB
	Model	Etanorm RG 300-500
	Nominal power output	132 kW

Pumps As Turbines have been proposed as the most technically and economically viable technology for hydroelectric generation in the range of Micro or Pico hydropower ($P < 100$ kW) in developing countries. For instance, the Government of India has identified thousands of sites across the northern and north-eastern parts of the country which are potentially suitable for hosting MHP schemes and PATs would ideally be suitable in the context (Rawal & Kshirsagar, 2007).

As a practical outcome of his PhD project, Singh designed and supervised a MHP off-grid scheme in a rural area of the Darjeeling region in India. While the available gross head was fixed at 95.3m, the water flow experienced high seasonal variability ranging from zero during winter months up to 220l/s at the peak of monsoon season in July (Singh, 2005). In order to cope with the flow variability and power requirements from local consumers the final plant design consisted of a forebay, a single penstock, a distributor and a set of three PATs fitted with a draft tube powering a single generator unit through a belt drive. The details of the scheme are summarised in Table 3-7. Each of the PATs chosen for the site has been previously machined in a workshop to round the inlet impeller and remove the casing eye rib as means of achieving a higher efficiency in turbine mode

(Singh, 2005). A schematic view of the designed plant can be seen in Figure 3-29 (a), while the built powerhouse is shown in Figure 3-29 (b).

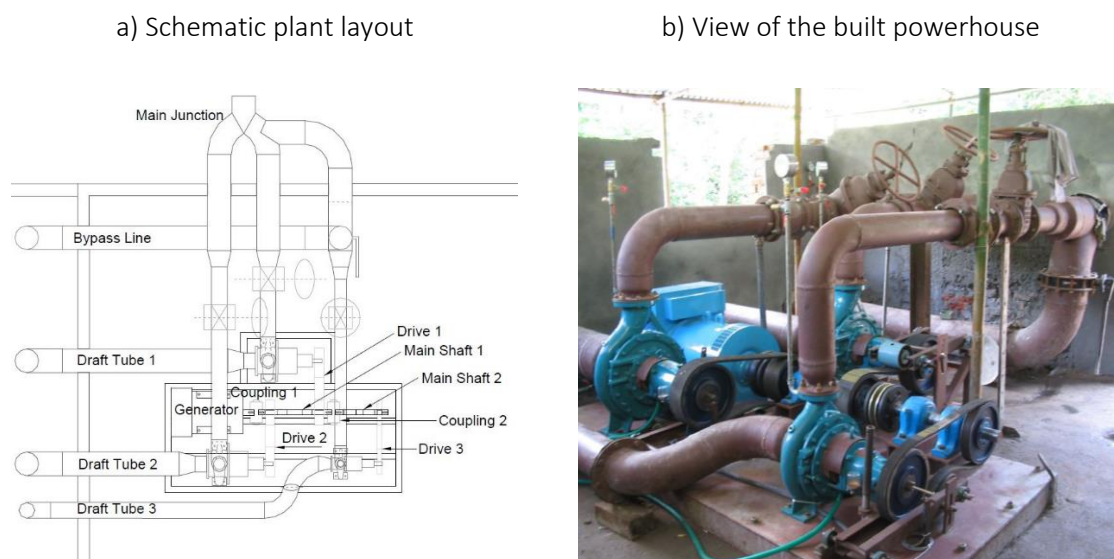


FIGURE 3-29. PLANT LAYOUT (A) AND VIEW OF THE POWERHOUSE (B) OF THE AMBOOTIA HYDRO PLANT (SINGH, 2005).

TABLE 3-7. DESIGN PARAMETERS FOR PATS CHOSEN FOR THE AMBOOTIA HYDRO PLANT (SINGH, 2005).

PAT 1 and 2	Type	Centrifugal
	Manufacturer	Kirloskar
	Nominal head	83 m
	Nominal discharge	134 l/s
	Nominal efficiency	76%
	Nominal N	1237 RPM
PAT 3	Type	Centrifugal
	Manufacturer	Kirloskar
	Nominal head	77 m
	Nominal discharge	51 l/s
	Nominal efficiency	71%
	Nominal N	1472 RPM

Other smaller PAT installations for rural electrification in developing countries described in the literature include:

- a 2.2 kW community-owned pico hydro scheme based on a mono-bloc centrifugal PAT installed in December 2001 in the Kenyan village of Thima (Maher & Williams, 2003). The flow-to-wire efficiency was evaluated as 45%, and the energy is produced through a 2-poles induction generator. The total project cost was equal to 7,865 \$ of which the 22%

went for the turbine, generator, controller and protection devices and the 11% for the civil works and penstock. The remaining costs were necessary to implement the local micro-grid able to power 110 houses (Maher, et al., 2003);

- a 10 kW PAT installed within a rural hydro scheme serving around 100 households of a village in rural Tanzania. Despite the initial successful commissioning of the plant in July 2006, the energy generation has been frequently discontinued and the financing entity has deemed the site production “unsustainable” due to a series of technical and managerial issues incurred (United Nations Industrial Development Organization, 2010);
- a second 3 kW mono-bloc centrifugal PAT was installed in 2008 in the Mae Wei village within the Tha Song Yang district in Thailand. The hydro installation was developed as stand-alone, and included an induction generator, a controller and a resistive load. The power output was fed to the local school building and used for lighting and powering other electric appliances (Greacen, 2008). The operations of the scheme encountered difficulties due to the fact that the local community did not assume responsibility for the periodic maintenance and financial support after the plant implementation (personal communication, 2018). The electric diagram of the scheme and some pictures of the installation are shown in Figure 3-30 and Figure 3-31 respectively;
- a centrifugal PAT was installed as a prime mover (without a generator) to power a mechanical mill at the Mbangamao village in Tanzania (Greacen, 2008).

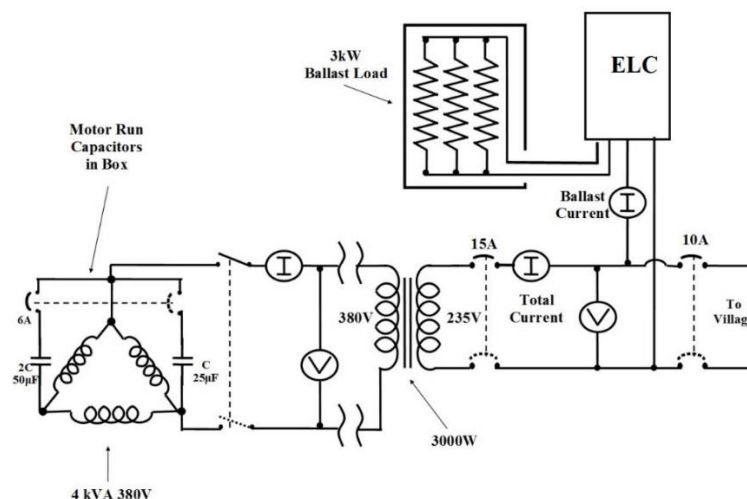


FIGURE 3-30. ELECTRIC CIRCUIT DIAGRAM FOR THE PAT INSTALLATION AT THE MAE WEI VILLAGE IN THAILAND (GREACEN, 2008).

a) Electric cabinet



b) Electric cabinet and dump resistors



c) Turbine delivery on site



FIGURE 3-31. PAT INSTALLATION AT THE MAE WEI VILLAGE IN THAILAND: A) ELECTRIC CABINET; B) ELECTRIC CABINET AND DUMP RESISTORS; C) TURBINE DELIVERY ON SITE (GREACEN, 2008).

3.6.3.2. Residual water utilisation in small hydropower stations or non-powered dams

In 2012 South West Water refitted its Morwellham Quay power station dating back to the 1930's by installing a KSB PAT in addition to the existing generating capacity. The new PAT is designed to work under variable speed and operate at the low-end of the range of the existing turbines thus avoiding water stagnation during periods of low available flow and increase the plant flexibility. The main characteristics of the installed PATs are described in Table 3-8, while two images are displayed as Figure 3-32. The installed PAT was reported to be more economically convenient than

any other conventional turbine solution in terms of both initial investment and maintenance (KSB, 2012b).

TABLE 3-8. MAIN CHARACTERISTICS OF MORWELLHAM QUAY PAT (KSB, 2012B).

Type	Split case, double entry PAT
Model	Omega 125-365A
Manufacturer	KSB
Nominal head	76 m
Nominal discharge	123.5 l/s
Hydraulic efficiency	83.6 %
Nominal power output	77 kW

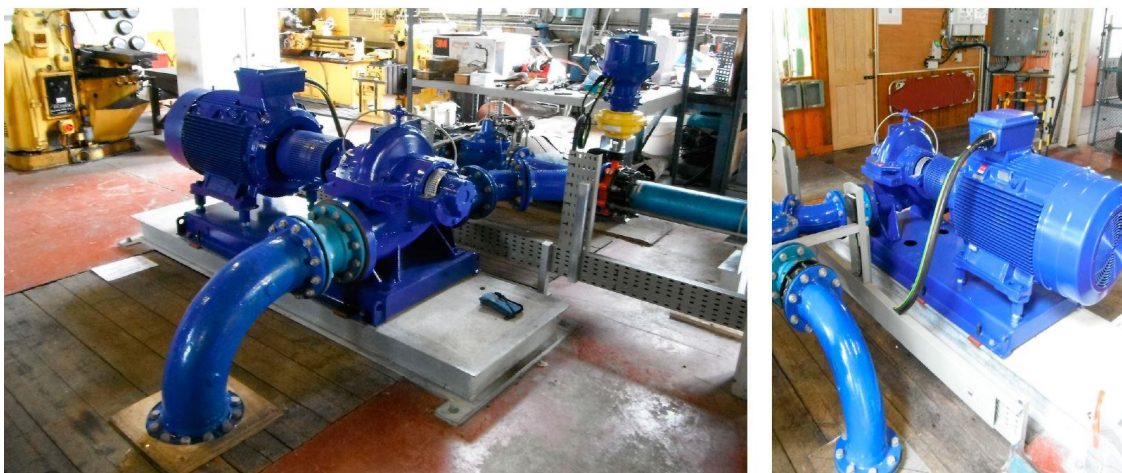


FIGURE 3-32. KSB PAT INSTALLED BY SOUTH WEST WATER AT MORWELLHAM QUAY (KSB, 2012B).

Another documented installation made by KSB together with Kier for South West Water consists of one 11 and one 140 kW PAT exploiting respectively the 24-h compensation flow and the seasonal spill flow occurring from the Avon Dam in Devon, UK. The global power output from the two units amounts to 454 MWh per typical year, thus utilizing a potential previously untapped. The units were commissioned in June 2012, and the main characteristics of the installed PATs are described in Table 3-9, while pictures are in Figure 3-33 and Figure 3-34 (KSB, 2012a).

TABLE 3-9. MAIN CHARACTERISTICS OF THE INSTALLED PATS AT AVON DAM (KSB, 2012A).

PAT 1 (compensation flow)	Type	Centrifugal
	Manufacturer	KSB
	Model	Etabloc MN 100-200
	Nominal head	27 m
	Nominal discharge	60 l/s
	Hydraulic efficiency	78%
PAT 2 (spill flow)	Type	Split case, double entry PAT
	Manufacturer	KSB
	Model	Omega 350-430
	Nominal head	27 m
	Nominal discharge	650 l/s
	Nominal efficiency	86.4 %



FIGURE 3-33. IMAGES OF THE TWO PATS INSTALLED AT AVON DAM: SPLIT-CASE DOUBLE FLOW UNIT (ON THE LEFT) AND CENTRIFUGAL UNIT (RIGHT) (KSB, 2012A).



FIGURE 3-34. CENTRIFUGAL PAT INSTALLED AT THE AVON DAM (LEFT) AND VIEW OF THE POWERHOUSE FROM THE DAM CREST (RIGHT) (KSB, 2012A).

3.6.3.3. Energy recovery from Water Supply Systems (WSS) and sewage systems

The earliest application that could be traced dates back to 1981, when in the water supply system of the city of La Habra in California (USA) a 112 kW Pump As Turbine was installed in parallel to an existing PRV. In order to accommodate the large daily flow fluctuations a control valve was placed in series to the turbine and the bypass branch with the previously existing PRV was automatized to open during periods of large flow. The cost of the project for the city was 160,000 \$, and the design power output was equal to 760 MWh per year (Dandekar & N., 2013).

When building the new municipal WWTP for the Swiss city of Nyon in the 1990's a MHP plant was commissioned to recover the potential energy of treated wastewater being discharged into Lake Geneva (European Directorate for Transport and Energy - FP6 Project Shapes, 2010). The chosen technology was that of reverse running pumps, and a 220kW PAT working under a fixed discharge and intermittent operations was designed as summarised in Table 3-10. A picture of the energy recovery installation is shown in Figure 3-35.

From Figure 3-35 it can be seen that the installed PAT is of the double flow/split case type and a flywheel has been placed between the turbine and the generator to ensure smooth transitions during intermittent operations.

TABLE 3-10. DESIGN PARAMETERS OF PAT
FOR NYON WWTP OUTFALL (EUROPEAN
DIRECTORATE FOR TRANSPORT AND ENERGY -
FP6 PROJECT SHAPES, 2010).

Type	Centrifugal PAT
Manufacturer	Sulzen Pumpen, CH
Nominal head	94 m
Nominal discharge	293 l/s
Nominal efficiency	82%
Nominal shaft power	220 kW
Annual energy yield	700,000 kWh/year



FIGURE 3-35. PAT INSTALLED AT THE OUTFALL
OF NYON WWTP.

As of 2010, the operator had commissioned a study to evaluate the replacement of the PAT unit with a conventional water turbine in order to increase plant flexibility, reduce noise emissions and increase energy production (European Directorate for Transport and Energy - FP6 Project Shapes, 2010). The new proposed turbine would be additionally fed with water adduced from the nearby

Asse river and thus increase the energy yield between +150 and +250% with respect to the reported value (Municipalité de Nyon, 2011).

Another noticeable use of Pumps as Turbines in the water industry is related to the water treatment plant located in Beliche, in the Algarve region of Portugal. One of the main water surface reservoirs of the region is the 48hm³ Beliche artificial basin contained by a 54m high earth and rockfill dam inaugurated in 1986 (Alonso Pérez de Agreda, et al., 2005). Water is withdrawn from the reservoir through an intake tower, and a 390m long tunnel and a concrete pipe and fed to the water treatment plant built for the purpose at the base of the dam. Here, within the valve chamber it was decided to replace the original inlet valves with two identical PATs in parallel able to exploit the energy potential of water flowing from the reservoir to the treatment facility. The plant was built in 2008 and was connected to the national grid in 2011. The main characteristics of the installed PATs are described in Table 3-11 (Ramos, et al., 2014). Two pictures of the powerhouse can be seen in Figure 3-36.

TABLE 3-11. DESIGN PARAMETERS OF TWIN PATS FOR BELICHE WATER TREATMENT PLANT (SALVADOR, 2014)

Type	Centrifugal
Model	KSB ETANORM 100-200
Nominal head	17.6 m
Nominal discharge	48 l/s
Nominal efficiency	76%
Nominal shaft speed	1,500 RPM
Nominal shaft power	5.5 kW



FIGURE 3-36. VIEW OF PAT INSTALLATION AT BELICHE WATER TREATMENT PLANT (RAMOS, ET AL., 2014).

A 2014 study analysed and compared three different strategies concerning the operation of PATs installed in Beliche water treatment plant, namely:

- A. water flow is equally split between the two identical PATs;
- B. one of the two PATs is allowed to start operations only after that the other unit has reached a maximum nominal flow rate;
- C. best operation mode between A and B, but in addition turbines are switched on only when the available flow allows them to operate with efficiency higher than 35%.

Results showed that strategy B should be preferred, leading to highest energy yield with respect to other two case studies corresponding to 55.8MWh along a typical year. The difference between case B and C in terms of energy production was found to be extremely low, hence the author suggested for simplicity's sake not to impose any working restriction on daily turbine operations (Salvador, 2014). Within the same research, the possibility of installing a third identical turbine in Beliche site was also investigated and found as economically viable for both the scenarios of energy sale to the national grid and energy in-situ self-consumption (Salvador, 2014). However, since the operations of such additional unit would be limited to a small number of hours per year concentrated in summer peak demand period it would have been appropriate to evaluate the installation of a third PAT having smaller size than the two existing ones.

Finally, in order to increase the energy generation of the site the author evaluated the possibility of building a raw water storage tank downstream of the Beliche powerhouse which could allow for operating intermittently the existing turbines at their constant nominal flow rate while cyclically refilling the created reservoir. Even if it was found that the a 100,000m³ buffer tank would increase the yearly energy generation by 43%, the author didn't perform an economic analysis to assess the viability of such system modification under financial terms (Salvador, 2014).

A further example of PAT implementation for energy recovery in WSS is that of Breech power station (DE) commissioned in 2006, where eight PATs operated in parallel are employed to reduce pipeline pressure and at the same time generate a maximum of 300kW. The design of a modular system based on several units operated independently resulted in a large operational flexibility, and the plant was able to process flow rates in the range between 175 and 1,100l/s while dissipating a net head equal to 52 and 35m respectively. The chosen technology corresponded to eight identical centrifugal PATs from KSB manufacturer, and the payback of the installation was

found to be lower than three years (KSB, 2008). A picture of the powerhouse is shown in Figure 3-37.



FIGURE 3-37. POWERHOUSE OF BREECH ENERGY RECOVERY STATION (KSB, 2008).

The UK water utility South West Water has installed a PAT on a pipeline conveying water from an elevated source (Erme stream) into the Littlehempston water treatment work in Devon. The PAT has been placed in parallel to a Pressure Reducing Valve which was used to produce a 9 bar pressure drop by dissipating it as heat and noise. The inlet design pressure of the PAT is 15 bar and the outlet pressure is of about 6 bar, and the turbine has been enclosed by an acoustic casing in an effort to further reduce its noise signature given the sensitivity of the natural environment around the powerhouse. The unit was commissioned in June 2012, and the main characteristics of the installed PATs are described in Table 3-12, while a few pictures are shown in Figure 3-38 (KSB, 2013).





FIGURE 3-38. PICTURES OF THE PAT INSTALLED BY SOUTH WEST WATER ON THE ERME STREAM (KSB, 2013).

TABLE 3-12. DESIGN PARAMETERS OF THE PAT INSTALLED ON THE ERME STREAM (KSB, 2013).

Type	Multistage PAT
Model	Multitec A 125/2-9.1
Manufacturer	KSB
Nominal head	90 m
Nominal discharge	55.3 l/s
Hydraulic efficiency	75 %
Nominal power output	37 kW

The Spanish firm Tecnoturbine has installed a 37 kW inline PAT exploiting the overpressure within a water pipeline between two reservoirs located at different heights in the municipal water supply system of Valencia. The turbine design point corresponds to 4 bar and 100 l/s and the electricity is fed into the national grid (Lledo, 2016).

3.6.3.4. Energy recovery from irrigation networks, exploiting overpressures occurring in the lowest network branches

A research team from Spain and Portugal developed a methodology to assess the energy recovery potential from irrigation networks in which flow meters are not available. Thanks to the use of the public domain software EPANET it is possible to calculate the instant flow rate and available pressure at each system nodal point if only a record of historical trends and patterns is available (Pérez-Sánchez, et al., 2016). The authors applied their method to an irrigation network in the

Spanish region of Valencia and found that the most promising point for installing a hydraulic turbine possessed a potential for recovery equal to approximately 90MWh per year (Pérez-Sánchez, et al., 2016).

3.6.3.5. Direct coupling with pumps for water supply

Designed by Karlsruhe Institute of Technology and commissioned in 2009, a stand-alone PAT scheme for water supply has been implemented on the Indonesian island of Java (KIT, 2008). The problem addressed is the lack of surface runoff occurring during dry months, which limits the quantity of water available to inhabitants to values lower than 10 litres per day. Additionally, such available water has been previously collected in open ponds or cisterns are often polluted. The idea was to utilize the perennial water that flows in underground rivers running through the island's karst cave system and pump it into a surface reservoir. Given the fact that ideally a pump must be placed around 100 meters below the terrain and a connection to the electric grid was deemed unfeasible, it was decided to couple the pump directly to a PAT meant to exploit the potential of the underground river (Baumgarten & Guder, 2005). The modular designed PAT-gearbox-pump assembly is shown in Figure 3-39 (a), while in Figure 3-39 (b) is displayed a basic sketch of the final configuration of the underground hydropower plant.

a) PAT-gearbox-pump assembly (Baumgarten & Guder, 2005)

b) basic plant sketch
(KIT, 2008)

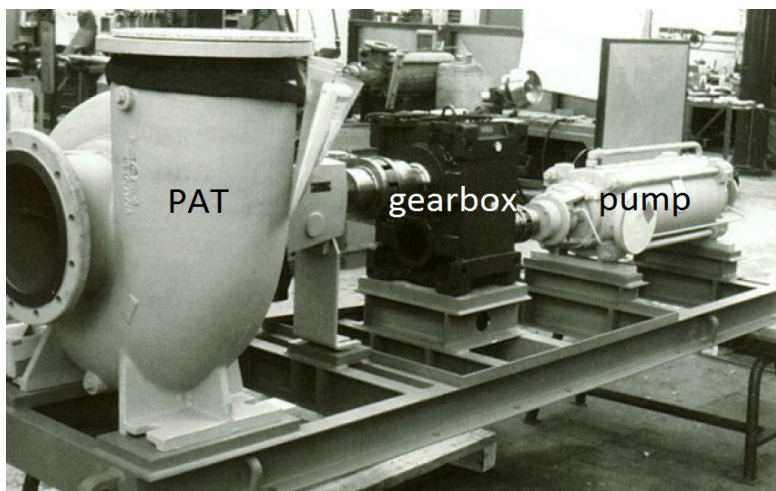


FIGURE 3-39. ASSEMBLED PAT-GEARBOX-PUMP MODULE AND LAYOUT SKETCH OF JAVA ISLAND HYDROPOWER PLANT.

In order to accommodate the seasonal variability of underground water flow the hydropower and pumping station has been divided into modular units whose characteristics are summed up in Table 3-13.

A 2010 report from Stellenbosch University (South Africa) evaluated different design possibilities for a MHP scheme meant to power a water pump supplying ocean water to a sea farm located in Western Cape Province. The most suitable identified design consisted of a PAT coupled to a direct drive supply pump, which proved to be more cost-effective than layouts with conventional water turbines and layouts with indirect electric PAT-pump coupling (Teuteberg, 2010). The main characteristics of chosen PAT are summed up in Table 3-14.

A 2017 research based on experimental data demonstrated the concept of a PAT-pump turbocharger able to harvest energy for pressure reduction and transfer it to a separate pressurized stream. Such configuration is suggested by the authors to be simple, reliable and convenient besides leading to significant energy savings with respect to a traditional pumping station (Carravetta, et al., 2017a).

TABLE 3-13. PAT/GEARBOX/PUMP ASSEMBLY (BAUMGARTEN & GUDER, 2005).

PAT unit	Type	Mixed flow
	Model	KSB Etanorm R 300-400
	Nominal head	15 m
	Nominal discharge	375 l/s
	Nominal efficiency	81%
	Nominal N	1,200 RPM
	Nominal shaft power	45 kW
gearbox	ratio	1 : 1.83
pump unit	Type	Multistage ring-section
	Model	KSB Multitec D 65/09
	Nominal head	190 m
	Nominal discharge	17 l/s
	Nominal efficiency	75%
	Nominal N	2,200 RPM

Andritz installed two identical sets of PAT-pump assembly at the waste water treatment system of a LEIPA paper mill in Germany, in which a 53 kW PAT driven by the pressurized wastewater is connected to a motor which powers a 127 kW booster pump. All three units (PAT-motor-pump) are assembled on the same shaft, and allow the re-utilization of more than 42% of the energy contained in the waste water stream (Andritz, 2018). An image of the setup is presented in Figure 3-40.

TABLE 3-14. DESIGN PARAMETERS OF PAT FOR ROMAN BAY SEA FARM (TEUTEBERG, 2010).

Type	Axially split, double flow PAT
Model	KSB Omega 350-510A
Nominal head	23 m
Nominal discharge	520 l/s
Nominal efficiency	82%
Nominal shaft power	97 kW



FIGURE 3-40. ENERGY RECOVERY INSTALLATION USING A PAT AT A PAPER MILL IN GERMANY (ANDRITZ, 2018).

3.6.3.6. Energy recovery in desalination plants based on reverse osmosis principle

Desalination plants commonly rely on the reverse osmosis principle to separate marine water from its contained mineral solutes. The by-product of such separation is a stream of highly pressurized brine (typically at 40-80 bar) whose energy content may be recovered through a dedicated turbine in order to partially offset the energy demand of the desalination plant itself (European Directorate for Transport and Energy - FP6 Project Shapes, 2010). The main challenges in the design of an energy recovery device working in such environment are the requirement for withstanding abrasive fluid action and resisting to wide temperature and pressure variations, and PATs were among the first types of devices used in the field because of their resistance to seawater corrosion, simple maintenance and ease of operations (Farooque, et al., 2004; Baig & Al Kutbi, 1998). The Al-

Jubail reverse osmosis plant situated in Saudi Arabia along the Persian Gulf coastline was designed to provide the namesake city with nearly 91,000m³ of fresh water per day. For energy recovery purposes a reverse running radial pump was selected, whose main design parameters are shown in Table 3-15.

TABLE 3-15. DESIGN PARAMETERS OF PAT FOR AL-JUBAIL DESALINIZATION PLANT (BAIG & AL KUTBI, 1998).

Type	Centrifugal PAT
Manufacturer	Ingersoll Dresser, UK
Nominal head	730 m
Nominal discharge	135 l/s
Nominal efficiency	79%
Nominal shaft power	760 kW

3.6.3.7. Replacement of throttling valves in various industrial applications

The boiler of the municipal heating power plant in Łomża (PL) was designed to supply water at a temperature of 120°C which was too high to be managed by the heating network. Therefore, outlet water needed to be mixed with cold water prior to distribution to final users and the pressure difference between the hot and cold stream was dissipated by means of a throttling valve. In 1997 a PAT whose characteristics are shown in Table 3-16 was installed in place of the throttling valve, and is reported to have operated continuously until 2005 (European Directorate for Transport and Energy - FP6 Project Shapes, 2010). A picture of the described PAT installation is displayed in Figure 3-41.

TABLE 3-16. DESIGN PARAMETERS OF PAT FOR ŁOMŻA MUNICIPAL HEATING PLANT (EUROPEAN DIRECTORATE FOR TRANSPORT AND ENERGY - FP6 PROJECT SHAPES, 2010)

Manufacturer	Leszno Pump Manufacturers, PL
Nominal head	30.5 m
Nominal discharge	92 l/s
Nominal efficiency	73%
Electrical output	20 kW



FIGURE 3-41 - PAT INSTALLED IN ŁOMŻA MUNICIPAL HEATING PLANT (EUROPEAN DIRECTORATE FOR TRANSPORT AND ENERGY - FP6 PROJECT SHAPES, 2010).

Apart from the water sector, being capable of processing dual-phase flows and of working under explosive atmosphere PATs are particularly suitable for energy recovery applications in industrial processes involving natural gas treating, oil refining or fertilizers production (Alatorre-Frenk, 1994). A Brazilian oil firm chose to implement a reverse running pump to recover energy along a hydrocarbon processing line, when a high pressure liquid containing dissolved gas particles was previously throttled from 74.8 to 14.8 bar. The continuous recovered power offered by the PAT corresponded to 258kW and was used to assist a second 870kW electric motor which drives the primary circuit pressure increasing pump (Sulzer Pumps, 2011).

3.6.3.8. Micro-energy pumped hydro energy storage

Pumped hydropower is one of the most mature and cost-effective technologies for large-scale energy storage and grid balancing. While the common size of such installations is in the range of hundreds of Megawatts, micro and pico pumped hydro storage schemes have been proposed as a means to balance micro-grids based on intermittent renewable energy sources, namely solar Photovoltaic (PV) and wind. A 2017 research team has evaluated the installation of a Pump As Turbine for a 17 kWh hydro storage scheme in a research centre near Froyennes, Belgium (Morabito, et al., 2017). The quoted research proposed a methodology to design micro-hydropower pumped storage plants and applies it to a specific case study employing a single PAT rated for 10.5 kW as a pump and up to 7.5 kW as a turbine.

3.6.3.9. Final remarks on PAT case studies

The presented case studies give a significant insight on the variety of PAT applications that have been trialled up to date. The compilation of such a list has proved very challenging due to the fragmented nature of the sources of information, which mostly consisted of commercial pamphlets published by the equipment suppliers. It must be stressed that 8 out of the 19 practical PAT applications described adopted machines manufactured by KSB, which appears to be the only pump maker which possesses a degree of knowledge of the performance of its products in turbine mode and has made some kind of marketing efforts. Instead, the other seven manufacturers mentioned in the previous sub-chapters seem to have been involved only in the realization of sporadic pilot plants but have lacked the knowledge, resources and/or the interest to become players in the PAT market. As a last remark, the above collection of PAT installations represents a minuscule portion of the potential sites in which such machines may be adopted worldwide.

3.6.4. PREDICTION OF PAT PERFORMANCES

One of the major limitations to the application of PATs is that manufacturers do not commonly provide clients with the characteristic curves of pumps running in reverse mode, thus their BEP must be estimated by the purchaser (see Section 4.2 for further details). For this purpose, several non-dimensional 1D methods exist that relate the head and flow rate values at the BEP of a PAT with those of the same machine operating as pump by means of two coefficients h and q introduced as:

$$h = \frac{H_{t,BEP}}{H_{p,BEP}} \qquad q = \frac{Q_{t,BEP}}{Q_{p,BEP}} \qquad (10)$$

Even if a large number of numerical methods to calculate such h and q can be found in the literature, it has been shown that none of them can give an accurate estimate of BEP location for any pump (Singh, 2005).

3.6.5. PAT MODIFICATIONS

In order to increase the efficiency of pumps in turbine mode, some authors have suggested modifications to the PAT geometry with respect to the standard factory design. Among the proposed modifications are:

- Substitution of the original impeller with another one having different diameter or shape. Such modifications can potentially shift the characteristic curve of the machine to better

adjust the flow conditions at which the machine is expected to work. The maximum diameter of impeller that can be fitted in a PAT depends upon its dimensions, while selection of an impeller having too low dimensions would reduce dramatically the hydraulic efficiency (Alatorre-Frenk, 1994; Williams, 1996)

- Impeller trimming (Singh, 2005; Lydon, 2016)
- suction eye modifications (Singh, 2005)
- machining the impeller tip (Alatorre-Frenk, 1994)
- modifications of the volute/guide vanes (Singh, 2005)

3.6.6. PAT COST IN LITERATURE

Another limitation to the widespread application of PATs is the absence of a comprehensive and up-to-date cost model enabling the prediction of purchase price for PATs in a European context, given their characteristics. Indeed, such information is fundamental in the preliminary design phase when choosing the most suitable hydraulic converter for a specific site. The absence of reliable cost data make recommendations for PAT installation, uncertain and unattractive. In 2009 the PAT unit cost (pump + generator) was calculated as ranging from 220 to 370€/kW for a nominal power lower than 40kW as shown in Figure 3-42 (Ramos, et al., 2009).

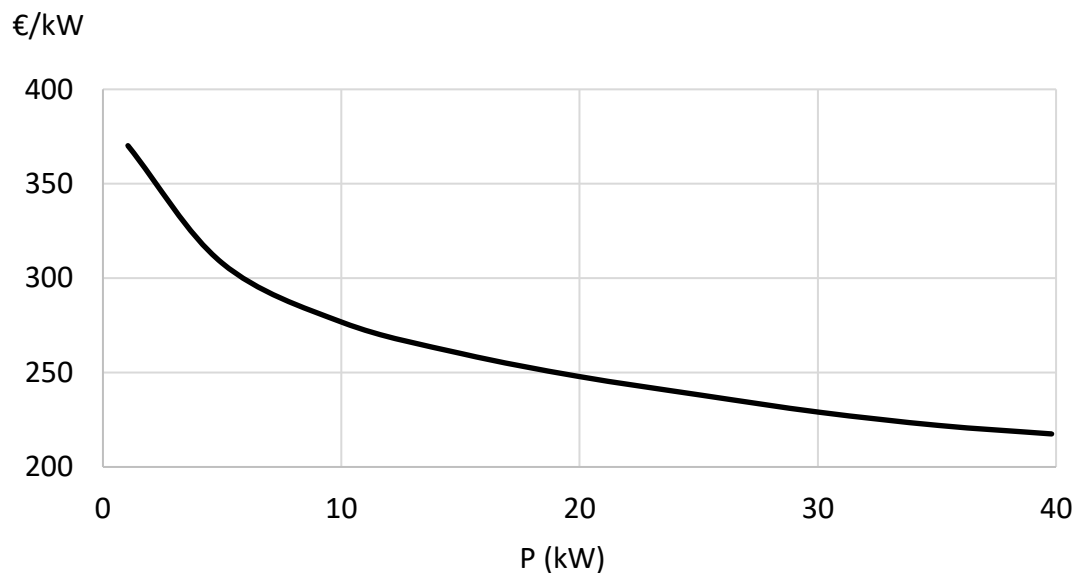


FIGURE 3-42. ESTIMATED PAT PURCHASE PRICE VS. NOMINAL POWER SHOWING ECONOMIES OF SCALE (RAMOS, ET AL., 2009).

Despite the lack of detailed information on the data source, the cost model presented in Figure 3-42 shows a decrease in unit equipment cost along the increasing nominal power as expected in the context of economies of scale.

Instead, Carravetta et al. (2013) provided more recently an estimation of PAT cost as the sum of two constant factors C_{PAT} and C_{gen} described as:

$$C_{PAT} (\text{€}) = 230 (\text{€/kW}) \cdot P_{BEP} (\text{kW}) \quad (11)$$

which accounts for the pump/turbine unit alone, proportional to the power at BEP;

$$C_{gen} (\text{€}) = 115 (\text{€/kW}) \cdot P_{max} (\text{kW}) \quad (12)$$

which accounts for the cost of generator unit and is proportional to maximum PAT power.

Another PAT cost estimate available in literature corresponds to (De Marchis, et al., 2014):

$$C (\text{€}) = 2,000 (\text{€/kW}) \cdot P_{BEP} (\text{kW}) \quad (13)$$

An obvious limitation of Equation (13) is that authors do not provide information on characteristics of pumps upon which such figure was inferred, nor the temporal validity of the presented information.

Both models proposed by (Carravetta, et al., 2013a) and (De Marchis, et al., 2014) present a fixed value of equipment unit specific cost, and therefore their applicability is expected to be limited to narrow intervals of turbine nominal powers. On the contrary, the method by (Ramos, et al., 2009) despite being slightly more outdated offers a more realistic prospective. However, all the cited PAT cost models compare favourably with the investment cost per kW of nominal power of traditional water turbines for Micro-hydro applications, which is in the range $1,000 < \text{€/kW} < 6,200$ as can be seen in Figure 3-19. As a comparison, Alatorre-Frenk in 1994 suggested PATs being five times less expensive than conventional hydro turbines working under similar conditions (Alatorre-Frenk, 1994).

3.6.7. DESIGN OF HYDROPOWER SCHEMES ADOPTING PATs

Any hydraulic installation is commonly required to face two kinds of flow and head variability: short-term variability due to sudden changes in flow conditions and long-term or “seasonal” variability (Alatorre-Frenk, 1994). A well-designed standard hydro turbine may adapt to most flow

and head conditions by means of its embedded hydraulic controls (e.g. spear valve regulation, flow deflector, guide vanes pitch). However, a PAT does not possess flow control devices and is characterized by a unique flow-head characteristic curve. Therefore, the first kind of variability is usually addressed with the implementation of an automated by-pass line to process excess flow or a throttling valve to reduce excessive pressures. Conversely, different design strategies have been proposed to accommodate a PAT in scenarios featuring significant seasonal flow variability.

- a) Design of a turbine for working at a **determined H-Q point** based on the permanent minimum flow rate (e.g. the reserved flow which must constantly be released from the foot of dams, which is unvaried along the year);
- b) Design of a single PAT meant for **intermittent operations**, thanks to the creation of a small upstream reservoir periodically emptied at once. The advantages of such approach include the ease of operations, a higher energy yield since no water is bypassed or pressure throttled and the fact that a single large machine would be more efficient and cost-effective than several smaller ones. Oppositely, the disadvantages of a system designed for intermittent operation may include the likely unsuitability of generation pattern for direct energy use, the additional cost of valve or siphon and the potential damages given by water hammer and cavitation phenomena that may occur during operations (Alatorre-Frenk, 1994). The MHP scheme at Nyon (CH) wastewater treatment plant outfall described in Chapter 3.6.3.3 relied on intermittent operations, and its operator was reported to be unsatisfied about maintenance problems (e.g. the upstream butterfly valve was damaged by cavitation and required substitution before the planned time) as well as noise and vibration levels (European Directorate for Transport and Energy - FP6 Project Shapes, 2010).
- c) Design of a **modular hydro plant** composed of a certain number of turbines in parallel operated according to instantaneous flow and head availability. Within the design options of modular PAT systems two further sub-strategies can be identified:
 1. **synchronized** PAT operations, in which several PATs are connected via a synchronous belt to a single shaft driving a generator. The main advantage of having a single generator unit is the higher efficiency and the lower purchase cost with respect to multiple smaller units, while disadvantages include the powertrain energy losses and the need for rigid procedures for PAT start-up and shutoff (Singh, 2005). The Ambootia

hydro plant described in Chapter 3.6.3.1 consists of three PAT units driving a single generator;

2. **independent** PAT operations, in which all turbines are coupled to a generator and rotate independently from each other. An example of a hydropower plant relying on multiple independent units in parallel configuration is that of Breech power station described in Chapter 3.6.3.3. Such configuration despite increasing the upfront investment and maintenance costs is normally preferred since due to its reliability and ease of operations.

Besides, further PAT control strategies can be implemented to accommodate short-term variability of head and flow rate.

- a) **Single speed operations**, if the velocity at the shaft is fixed. Such stationary shaft speed may be a function of grid frequency and generator poles as described in Equation (9), or could be adjusted through the use of a speed drive as proposed by (Fecarotta, et al., 2015). The reason behind adjusting the values of stationary rotational speed is that for a determined flow conditions a PAT could have different efficiency and power output as seen in Figure 3-43 (Livramento, 2013).

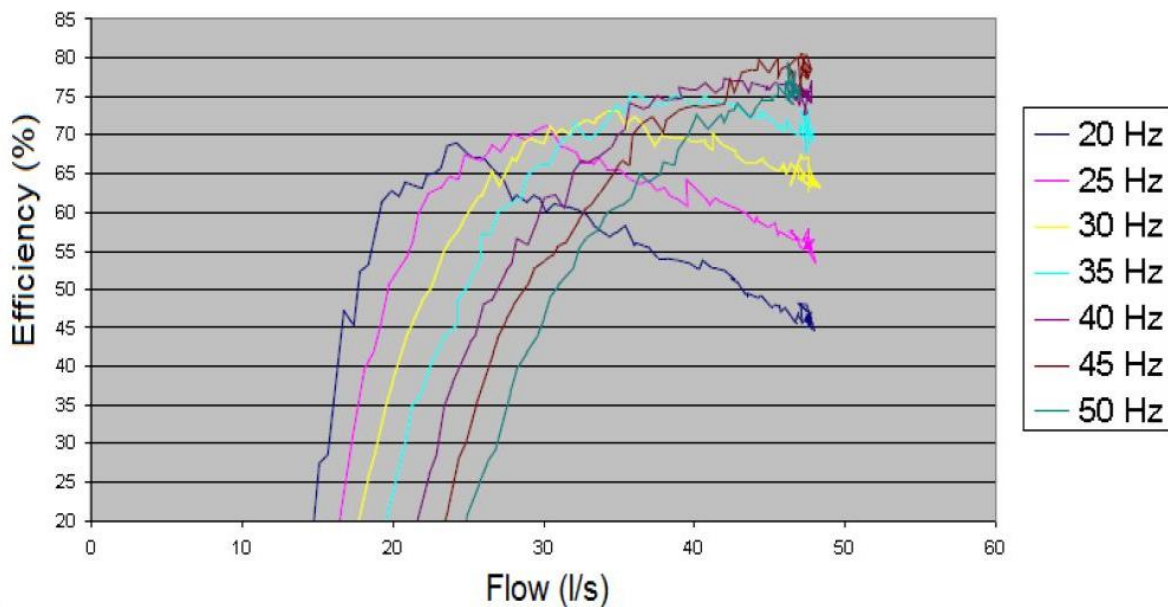


FIGURE 3-43. RESULTS OF EXPERIMENTAL TESTS OF A PAT FOR ATMOSPHERIC DOWNSTREAM PRESSURE CONDITIONS (LIVRAMENTO, 2013)

b) **Variable speed PAT operations**, in which a variable speed drive is employed to shift the characteristic curve of a turbine. In this case, a digital inverter is usually used to change the voltage and thus the synchronous rotational speed of induction generators (Carravetta, et al., 2014). An experimental chart depicting the variation of a PAT's characteristic curve and efficiency when varying the shaft speed is shown in Figure 3-44 as tested by (Singh, 2005). The upper limit to generator frequency regulation corresponds to 60Hz in order to avoid shaft speeds higher than the design velocity range, while the lower limit consists of a 50% reduction of excitation frequency and is due to the need of maintaining the efficient operation of the cooling system of the generator (Carravetta, et al., 2013a).

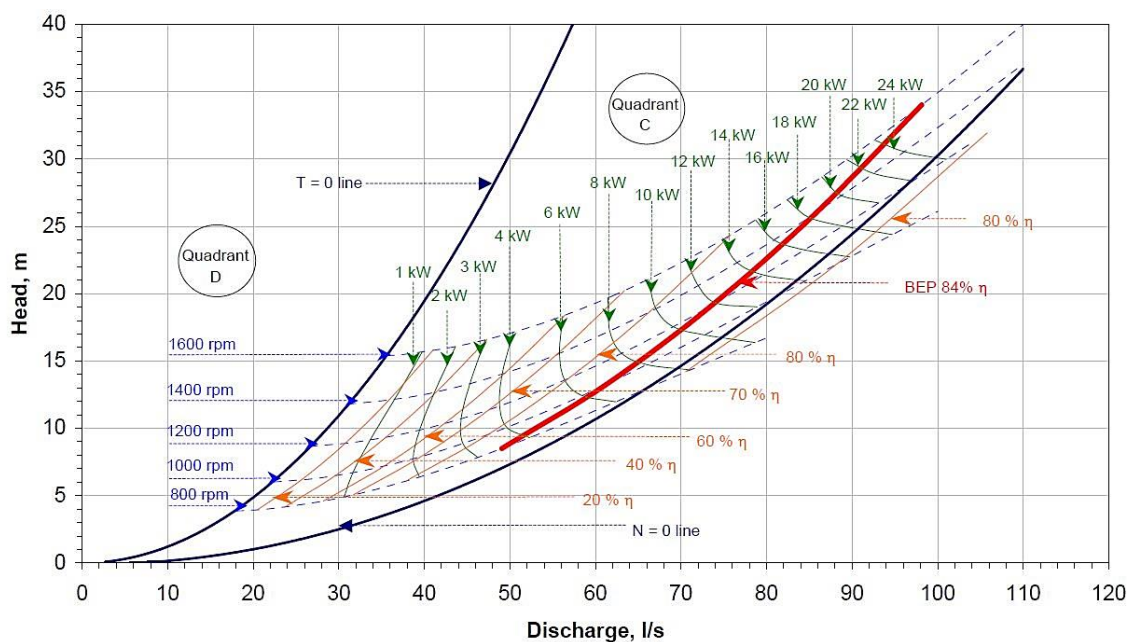


FIGURE 3-44. CHARACTERISTIC CURVES AND EFFICIENCY OF A PAT AT DIFFERENT SHAFT SPEEDS (SINGH, 2005)

Previous research compared the adoption of an automated bypass (Hydraulic Regulation, HR) with the use of a variable speed drive to modulate rotor speed (Electric Regulation, ER) for PATs installed along water distribution networks, based on data from an existing network. Results showed that schemes with HR proved to be more efficient, flexible and cost-effective than ER (Carravetta, et al., 2013a). A hypothetical system layout is displayed in Figure 3-45, while PAT operations under HR and ER mode are shown schematically in Figure 3-46.

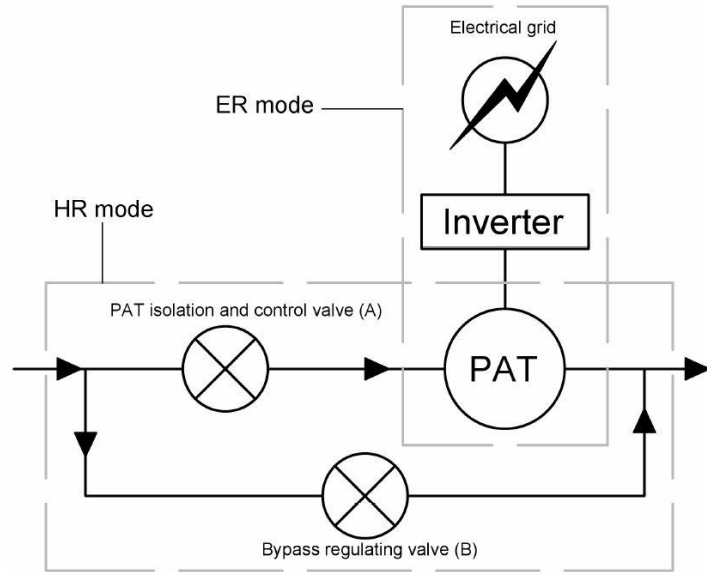


FIGURE 3-45. INSTALLATION SCHEME FOR A PAT WORKING UNDER HR OR ER MODE (CARRAVETTA, ET AL., 2013A)

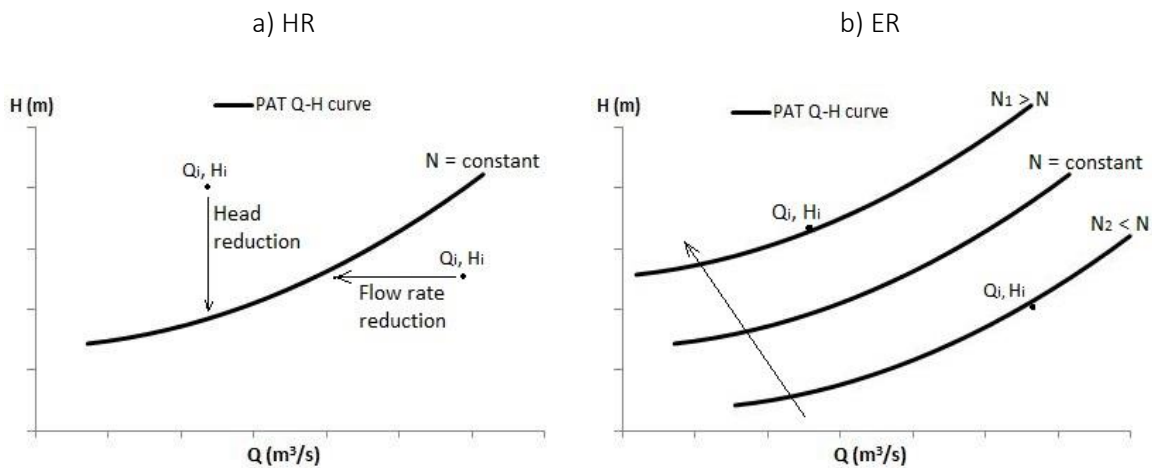


FIGURE 3-46. PAT OPERATIONS IN HR (A) AND ER (B) MODE, ADAPTED FROM (CARRAVETTA, ET AL., 2014)

Given the discussed lack of flow/head control devices, the nominal efficiency of a PAT is not a representative parameter to compare it with conventional hydraulic turbines since the latter are easily adaptable and can exploit larger portions of hydro potential at any given site. Instead, a fairer means of comparison is the ratio between the energy producible by a certain hydro generating set along a precise timeframe over the available hydraulic potential energy over the same interval (Alatorre-Frenk, 1994). An extension of such a concept for in-pipe PAT schemes was proposed by (Carravetta, et al., 2012) under the name Variable Operating Strategy (VOS), consisting of designing

and sizing the machinery for a determined site according to the maximization of plant overall efficiency η_p defined as:

$$\eta_p = \frac{\sum_{i=1}^n 9.8 \eta_i Q_{t,i} H_{t,i} \Delta t_i}{\sum_{i=1}^n 9.8 Q_i H_i \Delta t_i} \quad (14)$$

where:

Q, H : volumetric flow rate and available head present in the system ($\text{m}^3/\text{s}, \text{m}$)

Q_t, H_t : volumetric flow rate and head drop through the PAT ($\text{m}^3/\text{s}, \text{m}$)

η : PAT efficiency (-)

Δt : time interval (h)

n : number of system working points corresponding to the daily profile of available flow and head.

3.7. CONCLUSIONS

The main findings from the literature review can be summed up as

- Increasing the amount of electricity generation from renewable sources is a key worldwide challenge dictated by the needs to mitigate climate change, ensure reliability of supplies and cope with the decline in fossil fuels reserves.
- The water sector is amongst the highest energy-intensive ones as electricity is required during all phases of water extraction, purification, distribution and eventually collection and treatment of effluents.
- Hydropower is a well-established, reliable and environmental-friendly technology, with the remarkable exceptions of environmental and social impacts generated by large-scale hydraulic infrastructures.
- Increases in hydropower generation capacity in the context of climate change have been found to have the potential to increase the resilience of communities depending upon the same water resources, provided that an integrated resource management strategy is adopted under the framework of Water-Energy-Nexus (WEN).
- The application of hydropower technologies to recover energy from water networks has the potential to partially offset the energy demand of water utilities.

- Besides conventional hydro turbines, a large number of hydropower converters in the Pico and Micro scale has been proposed or tested in recent years, each working according to certain physical principles.
- Pumps As Turbines (PAT) are a low-cost and easy to maintain alternative to conventional hydro turbines, and as they can accommodate a residual downstream pressure they are suitable for use within pressurized networks. However, their major drawbacks are given by the lack of knowledge over the performance of pumps in reverse mode and by their poor performances at part-load when operated at fixed speed.
- A number of operational PAT installations can be found worldwide, working under different configurations and applied either as conventional hydropower converters or as energy recovery devices. Despite this, the majority of case studies collected refers to pilot PAT installations which have not been replicated on a large scale and in several cases have since been decommissioned due to the aforementioned difficulties in predicting their performance.
- In order to cope with long-term or short-term flow and head variability a variety of installation strategies have been suggested for PATs.

4. IDENTIFICATION OF BOTTLENECKS TO PAT IMPLEMENTATION

Why aren't PATs more used on the market? Despite their known shortcomings (e.g. no in-built flow rate regulation and low part-load efficiency, difficult availability of characteristic curves in turbine mode given from manufacturers) they still pose matchless advantages in terms of cost per installed power, maintenance simplicity and availability. Despite this, the up-to-date adoption of PATs is marginal in two sectors for which they would be ideally suitable such as stand-alone MHP (with a few exceptions in developing countries) and energy recovery from water infrastructure. In order to identify the bottlenecks to PAT implementation the initial efforts of the research were aimed at getting a better understanding on the topic by interacting with representatives of four categories of companies: potential PAT users, hydropower designers, pump manufacturers and academia.

The idea behind this was to use such interviews with experts and people working in the field to complement the findings from the literature review consisting of a collection of case studies and "best practices".

4.1. SURVEY ABOUT PAT UP-TO-DATE UTILIZATION

After having investigated the available literature and personally interviewed several experts and professionals in the field, awareness was gained of a number of possible factors that may bring about the phenomena mentioned as research questions: *which are the main bottlenecks to energy recovery in the water industry and to the large-scale diffusion of Pumps as Turbines (PATs)?*

Four different categories of firms were interviewed in the process of the survey:

- Water utilities as potential PAT end users:
 - Irish Water
 - Welsh Water
 - South West Water
 - Consorzio Comuni dell'Acquedotto del Monferrato (CCAM)
 - Asti Servizi Pubblici (ASP)
 - a number of Group Water Schemes across Ireland

- Hydropower developers/designers:
 - Dulas Ltd
 - Mhylab
 - Sasso Srl
 - Cat Engineering
 - Rentricity
 - Tecnoturbines
 - Sea-Lix
 - GIZ
- pump manufacturers:
 - Gilkes
 - Grundfos
 - KSB (Ita / UK)
 - SPP Pumps
 - Garbarino Pompe
 - Salvatore Robuschi
- Academia:
 - Federico II University (Naples, Italy)
 - Instituto Superior Tecnico (Lisbon, Portugal)

4.2. INSIGHTS GAINED

The following list shows the main findings, which are also graphically displayed in Figure 4-1.

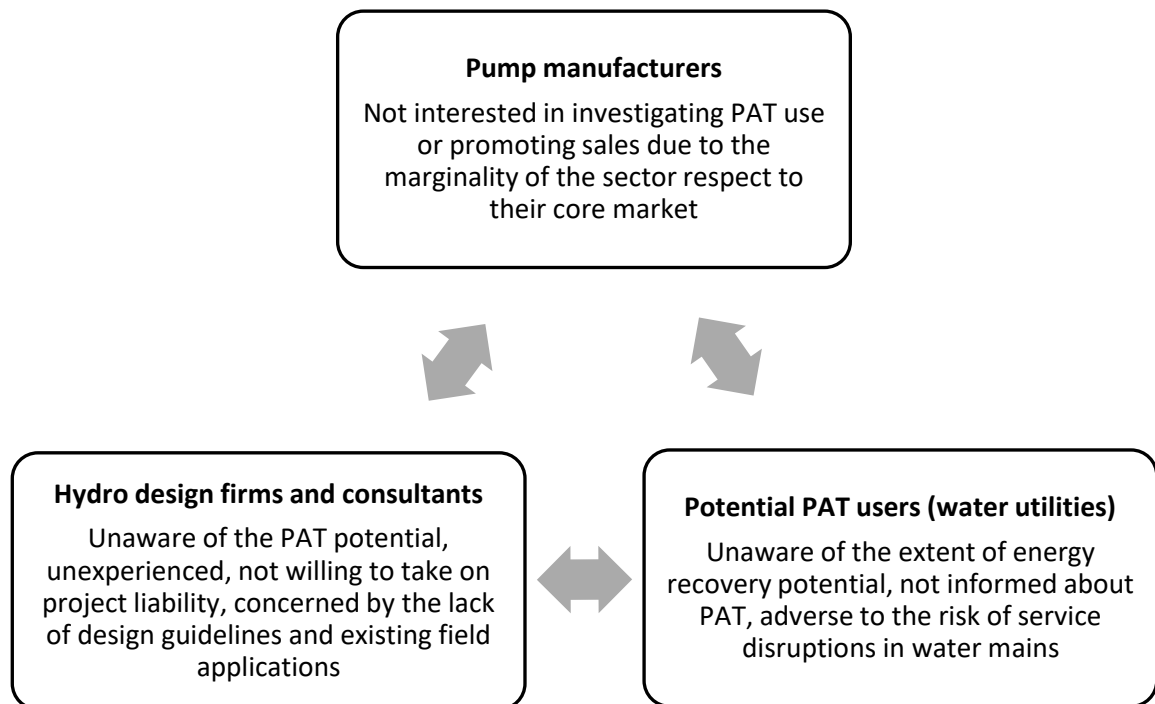


FIGURE 4-1. INVESTIGATED BOTTLENECKS TO PAT IMPLEMENTATION AND ENERGY RECOVERY FROM WATER NETWORKS

1. Pumps as Turbines are particularly suited for tapping into small hydro potentials, which has come into general interest only in recent times. Often **small-scale applications weren't even considered in the past**, because of economic aspects and perhaps cultural elements (e.g. expansive economies and little considerations of climate change).
2. Hydro design firms and consultants possess **extensive experience and established commercial channels with suppliers of conventional turbines**, which are in turn a very mature and reliable technology which haven't evolved much over the last decades.
3. Another factor could be a **limited knowledge transfer from academia to industry**. Despite the growing trend of studies on PATs published every year especially in the last decade as seen in Figure 4-2, there seem to be a lack of understanding of the PAT technology itself from many people in the sector. And a key concept to push forward any innovative solution is the customers need to understand the technology behind it.

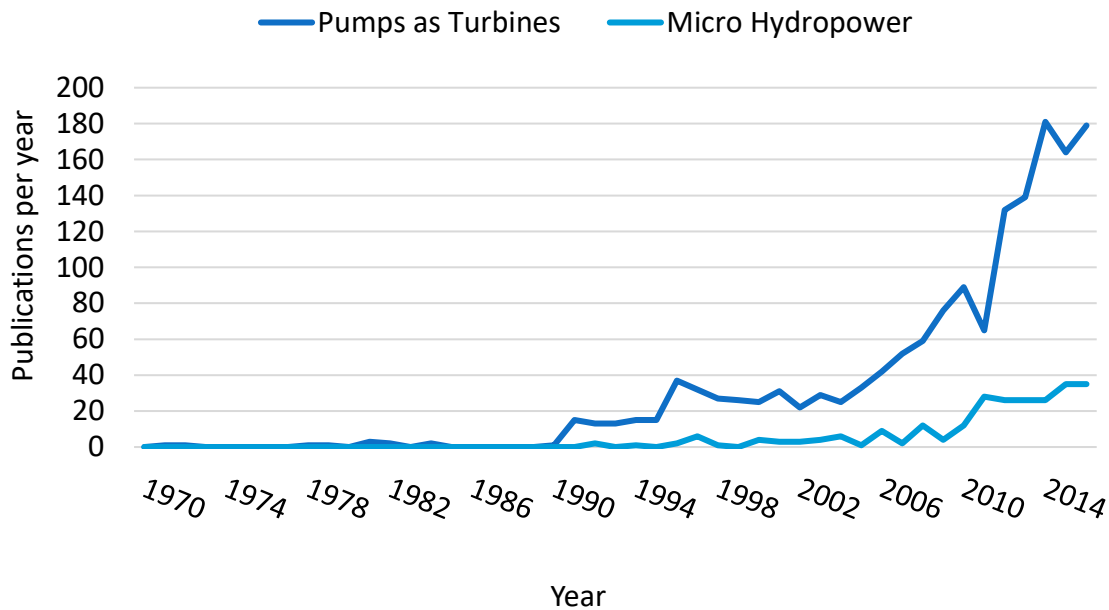
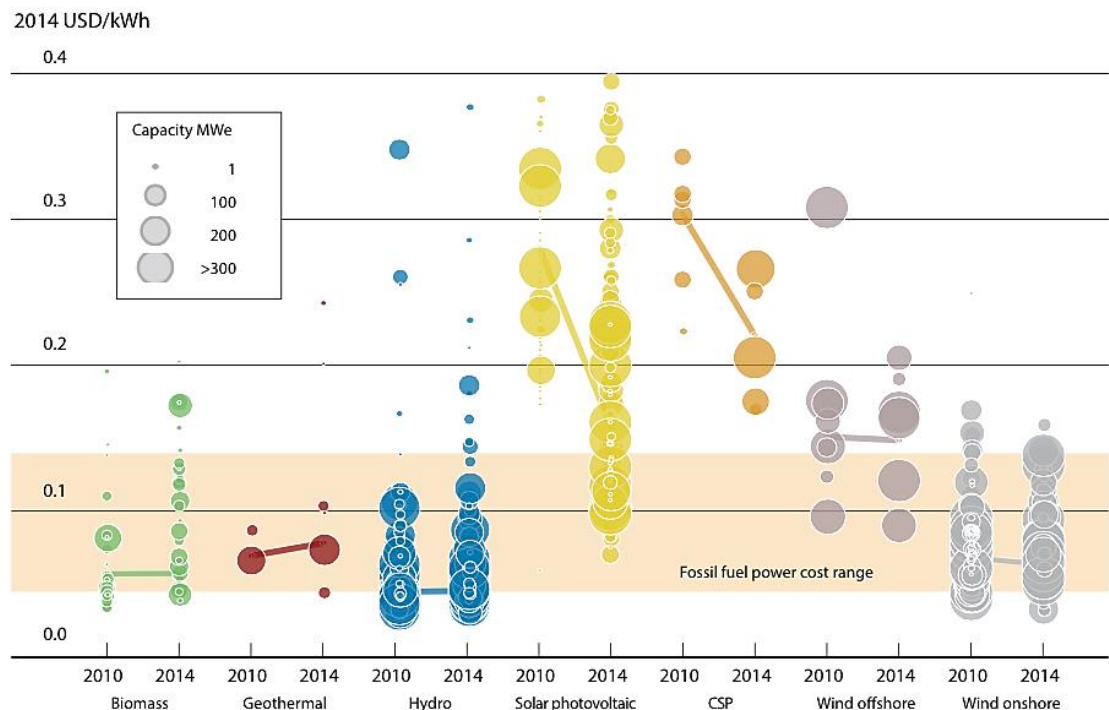


FIGURE 4-2. NUMBER OF PUBLICATIONS PER YEAR CONTAINING THE KEYWORDS “PUMP AS TURBINES” AND “MICRO HYDROPOWER” (THOMSON REUTERS’ WEB OF SCIENCE PORTAL, [HTTPS://CLARIVATE.COM/PRODUCTS/WEB-OF-SCIENCE/](https://clarivate.com/products/web-of-science/)).

4. An important factor is likely to be the **lack of sound design guidelines for PAT installations**. The only comprehensive PAT design guidebooks (Chapallaz, et al., 1992; Williams, 1996) date back to the 1990s and are uniquely addressed at the use of PATs for micro hydro without considering the constraints and opportunities arising from energy recovery within water networks. Besides, many recent developments in control systems and electrical conversion devices, especially in the area of power electronics, are not accounted for in the two mentioned resources.
5. **Project risk and liability due to design uncertainties**. Most hydro designers will uniquely look for a “proven” design strategy, possibly already applied to a few working sites. Also, it is important for them to get a measure of the uncertainties involved in the design of a hydro scheme already at early stages of a project in order to secure investments. Unfortunately, the lack of experience regarding PAT application prevents designers for taking such informed decision.
6. The presence of **Feed-In Tariffs (FIT) in the past favoured traditional turbines over PATs**. In fact, conventional purpose-made turbines can attain higher peak efficiency and thus maximise the energy production and consequent revenues. Despite this, given the gradual disappearance of FITs on the European scene several design firms are now struggling to make hydropower

competitive again. In the last decade the Levelized Cost Of Electricity (LCOE) of photovoltaic cells and wind turbines dropped significantly thanks to technological and logistical improvements, while the LCOE of hydro installations has remained almost constant as shown in Figure 4-3. This may open a new marketplace for low-cost and innovative technologies of hydro turbines.



Source: IRENA Renewable Cost Database.

Note: Size of the diameter of the circle represents the size of the project. The centre of each circle is the value for the cost of each project on the Y axis. Real weighted average cost of capital is 7.5% in OECD countries and China; 10% in the rest of the world.

FIGURE 4-3. THE LEVELIZED COST OF ELECTRICITY FROM UTILITY-SCALE RENEWABLE TECHNOLOGIES, 2010 AND 2014 (IRENA, 2015)

7. Referring particularly to the PAT application for energy recovery in Water Supply Systems (WSS), in some cases the **potential costs and risks due to the service disruption** that may occur during the installation or due to a malfunctioning of the equipment are considered unbearable by water utilities. Additionally, some organisations may view the benefits for energy recovery as marginal and not worth allocating human and financial resources on such projects.
8. **Pump manufacturers** consider the market opportunities for their own products used as turbines as a small niche compared to conventional pump market. As a reference, according to the 2014 UN COMTRADE database the total world export value of centrifugal pumps

corresponded to €10.7 billion while the 2016 market size of in-pipe energy recovery turbines – of which PATs are one of the available technologies – was estimated as 6.6 M€ which is the 0.06% of the previous value (Transparency Market Research, 2017). As a consequence, pump makers most often do not invest in research or employees formation on the topic and are not willing to promote their products as PATs. The vast majority of pump manufacturers fall into this category, with the exception of a few firms who devoted limited resources to the testing and commercialization of PATs but kept most of the findings undisclosed to the general public (e.g. Caprari, KSB).

4.3. RESPONSE TO THE INSIGHTS GAINED

While they provided a useful validation of the relevance of the research targets initially stated in Chapter 1.1, the results from the survey helped to formulate a few additional key concepts guiding the continuation of the research work. According to the lessons learned from the survey and the literature review the following principles were adopted regarding the development of the planned PAT selection software:

- Need for a stochastic framework (the problem of induction): any analysis on PAT performances based on a small number of experimental tested units cannot be representative of all such turbines. Since the performance of the multitude of pumps sold on the market when used as turbines may vary even between units of the same class made by different producers, it is impossible to make a general model about PAT performance prediction based on a small amount of samples. Instead, the analysis shall be carried out considering stochastic deviations from an ideal machine behaviour and it needs to include information from a statistically significant sample of PATs.
- Since multiple and mutually independent variables do affect the overall performance of a PAT (e.g. location of the Best Efficiency Point, steepness of the characteristic curve, variation of available flow rate with respect to the predicted values), it is appropriate to decompose the whole analysis in order to create first a series of smaller representative models able to predict such variables singularly and later bring them together in the final model/software.
- Not many businesses are aware of PAT existence and potentials, and almost none knows of a practical application of this technology in an area nearby its headquarter. As part of the

work advice should be offered to water companies and hydropower professionals resulting in case-studies comparing the adoption of PATs to that of other turbine types for determined sites. This shall both increase the general awareness about PATs and potentially result in additional field deployment of the technology being researched.

5. DEVELOPMENT OF A COMPREHENSIVE PAT SELECTION SOFTWARE

As outlined in Chapter 2, and in particular in Figure 2-1, in order to build a comprehensive PAT selection software representing a significant improvement over the limitations of the existing sizing methods several inter-related desk studies must be carried out. As discussed in Chapter 4.3, a stochastic framework built upon as much experimental data as can be possibly gathered or generated has to be adopted in order to derive significant correlations upon which the sizing algorithm shall be based. The different desk studies described hereby in Chapter 5 consist of the following elements:

- 5.1. comparison of existing PAT performance prediction methods
- 5.2. development of a model for PAT efficiency estimation
- 5.3. prediction of PAT performance away from BEP
- 5.4. prediction of PAT cost

Finally, Section 5.5 assembles the various discussed elements leading to the formulation of a functional PAT decision support and selection software.

5.1. COMPARISON OF PAT PERFORMANCE PREDICTION METHODS

As mentioned in Chapter 3.6.4, a large number of methods have been proposed over the last few decades to predict the performances of a pump in turbine mode in the absence of experimental tests. Most of them rely on empirical correlations and require as input: the pump maximum efficiency and/or its specific speed as a turbine N_s . Apart from these, some other methods instead require a specific knowledge of the internal geometry of the machine which is not normally disclosed to the common public while a few others are based on original semi-empirical equations. A summary of such methods is displayed in Table 5-1 while the equations behind the main 1D mathematical methods is listed in Table 5-2.

TABLE 5-1. SUMMARY AND BRIEF DESCRIPTION OF PREVIOUSLY PROPOSED PAT PERFORMANCE PREDICTION METHODS.

Method based on:	Author(s)	Empirical	Theoretical	Amount of information required
η_{MAX}	• Stepanoff (Stepanoff, 1957) • Childs (Childs, 1963) • Sharma (Sharma, 1985) • Thome, McClaskey & Lundqvist, Garay, Hancock (Singh, 2005) • Engeda (Singh, 2005) • BUTU (Alatorre-Frenk & Thomas, 1990) • Yang (Yang, et al., 2012) • Williams (Alatorre-Frenk, 1994)	✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓		Minimal
Ns	• Grover (Grover, 1980) • Hergt (Singh, 2005) • Barbarelli (Barbarelli, et al., 2017) • Diederich (Singh, 2005)	✓ ✓ ✓ ✓		Minimal
η_{MAX} & Ns	• Nautiyal (Singh, 2005) • Alatorre-Frenk (Alatorre-Frenk, 1994) • SchmielDI (Alatorre-Frenk, 1994) • Chapallaz (Alatorre-Frenk, 1994)	✓ ✓ ✓ ✓		Minimal
Geometric parameters	• Jain (Jain, et al., 2015) • Ventrone (Ventrone, et al., 2000) • Barbarelli (Barbarelli, et al., 2016) • Amplification coefficient (Shi, et al., 2017) • Huang (Huang, et al., 2017)	✓ ✓ ✓ ✓ ✓	✓ ✓ ✓ ✓ ✓	High
Other approaches	• Derakhshan & Nourbakhsh (Derakhshan & Nourbakhsh, 2008) • Xu Tan (Tan & Engeda, 2016)	✓ ✓	✓ ✓	Variable

TABLE 5-2. EQUATIONS BEHIND THE MAIN 1-D NUMERICAL METHODS FOR PAT BEP PREDICTION.

Method	h	q	Other Correlations
Stepanoff (Stepanoff, 1957)	$\frac{1}{\eta_{max,p}}$	$\frac{1}{\sqrt{\eta_{max,p}}}$	/
Childs (Childs, 1963)	$\frac{1}{\eta_{max,p}^2}$	$\frac{1}{\eta_{max,p}^2}$	/
Sharma (Sharma, 1985)	$\frac{1}{\eta_{max,p}^{1.2}}$	$\frac{1}{\eta_{max,p}^{0.8}}$	/
Thorne / McClaskey & Lundqvist / Garay / Hancock (Singh, 2005)	$\frac{1}{\eta_{max,p}}$	$\frac{1}{\eta_{max,p}}$	/
Chapallaz (Alatorre-Frenk, 1994)	$\frac{1.1 \cdot (1 + (0.3 + \ln(\Omega)^2)^{0.3})}{(\eta_{max,p})^{0.8}}$	$\frac{1.12 \cdot (1 + (0.4 + \ln(\Omega)^2)^{0.15})}{(\eta_{max,p})^{0.6}}$	$\eta_{max,t}$ = $\eta_{max,p}$ – 0.03
Alatorre-Frenk 2 (Alatorre-Frenk, 1994)	$\frac{1.21 \cdot (1 + (0.6 + \ln(\Omega)^2)^{0.3})}{(\eta_{max,p})^{0.8}}$	$\frac{1.21}{(\eta_{max,p})^{0.6}}$	/
Schmielidl (Alatorre-Frenk, 1994)	$-1.516 + \frac{2.369}{(\eta_{max,t} \cdot \eta_{max,p})^{0.5}}$	$-1.378 + \frac{2.455}{(\eta_{max,t} \cdot \eta_{max,p})^{0.25}}$	$\eta_{max,t}$ = $\eta_{max,p}(1.158 - 0.265\Omega_p)$
Hergt (Singh, 2005)	$1.3 - \frac{1.6}{(N_{s,t} - 5)}$	$1.3 - \frac{6}{(N_{s,t} - 3)}$	/
Williams (Alatorre-Frenk, 1994)	$\frac{1.1}{\eta_{max,p}^{1.2}}$	$\frac{1.1}{\eta_{max,p}^{0.8}}$	/
Diederich (Singh, 2005)	$\frac{1.556}{\Omega^{0.174}}$	$\frac{1.402}{\Omega^{0.171}}$	/
BUTU (Alatorre-Frenk & Thomas, 1990)	$\frac{1}{0.85 \eta_{max,p}^5 + 0.385}$	$\frac{0.85 \eta_{max,p}^5 + 0.385}{2 \eta_{max,p}^{9.5} + 0.205}$	$\eta_{max,t}$ = $\eta_{max,p}$ – 0.03

(Yang, et al., 2012)	$\frac{1.2}{\eta_{max,p}^{1.1}}$	$\frac{1.2}{\eta_{max,p}^{0.55}}$	/
Gopalakrishnan (Singh, 2005)	$\frac{1}{(\eta_{p,BEP})^2}$	$\frac{1}{\eta_{p,BEP}}$	/
(Grover, 1980)	$2.693 - 0.0229N_{s,t}$	$2.379 - 0.0264 N_{s,t}$	/
Nautiyal (Singh, 2005)	$41.667 \frac{\eta_{max,p} - 0.212}{\ln(N_s)} - 5.042$	$30.303 \frac{\eta_{max,p} - 0.212}{\ln(N_s)} - 3.424$	/

While the methods relying on the knowledge of η_{MAX}/N_s described in Table 5-2 consist of a limited set of equations each, there are two additional methods introduced in Table 5-1 with the label “Other approaches” which are each composed of a larger set of equations described in the following sub-chapters.

5.1.1. DERAKHSHAN & NOURBAKHS METHOD DESCRIPTION

The present methodology has been elaborated by the authors on the basis of an experimental study of four centrifugal pumps with different N_s (always < 60), and it follows a series of steps (Derakhshan & Nourbakhsh, 2008):

1. acquiring knowledge of the BEP of a PAT in turbine mode (Q_t , H_t , η_{MAX});
2. calculating the parameter N_s (kW, m) as per Equation (5) at Page 45;
3. obtaining the specific speed of the PAT in pump mode $N_{s,p}$ (m, m³) as:

$$N_{s,p} = 0.3705 N_{s,t} + 5.083 \quad (15)$$

4. calculating the dimensionless parameter γ through the following formulation:

$$\gamma = 0.0233 \frac{N_{s,p}}{g^{0.75}} + 0.6464 \quad (16)$$

5. provided that the PAT rotational speed in turbine mode equals the speed in pump mode, the nondimensional head ratio h can be calculated as:

$$h = \frac{1}{\gamma^2} \quad (17)$$

6. after calculating H in pump mode, knowing N and $N_{s,p}$ (m, m^3) it is possible to obtain the value of Q (and hence the nondimensional coefficient q);
7. in order to cater for the difference in impeller diameter between the PATs tested by the authors and other models, three additional correlations have been proposed to adjust the values of h and q coefficients:

$$h_{new} = h \left(\frac{0.25}{D} \right)^{1/4} \quad q_{new} = q \left(\frac{0.25}{D} \right)^{1/6} \quad (18)$$

5.1.2. XU TAN METHOD DESCRIPTION

This method was developed in the Michigan State University (US) upon data collected from four PAT experimentally tested (Tan & Engeda, 2016). It consists of the following steps:

1. knowing the speed (ω in rad/s) and BEP coordinates of a centrifugal pump, calculate its N_s as per Equation (3) at Page 45;
2. derive the PAT N_s in turbine mode as:

$$N_{s,t} = 0.752 \cdot N_{s,p} + 0.0883 \quad (19)$$

3. calculate the specific diameter of the PAT in pump mode ($D_{s,p}$) defined as:

$$D_{s,p} = D(mm) \frac{H_{BEP,p}^{0.25}}{\sqrt{Q_{BEP,p}}} \quad (20)$$

4. estimate the specific diameter of the PAT in turbine mode ($D_{s,t}$) as:

$$D_{s,t} = 1.072 \cdot D_{s,p} - 0.1419 \quad (21)$$

5. derive the coordinates of the BEP in turbine mode as:

$$H_{BEP,t} = \left(\frac{\omega \cdot D}{N_{s,t} \cdot D_{s,t} \cdot g^{3/4}} \right)^2 \quad Q_{BEP,t} = \frac{\omega \cdot D^3}{N_{s,t} \cdot D_{s,t}^3 \cdot g^{3/4}} \quad (22)$$

6. finally, the maximum efficiency of the PAT in turbine mode can be estimated as:

$$\eta_{max,t} = \frac{\eta_{max,p}}{0.2267 \cdot N_{s,p} + 0.8057} \quad (23)$$

5.1.3. ASSESSMENT OF BEP PREDICTION MODELS

A large database containing the information on BEP in pump and turbine mode of 119 commercial pumps of different type and manufacturer has been collected here and organized based on publicly available material as shown in Annex 4.1 (Alatorre-Frenk, 1994; Derakhshan & Nourbakhsh, 2008; Calado, 2014; Teuteberg, 2010; KSB and Caprari pers. communication, 2018). The various methods proposed to correlate the BEP in turbine mode to that in pump mode described in Table 5-1 and Table 5-2 (with the exception of those relying on geometric parameters) have been tested against them and the case-by-case results of the analysis are displayed in Figure 5-1. All methods were used within the range of applicability stated by its inventors in terms of range of specific speed.

The error between the predicted and the actual H at the BEP in turbine mode for each PAT has been calculated as:

$$E_h = \frac{h_p - h_a}{h_a} \quad (24)$$

where h_p and h_a are the predicted and actual nondimensional h coefficients respectively. A similar formulation was used to calculate the errors associated with Q at the BEP of each PAT:

$$E_q = \frac{q_p - q_a}{q_a} \quad (25)$$

Hence the overall error associated with each method was eventually computed as:

$$E = \sqrt{E_h^2 + E_q^2} \quad (26)$$

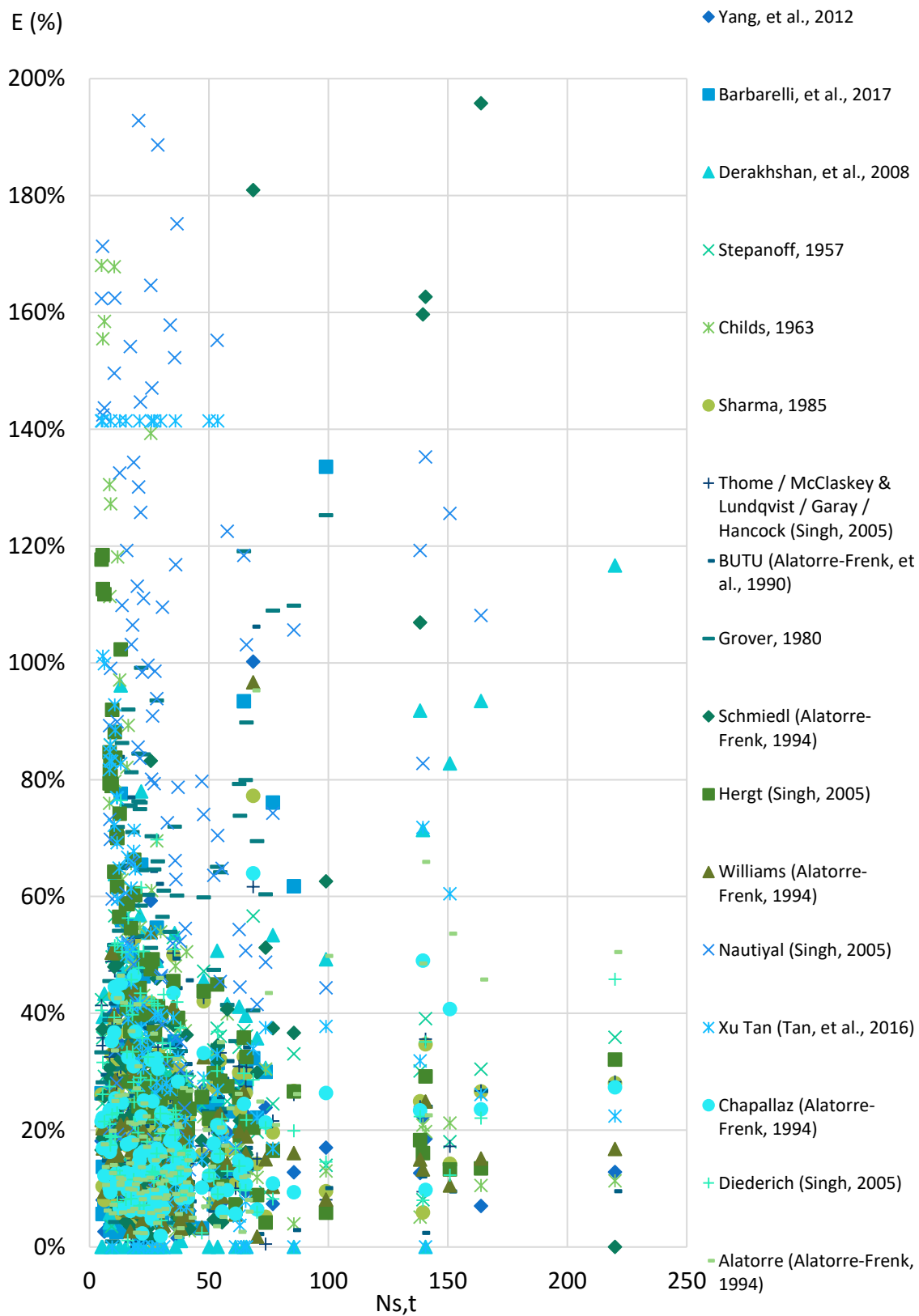


FIGURE 5-1. PERCENTAGE ERROR BETWEEN THE MEASURED AND CALCULATED BEP OF THE 119 ANALYSED PATS.

As can be noticed from Figure 5-1 the error associated with some of the methods, in particular those by Nautiyal, Childs and Schmiedl, can reach up to nearly 200% for a few machines which is by no means acceptable from a hydro design or performance point of view. Another indicative result from Figure 5-1 is that the accuracy of the analysed methods can vary greatly from one PAT to another even if the two of them are characterized by a similar specific speed (N_s).

In order to assess which methods performed better or worse than others when applied to the whole set of sample PATs the mean error across all analysed machines has been derived in Figure 5-2 for $N_s < 60$ and in Figure 5-3 for $n_s > 60$.

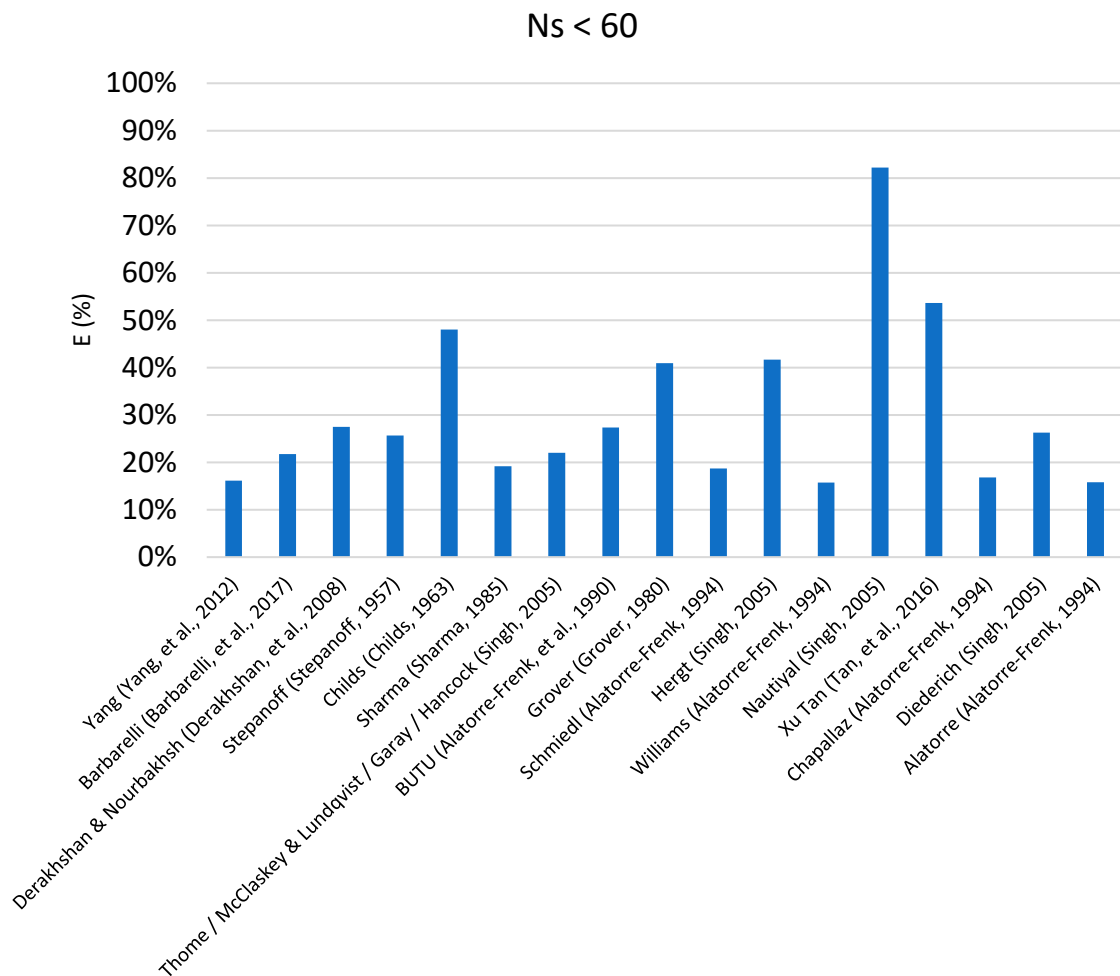


FIGURE 5-2. MEAN ERROR FROM EACH METHOD ACROSS THE ANALYZED DATASET OF PATS WITH $N_s < 60$.

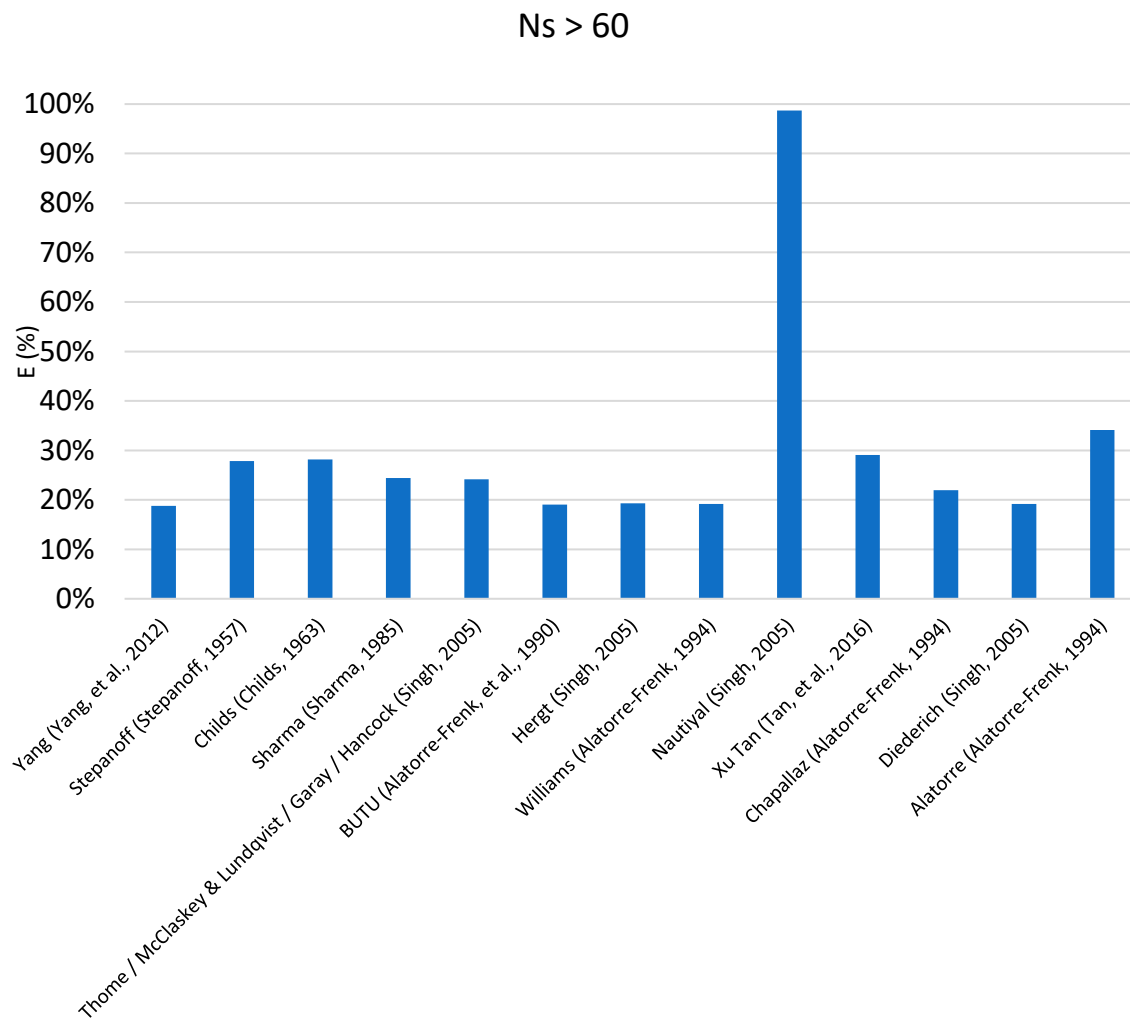


FIGURE 5-3. MEAN ERROR FROM EACH METHOD ACROSS THE ANALYZED DATASET OF PATS WITH NS > 60.

The bar chart of Figure 5-2 shows clearly that the Nautiyal, Xu Tan and Childs methods performed much worse than the remaining ones, generating mean errors up to 82%. Conversely, six methods generated mean errors lower than the 20% threshold which would be acceptable on a single machine. Among those the best results derived from the ones by Williams and Alatorre (15.8%) followed up by Yang (16.2 %) and Chapallaz (16.8 %).

From the chart of Figure 5-3 instead it can be noticed that the Nautiyal method was again the worst one in the set and on average it led to a mean error of nearly 100% between the predicted and actual BEP coordinates of PATs with Ns < 60. Conversely, the best methods for this set of machines were the ones by Yang (18.8 %), BUTU (19 %), Williams (19.2 %) and Hergt (19.3 %).

In conclusion, the Yang method was the one which performed consistently better than others across the whole data set for PATs with both high and low N_s .

5.2. DEVELOPMENT OF A MODEL FOR PAT EFFICIENCY ESTIMATION

The content of the present chapter has been partially published as:

Novara, D., S. Derakhshan, A. McNabola, and H. Ramos. "Estimation of unit cost and maximum efficiency for Pumps as Turbines." *Young Water Professionals* (2017).

The full research paper is hereby attached as Annex 3.1.

5.2.1. INTRODUCTION AND METHOD

The available literature shows that the maximum hydraulic efficiency of PATs ranges between values as low as 40% up to nearly 90% but no practical mathematical tool or chart exists to readily estimate the rated efficiency of a PAT under given nominal conditions.

A database containing the Best Efficiency Point (BEP) of 280 pumps coupled with asynchronous generators was gathered, and their performance as turbine was estimated through the Yang methods following the recommendations of Section 5.1. The resulting PAT database is attached as Annex 4.2 (Grundfos, 2016). Based on this information a 3D function has been developed relating the maximum efficiency of a PAT + generator unit to its nominal flow rate and specific speed by modifying an equation previously suggested in the literature for the prediction of pump efficiency (Anderson, 1977). All PATs have been considered as coupled to IE3 efficiency class asynchronous generators whose efficiency requirements are shown in Figure 5-4.

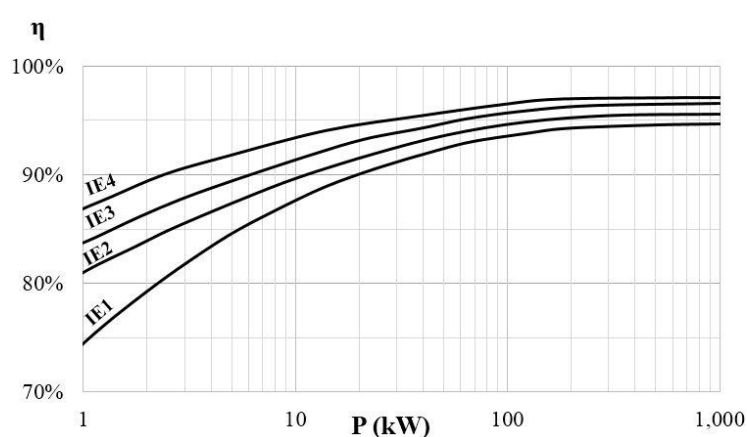


FIGURE 5-4. PERFORMANCE OF ASYNCHRONOUS MOTORS ACCORDING TO RATED POWER AND EFFICIENCY CLASS (ABB, 2014)

The dependence of maximum efficiency of pumps on specific speed and flow rate has been widely acknowledged, and often represented on 2D charts (Chapallaz, et al., 1992; Łazarkiewicz & Troskołański, 1965). Another author proposed a mathematical correlation to estimate the rated efficiency of a hydraulic pump (Anderson, 1977):

$$\eta_{max} = a - \frac{b}{Q_{BEP}^c} - d \left(e + \ln \frac{N_s}{52.933} \right)^2 \quad (27)$$

The term N_s refers to the specific speed defined as per the previous Equation (3) at Page 45. The value 52.933 in Equation (27) corresponds to the conversion factor between the specific speed computed as N_s and Ω as defined in Equation (4) at Page 45.

While Anderson suggested coefficients for Equation (27) adequate for pump performance prediction, new coefficients have been calculated here for PATs coupled to IE3-class induction generators by means of least squares method based on the 280 pumps coupled with asynchronous generators, shown as Annex 4.2.

5.2.2. RESULTS AND DISCUSSION

The resulting coefficients a , b , c , d and e for Equation (27) as well as the original ones for pumps proposed by Anderson are summarized in Table 5-3.

TABLE 5-3. COEFFICIENTS FOR EQUATION (27)

	a	b	C	d	e
Pumps (Anderson, 1977)	0.94	0.048	0.32	0.055	0.18
PAT + IE3 generator	0.89	0.024	0.41	0.076	0.22

Graphic results of the equation to estimate the efficiency of PAT + generator units are shown in Figure 5-5, while a 2-D contour chart similar to the one proposed for pumps by (Chapallaz et al., 1992) is presented in Figure 5-6.

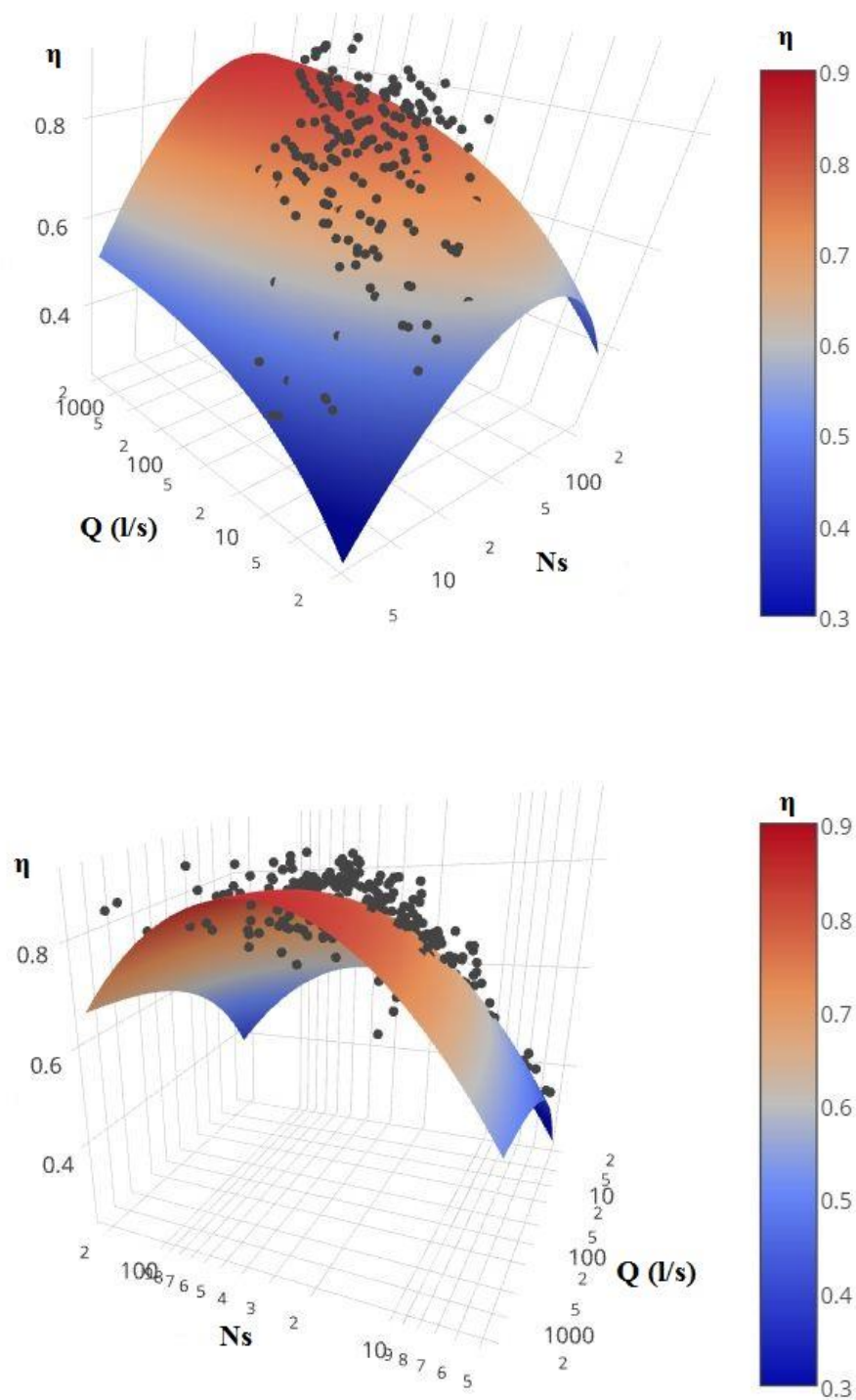


FIGURE 5-5. PROPOSED 3D FUNCTION FOR PAT EFFICIENCY ESTIMATION PLOTTED AGAINST THE ACTUAL EFFICIENCY OF SELECTED MACHINES.

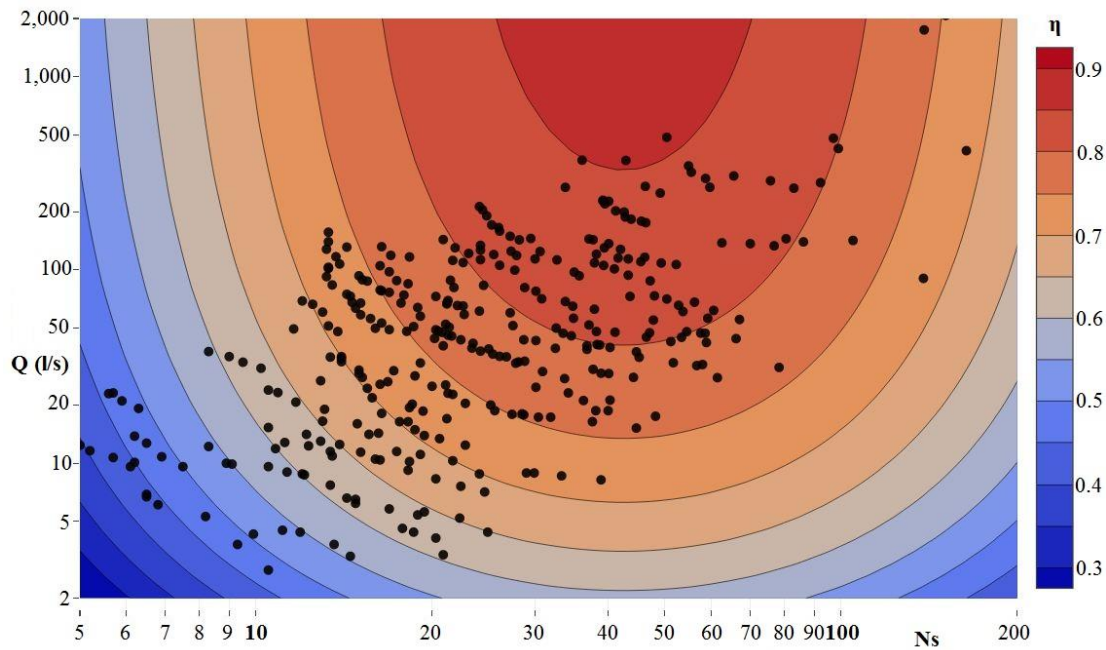


FIGURE 5-6. 2D CONTOURS OF THE PROPOSED FUNCTION FOR PAT EFFICIENCY ESTIMATION PLOTTED AGAINST THE NOMINAL FLOW RATE AND SPECIFIC SPEED OF SELECTED MACHINES.

Finally, Figure 5-7 shows the distribution of errors associated with the proposed function with respect to the actual efficiency of the analysed PAT + generator units. The mean error corresponded to 5.2%, with 86% of samples falling in the $\pm 10\%$ error interval and only four units showing a deviation from estimated efficiency higher than 20%.

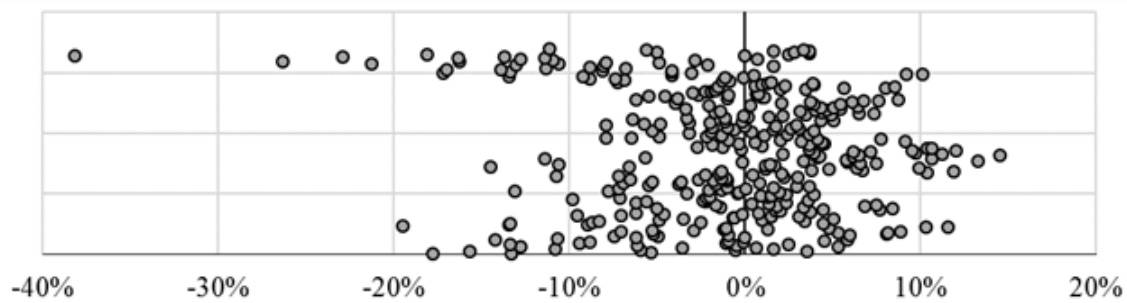


FIGURE 5-7. PERCENTAGE OF ERROR ON PAT H ESTIMATION THROUGH THE PROPOSED EQUATION.

The results showed that the turbine peak efficiency may vary greatly according to PAT specific speed and nominal flow rate as seen in Figure 5-5 and Figure 5-6 which explains the wide range of values found in literature. As a result a function was developed to predict the efficiency of a PAT generating set given its nominal flow rate and specific speed.

5.3. PREDICTION OF PAT PERFORMANCE AWAY FROM BEP

The content of the present chapter has been published as:

Novara, D.; McNabola, A. (2018). A model for the extrapolation of the characteristic curves of Pumps as Turbines from a datum Best Efficiency Point. *Energy conversion and Management*, 174, 1-7. <https://doi.org/10.1016/j.enconman.2018.07.091>

The full research paper is hereby attached as Annex 3.2.

5.3.1. INTRODUCTION

As discussed in Chapter 3.6, a major barrier to PAT implementation on a wide scale is the unavailability of the characteristic curves of pumps when operated as turbines since most manufacturers do not provide clients with such information and/or do not possess it. The most error-free method available to verify the performance of pumps in reverse running mode is their physical testing in a dedicated lab facility, which is an extremely resource-demanding process. Alternatively, some authors have proposed the use of Computational Fluid Dynamics (CFD) which in turn is prone to errors, requires considerable computational resources and needs an extensive knowledge of the internal geometry of each unit (Binama, et al., 2017).

Therefore, researchers and practitioners need to cope with the lack of knowledge involving the vast majority of the off-the-shelf available PATs through numerical models based on empirical data which are capable of estimating with reasonable accuracy the performance of any unit under turbine operations based on its known pump behaviour. For instance, several mathematical correlations have been proposed to correlate the location of the Best Efficiency Point (BEP) of a unit between pump and turbine mode, which is defined as the coordinates of head H (m) and flow rate Q (m^3/s) under which the machine works at maximum efficiency as introduced in Chapter 3.6.4. However, since most applications of PATs for in-pipe energy recovery require the units to accommodate large variations of flow and head (Jain, et al., 2015; Corcoran, et al., 2017) it is of vital importance to derive not only the location of the BEP in turbine mode but also the complete characteristic curve of a unit in order to predict its performances within its whole application range. This becomes very relevant when considering that the absence of in-built regulation devices in PATs commonly leads to poor part-load performances when compared to classic hydro turbines.

A key parameter for assessing the performance of a PAT is the specific speed N_s previously defined in Equation (3) at Page 45, and Chapallaz et al. (1992) suggested that the complete Q-H

characteristic curves of PATs having different N_s tend to have variable shapes according to the respective value of N_s as shown in Figure 5-8. However, they did not propose any mathematical correlation to approximate those curves. It was assumed that a properly designed PAT scheme would operate exclusively at the machine's BEP, and therefore knowing the full shape of the characteristic curves was unimportant. Such an assumption may arguably be valid in the context of traditional hydropower plants associated with a reservoir. However, this is not true in cases in which a PAT would need to cope with recurring variations in available flow rate and pressure head (e.g. in water-pipe networks).

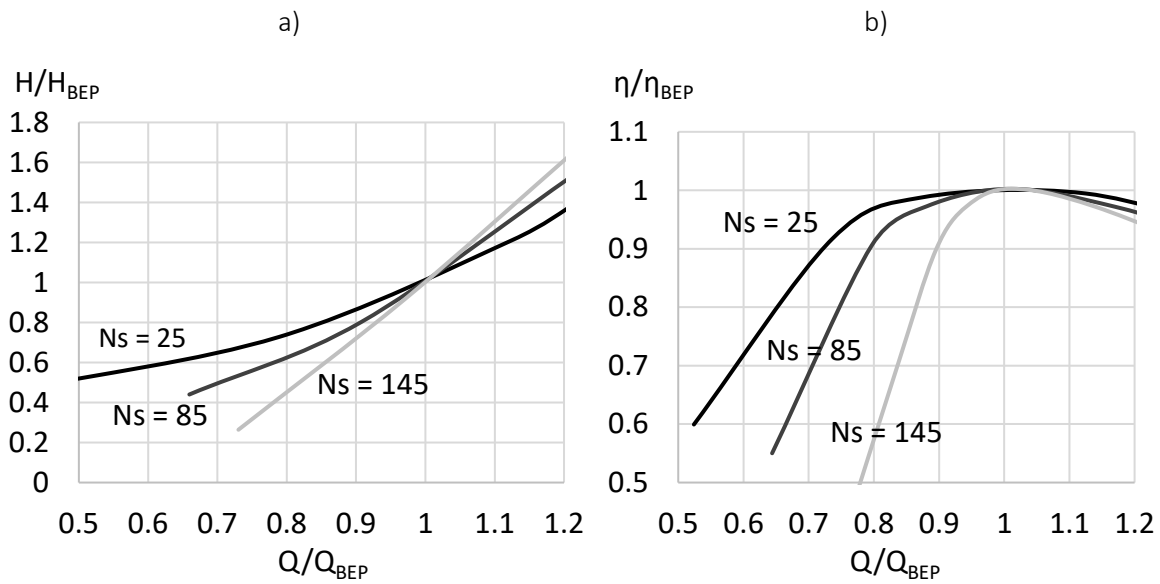


FIGURE 5-8. COMPARISON OF THREE PATs HAVING DIFFERENT N_s : (A) RELATIVE HEAD VERSUS RELATIVE FLOW RATE, (B) RELATIVE EFFICIENCY VERSUS RELATIVE FLOW RATE (ADAPTED FROM: CHAPALLAZ ET AL., 1992).

Acknowledging the importance of estimating the full head and efficiency characteristic curves within the whole PAT range, Derakhshan and Nourbakhsh (2008) suggested to approximate the relative head and power curves over the processed flow rate for any given machine using a second and third order polynomial respectively. The authors then suggested a set of empirical coefficients based on experimental testing of eleven PATs having N_s in pump mode in the range 14.6-55.6 (Derakhshan & Nourbakhsh, 2008), resulting in:

$$H_t / H_{t,BEP} = 1.0283 (Q_t / Q_{t,BEP})^2 - 0.5468 (Q_t / Q_{t,BEP}) + 0.5314 \quad (28)$$

$$P_t / P_{t,BEP} = - 0.3092 (Q_t / Q_{t,BEP})^3 + 2.1472 (Q_t / Q_{t,BEP})^2 - 0.8865 (Q_t / Q_{t,BEP}) + 0.0452 \quad (29)$$

Pugliese et al. (2016) reviewed the above equations based on laboratory testing of two radial PATs, one of which was a multistage, having N_s of 28.7 and 40.7. The authors confirmed the validity of Equation (28) relative to the head curve, but instead suggested that Equation (29) shall be considered valid only for values of non-dimensional flow rates lower than 0.4 (Pugliese, et al., 2016). Such non-dimensional flow rate is defined as:

$$\Phi = Q / (n D^3) \quad (30)$$

where Q (m^3/s) is the nominal flow rate, n the shaft speed (rad/s) and D (m) the impeller diameter. The updated third-degree polynomial approximating the power curves of PATs proposed as a replacement of Equation (29) is (Pugliese, et al., 2016):

$$P_t / P_{t,BEP} = 4.000 \cdot 10^{-3} (Q_t / Q_{t,BEP})^3 + 1.386 (Q_t / Q_{t,BEP})^2 - 0.390 (Q_t / Q_{t,BEP}) \quad (31)$$

Within the same paper the authors pointed out that the accuracy of such equations was questionable when dealing with multistage units, since the combined use of Equation (28) and (31) “under-estimated the experimental results by around 20-30%” regarding the tested two-stage pump. Subsequently, they proposed further experiments on multistage units to be performed in order to evaluate whether dedicated equations are needed or rather the measured inaccuracies have no statistical relevance in the context of larger experimental data (Pugliese, et al., 2016).

Barbarelli et al., (2017) also suggested alternative coefficients for head and power polynomials based on the formulation proposed by Derakhshan & Nourbakhsh (2008). Based on experimental data from 27 pumps tested as turbines the authors generated coefficients fitting the relative head and power curves for each machine and subsequently calculated their mean values to replace the original coefficients of Equations (28) and (29), resulting into:

$$H_t / H_{t,BEP} = 0.922 (Q_t / Q_{t,BEP})^2 - 0.406 (Q_t / Q_{t,BEP}) + 0.483 \quad (32)$$

$$P_t / P_{t,BEP} = 0.040 (Q_t / Q_{t,BEP})^3 + 1.185 (Q_t / Q_{t,BEP})^2 - 0.043 (Q_t / Q_{t,BEP}) - 0.183 \quad (33)$$

Finally, Fecarotta et al. (2016) proposed the approximation of the head and power curves of a PAT by means of two second-degree polynomials. The coefficients of such equations have been derived

by the experimental characterization of five PATs of different type: centrifugal, semi-axial and submersible. The resulting equations are (Fecarotta, et al., 2016):

$$H_t / H_{t,BEP} = 1.61 (Q_t / Q_{t,BEP})^2 - 1.41 (Q_t / Q_{t,BEP}) + 0.805 \quad (34)$$

$$P_t / P_{t,BEP} = 1.85 (Q_t / Q_{t,BEP})^2 - 0.858 (Q_t / Q_{t,BEP}) - 0.00567 \quad (35)$$

Given the unavailability of reasonable alternatives, researchers and practitioners who deal with PATs are in need of a set of mathematical correlations enabling them to select the most appropriate equipment based solely on the known machine characteristics in pump mode. Several methods exist to predict the values of flow rate and head across a PAT leading to maximum efficiency (i.e. the Best Efficiency Point). However, in most cases this information is not sufficient since practically most PATs will need to work under variable flow and head conditions. Therefore, as a further step it is necessary to develop a numerical model based on empirical data to predict the full shape of the characteristic curves of any pump in turbine mode once that its BEP has been calculated. Four such mathematical methods for PAT curves prediction have been proposed in the literature, but their accuracy is limited by the reduced sample of empirical PAT curves upon which the underlying calculations were based on. Besides, the use of fixed-coefficients polynomials for each of the four models does not capture the real behaviour of PATs having different N_s as shown in Figure 5-8.

The scope of the present sub-section is to derive a new set of polynomials with variable coefficients for the extrapolation of the full characteristic curves of PATs based on a significantly larger dataset of experimental PAT performance curves (113), than those used in previous research. The new model obtained will be compared to the previously existing ones in order to assess its accuracy. Secondly, thanks to the existing PAT curves database the part-load efficiency of PATs will be compared to that of conventional hydro turbines.

5.3.2. METHOD AND CALCULATIONS

The characteristic curves of 113 PATs of different sizes and from different manufacturers, tested in turbine mode have been compiled as shown in Figure 5-9 and detailed in Annex 4.3 (Alatorre-Frenk, 1994; Derakhshan & Nourbakhsh, 2008; Teuteberg, 2010; Calado, 2014; Scafa, 2014; KSB and Caprari pers. communication, 2018). The sample consisted of radial end-suction, semi-axial and axial PATs.

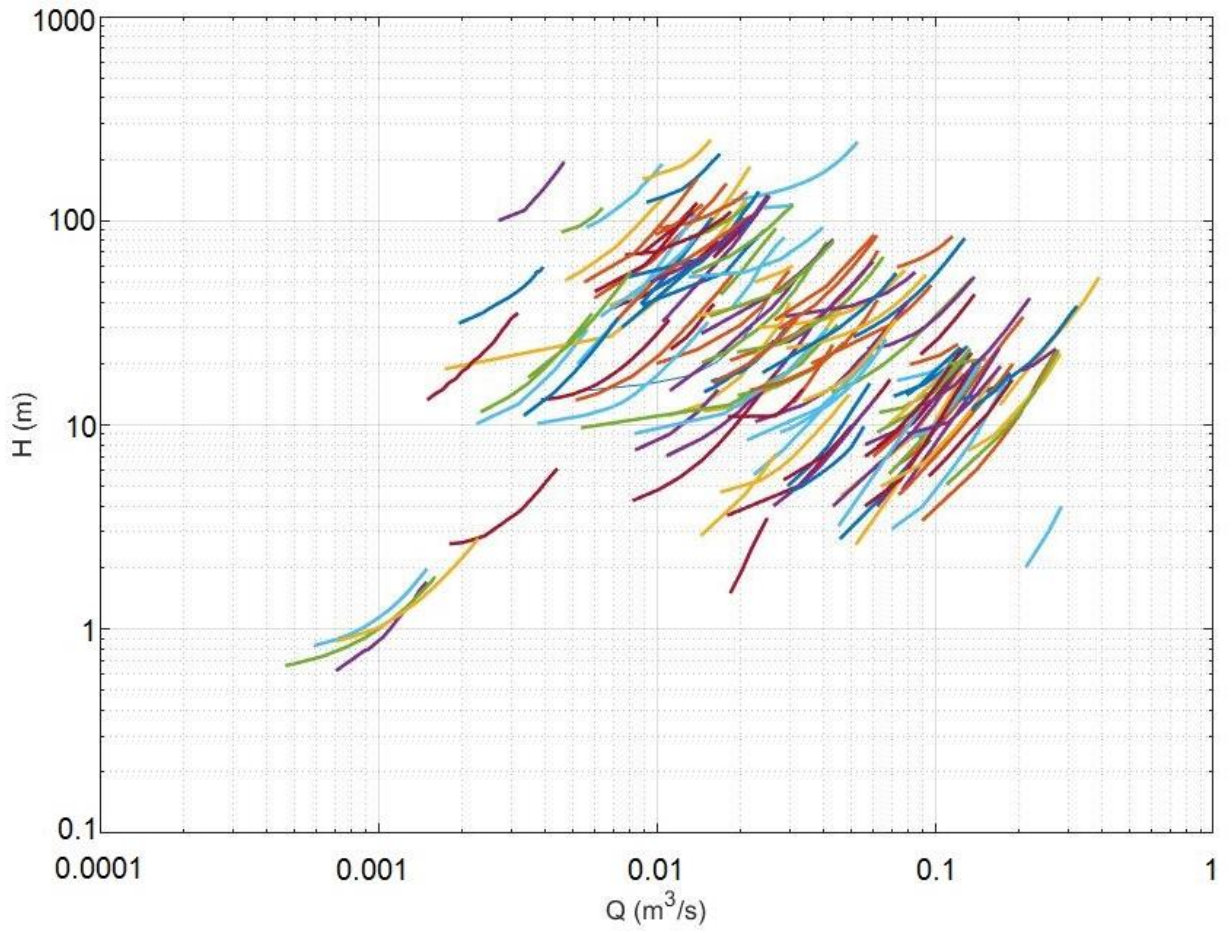


FIGURE 5-9. CHARACTERISTIC CURVES OF THE 113 PATS.

For each machine the experimental characteristic head and power curves were approximated as best fits of two second degree polynomials as shown in Equations (36) and Equation (37). The relative power curves were treated as second degree polynomials as suggested by (Fecarotta et al., 2016) rather than third degree as proposed by other authors since in both cases the resulting R^2 coefficient was found to be higher than 95%. Therefore, such choice allowed the reduction of the number of variable coefficients from a total of seven to six without an impact on the model accuracy.

$$H_t / H_{t,BEP} = a (Q_t / Q_{t,BEP})^2 + b (Q_t / Q_{t,BEP}) + c \quad (36)$$

$$P_t / P_{t,BEP} = d (Q_t / Q_{t,BEP})^2 + e (Q_t / Q_{t,BEP}) + f \quad (37)$$

For each machine, the above mentioned coefficients have been found by means of least squares linear regression from the tabulated points of characteristic curve. The non-dimensional Flow-Head

(Q-H) and Flow-Power (Q-P) characteristic curves of all the considered PATs are shown respectively in Figure 5-10 and Figure 5-11.

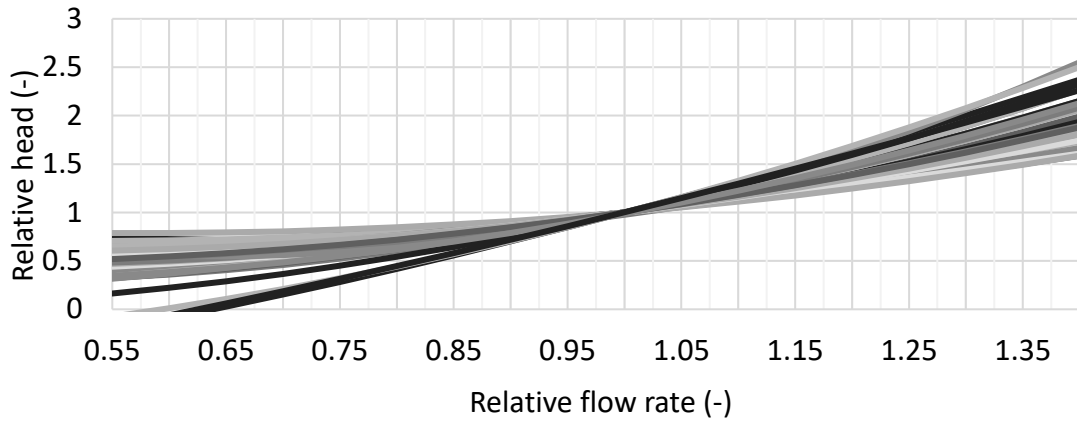


FIGURE 5-10. RELATIVE FLOW AND HEAD CHARACTERISTIC CURVES OF ALL THE 113 ANALYZED PATS

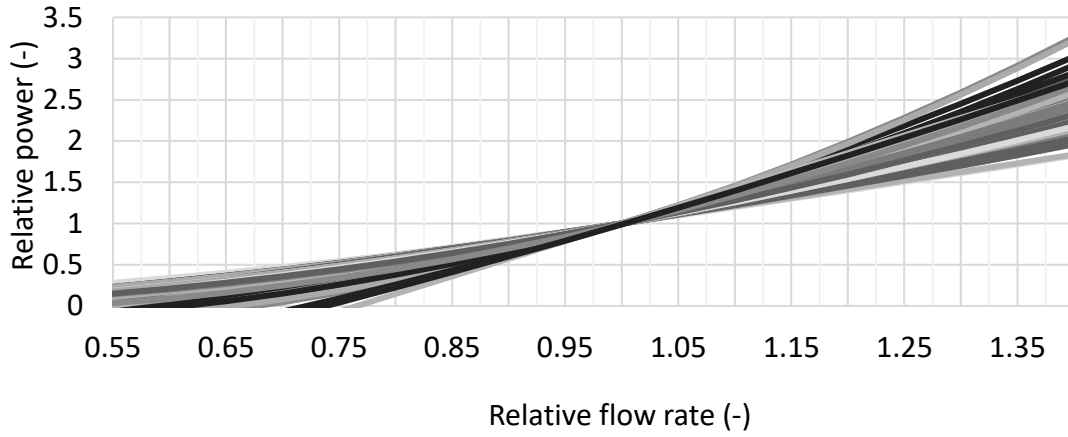


FIGURE 5-11. RELATIVE FLOW AND POWER CHARACTERISTIC CURVES OF ALL THE 113 ANALYZED PATS.

Looking at Figure 5-10 and Figure 5-11 it can be noticed that there is a significant dispersion of the sample curves away from the BEP, which corresponds to the point of coordinates (1,1). Hence, approximating the head and flow curves of a PAT with fixed polynomials may lead to errors becoming more and more significant in the regions away from the BEP. For each individual non-dimensional head and power curve of Figure 5-10 and Figure 5-11 the derivatives at the BEP have been calculated through the same coefficients previously introduced for Equation (36) and (37):

$$dH_{rel} / dQ_{rel} = 2a + b \quad (38)$$

$$dP_{rel} / dQ_{rel} = 2d + e \quad (39)$$

As a further step, it has been investigated whether the steepness of the non-dimensional head and power curves of the 113 analysed PATs increased on increasing the value of N_s as suggested by Chapallaz et al. (1992). In Figure 5-12 and Figure 5-13 the derivatives of the head and power relative curves at the BEP for each PAT calculated according to Equations (38)-(39) have been plotted against the N_s . The values of derivatives at the BEP of the non-dimensional PAT curves with fixed coefficients proposed by other authors and presented as Equations (28), (29) and (31)-(35) have also been included in Figure 5-12 and Figure 5-13 for comparison.

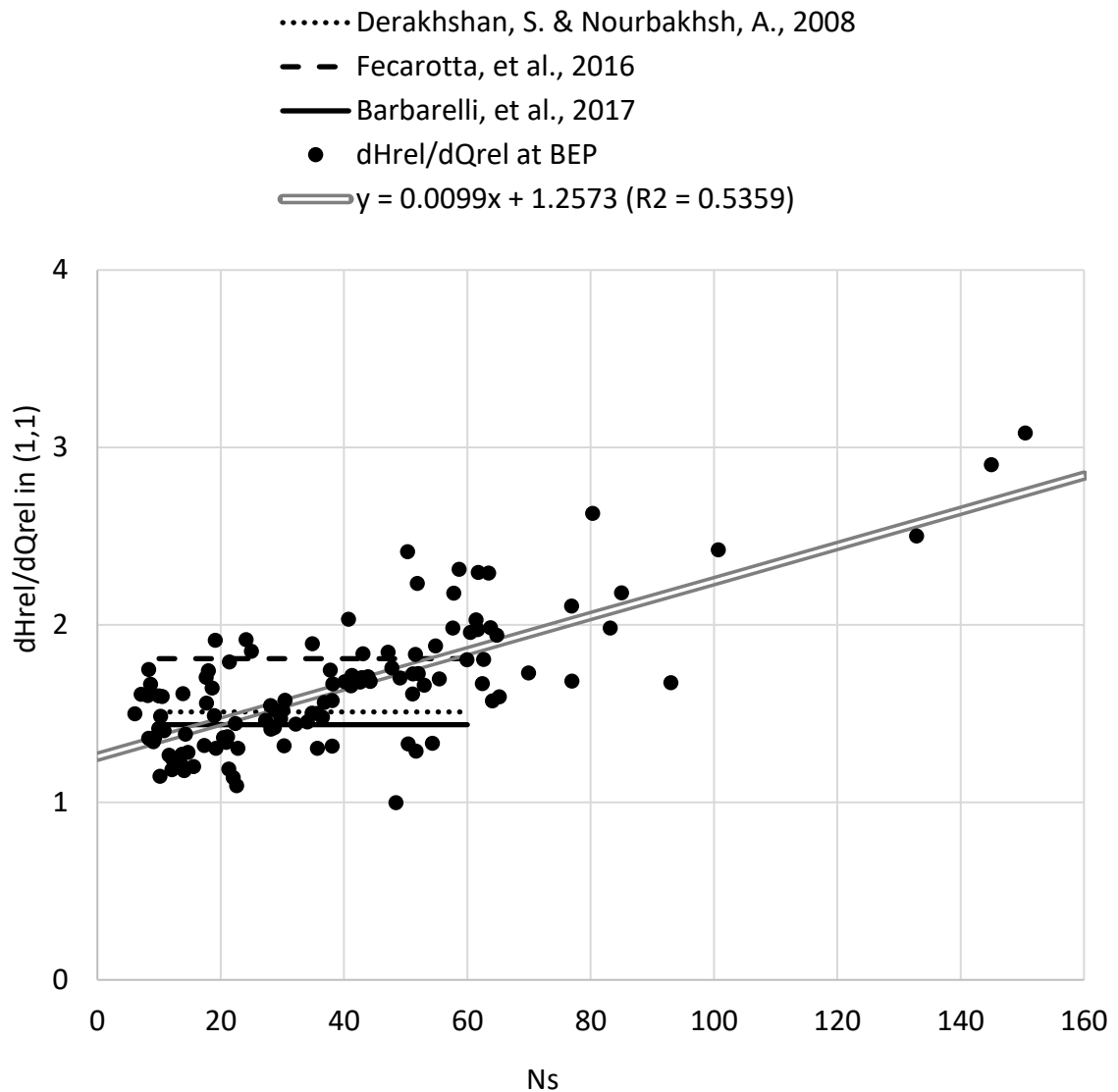


FIGURE 5-12. DERIVATIVES AT THE BEP (1,1) OF THE NON-DIMENSIONAL HEAD CURVES OF ALL 113 SAMPLE PATS.

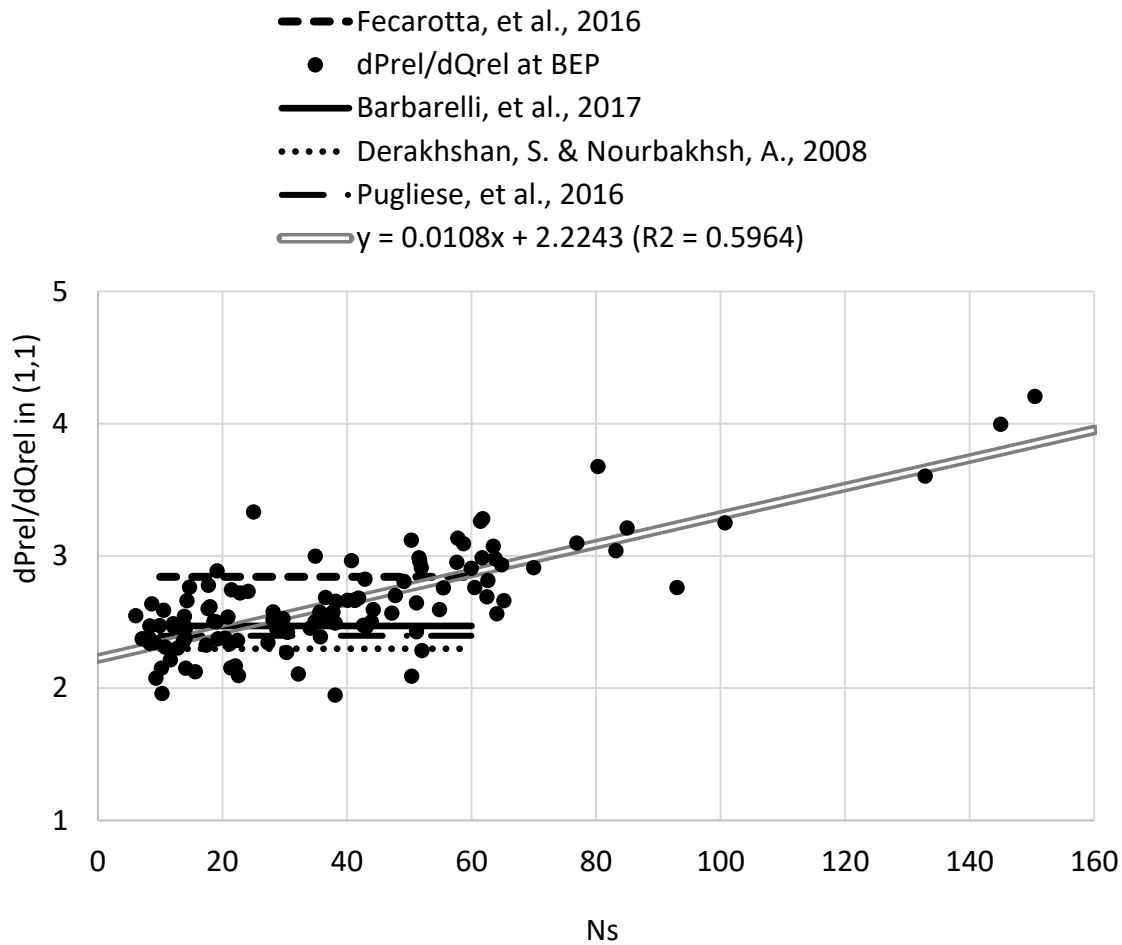


FIGURE 5-13. DERIVATIVES AT THE BEP (1,1) OF THE NON-DIMENSIONAL POWER CURVES OF ALL 113 SAMPLE PATS.

From the analysis of Figure 5-12 and Figure 5-13 it can be concluded that despite some dispersion of the samples a significant linear trend exists which correlates high values of N_s with a greater steepness of the non-dimensional head and power curves of a PAT. This behaviour in turn cannot be captured by the four fixed-coefficient polynomials existing in the literature and presented in the Introduction. Thus a new set of coefficients for Equation (36) and (37) based on the linear trend line displayed in Figure 5-12 and Figure 5-13 are proposed as shown in Table 5-4.

TABLE 5-4. TABULATED COEFFICIENTS FOR EQUATIONS (36) AND (37) DEPENDING ON N_s .

Non-dimensional head curve		Non-dimensional power curve	
a	1.160	d	1.248
b	$(0.0099 N_s + 1.2573) - 2a$	e	$(0.0108 N_s + 2.2243) - 2d$
c	$1 - a - b$	f	$1 - d - e$

The values for the coefficient “a” and “d” in Table 5-4 have been calculated as mean values of the 113 derived PAT curves.

5.3.3. MODEL ACCURACY VERIFICATION

The accuracy of the proposed model for the extrapolation of PAT full characteristic curves knowing the BEP has been assessed against the PAT database and compared to other methods described in the literature. Three widely accepted indicators showing the goodness of the fit were adopted as benchmarks: the Sum of Squares Error (SSE), the coefficient of determination R^2 and the Root Mean Square Error (RMSE). The SSE is the most basic indicator among the possible ones, R^2 is conveniently scaled between 0 and 1 and therefore easier to interpret while the RMSE is not scaled to any particular values and gives a more explicit measure of how much our predictions deviate, on average, from the actual values in the dataset. The results for the non-dimensional head curve extrapolation are displayed in Table 5-5, while those for the non-dimensional power curve are shown in Table 5-6.

TABLE 5-5. GOODNESS OF FIT OF THE PROPOSED CORRELATION COMPARED TO OTHER PROPOSED METHODS FOR THE APPROXIMATION OF THE RELATIVE HEAD CURVE OF A PAT.

Parameter	Proposed correlation	Derakhshan & Nourbakhsh, 2008	Fecarotta, et al., 2016	Barbarelli, et al., 2017
SSE	11.76	25.87	27.75	30.06
R^2	0.9701	0.9343	0.9295	0.9236
RMSE	0.0751	0.1113	0.1153	0.1200

TABLE 5-6. GOODNESS OF FIT OF THE PROPOSED CORRELATION COMPARED TO OTHER PROPOSED METHODS FOR THE APPROXIMATION OF THE RELATIVE POWER CURVE OF A PAT.

Parameter	Proposed correlation	Derakhshan & Nourbakhsh, 2008	Pugliese, et al., 2016	Fecarotta, et al., 2016	Barbarelli, et al., 2017
SSE	22.88	46.39	51.91	39.72	45.82
R^2	0.9776	0.9545	0.9491	0.9611	0.9551
RMSE	0.1057	0.1505	0.1592	0.1393	0.1496

The results from both Table 5-5 and Table 5-6 indicate that the proposed method, which considers the coefficients of the non-dimensional curves in relation to N_s , performs consistently better than all other methods available in the literature which adopt fixed coefficients. In particular, it enabled a reduction of the SSE indicator by over 60% and 55% with respect to the worst performing model regarding respectively the relative head and power curve prediction.

The RMSE indicator accordingly saw its value reduced with respect to the adoption of the four pre-existing models between a 24% and a 37%. Lastly, the overall R^2 coefficient was raised to over 97% from values in the range 92-93% regarding the relative head curve prediction and from a range of 94-95% as for the relative power curve prediction as displayed in Table 5-5 and Table 5-6.

The graph of fit of the proposed numerical model from which results from Table 5-5 and Table 5-6 were derived from the 113 analysed PATs can be seen in Figure 5-14 and Figure 5-15.

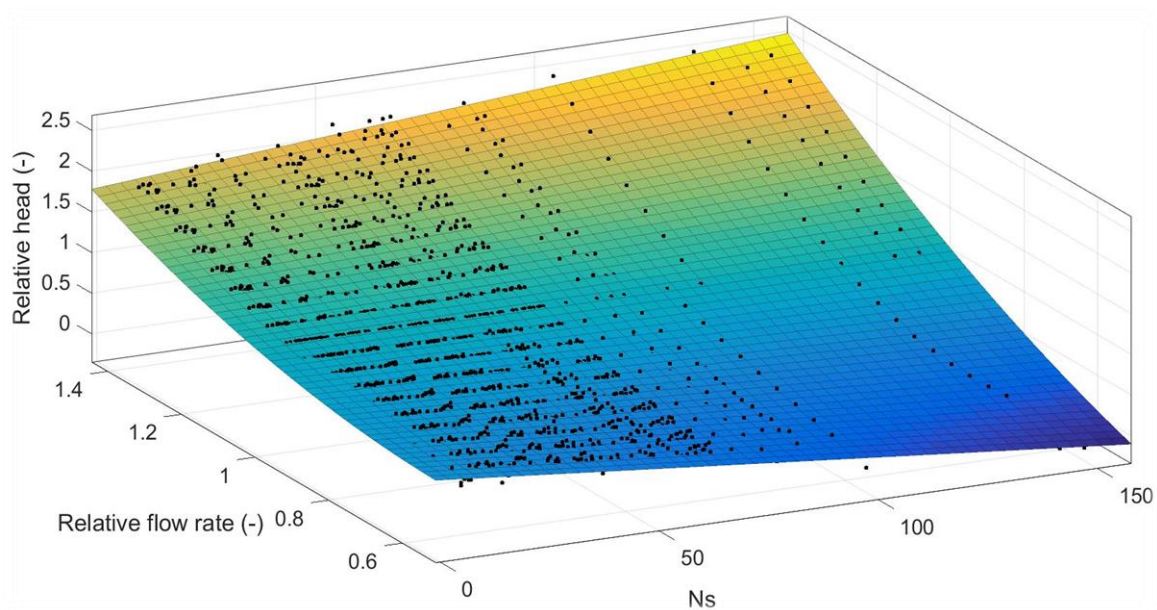


FIGURE 5-14. RELATIVE FLOW AND HEAD CHARACTERISTIC CURVES ACCORDING TO THE N_s OF ALL THE 113 ANALYZED PATs, DISPLAYED TOGETHER WITH THE PROPOSED INTERPOLATING 3D FUNCTION.

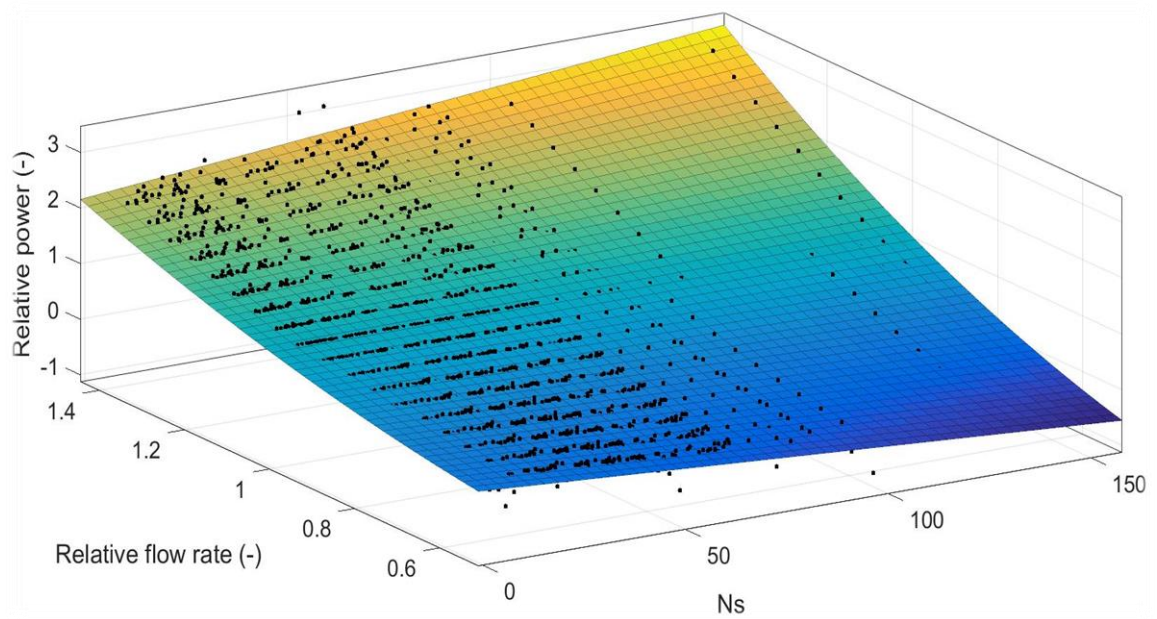


FIGURE 5-15. RELATIVE FLOW AND POWER CHARACTERISTIC CURVES ACCORDING TO THE NS OF ALL THE 113 ANALYZED PATS, DISPLAYED TOGETHER WITH THE PROPOSED INTERPOLATING 3D FUNCTION.

5.3.4. COMPARISON OF PAT EFFICIENCY AT PARTIAL LOAD

Using the same dataset of performance curves from 113 machines the mean mechanical efficiency of PATs under variable flow rates has been assessed and compared to that of conventional hydro turbines. For each PAT the relative efficiency was calculated as the ratio between the efficiency at a certain relative flow rate and the value of peak mechanical efficiency of the turbine defined as:

$$\eta = P / (9.81 Q H) \quad (40)$$

where P is the output power in kW, Q the flow rate in m³/s and H the head in m across the PAT.

Besides, the BEP of a PAT was fixed to the 75% of the maximum admissible flow rate through it which is consistent with the behaviour of the gathered PAT curves. The relative PAT efficiency with varying flow rate compared to that of other classes of hydro turbines is shown in Figure 5-16.

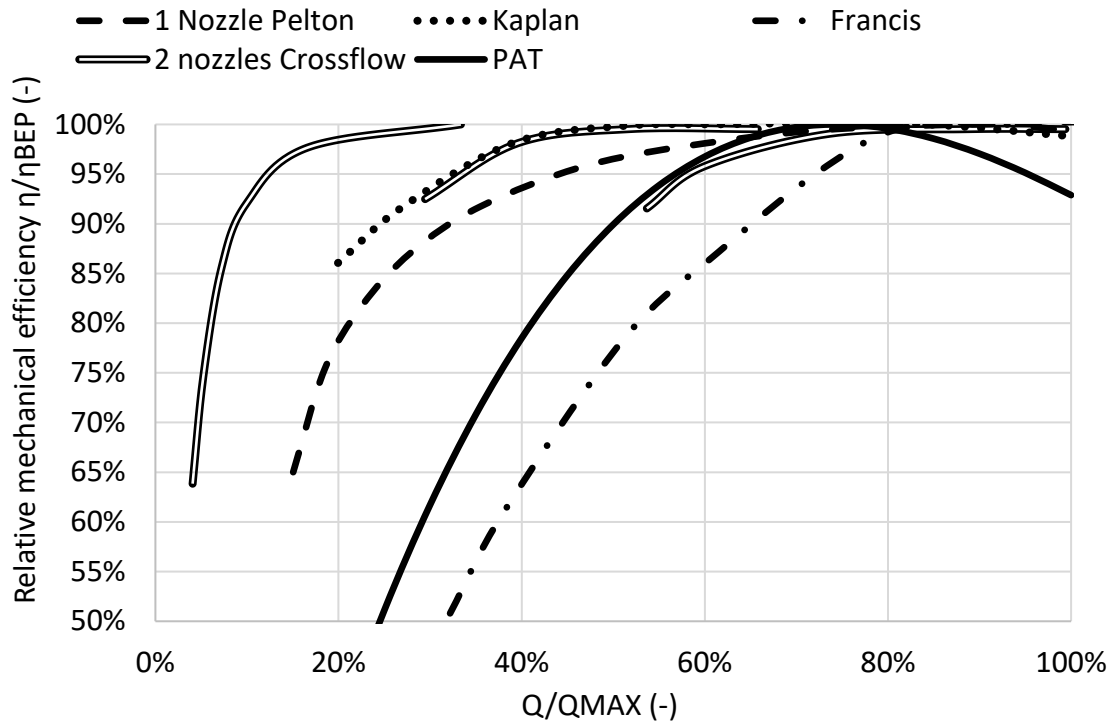


FIGURE 5-16. RELATIVE MECHANICAL EFFICIENCY VS. RELATIVE FLOW RATE OF PATS AND OTHER TYPES OF CONVENTIONAL HYDRO TURBINES (ADAPTED FROM: EUROPEAN DIRECTORATE FOR TRANSPORT AND ENERGY - FP6 PROJECT SHAPES, 2010).

The results of Figure 5-16 indicate that the relative mechanical efficiency of a PAT is comparable to that of a Francis turbine while at the same time Pelton, Crossflow or Kaplan turbines indicatively present a more flat efficiency curve over the whole operating range. However, such a comparison is not entirely fair because as opposed to a conventional turbine a PAT does not have an in-built regulation device but rather features a rigid volute and impeller geometry. This means that when a higher flow rate than the BEP is available in the system a portion of it must be bypassed, while if less flow rate than the BEP is available a portion of the fluid pressure must be dissipated by an in-series control valve. Ultimately this phenomenon reduces the energy output from a PAT system. This can be evaluated by introducing the concept of system efficiency as the ratio of the power output from a turbine to the available system power:

$$\text{System efficiency (-)} = P / (9.81 Q_{av} H_{av}) \quad (41)$$

where P is the output power from the turbine in kW, Q_{av} the flow rate in m^3/s and H the head in m available in the system. Figure 5-17 shows the indicative variation of system efficiency of different turbine technologies under different flow rates. The head curve of the system was considered

constant, while representative values of turbine efficiency have been selected as 90% for Kaplan and Pelton units, 82% for Crossflows, 90% for Francis machines and 75% for PATs.

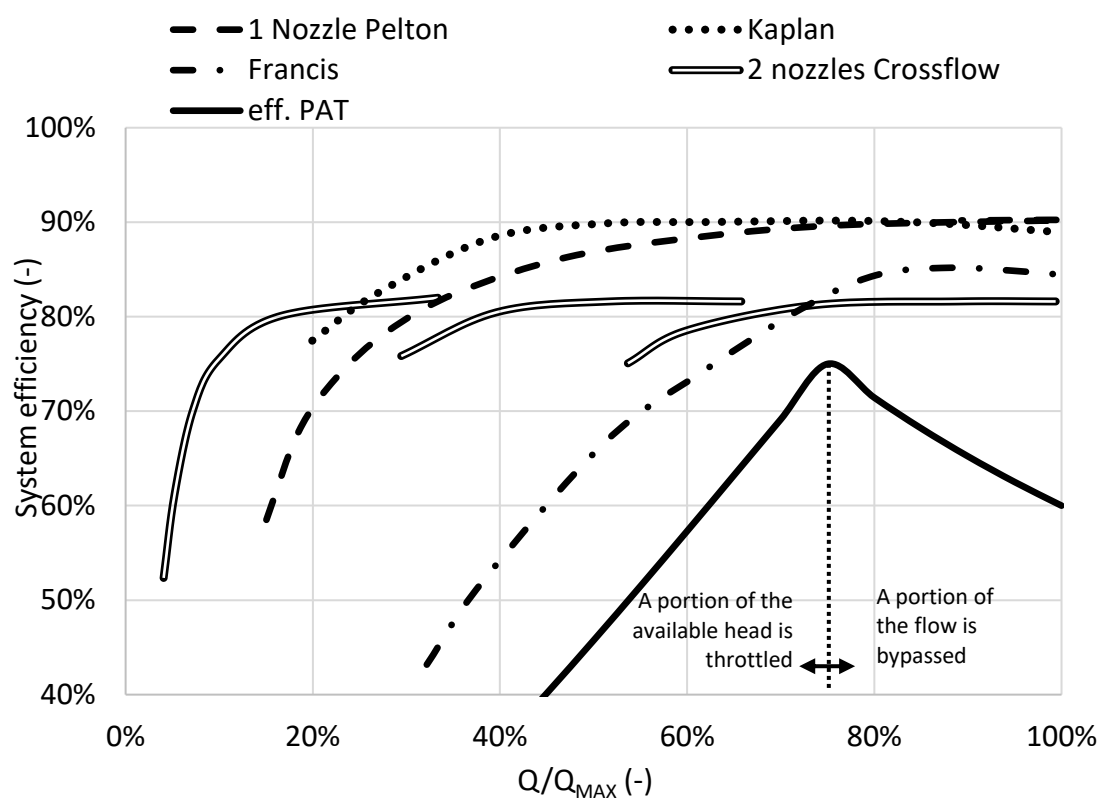


FIGURE 5-17. GLOBAL SYSTEM EFFICIENCY VS. RELATIVE FLOW RATE OF PATS AND OTHER TYPES OF CONVENTIONAL HYDRO TURBINES (ADAPTED FROM: EUROPEAN DIRECTORATE FOR TRANSPORT AND ENERGY - FP6 PROJECT SHAPES, 2010).

From Figure 5-17 it can be concluded that in terms of system efficiency a PAT has a clear peak at the BEP around which its effectiveness declines steeply due to its inability to adapt to different flow and head conditions.

5.3.5. CONCLUSIONS

The extrapolation of full head and power characteristic curves of pumps as turbines is a fundamental step in order to correctly select the most effective machine working under variable load conditions for a determined site. Four existing methods in the literature based on fixed-coefficients and second and third degree polynomials have been evaluated against a large dataset of 113 PAT characteristic curves. Since the steepness of the head and power non-dimensional curves gathered in the database has been demonstrated to vary according to the specific speed of each machine, a new method for the full characteristic curves extrapolation was proposed in which

the coefficients of two second-degree polynomials are variable as functions of N_s . This proposed method was found to be statistically more accurate in reproducing the 113 PAT curves of the dataset with respect to the other four methods available in the literature. In particular, the accuracy regarding the prediction of relative head curves has been improved up to 60%, 37% and 5% when considering the SSE, RMSE and R^2 indicators respectively. Due to the limited number of PAT characteristic curves in the database with N_s higher than 100 the author recommends caution with adopting the proposed method for head and power relative curves extrapolation in the mentioned range.

Secondly, based on the same database the average mechanical efficiency of PATs with varying flow rate has been shown to compare favourably to the behaviour of other classes of conventional hydro turbines (Francis). However, when considering the global system efficiency the average performance of PATs are inferior to that of other turbine classes with a clearly identifiable peak preceded and followed by a steep decline in efficiency. This suggests that the process of selecting a hydraulic pump to work as a turbine for a given site is a highly critical operation, since even a small assessment mistake could have significant repercussions on the system effectiveness.

As a scope for future research it is suggested to experimentally test PATs having N_s in the range 60-150 and to integrate their derived characteristic curves into the database utilized for the present calculations. Indeed, the vast majority of PATs whose performance curves are known to the authors present N_s lower than 60.

5.4. PREDICTION OF PAT COST

The content of the present chapter has been published as:

Novara, D.; Carravetta, A.; McNabola, A. & Ramos, H. M. "Cost Model for Pumps as Turbines in Run-of-River and In-Pipe Microhydropower Applications". J. Water Resour. Plann. Manage., 2019, 145(5): 04019012. DOI: 10.1061/(ASCE)WR.1943-5452.0001063

García, I.F.; Novara, D. & McNabola, A., 2019. "A Model for Selecting the Most Cost-Effective Pressure Control Device for More Sustainable Water Supply Networks". Water, 2019, 11(6), p.1297.

The full research papers are hereby attached as Annexes 3.3 and 3.4.

5.4.1. INTRODUCTION

Among the various applications of PATs, these have proved to be very effective if used for micro hydropower plants and in-pipe energy recovery with associated power output lower than 100 kW (Williams, 1996; Lahimer & al., 2012; Carravetta, et al., 2012; McNabola, et al., 2014a; Pérez-Sánchez, et al., 2017). The cost of such small-scale hydro schemes is heavily influenced by the value of the generating set, consisting of the turbogenerator group. Such equipment commonly represents between 30 and 40% of the total required investment cost (Chapallaz, et al., 1992; Ogayar & Vidal, 2009). Furthermore, for plants having capacity in the range of a few kilowatts this figure could be up to 70% (Gallagher, et al., 2015b). This fact makes the PAT technology a natural choice for powering small-size hydro schemes, since cost savings on the purchase of the generating set could make a large impact on the final project viability.

Indeed, in the literature it is universally acknowledged that the mass manufacturing process allows for the purchase price of pumps to be considerably less expensive than conventional hydro turbines (e.g. Pelton, Francis, Crossflow/Banki and Kaplan). However, little detailed information is available in the literature about the purchase price of a PAT and what is there is often out-of-date and contradictory as displayed in Chapter 3.6.6. Some authors in 2009 calculated the cost of a PAT to vary between 200 and 400 €/kW for a nominal power lower than 40 kW (Ramos & Borga, 1999) as displayed in Figure 3-42 at Page 83. Other authors provided an estimate of PAT turbogenerator unit cost as the sum of two components: 230 €/kW to be multiplied by the nominal power accounting for the turbine alone, plus 115 €/kW multiplied by the maximum PAT power accounting for the cost of the generator (Carravetta, et al., 2013b) as per Equations (11) and (12) at Page 84. Conversely, others proposed a fixed unit cost of 2,000 €/kW for assembled turbine + generator PAT units as per Equation (13) at Page 84 (Carravetta, et al., 2014).

The cost model proposed by Ramos & Borga (1999) described in Figure 3-42 shows a realistic decrease of unit cost with increasing PAT size due to economies of scale. However, the validity of this cost model is hampered by the lack of information on its data sources, by the length of time passed since its publication and by its applicability which is limited to a relatively narrow range of PAT sizes. Conversely, the other two cited cost models assume a constant unit cost of equipment. Using these cost models outside their range of applicability would be an oversimplification and subject to significant errors. Additionally, a common limitation of all the presented cost models is their independence from values of PAT nominal pressure head and power. Previous investigations

aimed at predicting the purchase price of conventional hydraulic turbines or pumps have found these factors to be significant for the estimation of the equipment cost (Couper, et al., 2009; Ogayar & Vidal, 2009). Moreover, the aforementioned studies by Ramos et al. (2009), Carravetta et al. (2013) and De Marchis et al. (2014) did not have as their primary objective, the development of a PAT cost model with wide applicability. Instead, they examined the technical performance of PATs in specific scenarios.

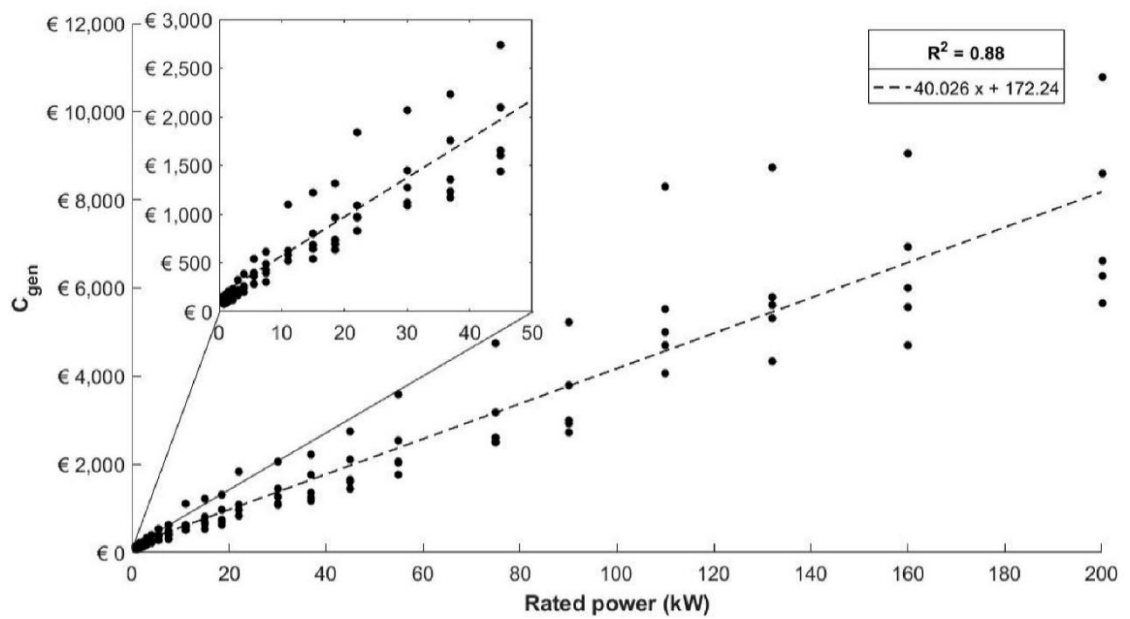
Furthermore, in the literature there is no agreement on the extent of the gap between the unit cost of PATs and that of conventional hydro turbines. Some authors suggested that PATs are five times less expensive than equivalent conventional hydro turbines (Alatorre-Frenk, 1994), while others pointed out that for $P > 40$ kW the price difference between them is lowered, to be below 35% as seen in Figure 3-18 at Page 54 (Ramos & Borga, 1999).

The lack of reliable information on the cost of micro hydropower plants has been deemed as one of the obstacles to the diffusion of such plants (Carravetta, et al., 2014). Indeed, in order to assess the financial viability of a hydropower project adopting the PAT technology it is fundamental to have access to realistic estimates of the cost of the generating set. The lack of this information is a clear barrier to the exploitation of this low-cost technology in natural or piped water resource systems. Therefore, the aim of the present section is to provide a comprehensive estimate of PAT + generator unitary purchase prices and to compare it with prices of conventional hydropower generating sets as well as previously published PAT cost figures.

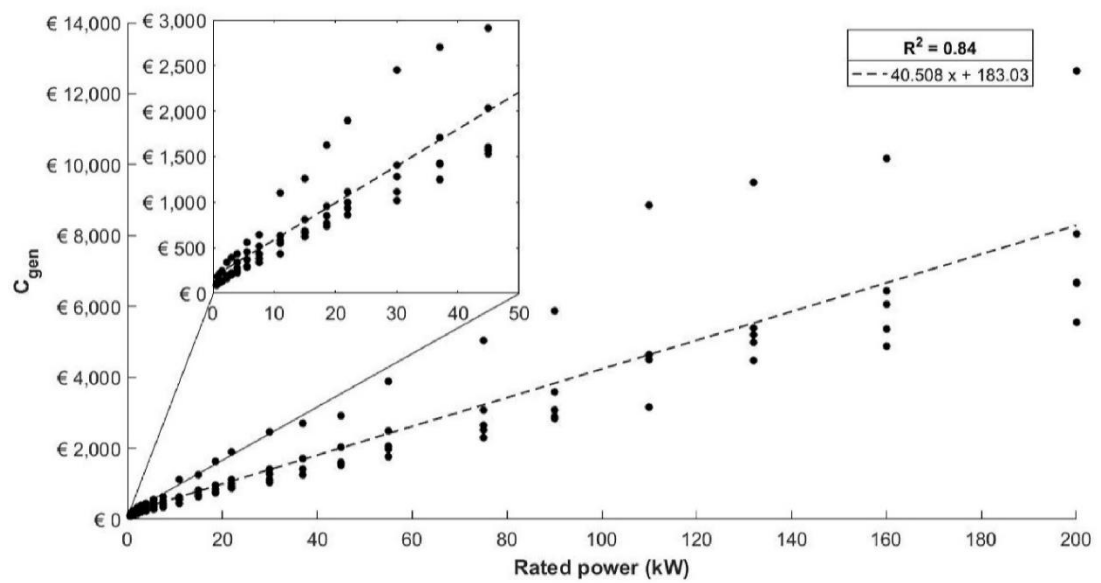
5.4.2. COST DETERMINATION OF ASYNCHRONOUS GENERATORS

As a first step, the price list (excluding VAT) of 286 asynchronous induction motors belonging to the IE3 efficiency class and rated for nominal power in the range 1 – 200 kW has been compiled as shown in Annex 4.4.1. The analysed machines belonged to four different manufacturers and their listings were relative to the years 2009 - 2018 (Omec Motors, 2015; Speroni S.p.a., 2009; Koupas Motors, 2016; Vallorani, 2016; MA.PI S.a.s., 2018). All generators were selected to have 1, 2 or 3 pairs of magnetic poles (pp) which when operated at a 50 Hz corresponded to a nominal shaft speed of 3,000, 1,500 and 1,000 RPM respectively. Generators rotating at lower speeds (e.g. 8-poles units, rated for 750 RPM at 50 Hz) have not been considered since typical pump performance charts only include curves relative to motors having $pp = 1, 2$ and 3 . The cost of the generators (C_{gen}) has been found to be approximately proportional to the nominal power rating of the machine as shown in Figure 5-18.

a) pp = 1



b) pp = 2



c) pp = 3

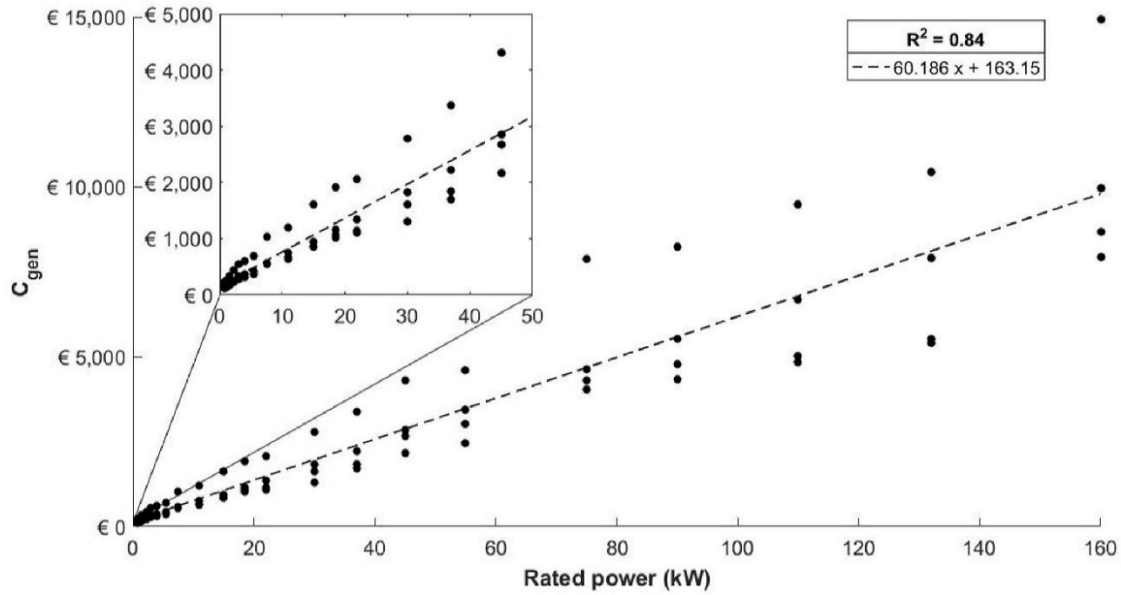


FIGURE 5-18. COST OF 286 ASYNCHRONOUS GENERATORS AS FUNCTION OF THEIR NOMINAL POWER FOR MACHINES HAVING A DIFFERENT NUMBER OF MAGNETIC POLE PAIRS (PP): A) PP = 1; B) PP= 2; C) PP = 3.

In Figure 5-18 it can be noticed that for a given power rating a motor from one manufacturer can be more than two times as expensive as an equivalent one from another manufacturer which is in line with the notion that the market for electric motors is a very competitive one and large discounts on the price listings are possible (De Almeida, et al., 2008). For instance, in Figure 5-18 (c) we can notice that a 160 kW generator with pp = 3 can cost between 7,500 and 15,000 € according to the manufacturer. The set of linear equations obtained from fitting linearly the cost versus power of the induction generators having one, two or three pairs of magnetic poles (pp = 1, 2 & 3) as seen in Figure 5-18 is displayed in Equations (42)-(44):

$$\text{for pp} = 1: \quad C_{gen} = 40.03 \cdot P(\text{kW}) + 172.24 \quad (42)$$

$$\text{for pp} = 2: \quad C_{gen} = 40.51 \cdot P(\text{kW}) + 183.03 \quad (43)$$

$$\text{for pp} = 3: \quad C_{gen} = 60.19 \cdot P(\text{kW}) + 163.15 \quad (44)$$

The results show that on increasing the number of magnetic poles the purchase price of the motor/generators rises as well, even though the difference between $pp = 1$ and $pp = 2$ is not marked.

5.4.3. COST DETERMINATION OF PATs

The estimation of cost per kW of centrifugal PATs coupled with induction generators having 2, 4 or 6 magnetic poles ($pp = 1, 2 \text{ \& } 3$) has been carried out based on commercially available data (Grundfos, 2016; Abacus Resale, 2018). In particular, the cost (excluding VAT) of 343 radial end-suction and vertical multistage close-coupled pumps connected to IE3 class generators has been compiled and related to their predicted power output when operated as turbines at their Best Efficiency Point (BEP). The features of such pumps are summarized in Table 5-7 and the complete set of information can be found in Appendix 4.4.2.

TABLE 5-7. FEATURES OF THE SELECTED PUMPS/PATs.

Pump type	Radial, end-suction	Vertical multistage
Sample size	301	42
Volute material	Cast iron	
Shaft material	Stainless steel	
Motor/generator class	Induction type, IE3 class efficiency, having 1, 2 or 3 pairs of poles (pp)	
Head range as pumps (m)	2 – 148	5 - 200
Capacity range as pumps (l/s)	2 – 240	1 - 43
Head range as PATs (m)	3 – 352	10 - 311
Capacity range as PATs (l/s)	3 – 320	1 - 62

For every unit considered, the main parameters obtained from the pump technical specification were the pump mode peak efficiency (η_{BEP}), shaft speed (N) and the coordinates of the pump mode BEP in terms of nominal flow rate and head ($Q_{p,BEP}$, $H_{p,BEP}$).

Given the difficulty in obtaining information about PAT performance, the characteristic parameters of pumps operated in reverse have to be estimated by the designers or final users. Several methods exist to predict the Best Efficiency Point (BEP) of a PAT as explained in Chapters 3.6.4. The comparative analysis presented in Chapter 5.1 showed that among the existing numerical methods the one proposed by Yang, et al. (2012) proved to be the most consistent across a large pump/PAT

database, and therefore such method was hereby applied to estimate the values of $H_{t,BEP}$ and $Q_{t,BEP}$ of the chosen set of pumps when running in turbine mode. Even though such a 1D mathematical method may occasionally result in under/overestimation of the BEP location for a determined machine, it was assumed that their application to a large number of machines, 343 in this case, should dampen such scattered errors. Finally, the maximum efficiency of pumps operating in turbine mode was considered equal to that in pump mode, which many authors have shown to be a good approximation (Garay, 1990; Derakhshan & Nourbakhsh, 2008).

The BEP values in terms of predicted nominal flow rate and head ($Q_{t,BEP}$, $H_{t,BEP}$) of the investigated PATs are displayed in Figure 5-19. This shows that the PATs selected for building the cost model may generate from as little as 300 W up to more than 300 kW. Additionally, when compared to the PAT application range chart proposed by Chapallaz et al. (1992), the selected set of PATs fit well within the proposed application boundaries.

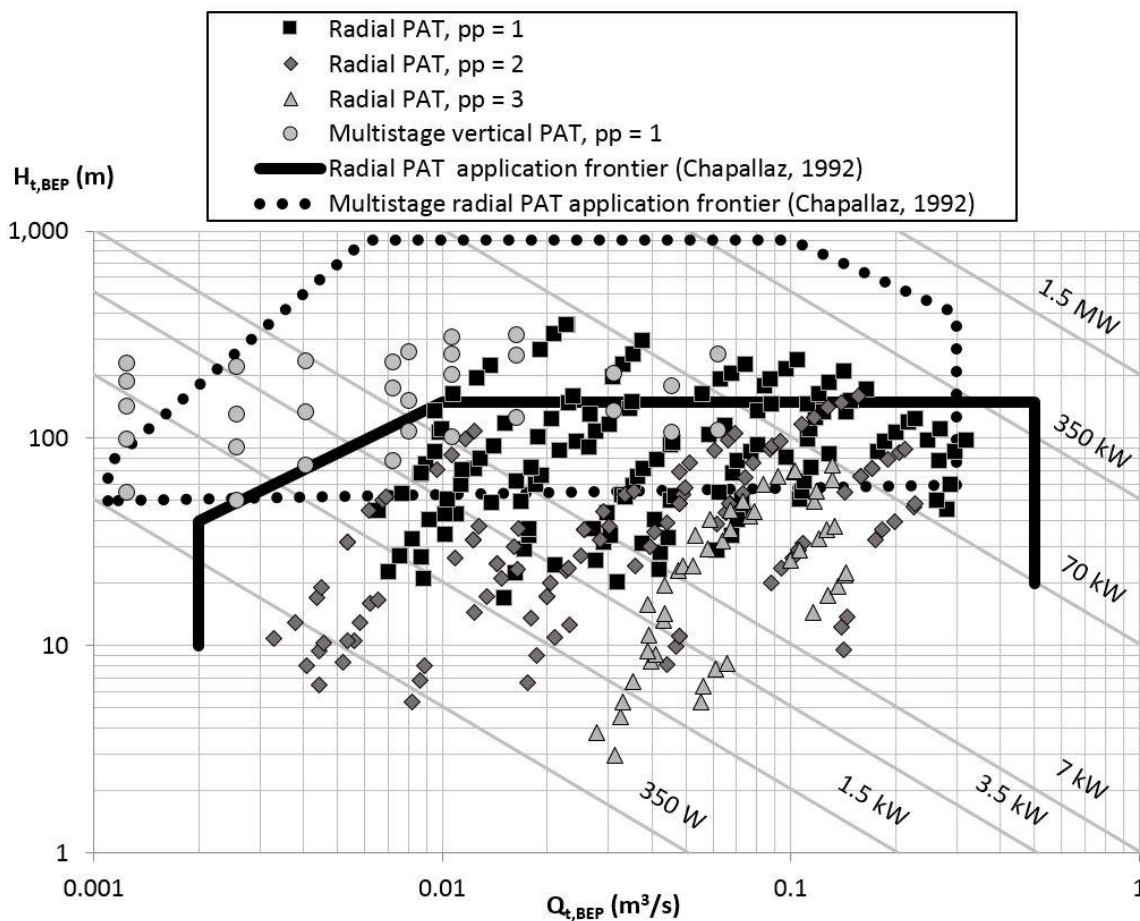


FIGURE 5-19. LOCATION OF THE ESTIMATED PAT BEST EFFICIENCY POINTS (BEP) FOR THE SELECTED 343 MACHINES OVER THE H-Q SPACE.

As a general rule, a pump is able to process a larger flow rate and hydraulic head when operated in reverse as a turbine compared to its performance as a pump. Therefore, when a pump is bought to be used as a PAT its attached motor unit will need to be replaced by another one rated for a higher nominal power. Besides, when using an asynchronous induction motor as the generator it has been suggested to de-rate it by 20% of the nominal capacity as a motor in order to avoid overheating and reliability issues (Williams, 1996). In order to account for this, Equations (42)-(44) have been used to calculate the price of the motor originally sold with pumps ($C_{gen,p}$) and the price of the larger unit able to run as generator at 80% of its nominal capacity ($C_{gen,PAT}$). The resulting cost of a PAT+generator has been therefore calculated as:

$$C_{PAT+gen}(\text{€}) = C_p(\text{€}) - C_{gen,p}(\text{€}) + C_{gen,PAT}(\text{€}) \quad (45)$$

where C_p stands for the selling price of a pump with its original motor. The nominal power of PATs was subsequently calculated per Equation (1) at Page 44. The PAT + generator unit cost ($CS_{PAT+gen}$) was then computed as:

$$CS_{PAT+gen}(\text{€/kW}) = C_{PAT+gen}(\text{€})/P(\text{kW}) \quad (46)$$

The resulting figures of PAT specific cost ($CS_{PAT+gen}$) have been compared with cost per rated power of conventional hydro turbines - Francis, Pelton and Kaplan - as gathered by the authors in 2017 and by Ogayar and Vidal (2009). The cost figures proposed by Ogayar and Vidal (2009) have been updated to account for the inflation which has occurred between 2008 (year of original data collection) and the end of 2016. Also, the specific cost of PAT generating sets has been compared with other previously published costs. The resulting chart is displayed in Figure 5-20, from where it can be noticed how for a given installed power according to the choice of PAT type (radial or multistage) and shaft speed ($pp = 1, 2$ or 3) the cost of a turbogenerator may even double. For instance, at 3 kW the unit cost $CS_{PAT+gen}$ varied from 500 to 1,000 €/kW and at 20 kW it was in the range 200 – 500 €/kW.

The PAT and generator assembly unit cost shown in Figure 5-20 decreased with the increase of nominal power output, showing realistic economies of scale. At the same time, the previously published cost estimates (Ramos & Borga, 1999; Carravetta, et al., 2013b; De Marchis, et al., 2014) can be considered accurate only within specific intervals of nominal power. On the other hand, generating sets relying on PAT technology proved to be less expensive per unit of installed power than conventional hydraulics turbines by a factor of 5 for $P > 20$ kW and up to 15 for lower powers.

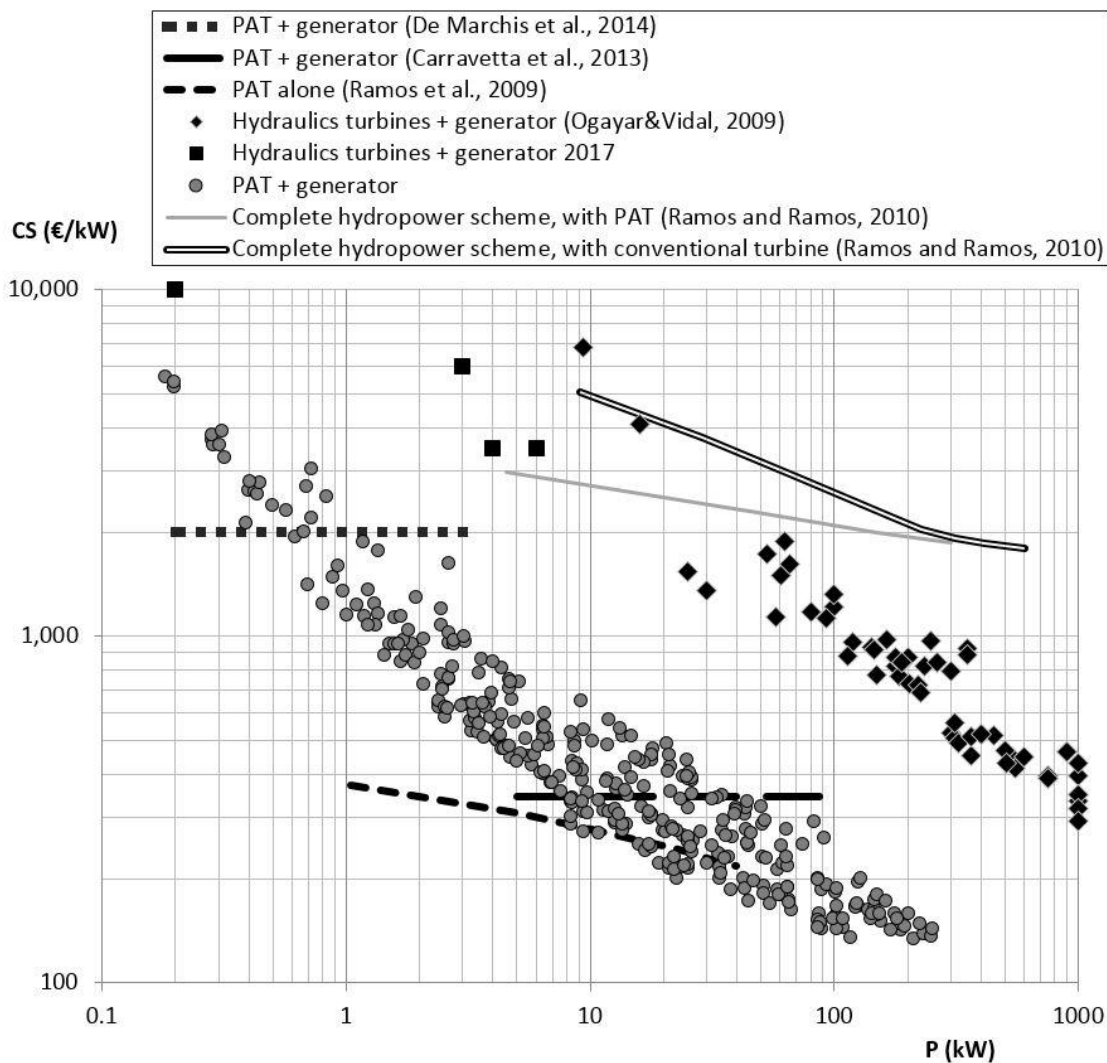


FIGURE 5-20. UNIT COST PER RATED POWER $CS_{PAT+GEN}$ OF THE 343 EVALUATED PATS COUPLED TO IE3 ASYNCHRONOUS GENERATORS, AS COMPARED TO THE PREVIOUSLY AVAILABLE PAT COST FIGURES AND UNIT COST OF CONVENTIONAL HYDRO TURBINE + GENERATOR UNITS.

In the literature the existence of a linear relationship between the parameter $Q_v(H)$ and the price of centrifugal pumps has been validated (Couper, et al., 2009). Plotting graphically such parameters for the analysed set of PATs, a significant linear correlation was also found, relating the $Q_{t,BEP}$ (m^3/s) and $H_{t,BEP}$ (m) of a machine to its purchase price.

For every considered PAT (single stage radial or vertical multistage, having generators with $pp = 1, 2$ or 3) a best linear fit for the cost function was derived as shown in Figure 5-21.

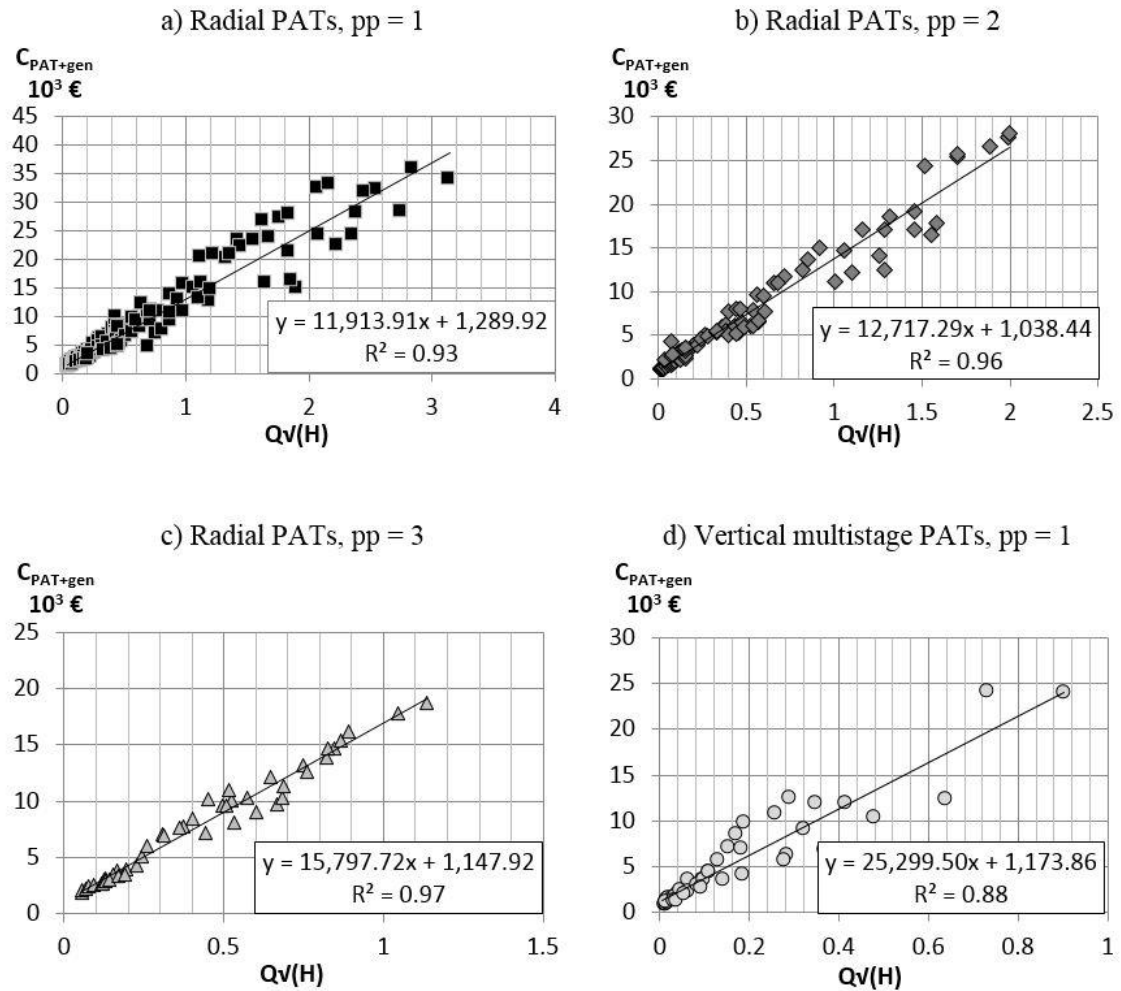


FIGURE 5-21. LINEAR FUNCTIONS PREDICTING THE PURCHASE PRICE OF A) RADIAL PATS, PP=1; B) RADIAL PATS, PP=2; C) RADIAL PATS, PP=3; AND D) VERTICAL MULTISTAGE PATS, PP=1.

The results of the linear regressions shown in Figure 5-21 proved to be a reasonable estimate in determining the purchase price of radial PATs with 2 and 3 pole pairs ($R^2 = 96\%$ and 97% respectively). The slightly reduced accuracy of the linear fit of radial PATs with pp = 1 generators was mostly due to the large number of data points available for this category in comparison to other types of machines.

Instead, the value of R^2 relative to vertical multistage PATs has been calculated as 88% . The set of linear equations arising from Figure 5-21 are displayed as Equations (47)-(50).

$$\text{for } pp = 1: \quad C_{PAT+gen}(\text{€}) = 11,913.91 \cdot Q\left(\frac{m^3}{s}\right) \sqrt{H(m)} + 1,289.92 \quad (47)$$

$$\text{for } pp = 2: \quad C_{PAT+gen}(\text{€}) = 12,717.29 \cdot Q\left(\frac{m^3}{s}\right) \sqrt{H(m)} + 1,038.44 \quad (48)$$

$$\text{for } pp = 3: \quad C_{PAT+gen}(\text{€}) = 15,797.72 \cdot Q\left(\frac{m^3}{s}\right) \sqrt{H(m)} + 1,147.92 \quad (49)$$

$$\text{for Vertical multistage PAT, } pp = 1: \quad C_{PAT+gen}(\text{€}) = 25,299.50 \cdot Q\left(\frac{m^3}{s}\right) \sqrt{H(m)} + 1,173.86 \quad (50)$$

As could be expected, PATs linked to a generator with a higher number of magnetic poles, i.e. rotating at a lower shaft speed, also had a higher purchase price.

Comparing the real cost of the analysed PAT + generator units to the cost estimate obtained by applying the set of Equations (47)-(50), the distribution of errors displayed in Figure 5-22 can be obtained. The mean error corresponded to -5.1%, meaning that the application of Equations (47)-(50) may lead to an average overestimation of the PAT + generator purchase price of about 5%.

However, in 81% of cases the relative error between actual and estimated prices was found to be within the $\pm 25\%$ interval which is generally acceptable at a pre-design phase.

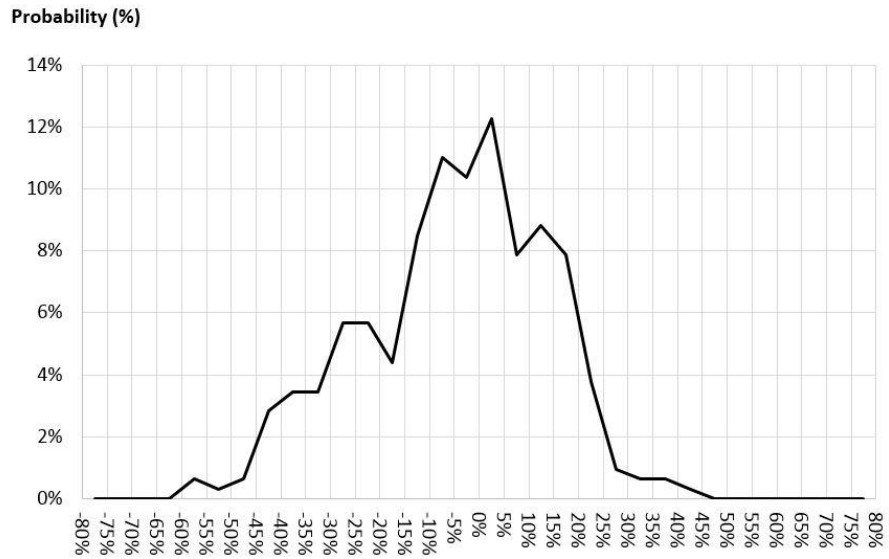


FIGURE 5-22. DISTRIBUTION OF THE PERCENTAGE ERRORS RESULTING FROM APPLYING EQUATIONS 9-12 TO THE 343 ANALYSED PATS.

The developed PAT cost prediction model has been compared to the selling price of nine radial, end suction pumps to be used as turbines as gathered in Ireland in the first trimester of 2018. The investigated units belonged to different manufacturers (KSB, Ebara, Johnson Pumps, DAB), and as shown in Figure 5-23 their cost per unit power agrees well with the proposed PAT purchase price prediction model.

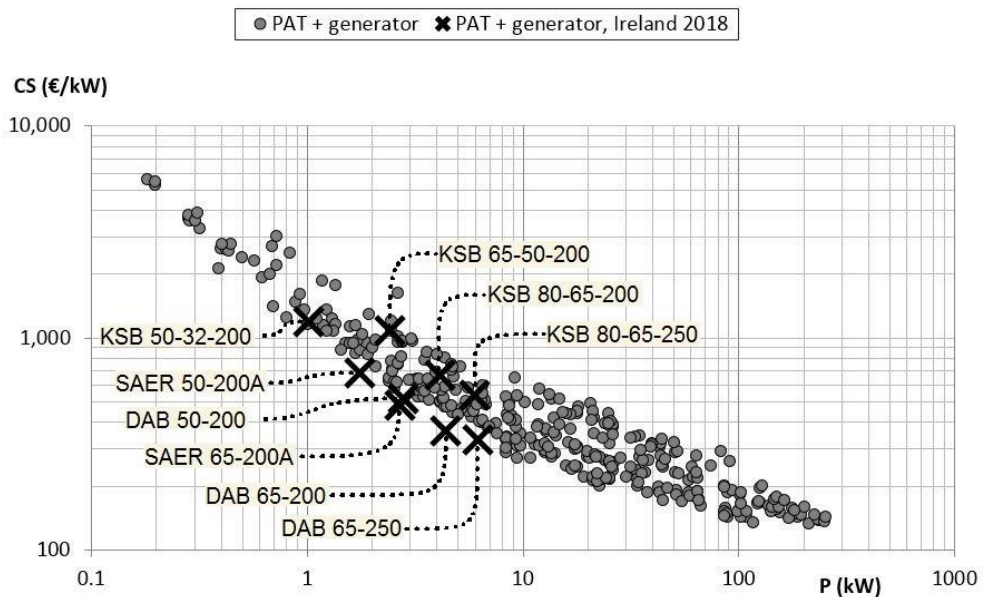


FIGURE 5-23. COMPARISON OF THE UNIT COST OF NINE PATS FROM DIFFERENT MANUFACTURERS AS GATHERED IN IRELAND, 2018, WITH THE PROPOSED COST MODEL.

5.4.4. REVIEW OF OTHER PROJECT COSTS IN A PAT INSTALLATION

The expenditure necessary for the purchase of the turbine and generator is only a portion of the total project cost of any hydropower scheme. Some of the additional required cost items such as civil works, engineering design, site management and grid connection fees can reasonably be considered as highly site-specific and independent of the chosen turbine technology. By contrast, the implementation of a control system with different degrees of automation is a reflection of the operator requirements and the choice of turbine type. The cost of such control systems can have a strong impact on the total cost figure, as reviewed by Ogayar & Vidal (2009) who established that on average for hydro schemes having capacity lower than 2 MW and featuring conventional turbine types (Francis, Pelton, Kaplan) the control system accounted for 22% of the total investment, representing nearly 2/3 of the turbogenerator cost. Since a turbogenerator unit featuring a PAT would be less expensive than an equivalent one based on a conventional turbine technology as seen on Figure 5-20, it is reasonable to assume that the share of the control system cost over the total investment may grow to above 22% if a PAT is used. Additionally, given its rigid geometry and lack of in-built regulation devices, in some contexts the adoption of a PAT would require the installation of a relatively sophisticated and expensive control system in order to achieve a degree of system flexibility comparable to that of a conventional turbine. An example of this is the application of PATs as energy recovery devices within drinking water networks designed to exploit local excesses of water pressure otherwise dissipated by means of Pressure Reducing Valves (PRVs). Due to operational reasons the network operators may require a specific backpressure level to be maintained downstream of the PAT independently from the instantaneous flow rate and upstream pressure, which the machine alone would not be capable of accomplishing. In order to achieve this, two alternative designs are possible: Hydraulic Regulation (HR) and Electric Regulation (ER) as described in Chapter 3.6.7 (Carravetta et al., 2012, 2013). Both designs require a closed-loop control device able to perform a real time reading of the connected sensors (typically, flow and pressure meters) and adjust accordingly the opening of the bypass in case of HR and the turbine shaft speed in case of ER (Carravetta et al., 2012, 2013).

Very little information can be found in the literature regarding existing PAT schemes, especially concerning their economics and construction details. An example of PAT installations within water networks featuring an HR control strategy are the twin power plants of Cartuja and Conejeras near Granada, Spain. The two installed PATs were designed for a flow of 278 l/s and a head of 39 m (Cartuja) and 36 m (Conejeras) and were coupled to a 90 kW generator each (Carravetta, et al.,

2017b). In both cases the reported cost of the control system was 1.2 times the purchase cost of the PAT and generator, which is higher than the ratio suggested by Ogayar and Vidal referred to hydro schemes with conventional turbines. It is reasonable to assume that this phenomenon is due partially to the lower acquisition cost of the PATs and partially to the relative sophistication of the control system needed to ensure HR. However, the in-pipe installation of PATs within water networks is a very specific application field of the technology which requires special precautions in order to ensure the essential reliability of water supply. Instead, in the vast majority of cases where flow and backpressure control is not required (e.g. run-off-river schemes) the control system of a PAT can be significantly less elaborate and expensive and based on a standard on/off operating procedure.

In addition, given the lower peak efficiency of a PAT compared to that of conventional turbines and the relatively more sophisticated control system that circumstances may require when using a PAT it is reasonable to assume that above a certain threshold value of installed power the total cost of a scheme adopting either a PAT or a conventional turbine could match. Previous research which analysed the cost per installed kW of several hydro schemes suggested that such a threshold corresponded to 200-300 kW as seen in Figure 3-18 at Page 54 (Ramos, et al., 2010a).

In order to evaluate more accurately the total construction cost of PAT installations, data from 9 energy recovery schemes in water networks built in different countries have been compiled (Livramento, 2013; Lledo, 2016; Carravetta, et al., 2017b; Kramer, et al., 2018). All of the selected plants adopted a PAT as a generating device, and had nominal powers ranging from 9 to 120 kW. According to the available information, it was possible to sub-divide the total cost of the installation into single components:

- turbogenerator alone;
- turbogenerator and control system;
- commissioning;
- civil works and hydraulic equipment;
- grid connection;
- other project costs.

A few additional information on the 9 selected PAT installations is listed in Table 5-8.

TABLE 5-8. LOCATION, CAPACITY AND TYPE OF THE 9 SELECTED PAT INSTALLATIONS.

N.	Name	Nation	Capacity (kW)	Cost per kW (€/kW)	Description
1	San Vito di Cadore	Italy	9	17,778	At a service reservoir
2	Beliche	Portugal	11	6,818	At the inlet of a WTP
3	Sörenberg	Germany	15	2,403	At a service reservoir, planned
4	Annonay	France	26	5,769	At the inlet of a WTP
5	Casinos, Valencia	Spain	39	1,494	PRV substitution
6	Conejeras	Spain	73	2,981	PRV substitution
7	San Antonio	Chile	86	2,388	PRV substitution
8	Cartuja	Spain	89	2,353	PRV substitution
9	Malecón, Murcia	Spain	120	1,827	PRV substitution

The cost breakdowns from the 9 evaluated PAT schemes according to the above listed categories did not lead to any identifiable trend according to the nominal power of the scheme as displayed in Figure 5-24. Instead, it is reasonable to assume that a number of site-specific conditions (e.g. amount of civil works required, distance from the electric grid, and effectiveness of the design) resulted in varied percentages of each analysed cost category.

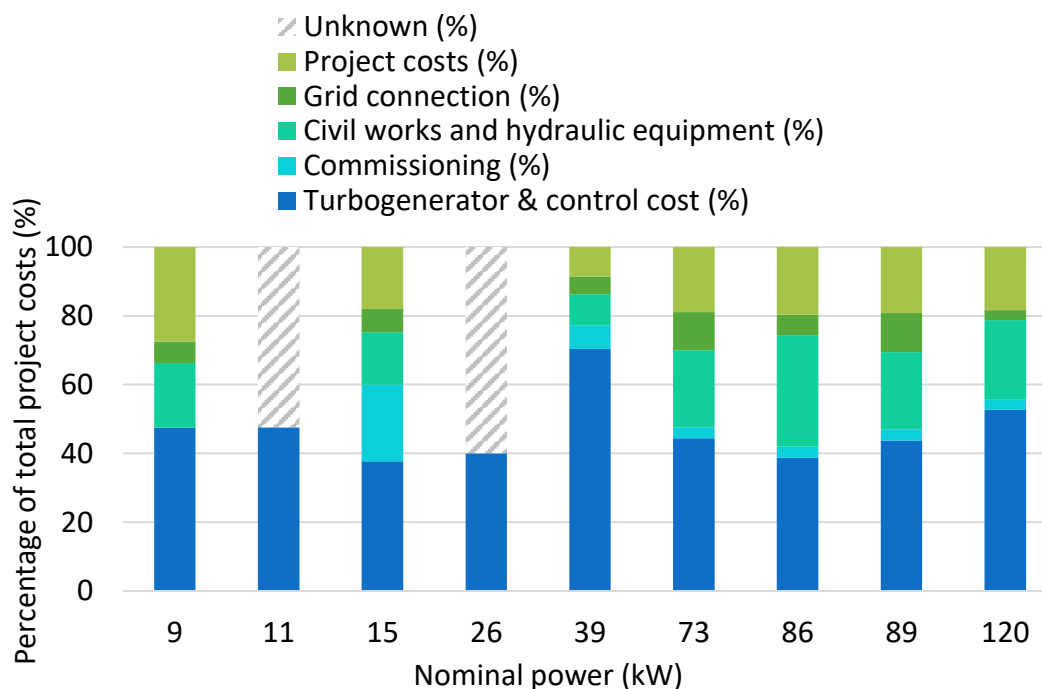


FIGURE 5-24. COST BREAKDOWN OF THE 9 ANALYSED PAT INSTALLATIONS IN WATER NETWORKS PER NOMINAL POWER.

The overall results of the analysis are shown in Figure 5-25, where the vertical error bars indicate the minimum and maximum percentages for each cost category.

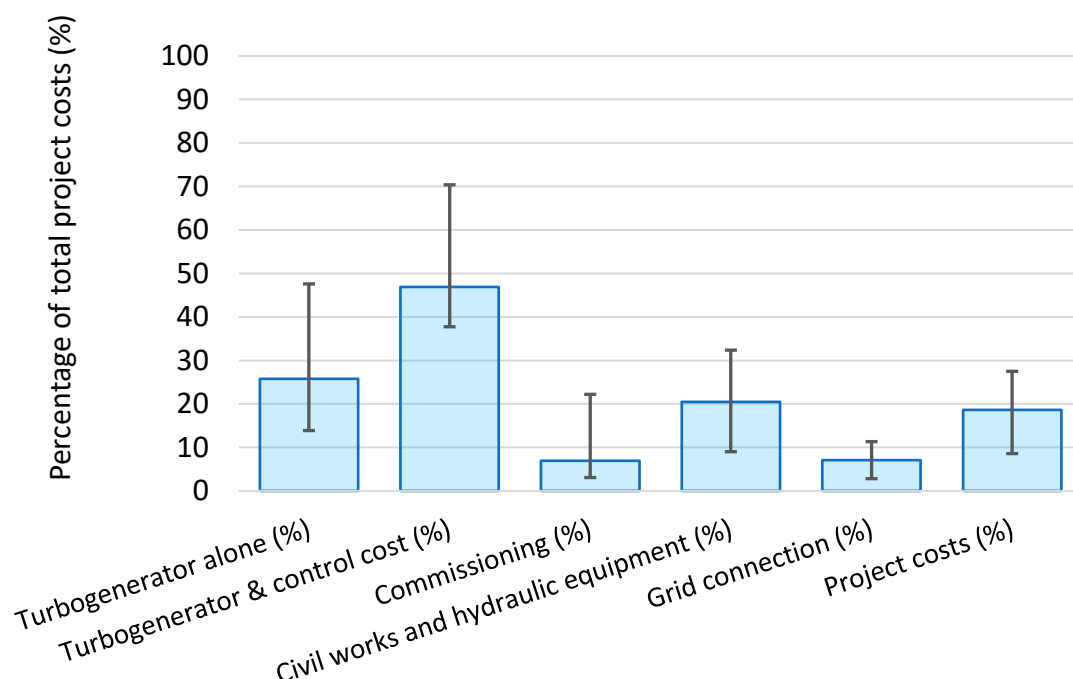


FIGURE 5-25. AVERAGE BREAKDOWN OF THE TOTAL INSTALLATION COST FOR THE 9 ANALYSED PAT SCHEMES.

The results from Figure 5-25 showed that across the 9 analysed schemes the purchase of the PAT and generator accounted on average for 26% of the total cost of the installation, ranging from a minimum of 14% to a maximum of 48%. Apart from the turbine and generator, the PAT control devices (with an average cost share of 24%) the civil work and hydraulic equipment (involving an average of 21% of the total cost) and the other project costs (accounting for the 19% of the total cost) were found to be the most expensive elements in a typical PAT installation.

As reviewed in Chapter 3.5.7 some researchers have suggested that in MHP applications the acquisition cost of the turbogenerator - not necessarily a PAT - typically accounts for a portion between 30 and 40% of the overall project budget (Ogayar & Vidal, 2009; Jain & Patel, 2014). Such a figure is slightly higher than the median value of 26% calculated for the analysed PAT installations as seen in Figure 5-25 but still falls within the minimum and maximum reviewed values. Two of the research studies described in Chapter 3.4.2 which evaluated the techno-economic feasibility of PRV

replacement with PAT in urban water networks neglected several cost factors (control equipment cost, commissioning, grid connection and project costs) and evaluated the cost magnitude of civil works as 30% of the turbogenerator costs (Fontana, et al., 2011; De Marchis, et al., 2014). According to the findings from Figure 5-25, such assumptions may lead to the underestimation of global project costs by nearly two thirds of the amount which may be expected. This will likely have a severe impact on the results shown by the two studies, which unsurprisingly determined very favourable project payback times between 2 and 6 years (Fontana, et al., 2011; De Marchis, et al., 2014).

5.4.5. CONCLUSIONS

The PAT cost model produced here is an important enabling tool in the exploitation of low-cost hydropower in natural and piped water resource systems. The results obtained prove that previously published PAT + generator cost figures are accurate but only in a fraction of the whole micro-hydropower application range, and the unit cost may vary from 130 to 5,400 €/kW according to the machine rated power. In all cases, PAT + generator unit cost proved to be between 5 and 15 times lower than generating sets based on conventional hydro turbines. Given that cost models are generally highly sensitive it is recommended for inflation to be accounted for during its future use.

Linear correlations were produced to estimate the price of radial and multistage PATs connected to a motor-generator unit having 1, 2 or 3 magnetic pole pairs as function of their nominal head and flow rate in turbine mode. The proposed cost model appears to be a valid tool for estimating the PAT price during the design phase, but its accuracy is limited by several factors: the number sample of pumps and pump types on which it is based, the assumption of equal maximum efficiency in pump and turbine modes and the inaccuracies of the Yang et al (2012) method used for predicting the PAT operating points.

Future work to improve the accuracy of this PAT cost model should focus on the inclusion of more data on axial flow and other pump families not well covered by the current database. In addition, when more reliable PAT performance prediction methods become available, the cost model can be further refined. However, these must be widely applicable considering the often limited amount of input parameters available.

Furthermore, scholars and practitioners attempting to predict the total project cost of a PAT installation must be aware that the cost of the control system for in-pipe applications where

backpressure control is needed has been reported to account for up to 1.2 times the acquisition cost of the turbogenerator unit. From a cross analysis of 9 PAT-bas energy recovery schemes installed in water networks around the world it emerged that on average the acquisition cost of the turbine and generator accounted for the 26% of the total project costs. Other relevant cost elements identified were, in decreasing order of relevance: turbine control module, civil works, project costs, commissioning and grid connection.

5.5. PAT DECISION SUPPORT AND SELECTION SOFTWARE

The content of the present chapter has been partially published as:

Novara, D. and McNabola, A., 2018. The development of a decision support software for the design of Micro-Hydropower schemes utilizing a Pump As Turbine. In Multidisciplinary Digital Publishing Institute Proceedings (Vol. 2, No. 11, p. 678).

The full research paper is hereby attached as Annex 3.5.

5.5.1. INTRODUCTION

As mentioned in Chapters 3.6.7 and 5.3.4 one of the major limitations of the use of PATs is their generally low part-load efficiency, which makes the choice of the equipment and plant layout for a determined site critical. This is particularly true for drinking water or irrigation networks, which commonly experience marked fluctuations of water demand daily and seasonally (Alatorre-Frenk, 1994).

Another limitation is the uncertainty about the PAT performances, since pump manufacturers tend not to divulge the characteristic charts of their units when operated as turbines. Such uncertainties in the location of the Best Efficiency Point (BEP) of PATs are intrinsic to the act of utilizing an off-the-shelf piece of equipment for a use not originally envisaged by its maker. The effect of such uncertainties can be multiple, ranging from a lower energy yield than the expected power output up to failing to maintain the desired backpressure at the turbine's outlet. Another important aspect to be considered is that choosing the most appropriate PAT for a determined site is a matter of non-trivial trade-offs between ideal targets which are often contrasting. The typical design targets can be of technical or economic nature:

- technical aspects
 - maximization of the energy yield over a set period of time;
 - maximization of the power output;
 - maintenance of an adequate backpressure at the turbine's outlet;
 - maximization of CO₂ savings over a set period of time;
 - mitigating the risk of pressure surges under runaway conditions, which occur if the connection to the grid is temporarily lost during turbine operations;
 - space constraints in presence of physical obstacles to the installation.
- economic aspects
 - maximization of the Net Present Value (NPV) of the project;
 - minimization of the project payback time.

A practical example of contrasting design parameters is when a pump is selected to operate as PAT in order to have as short payback time as possible, which would most likely lead to the selection of an undersized PAT with respect to other models which may have resulted in a higher energy yield and CO₂ savings. Ultimately, the selection of a PAT is a problem of multi-objective nonlinear optimization. Since the priorities and the relative weight of the above mentioned parameters may be relative to specific cases, the aim of the software developed here is to enable the designer to select any parameter relevant to their needs and for each of them identify on a chart the most appropriate families of commercially available pumps. Besides, the choice of giving a visual output also enables the software user to assess the convenience of selecting a different equipment layout (e.g., more than one PAT in parallel or in series).

5.5.2. ARCHITECTURE OF THE SOFTWARE

The key concept around which the software revolves is that of Best Efficiency Point (BEP), which corresponds to the set of hydraulic working conditions in terms of flow rate and head drop across a machine that allow the same unit to work under the highest possible hydraulic conversion efficiency. The software generates a 2D contour chart in which each point represents a hypothetical turbine whose BEP corresponds to the Q (m³/s) and H (m) coordinates shown on the

horizontal and vertical axis respectively. For each hypothetical turbine the hydraulic behaviour and purchase cost are computed, and its performances are evaluated against the provided flow rate and available head profiles. The logic of the software applied to each hypothetical unit is schematically represented in Figure 5-26.

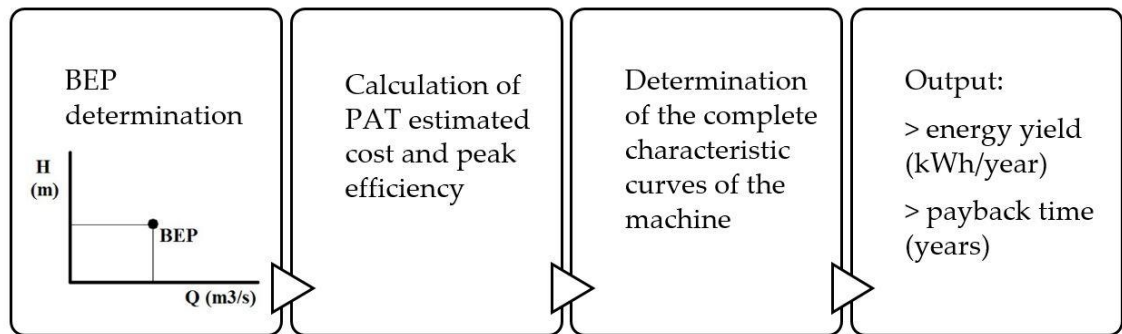


FIGURE 5-26. BASIC LAYOUT OF THE SOFTWARE LOGIC.

In order to enable the logical steps described in Figure 5-26, the following elements have been incorporated in the software following the scheme previously shown in Figure 2-1 at Page 21.

5.5.2.1. PAT Cost Model

The model to predict the purchase price of PATs connected to induction generators having 1, 2 or 3 pairs of magnetic poles (pp) developed in Chapter 5.4.3 and consisting of Equations (47)-(50) shown at Page 134 has been integrated in the PAT design software, allowing the calculation of the PAT + generator purchase price for a range of nominal shaft speed N of 3,020, 1,510 and 1,005 RPM respectively.

5.5.2.2. Efficiency Prediction

The model developed in Chapter 5.2.1 to predict the magnitude of peak hydraulic efficiency at BEP of a centrifugal PAT has been integrated in the software in the form of Equation (27) at Page 108.

5.5.2.3. Prediction of the BEP of a PAT

Since the graphic outputs generated by the software display several arrays of hypothetical PATs against their benefits (e.g. energy yield or payback time), it is evident that only a subset of them will represent machines that are effectively mass manufactured and available off-the-shelf.

Therefore, the BEP coordinates and maximum efficiency of the pumps featured in the catalogues by five manufacturers (KSB, SPP Pumps, Caprari, Ebara, Johnson Pumps) have been extrapolated in order to associate each unit to its predicted BEP as a turbine. Only units with full size impellers were selected as recommended by (Garay, 1990). In order to achieve this the semi-empirical nondimensional method developed by Yang et al. (2012) has been employed since among the alternative ones found in the literature and reviewed in Chapter 5.1 it has proved itself sufficiently accurate in a consistent manner. Besides, within the context of this analysis the peak efficiency of a pump has been considered equal to that of the very same unit in turbine mode, as suggested in the literature (Garay, 1990; Derakhshan & Nourbakhsh, 2008).

5.5.2.4. PAT Curve Extrapolation from BEP

In order to draw the approximate Q-H and Q- η full characteristic curves of all hypothetical PATs only knowing their Q, H and η at the BEP the model developed in Chapter 5.3 was applied. Such method is based on the application of Equations (36) and (37) at Page 115 with the set of coefficients shown in Table 5-4 at Page 118.

5.5.2.5. Prediction of PAT installation accessory costs and maintenance

In order to provide the software user with a rough estimate of the total project costs involved in the installation of a PAT within a water network, the results of the review carried out in Chapter 5.4.4 were considered. Following the cost breakdown displayed in Figure 5-25 at Page 139 an indicative estimate of the total expected cost for a PAT-based energy recovery scheme was calculated dividing by a factor of 0.26 the acquisition cost of PAT+generator alone previously obtained through Equations (47)-(50).

Finally, the yearly maintenance cost of the PAT & generator have been considered equal to the 5% of the acquisition cost of the two units themselves (Calado, 2014).

5.5.2.6. Development of a Graphical User Interface

In order to make the software more user-friendly and intuitive a Graphic User Interface (GUI) was designed as shown in Figure 5-27 and Figure 5-28.

The input tab displayed in Figure 5-27 requires the user to insert some key parameters, namely:

- PAT and generator shaft speed (RPM)
- If applicable, maximum power output from the scheme
- VAT rate (%) and value of the generated electricity in €/kWh (i.e. price of the imported electricity in case of self-consumption of the PAT output, or Feed-In Tariff in case of export to the grid)

Instead, the results tab relative to the yearly energy yield is displayed in Figure 5-28.

UI Figure

Input parameters Payback time Electricity Yield Numerical results

Shaft speed (RPM) 1500

Maximum power (kW) Inf

VAT (%) 20

Electricity cost (€/kWh) 0.18

Go

PaT selection software v1.0

FIGURE 5-27. GRAPHIC USER INTERFACE OF THE PAT SELECTION SOFTWARE: INPUT PARAMETERS TAB.

The presence of an automated by-pass duct has been supposed, in order to guarantee the network reliability under all operational regimes. Therefore, a portion of water can be diverted through the bypass instead of the turbine at any given time.

5.5.3. LIMITS OF THE ANALYSIS

In order to predict realistically the performance and cost of a continuous series of hypothetical PATs the model relies on a series of semi-empirical mathematical correlations whose applicability constraints and uncertainties consequently limit the results provided by the software.

Besides, although several pump—and therefore PAT—design families exist (e.g., multistage, submersible, double suction split-case, lobe pumps) by far the most common pump type corresponds to the centrifugal or radial units (Grundfos industry, 2004). Such category of pumps is not only the most easily available and sold worldwide, but also the one whose behaviour as turbine has been studied more in depth. Therefore, the present version of the software focuses solely on centrifugal PATs.

The permissible operating range of each PAT was considered as a precautionary measure to extend from 0.7 to 1.2 of Q/Q_{BEP} in order to ensure an acceptable reliability (Carravetta, et al., 2013a). Besides, the value of N_s for each PAT was limited to the range 6–150 since all of the units from which Equations (36) and (37) at Page 115 were derived fall within that interval.

5.5.4. CASE STUDIES AND RESULTS

The developed software has been preliminarily tested against two case-studies regarding energy recovery in water infrastructures available in the literature which refer to sites located in Italy and Portugal.

5.5.4.1. Water distribution network of Merano, Italy

A research study from the University of Trento (Italy) has evaluated the opportunity for energy recovery downstream of an elevated tank from which water is distributed to the city of Merano (Rossi, et al., 2016).

The flow duration curve follows a daily profile as displayed in Figure 5-29, showing a peak flow rate more than double than the minimum flow occurring during night time. In the absence of more detailed information, the available hydraulic head has been considered as constant and equal to 60

m (Rossi, et al., 2016). In reality, this value will change slightly according to the head losses in the piping.

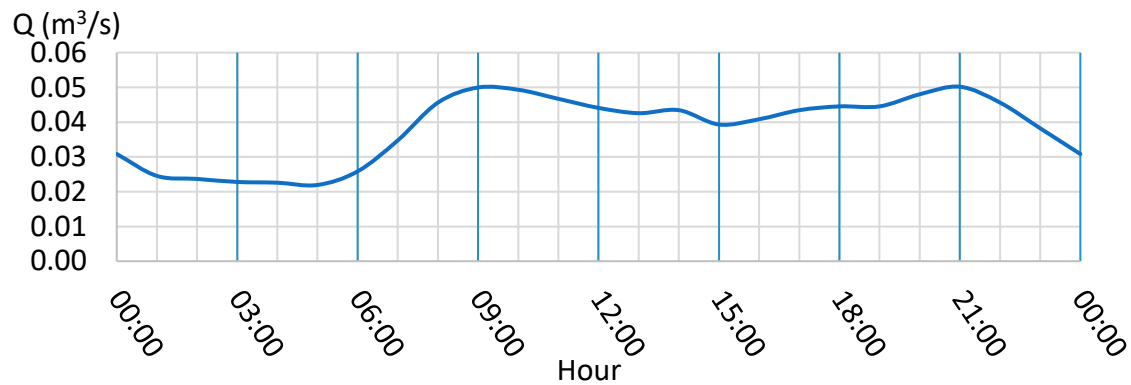


FIGURE 5-29. RECORDED FLOW RATE DAILY PROFILE AT THE OUTLET OF A WATER STORAGE TANK, ADAPTED FROM (ROSSI, ET AL., 2016)

The revenues from energy sales to the national grid were evaluated as 210 €/MWh which was the Feed-In Tariff from in-pipe MHP in Italy at the time of this study (GSE, 2016). Finally, in order to compare the results with those reported by the original study, the same shaft speed was selected (3,020 RPM). The results are shown in Figure 5-30.

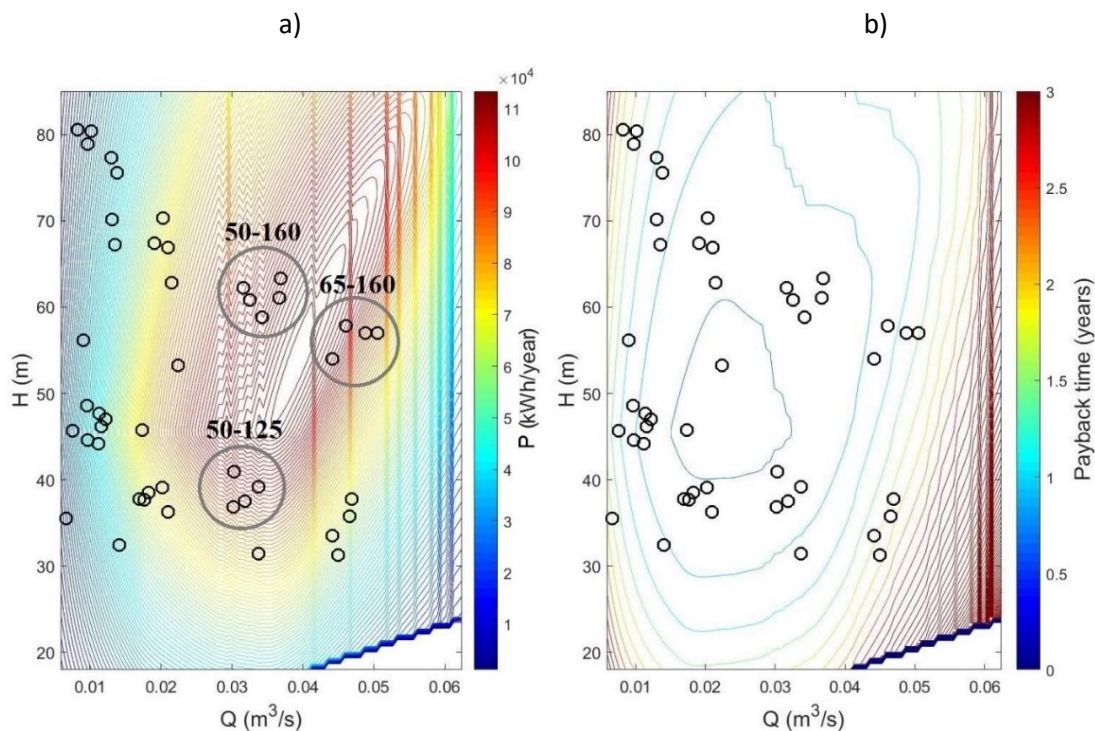


FIGURE 5-30. SOFTWARE RESULTS FOR THE PAT DOWNSTREAM OF MERANO AQUEDUCT: A) YEARLY ENERGY YIELD; B) SIMPLE PAYBACK TIME.

The chart of Figure 5-30 (a) indicates that the maximum yearly energy yield that can be obtained by a PAT rotating at 3,020 RPM in the selected location is equal to approximately 100 MWh/year which is close to the 123.4 MWh/year forecasted by the authors of the article (Rossi, et al., 2016). The families of standardized centrifugal pumps most likely to achieve the highest energy output are the 50–160, 50–125 and 65–150. The first two digits correspond to the nominal size in mm of the pump discharge nozzle (PAT inlet nozzle) while the last three refer to the nominal diameter of the impeller in mm. Instead, Figure 5-30 (b) shows that the payback of the investment would likely occur in less than three years with any PAT included in the range of the chart axis. However, the shortest possible payback time corresponds to a hypothetical turbine having nearly half of the nominal flow rate with respect to the ideal PAT that would maximize the energy yield as seen in Figure 5-30 (a).

5.5.4.2. Beliche WTP

The water treatment plant of Beliche previously introduced in Chapter 3.6.3.3 is located in the south of Portugal and derives water from an artificial reservoir. Two PATs were installed to recover energy from the residual pressure held by the water on entering the plant, and the energy recovery scheme started operations in January 2011 (Livramento, 2013; Ramos, et al., 2014)

The cumulative flow duration curve was calculated based on measured and predicted flows between the years 2006-2020 and is shown in Figure 5-31 (Livramento, 2013). Besides, a fixed available head of 17.6 m between the reservoir level and the treatment work was assumed (Livramento, 2013).

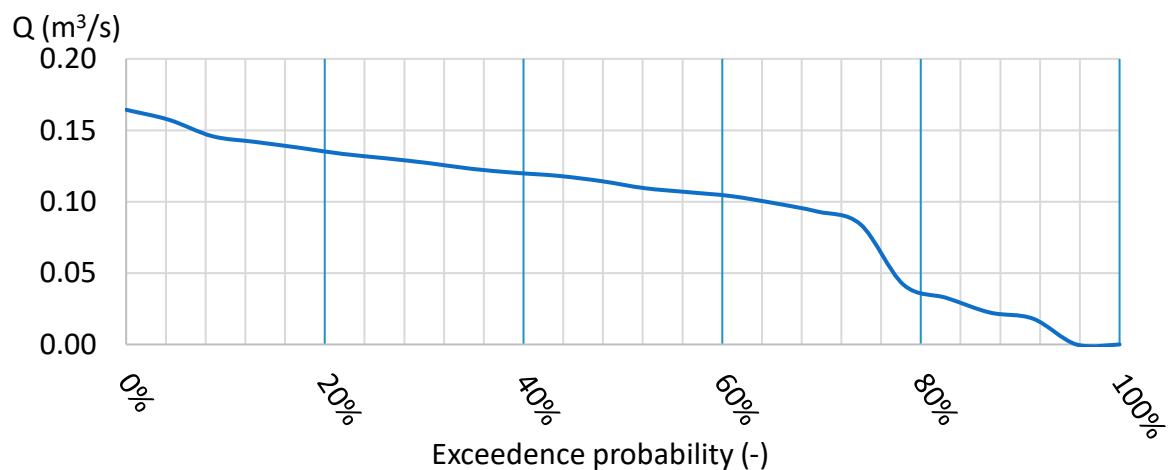


FIGURE 5-31. CUMULATIVE FLOW DURATION CURVE RELATIVE TO THE INFLOW TO THE BELICHE TREATMENT WORKS (LIVRAMENTO, 2013)

As in the previous case study, the turbine has been set to operate in parallel to an automated bypass channel to ensure system reliability. The revenues from energy sales to the national grid were set as 0.125 €/kWh (Livramento, 2013; Ramos, et al., 2014). The results are displayed in Figure 5-32.

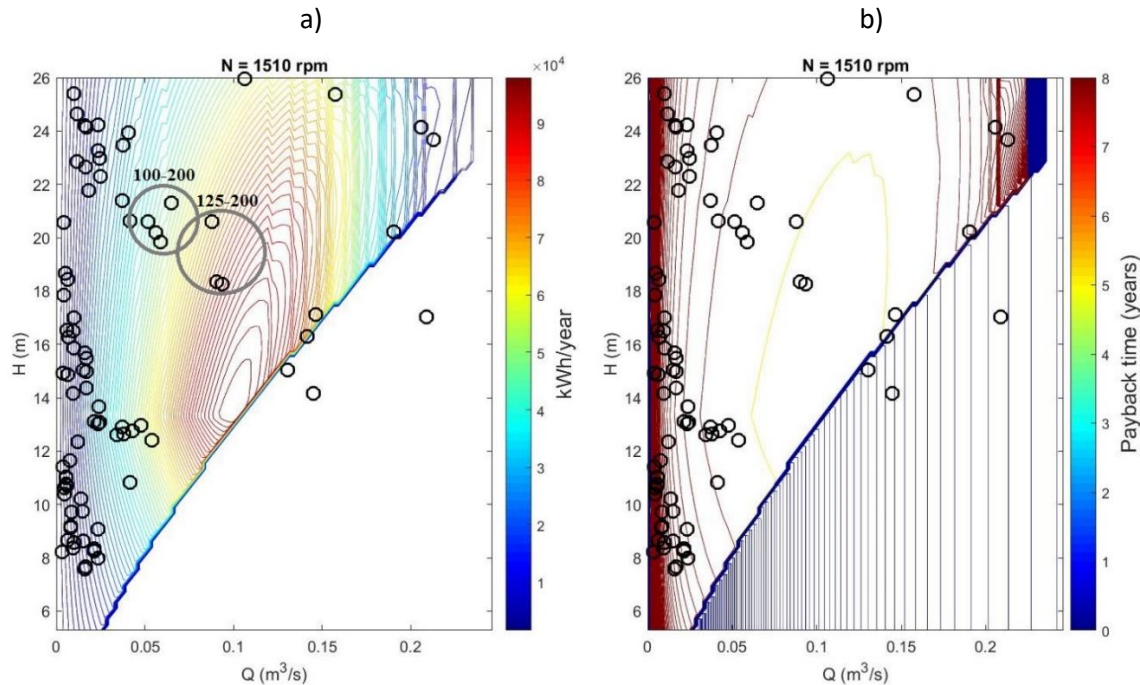


FIGURE 5-32. SOFTWARE RESULTS FOR A PAT AT THE BELICHE TREATMENT WORK INLET: A) YEARLY ENERGY YIELD; B) SIMPLE PAYBACK TIME.

The results chart of Figure 5-32 (a) and (b) shows that one single 125-200 PAT would likely maximize the energy production across the year and at the same time minimize the project payback time. Instead, it was decided for the actual installation to place two smaller 100-200 series PATs working in parallel. The two units in place can produce 5.5 kW each under nominal conditions while rotating as 1520 RPM (Livramento, 2013). This increase of equipment cost and system complexity may have had a relevant influence on the reported low profitability of the hydro scheme (Ramos, et al., 2014). In fact, one single 125-200 class PAT would be able to generate on average almost the same amount of electricity as two smaller 100-200 units over a year (about 70-80 MWh per annum).

The oblique cut that splits the plot areas in Figure 5-32 (a) and (b) is due to the imposed constraint of restricting the specific speed of PATs (N_s) in the 6-150 interval as described in Chapter 5.5.3.

5.5.5. CONCLUSIONS

The main scope of the proposed software is the selection of PAT family and system configuration for any given potential site in order to meet the existing constraints and satisfy the stakeholder's expectations. The generated results can be useful for a designer in order to shortlist one or more families of commercially available PATs for determinate locations. The software has been preliminarily tested against two previously investigated sites for in-pipe energy recovery within water networks whose data was found in the literature. The results compared favourably with the expectations of the authors of the original study and provided useful information for a successful choice of equipment. The software will be applied further in this thesis in the design of a large scale PAT lab test apparatus for validation purposes, and also in the design of two full scale field demonstrations (an in-pipe and run-of-river application).

6. LABORATORY CAMPAIGN

6.1. SMALL SCALE PAT TEST

6.1.1. DESCRIPTION OF THE EXPERIMENT

As a first experimental approach to the analysis of PATs it was decided to test a small 200W magnetic drive pump (brand IWAKI) in a variety of configurations in order to gather preliminary data and early learnings before proceeding with the design of the large scale PAT lab setup and the full scale pilot plants. The pump chosen is of the type commonly used for water circulation in swimming pools, has a simple design and is virtually leak-free thanks to the lack of a conventional seal.

The pump has been tested extensively in different configurations, both as a pump and as a turbine, namely:

- 1) **pump performances** at the rated speed;
- 2) **mechanical efficiency as a turbine**, after having it disconnected from its original motor and using a Prony brake to assess the torque;
- 3) **electric efficiency as a turbine**, after reconnecting it to its **AC induction motor** and using a grid-connected inverter to vary the shaft speed;
- 4) **electric efficiency as a turbine** when connected to a **three-phase AC brushless generator** under variable load;
- 5) when running connected to the DC brushless generator an Arduino shield has been programmed to act as a Maximum Power Point Tracker (**MPPT**) by varying the resistance of the dissipative circuit and thus changing the shaft speed to maximize the power output

The list of materials used along the experiments is shown in Table 6-1.

TABLE 6-1. LIST OF MATERIALS USED TO TEST THE IWAKI PAT.

Description	Make and model	Characteristics
600 W Supply pump	Pedrollo 3CRm 100	Multistage centrifugal pump run by a single-phase two-poles asynchronous motor
PAT	IWAKI MD-55R	Magnetic drive centrifugal pump powered by a 200W two-poles asynchronous motor

Piping	/	19 mm plastic piping
Valves	/	19 mm gate valves
Flow meter	IFM SM8000	Magnetic-inductive flow meter, measuring range: 0.2- 100 l/min
Pressure gauge	Gems Sensors 3500 Series	Absolute pressure transducer, max. pressure reading 4 bar, current output 4-20 mA
Tachometer	RS TM2011	Optical tachometer, measuring range: 60 - 19,999 RPM
Spring balances	/	Adjustable balances Measuring range: 0-3 Newton
Inverter	Invertek ODE-3-220105-1F42-01 Optidrive E3	1.1kW single-phase inverter with braking transistor
Braking load	TE Connectivity TJT150150RJ	Wire Leaded, 150 Ohm, 150 W, $\pm 5\%$
Multimeter	Megger M8035	Digital multimeter
Power meter	Efergy E2 Classic	Range of measurable current: 0.05 – 90 A
Visualizer	MECT MP1200 P6	Universal visualizer with programmable analog outputs
Brushless generator	Nanotec-DB59S0124035-A	Brushless DC motor, permanent magnet stator, with integrated Hall effect sensors
Rectifier	Semikron SKD-30	Power bridge rectifier
Control board	Quimat	Arduino UNO R3 type controller
8 channel switch	ELEGOO MEGA 2560 Project 1280 DSP ARM PIC AVR STM32	8 Channel DC 5V Relay Module with optocoupler for Arduino

Besides Equations (1)-(5), the following additional formulas have been used throughout the calculations:

$$H(m) = \Delta p(\text{bar}) \cdot 10.21 \quad (51)$$

$$P(W) = \frac{9,81 \left(\frac{m}{s^2} \right) \cdot H(m) \cdot Q \left(\frac{m^3}{s} \right)}{\eta} \quad (52)$$

(For the supply pump)

$$V(V) = R(\text{Ohm}) \cdot I(A) \quad (53)$$

$$P(W) = V(V) \cdot I(A) \quad (54)$$

The accuracy of the instrumentation employed is presented in Table 6-2 and has been evaluated in two ways:

1. the accuracy of the bespoke sensors employed (flow meter, pressure gauge, tachometer, spring balances) has been derived from their respective calibration certificates;
2. as for the other quantities measured through the Arduino shield, since such devices do not come with an official industrial calibration certificate the degree of uncertainty has been estimated case-by-case according to the byte/bit resolution available.

TABLE 6-2. ACCURACY OF THE INSTRUMENTATION EMPLOYED.

Unit	Device	Uncertainty	Notes
Q	Ultrasonic flow meter	Approx. $\pm 1.2\%$	$\pm (0.8 Q + 0.5 Q_{MAX})$
p	Pressure gauge	± 0.01 bar	0.25% of the scale (4 bar)
N	Tachometer	$\pm 0.05\%$	Best accuracy 0.05% + 1 digit
F	Spring balances	$\pm 3.3\%$	Scale of 0.1 N and range 0-3 N
V, N	Arduino voltage and Hall effect speed sensor	0.1%	$1/1023 * 100$
I	Arduino current	0.74%	0.46 (max. voltage reading) / $5 = 0.09$ $0.09 * 1023 = 135.04$ $1 / 135.04 = 0.74\%$

From the values of instrumental accuracy summarized in Table 6-2 the uncertainties on the main variables can be calculated as:

- Flow Q: $\pm 1.2\%$
- Head H: $\pm 0.35\%$
- Power available: 1.57%
- Power output P: $\pm 0.84\%$
- Efficiency: $\pm 2.05\%$

6.1.2. PUMP PERFORMANCES AT THE RATED SPEED

Some pictures of the experimental setup are displayed in Figure 6-1.

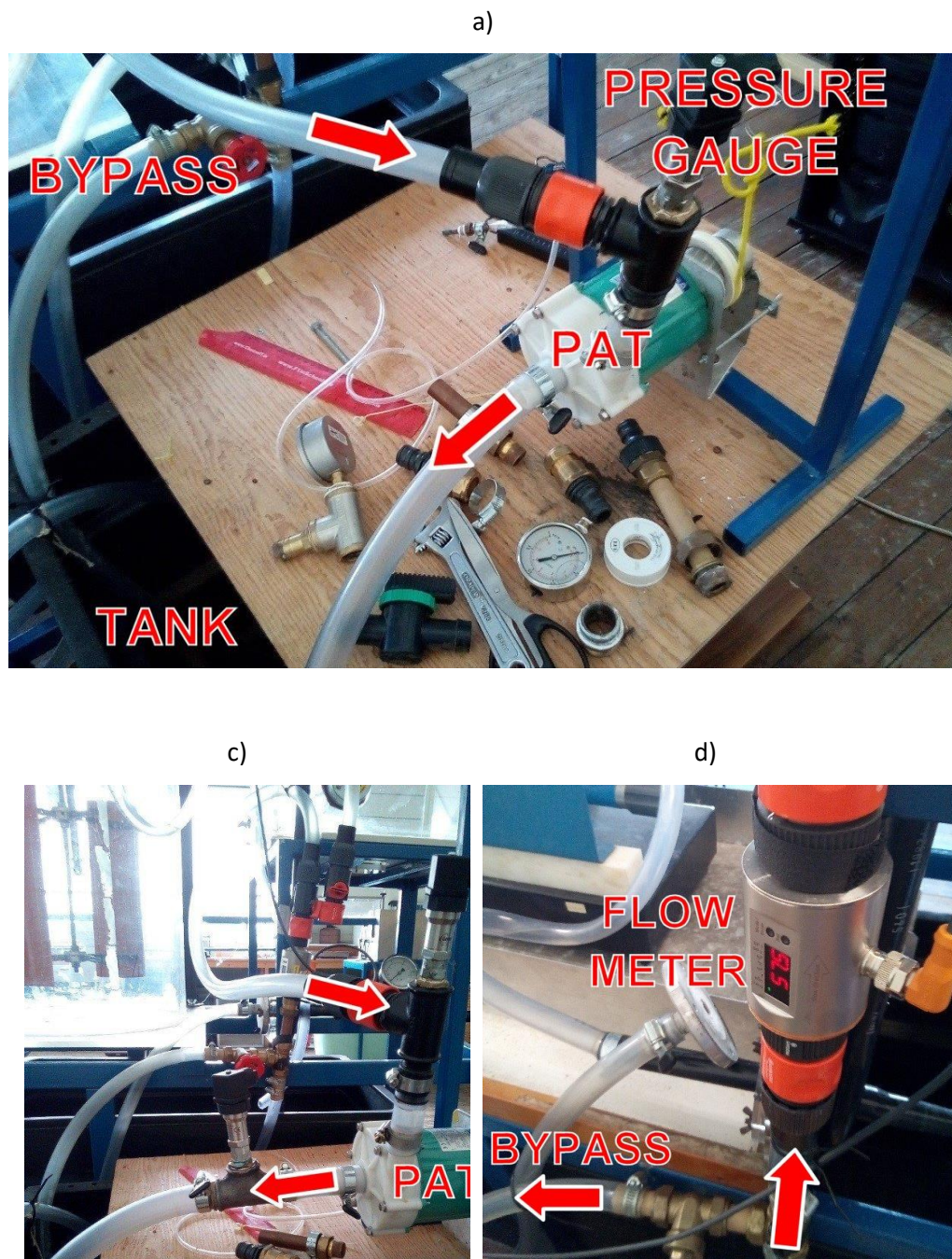


FIGURE 6-1. PICTURES OF THE MECHANICAL EFFICIENCY TESTS OF THE PAT: A) TOP VIEW OF THE PAT; B) SIDE VIEW OF THE PAT; C) DETAIL OF THE FLOW METER.

The first experimental setup to test the performances of the IWAKI unit as a pump is shown in Figure 6-2.

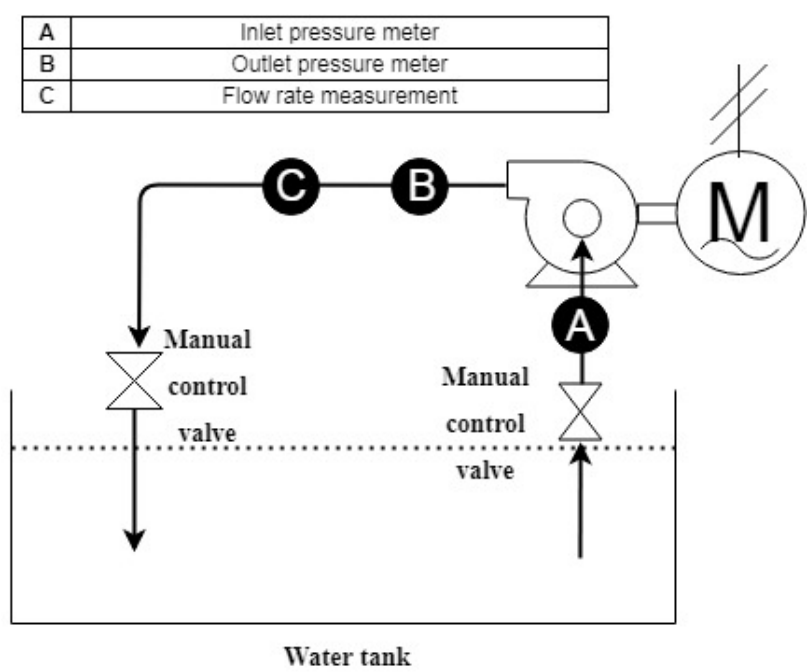


FIGURE 6-2. SCHEME OF THE EXPERIMENTAL SETUP TO TEST THE IWAKI UNIT AS A PUMP.

The results displayed in Figure 6-3 confirmed that the characteristic curve of the pump provided by the manufacturer was quite accurate.

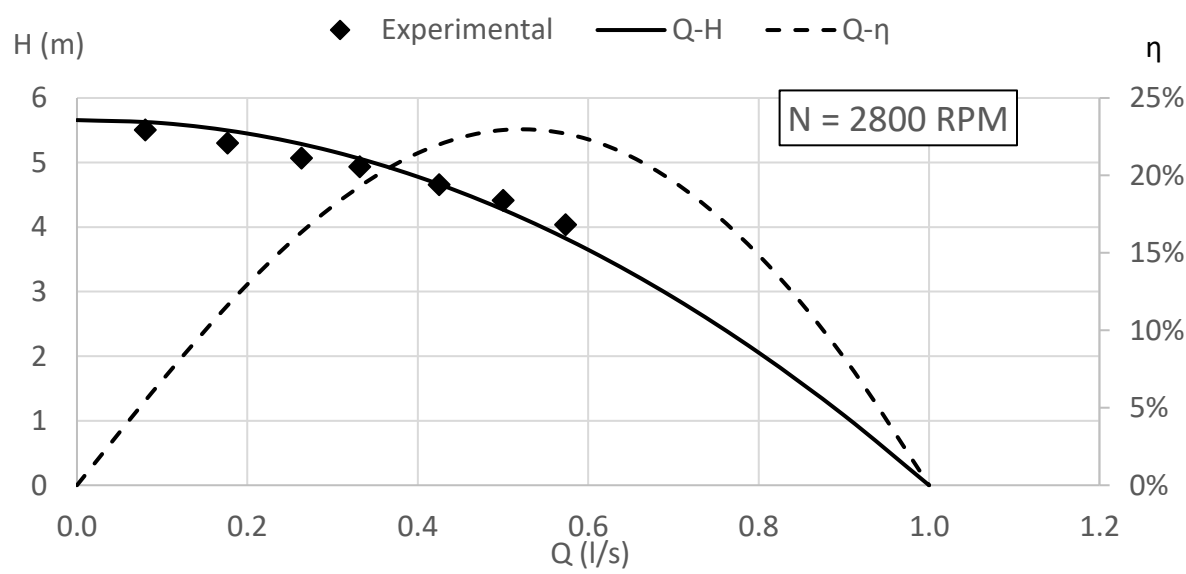


FIGURE 6-3. RESULTS OF TESTS OF THE IWAKI UNIT AS A PUMP VS. THE DECLARED PERFORMANCES AT NOMINAL SPEED (2800 RPM).

6.1.3. MECHANICAL EFFICIENCY AS TURBINE

As a second step, the pump was stripped of its motor stator and assembled on a frame as displayed in Figure 6-1 with a slip ring glued to the original rotor of the motor. A Prony brake device with adjustable tension springs has been utilized to measure the torque on the PAT under different speed, head and flow conditions. A scheme of the setup is shown in Figure 6-4.

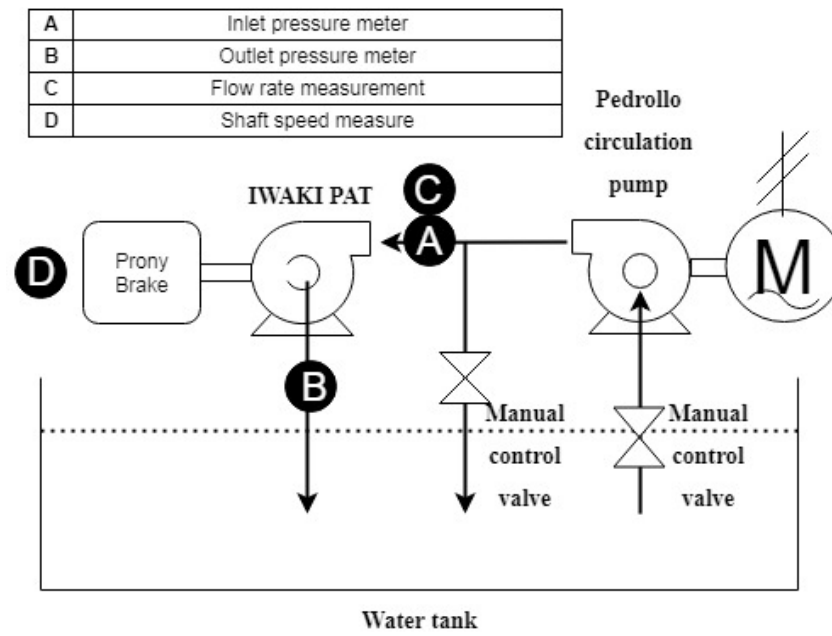


FIGURE 6-4. SCHEME OF THE EXPERIMENTAL SETUP TO TEST THE IWAKI UNIT AS A PAT WITH A PRONY BRAKE.

As a first stage, the locked rotor and runaway curves have been acquired as a succession of points displayed in Figure 6-5.

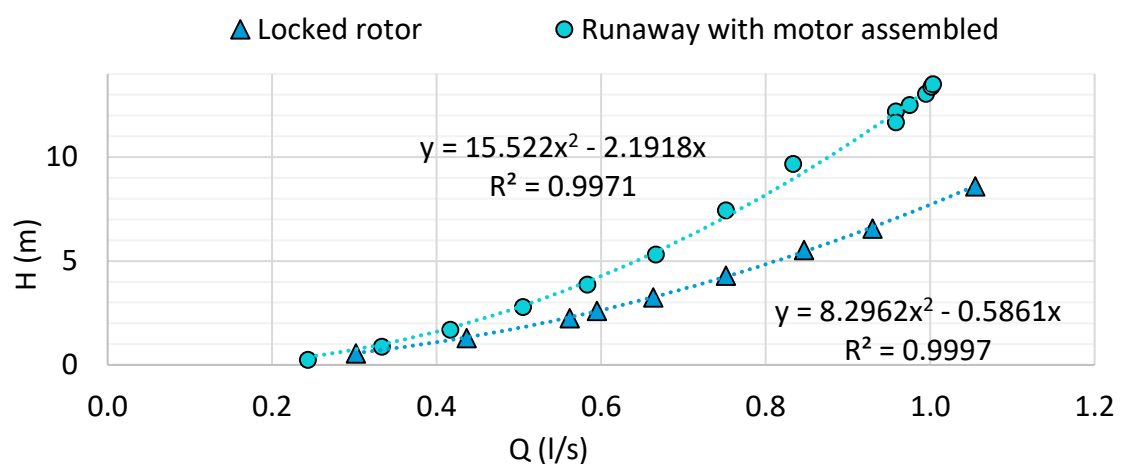
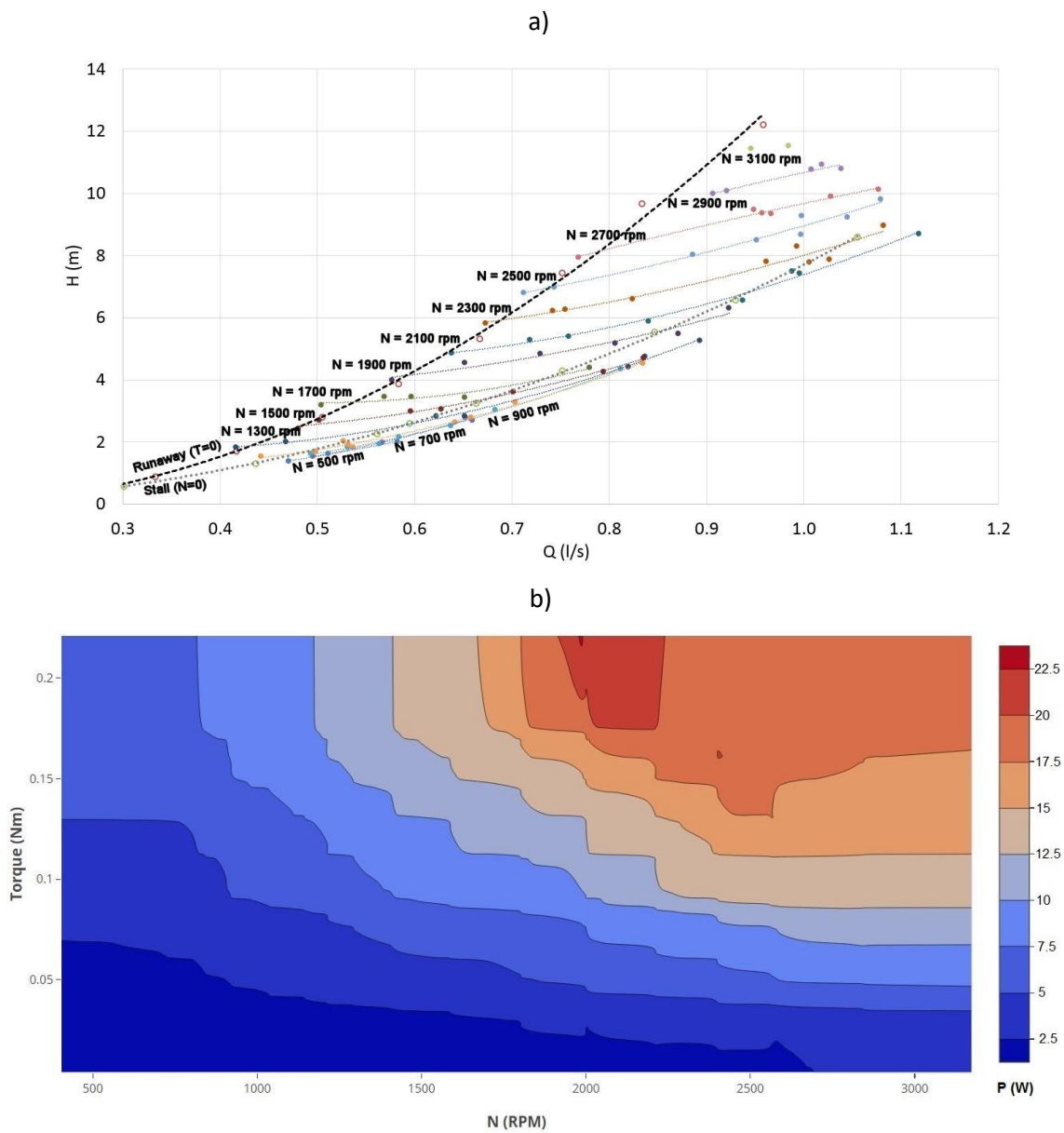


FIGURE 6-5. EXPERIMENTAL LOCKED ROTOR AND RUNAWAY CURVES.

Subsequently, the full characterization of the performances of the PAT was carried out through the following procedure:

- set the opening degree of the manual bypass valve, thus varying the flow and the head available at the PAT;
- gradually increase or decrease the tension on the spring balances;
- for each setting of the spring balances and bypass valve opening manually record the upstream and downstream pressure, flow rate, shaft speed, force on the brake springs.

The results in terms of head-flow, efficiency-flow and power-flow curves are displayed respectively in Figure 6-6 (a), (b), (c) and (d).



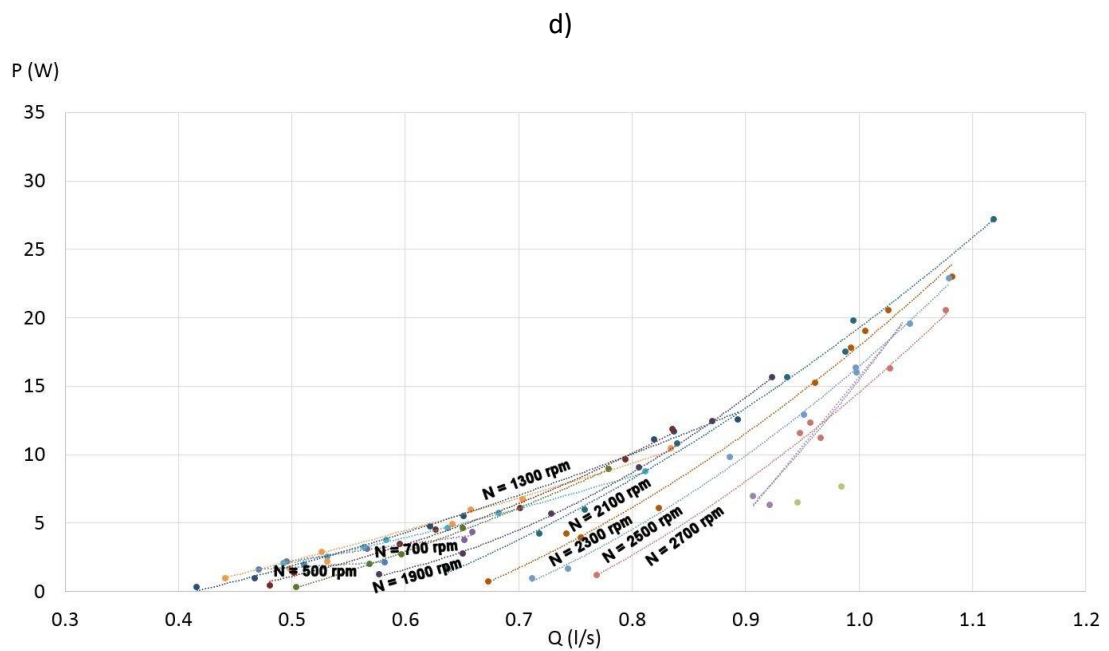
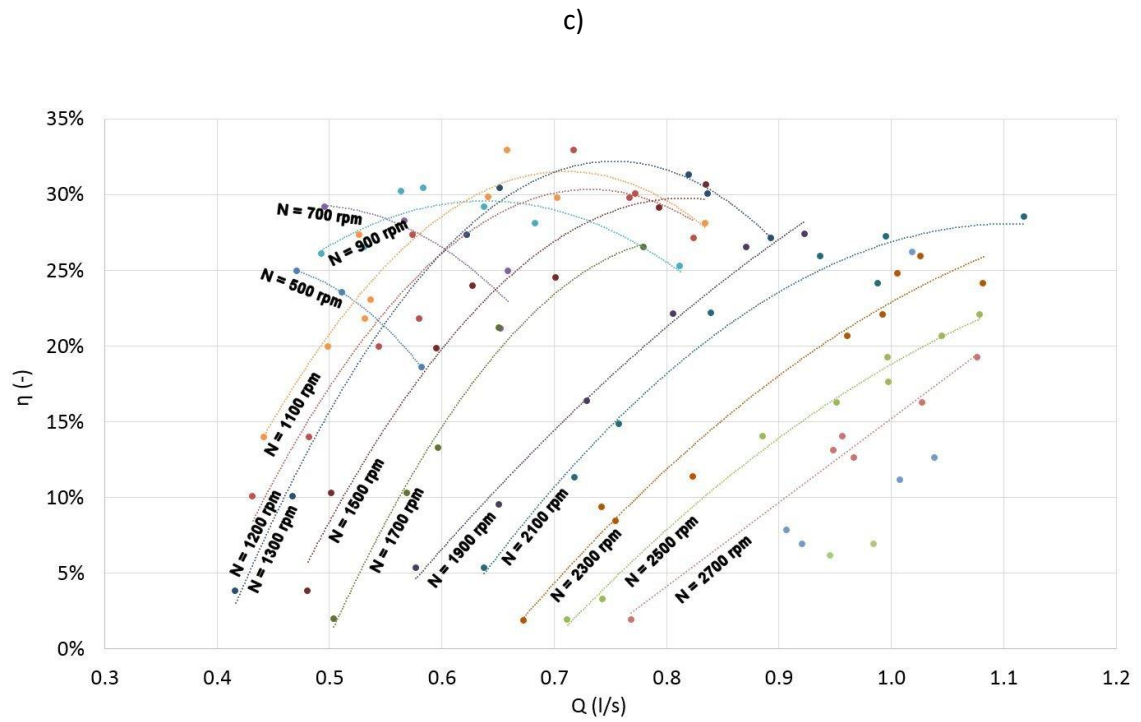


FIGURE 6-6. RESULTS FROM THE MAPPING OF THE IWAKI PAT THROUGH THE PRONY BRAKE: A) FLOW-HEAD CURVES AT DIFFERENT SPEED; B) POWER OUTPUT VS. TORQUE AND SPEED; C) EFFICIENCY VS. FLOW; D) POWER VS. FLOW.

The results shown in Figure 6-6 helped to determine the basic hydraulic behaviour of the IWAKI PAT, and were fully satisfactory as the trends of head, efficiency and power curves along the flow

rate variation closely resembled the ones of other PATs testes in the literature. The peak efficiency of the unit was rather poor (32.5 %) which could be expected due to the small scale of the machine and its low efficiency in pump mode.

6.1.4. ELECTRIC EFFICIENCY WITH AC INDUCTION MOTOR AND INVERTER

As a third step, the original 200W single-phase induction motor of the IWAKI PAT was reassembled and used as a generator in order to assess the combined performances of the unit when generating electricity. A few pictures of the setup are displayed in Figure 6-7.

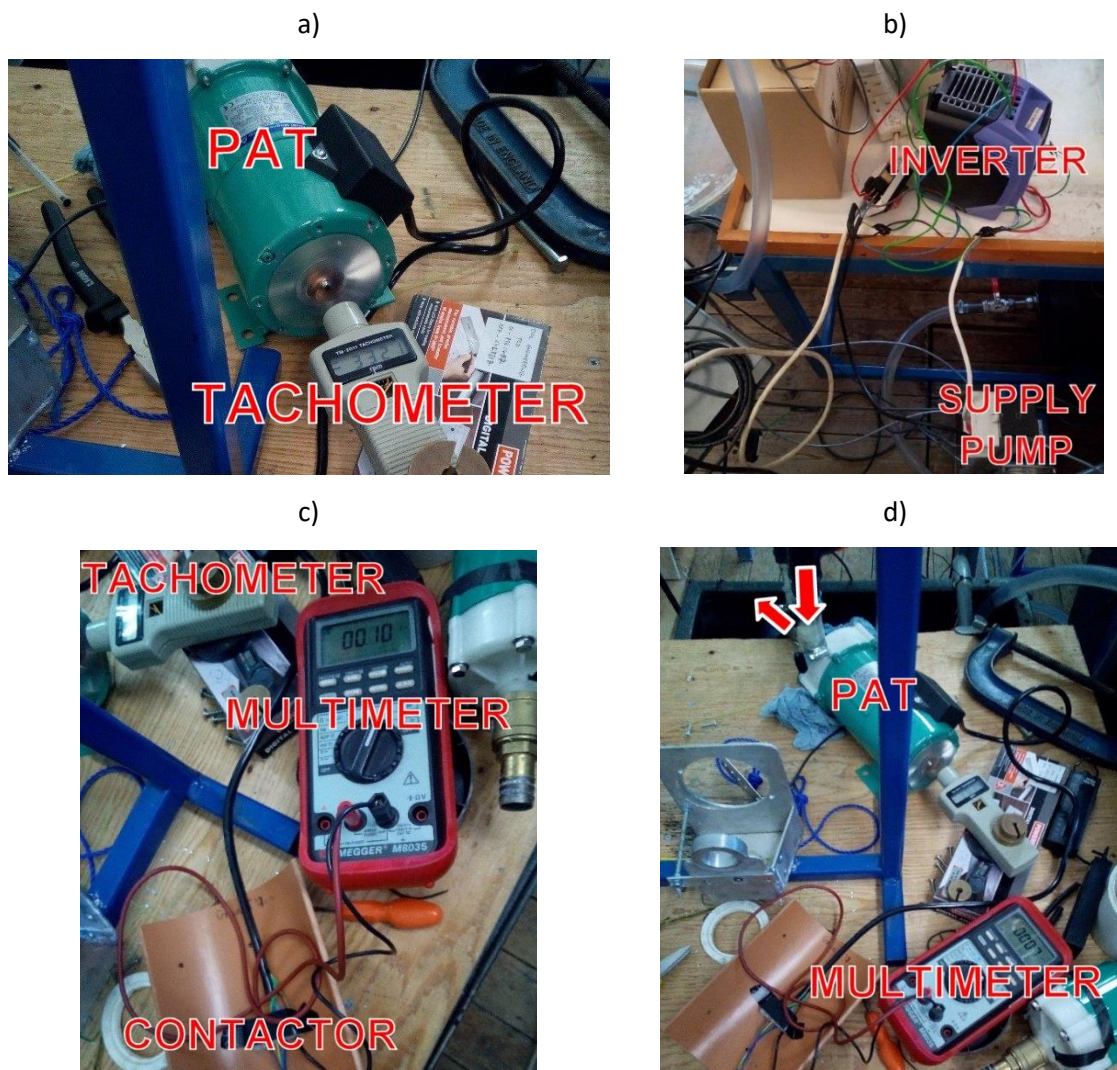


FIGURE 6-7. PICTURES OF THE PAT TESTED WITH CONNECTED INDUCTION MOTOR AS GENERATOR: A) TOP VIEW OF PAT AND TACHOMETER; B) INVERTER AND SUPPLY PUMP; C) AND D) MULTIMETER, PAT AND TACHOMETER.

At first the single-phase two-pole motor of the PAT was directly connected to the electric mains thus allowing uniquely for a fixed rotation speed of $\approx 3,020$ RPM in agreement with Equation (9) at Page 64. As seen from Figure 6-8 the PAT curve at the synchronous speed was for the most part above the runaway curve and therefore no power generation was measured by the instrumentation (multimeter and power meter). On the contrary, the PAT was effectively drawing energy from the grid in order to contrast the flow of water as the unit was operating in the D quadrant of Figure 3-21 at Page 59.

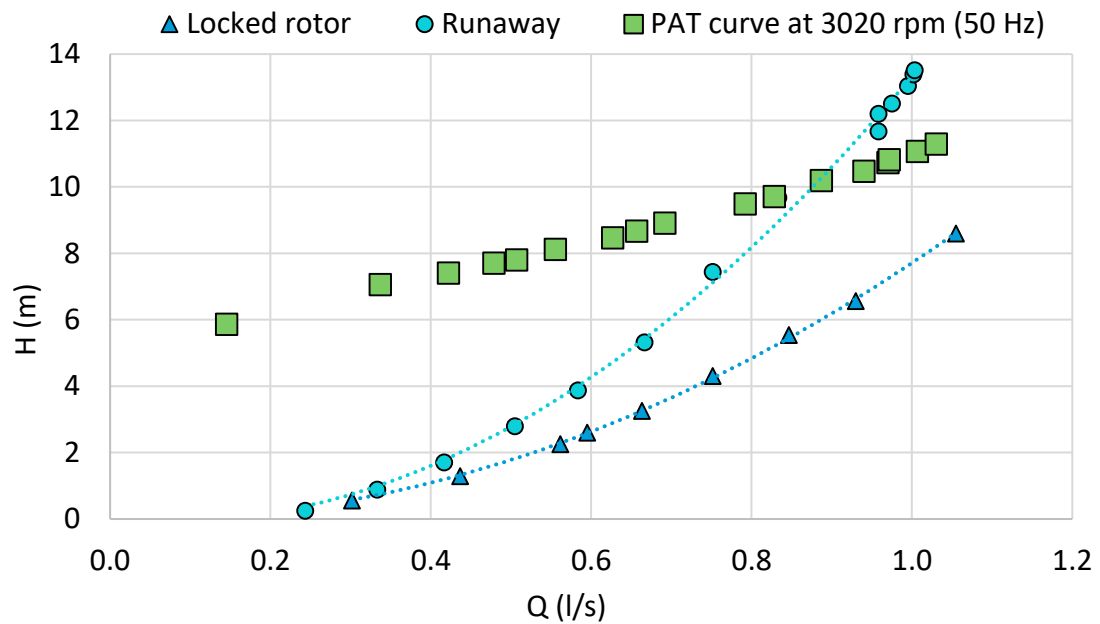


FIGURE 6-8. RESULTS AFTER USING THE IWAKI PAT WITH FIXED-SPEED INDUCTION GENERATOR.

In order to start generating electricity the solutions were twofold:

- install a more powerful supply pump other than the Pedrollo multistage unit utilized so far, able to work at higher flow rates and heads;
- install a single-phase variable frequency drive (inverter) placed between the PAT generator and the mains in order to vary the excitation frequency dispensed to the induction motor.

It was decided to proceed with the latter option, and a 1.1 kW single-phase inverter with braking transistor was purchased. Since regenerative inverters for such low power ratings are not available, the decision fell on a unit set up for regenerative braking and connected to a 150 Ohm braking resistor rated for high power inputs. A scheme of the final experimental setup is displayed in Figure 6-9.

A	Inlet pressure meter
B	Outlet pressure meter
C	Flow rate measurement
D	Shaft speed measure
E	Voltage and frequency measure (through the inverter)
F	Power meter

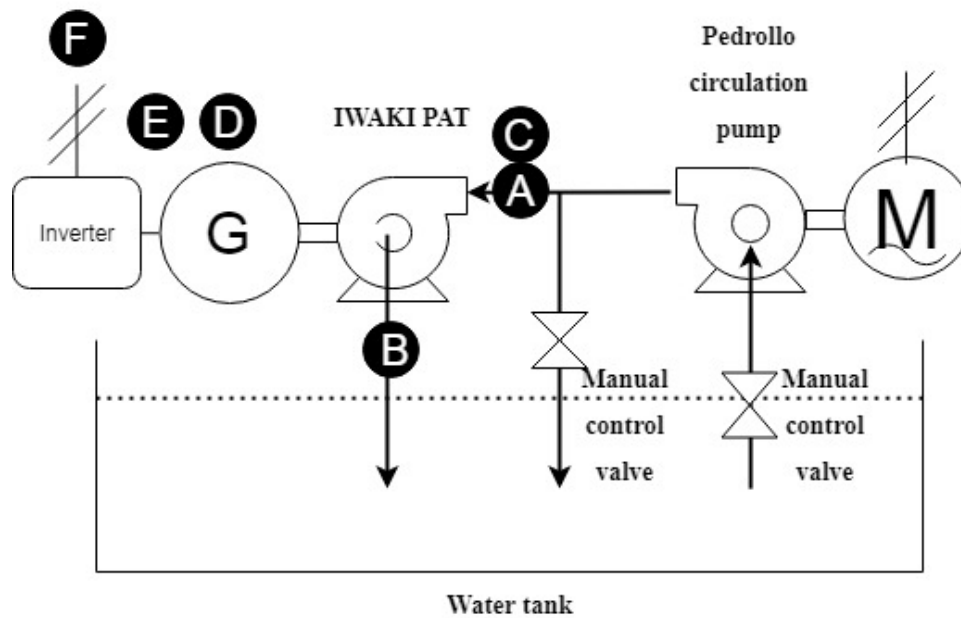


FIGURE 6-9. SCHEME OF THE EXPERIMENTAL SETUP TO TEST THE IWAKI UNIT AS A PAT WITH INDUCTION MOTOR AS GENERATOR AND INVERTER.

The inverter installed on the generation side was able to vary the voltage from 230 V down to about 90 V AC and the frequency from 70 to 20 Hz as displayed in Figure 6-10.

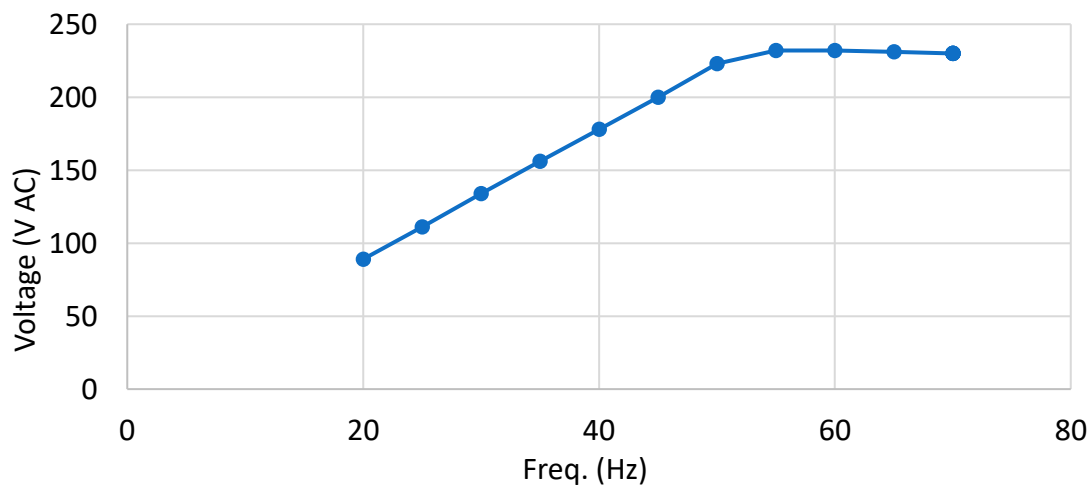


FIGURE 6-10. VOLTAGE AND AC FREQUENCY ACHIEVED BY THE INVERTER.

The whole setup was first tested keeping the inverter running at the highest admissible frequency (70 Hz) and slowly bringing the flow bypass valve from fully open to fully closed position. Subsequently, the frequency was lowered until the minimum reachable value (20 Hz) as shown in Figure 6-11. The test demonstrated that it was possible to operate the PAT under variable speed within the power generation area labelled as C in Figure 3-21.

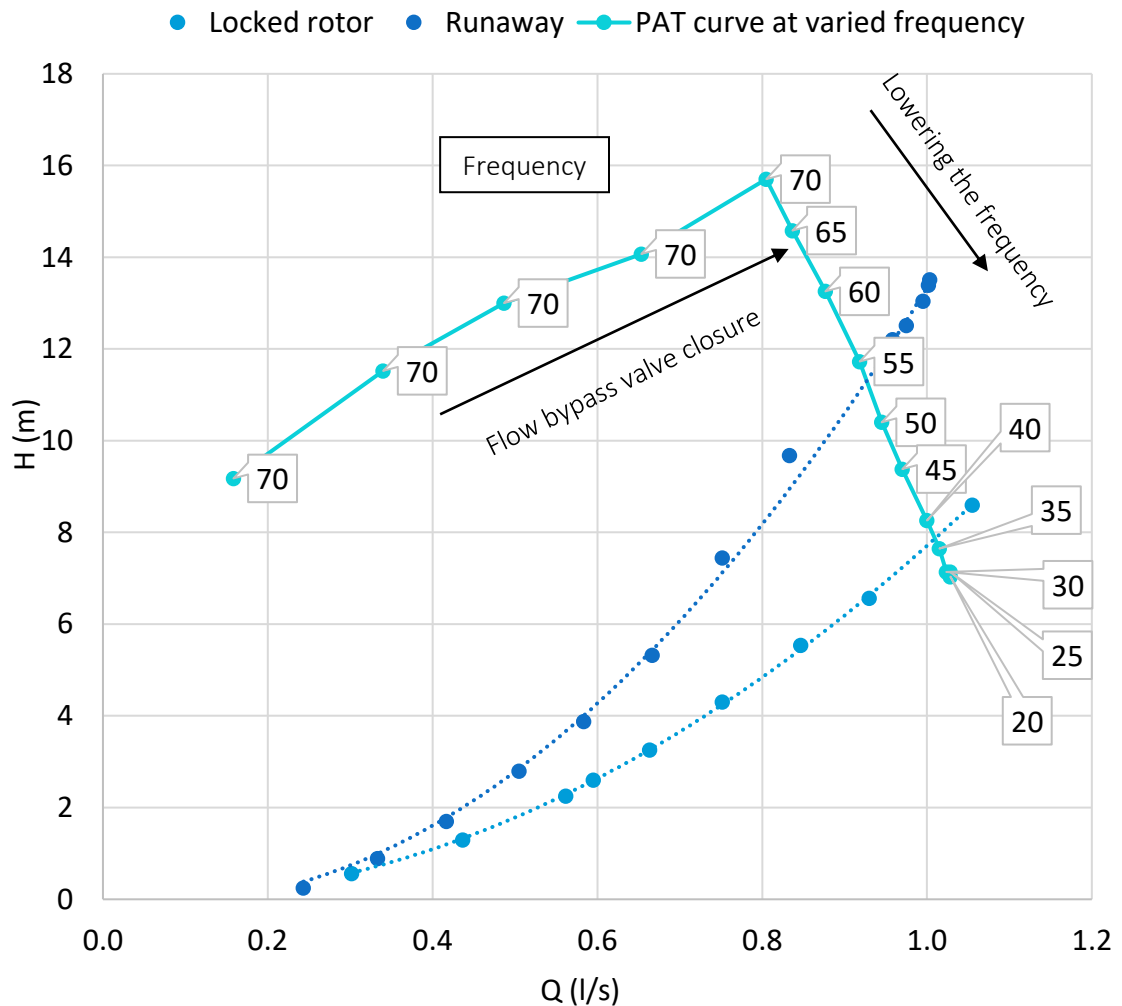


FIGURE 6-11. CHARACTERISTIC CURVE OF THE PAT VARYING THE FREQUENCY AT THE GENERATOR.

From the curve shape of Figure 6-11 we could therefore expect to reach power generation at frequencies below 50 Hz, which is coherent with the fact that we were previously not able to generate any power without the inverter when the generator was excited with standard 50 Hz current from the grid.

The same test was then repeated twice in order to measure the consumed or generated power by means of a digital multimeter (knowing the voltage imposed by the inverter) and a power meter

placed on the inverter supply cord. The results are displayed in Figure 6-12. The chart shows a good correlation between the readings made through the two instruments and validate the hypothesis that power generation could only occur with full bypass valve with frequencies below 50 Hz. The maximum recorded power generation equalled to around 20 W which is comparable with the 27 W of peak mechanical power generated by the PAT at the shaft measured with the Prony brake as can be seen in Figure 6-6.

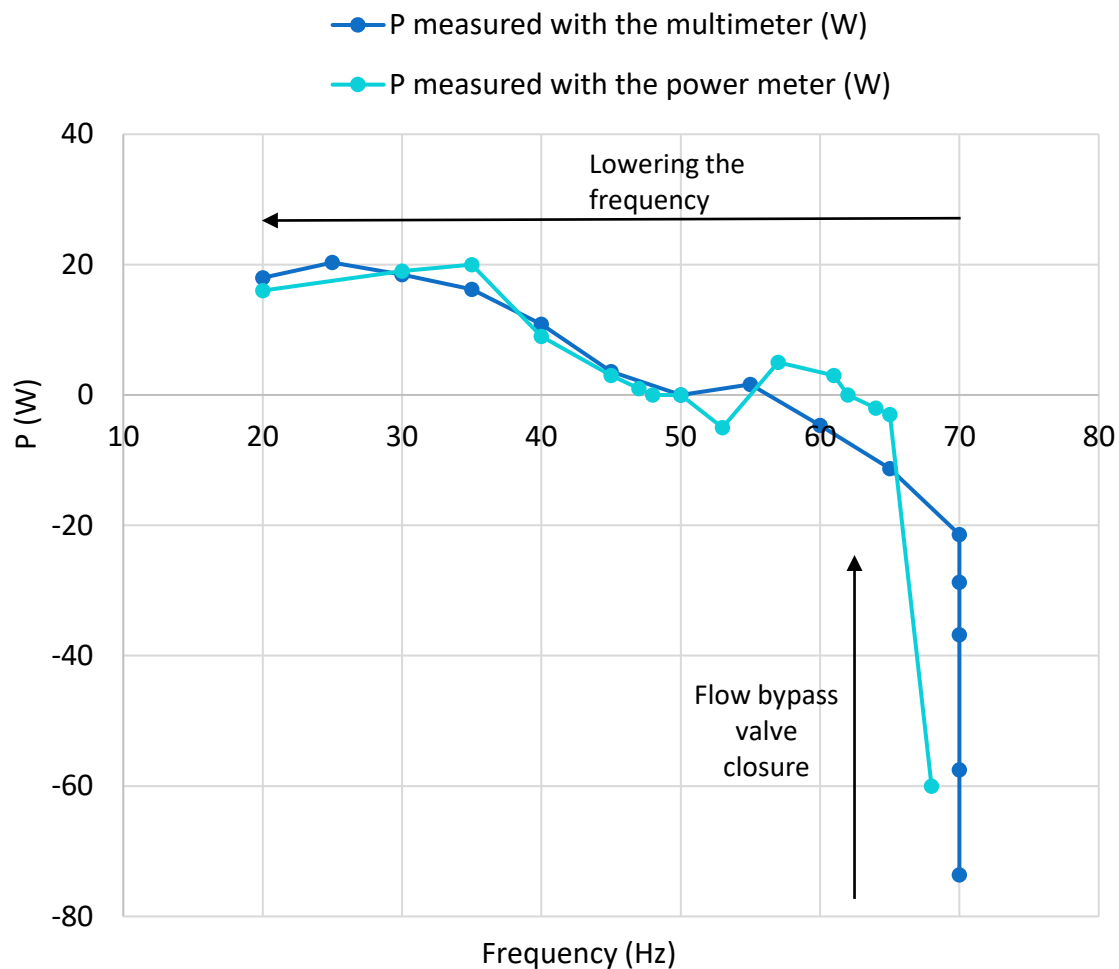


FIGURE 6-12. POWER ABSORBED (-) OR GENERATED (+) BY THE PAT CONNECTED TO ITS INDUCTION GENERATOR AT DIFFERENT FREQUENCIES.

6.1.5. ELECTRIC EFFICIENCY, DC BRUSHLESS MOTOR AND VARIABLE LOAD

In agreement with the project partners it was decided for the demo sites to be designed and built at Ty Mawr Wybrnant in Wales and to equip the PAT with a three-phase synchronous generator feeding power to a grid-connected solar type-approved inverter rather than a more common

induction generator as the one tested in the previous chapter. In order to gain confidence in the design of the installation, a three-phase brushless Permanent Magnet (PM) generator was purchased and long-coupled to the IWAKI PAT after stripping its original motor stator. Such generator is normally commercialized as a DC brushless motor according to its most common use, but fundamentally is a three-phase synchronous unit able to work as a generator. The unit purchased was a 6-poles one rated for 84 W at 24 V DC as a motor.

The PAT and the new generator were both mounted on a custom metal frame and the electricity produced fed into a diode full-bridge rectifier and subsequently dissipated by means of a variable resistive load. A scheme of the setup is displayed in Figure 6-13 and pictures of the experiment are shown in Figure 6-14.

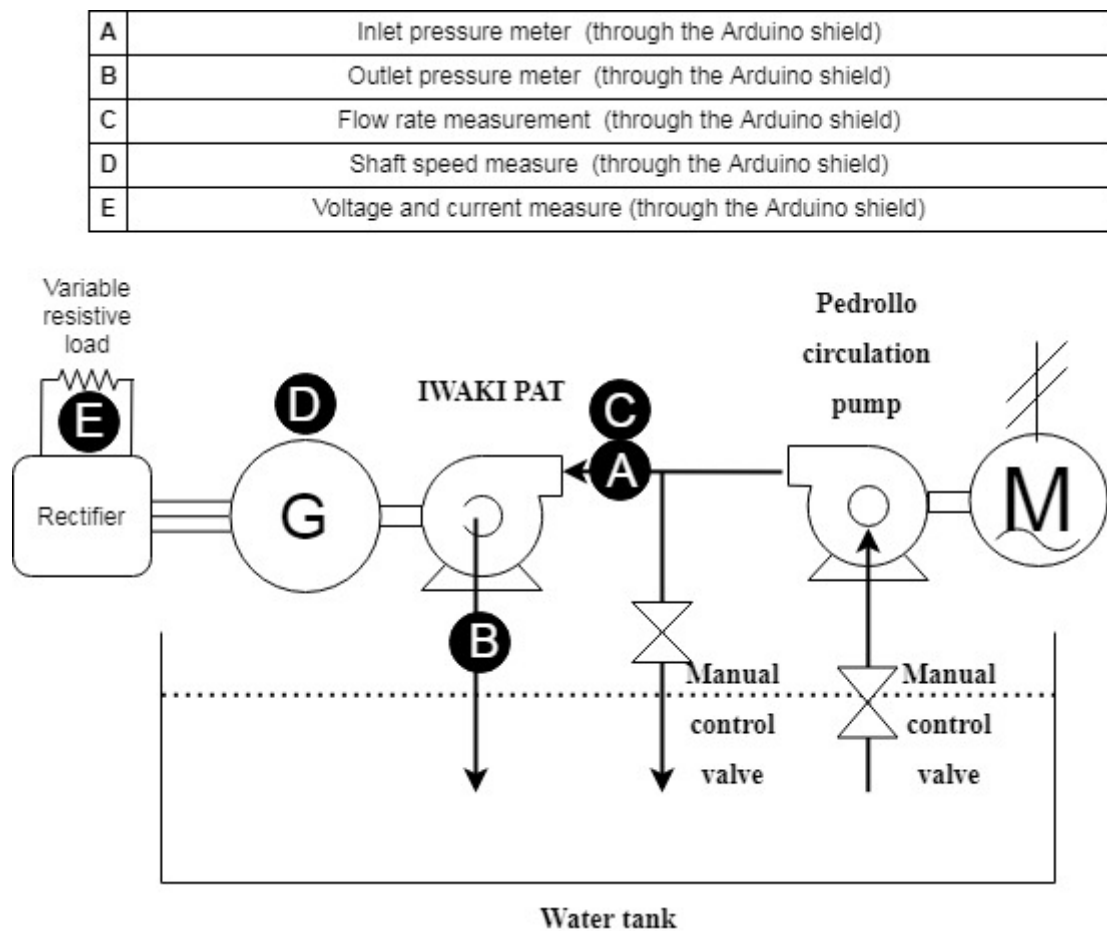


FIGURE 6-13. SCHEME OF THE EXPERIMENTAL SETUP TO TEST THE IWAKI UNIT AS A PAT WITH BRUSHLESS PM GENERATOR.

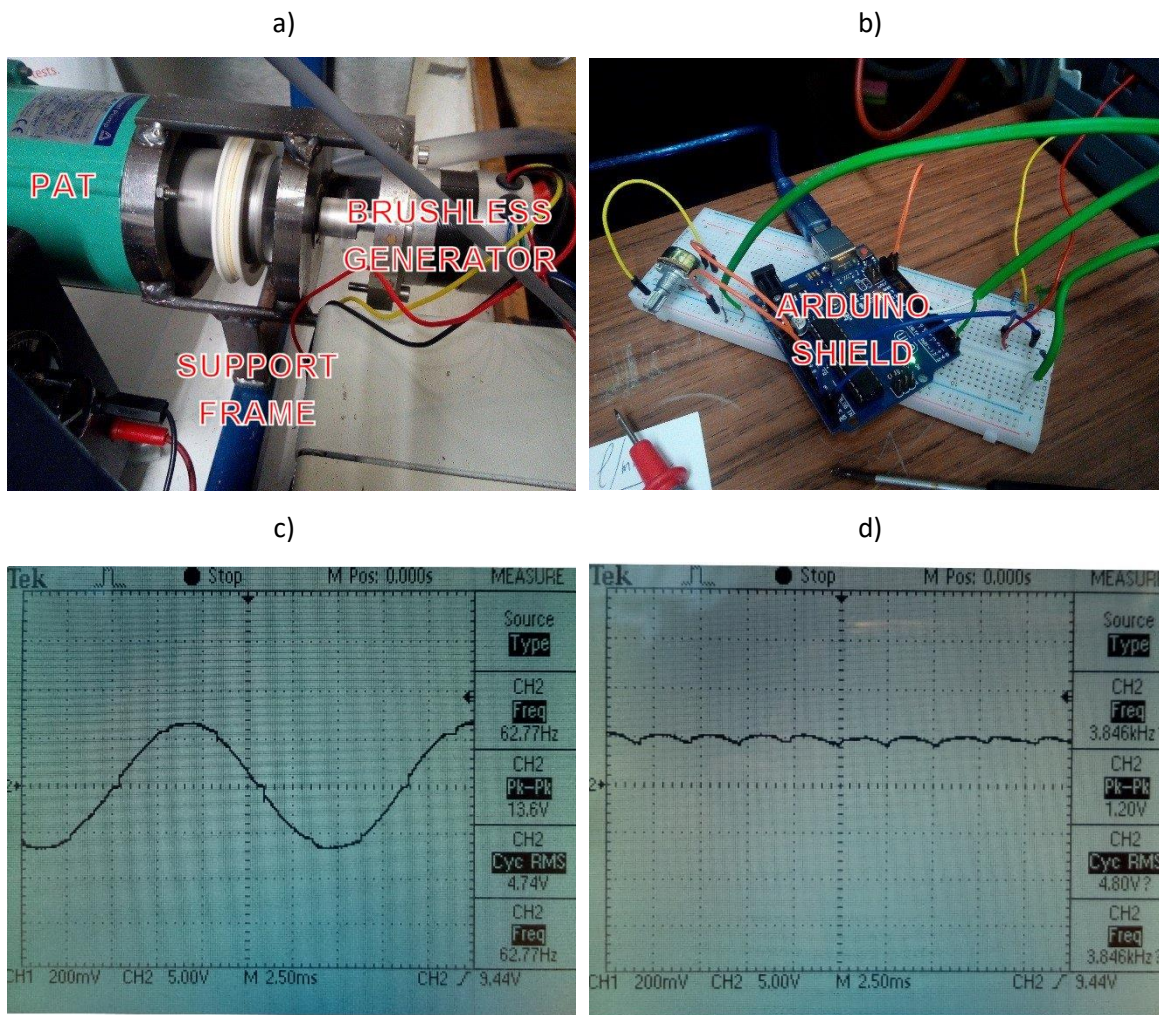


FIGURE 6-14. IMAGES OF THE IWAKI PAT TESTED WITH A BRUSHLESS AC GENERATOR: A) CLOSE VIEW OF THE PAT AND GENERATOR COUPLING; B) ARDUINO SHIELD USED AS A DATA COLLECTION UNIT; C) SINUSOIDAL VOLTAGE WAVE GENERATED BY ONE OF THE THREE PHASES OF THE BRUSHLESS GENERATOR; D) DC VOLTAGE AT THE POLES OF THE RECTIFIER.

From Figure 6-14 (c) and (d) it can be noticed how the bridge rectifier acts to convert an AC current of peak-to-peak amplitude of 13.6 V into an almost constant 4.8 V DC output made up of the crests of voltage peaks from all three phases originally shifted by 120° .

6.1.6. DATA LOGGING THROUGH THE ARDUINO SHIELD

As a substantial improvement from the manual readings of the various instrumentation performed during the previous experiments, it was decided to employ an Arduino shield to perform automatic

data logging task throughout the experiment. Since such microprocessor is only able to record voltage analog signals in the range 0-5 V the methodology adopted for each variable to be measured was:

- **pressure readings:** the 0-5 V analog signal generated by a MECT visualizer previously connected to the pressure gauge was directly fed into the Arduino pins;
- **flow readings:** the pins of the 4-20 mA output signal provided by the flow meter were directly connected to the Arduino in series with a resistor and the flow readings through the Arduino were calibrated by means of a multimeter and the LCD display of the meter as per Figure 6-15;

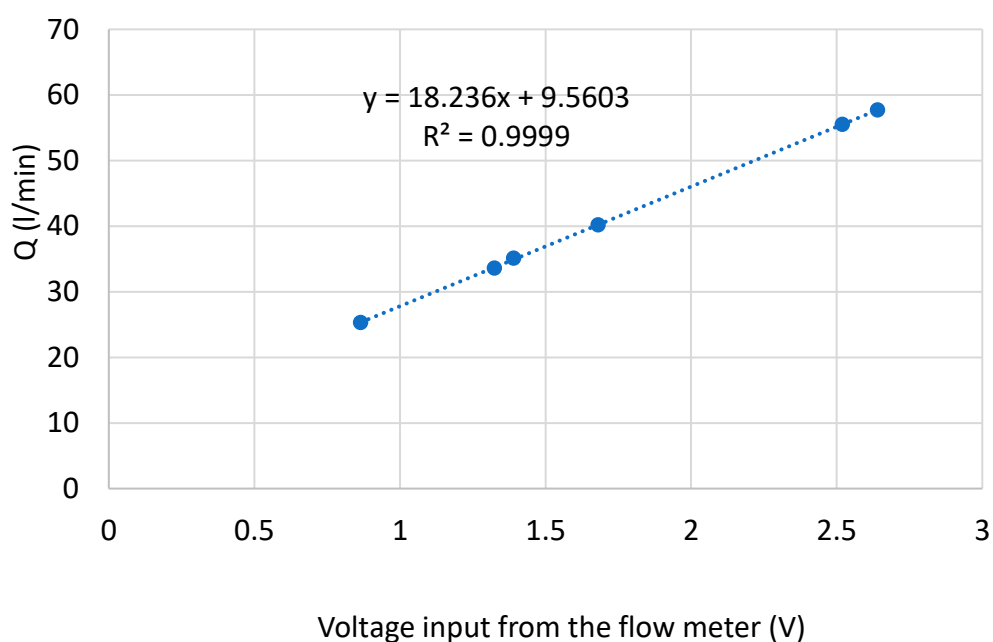
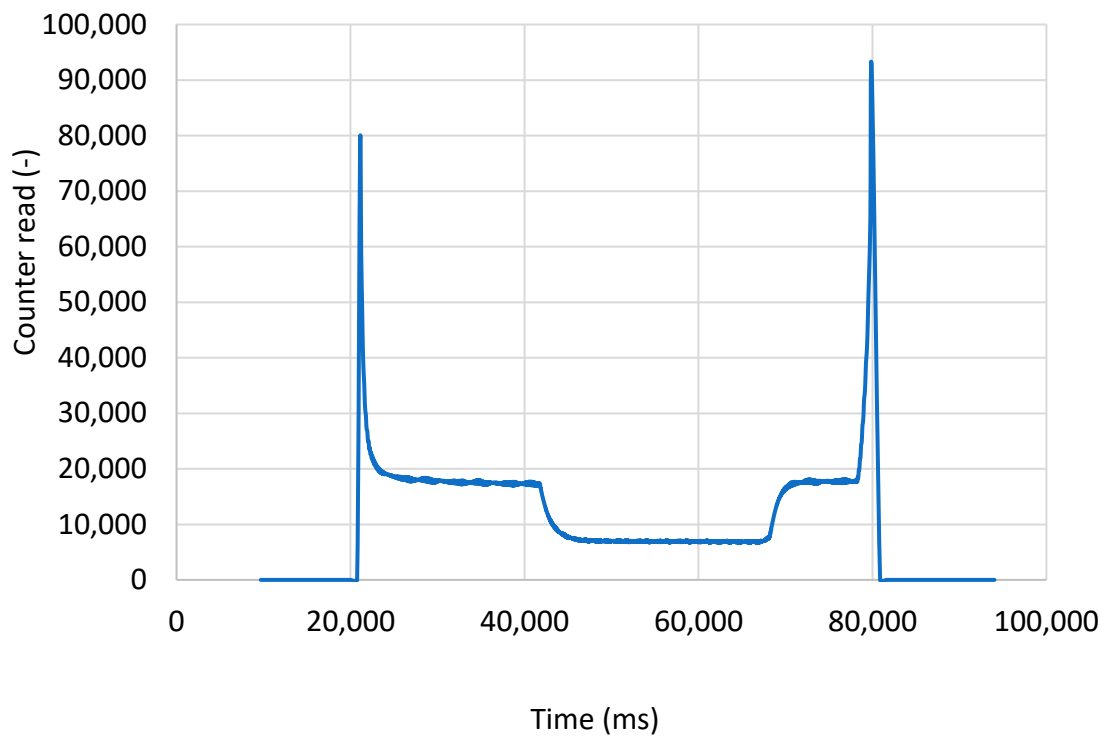


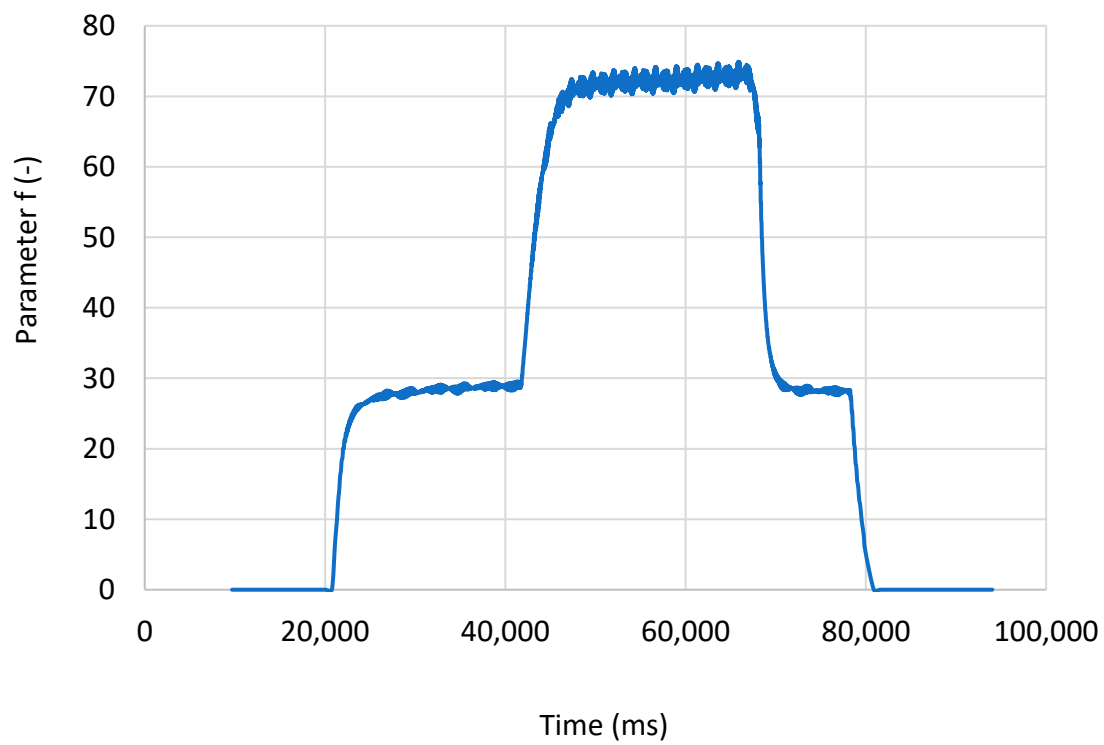
FIGURE 6-15. FLOW METER READINGS CALIBRATION THROUGH THE ARDUINO SHIELD.

- **shaft speed readings:** it was decided to utilize the Hall effect position sensor of the brushless generator powered by the 5 V in-built Arduino power supply to acquire readings of the instantaneous speed of the device through an Impulse Count function. At first the raw data from the Impulse Count function was acquired as seen from Figure 6-16 (a). Subsequently the reciprocal of the same variable was multiplied by an arbitrary value (500,000) in order to generate the f value displayed as Figure 6-16 (b). Lastly, by means of the manual tachometer the variable f was correlated to the actual shaft speed as seen in Figure 6-16 (c);

a)



b)



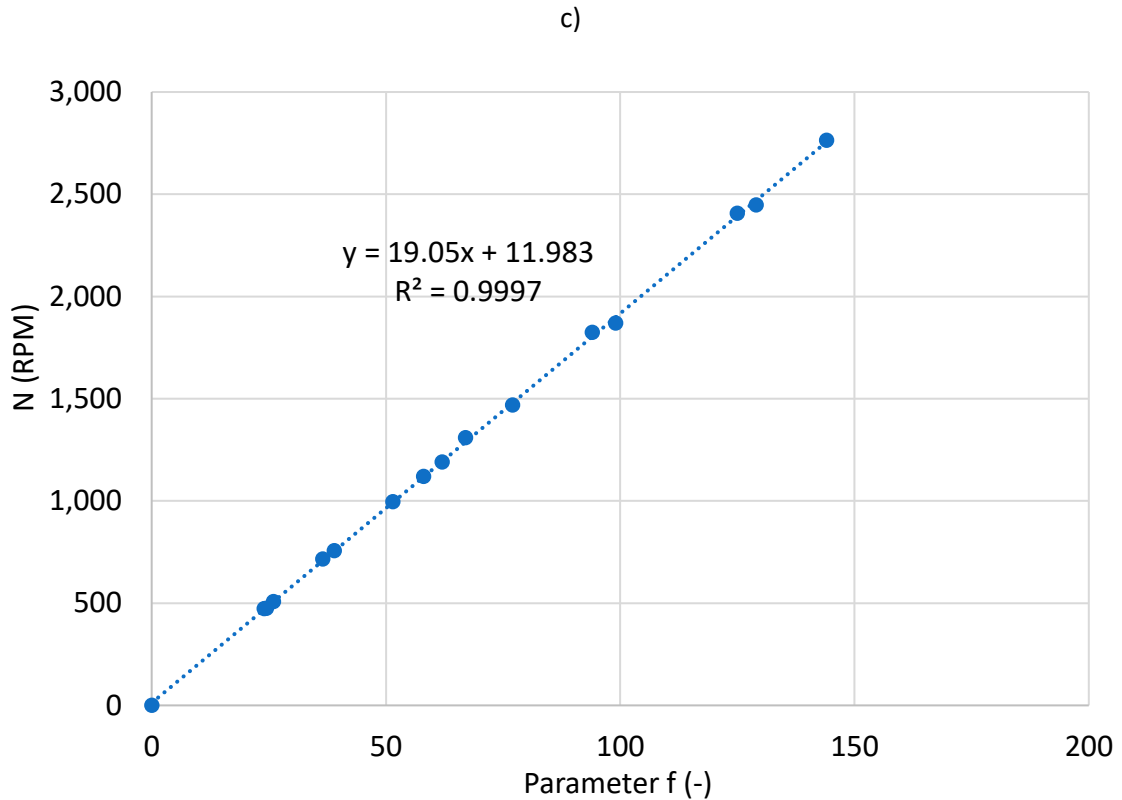
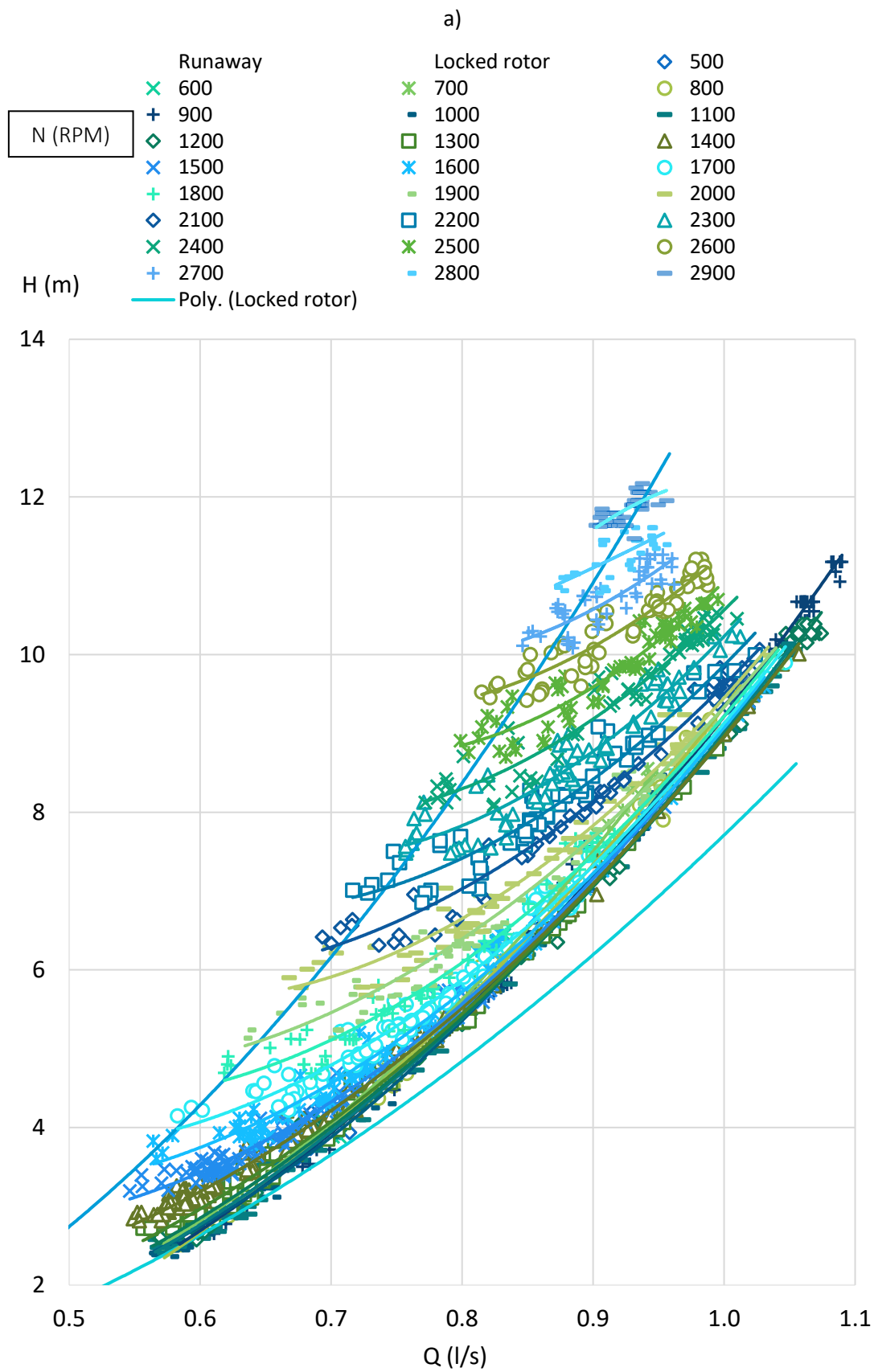


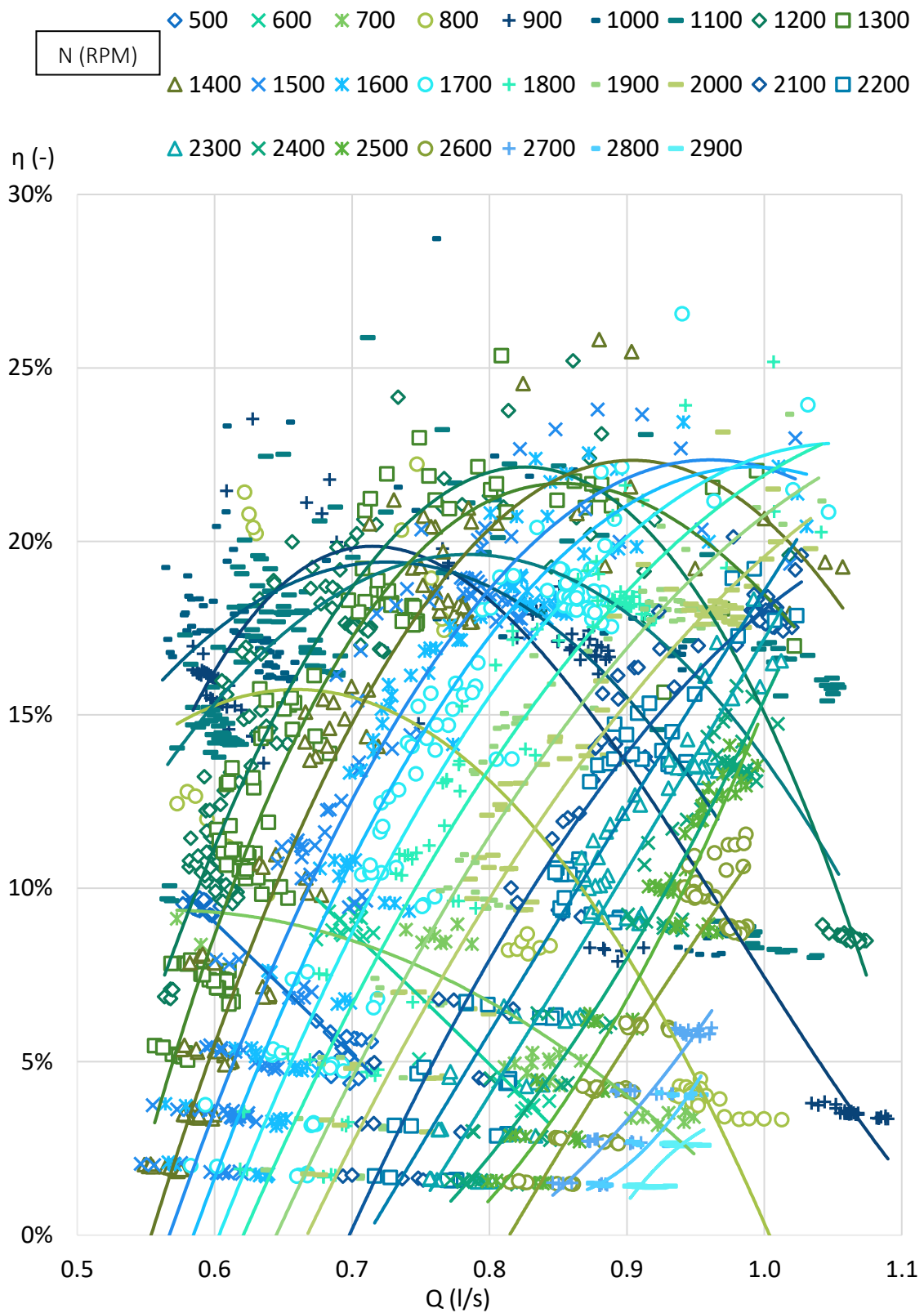
FIGURE 6-16. SHAFT SPEED READINGS CALIBRATION THROUGH THE ARDUINO SHIELD: A) READINGS FROM THE PULSE COUNTER; B) CALCULATED PARAMETER F; C) LINEAR CORRELATION BETWEEN F AND THE ACTUAL SPEED READ WITH THE TACHOMETER.

- **voltage readings:** the voltage at the DC poles of the rectifier was measured directly through the Arduino shield. Since the maximum voltage measured through the multimeter was around 14 V, a simple voltage divider circuit was built through three 10k Ohm resistors in order to keep the maximum input voltage to the Arduino pin within the 0-5 V range;
- **current readings:** the value of current through the dissipation circuit was measured by means of a 0.22 Ohm shunt resistor placed in series to the circuit. The size of the resistor has been chosen as a compromise, since an even smaller resistor would have dissipated less energy thus reducing its interference with the instrumentation, while a larger resistor would have improved the reading resolution of the Arduino and lowered the uncertainty. In fact, the 0.22 Ohm shunt resistor generated voltage drops in the range 0-0.66 V which means that the reading resolution was 135 bytes out of the 1023 bytes available in the whole 0-5 V range (1,023 bytes resolution, 204.6 bytes per V).

The results from the experimental campaign is displayed in Figure 6-17.



b)



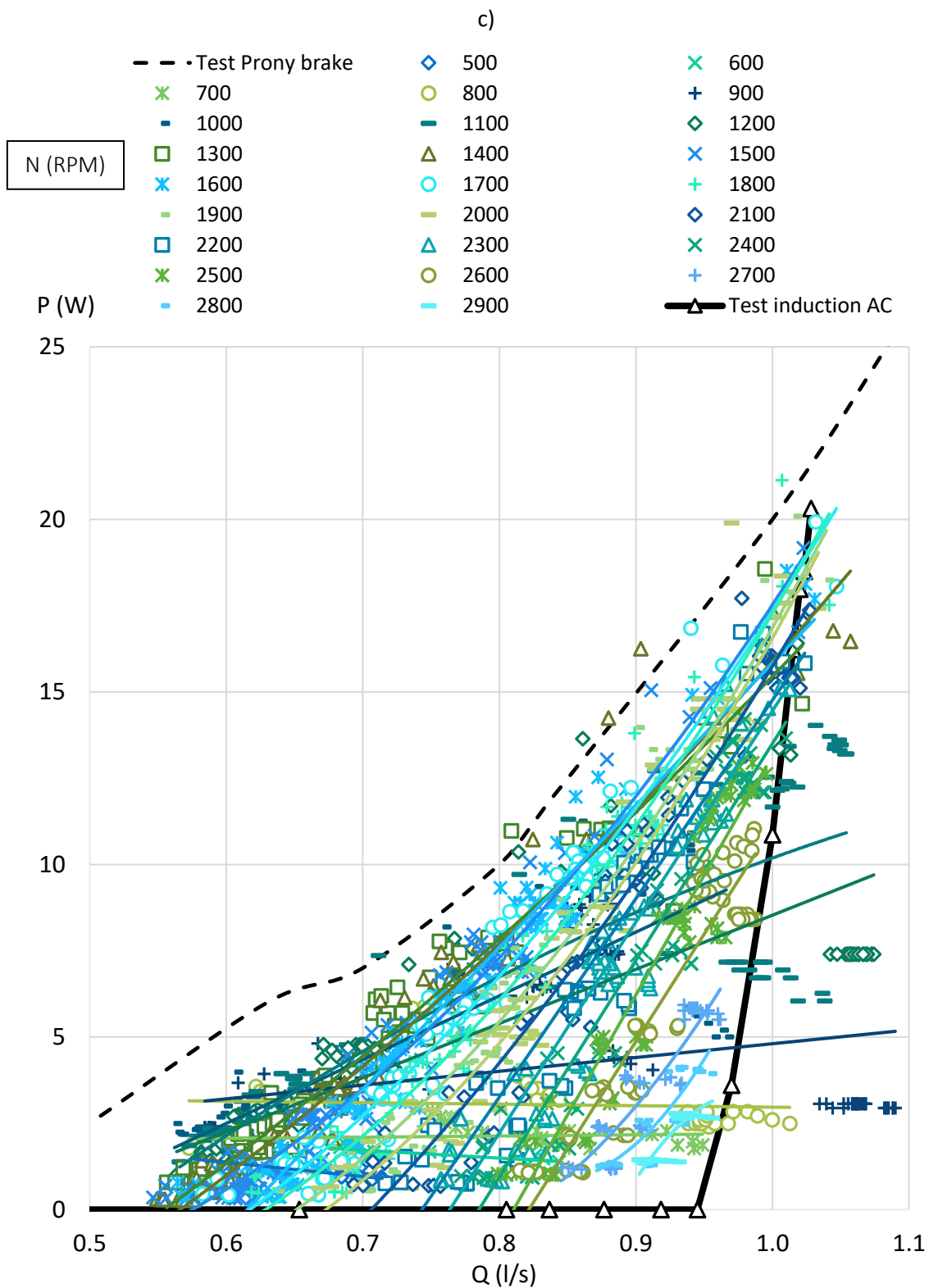


FIGURE 6-17. RESULTS FROM THE MAPPING OF THE IWAKI PAT WITH BRUSHLESS GENERATOR: (A) FLOW-HEAD CURVES; (B) FLOW-EFFICIENCY CURVES; (C) FLOW-POWER CURVES AND COMPARISON WITH THE PREVIOUS EXPERIMENTS.

The charts from Figure 6-17 (a) and Figure 6-17 (b) show the dependence of PAT head and efficiency from the flow rate and that these are closely comparable to the results of the PAT testing with Prony brake displayed at Figure 6-6. From Figure 6-17 (c) it can be seen that the power generated at different speeds with fully closed bypass valve and induction generator during the previous experiment is very similar to the power when the PAT is running with the synchronous generator. However, from the same plot is clear that under identical flow and speed the electricity output from the brushless generator is lower than the mechanical power output measured through the Prony brake. The difference between the two values can be ascribed to the mechanical-electric energy conversion efficiency of the generator which was found to be in the range 35-68% according to the rated speed as seen from Figure 6-18.

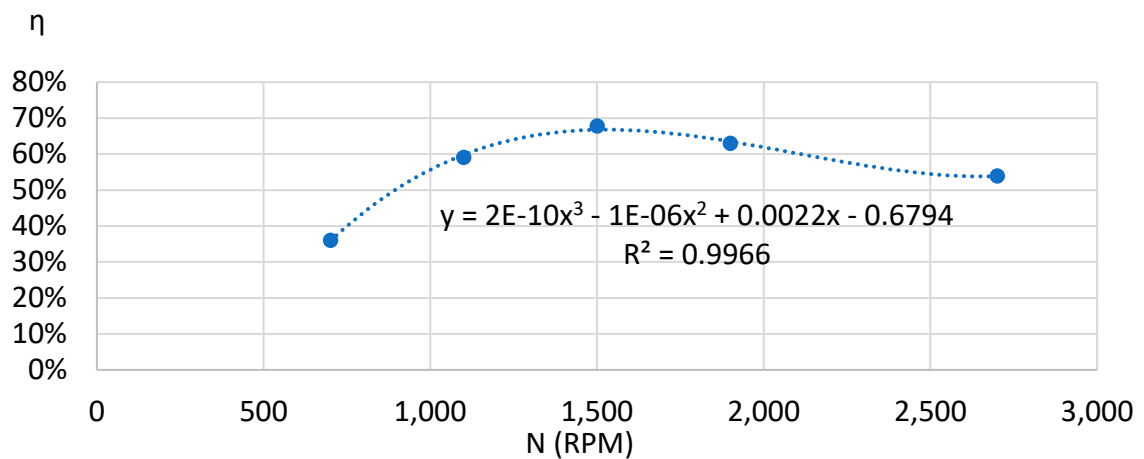


FIGURE 6-18. CALCULATED EFFICIENCY OF THE BRUSHLESS GENERATOR AT DIFFERENT SHAFT SPEEDS.

6.1.7. MPPT TRACKING ALGORITHM WITH DC BRUSHLESS MOTOR

A further step to gain confidence in the design of the Welsh demo site was to simulate on a small scale the logic of the Maximum Power point Tracker (MPPT) of the solar inverter proposed to operate with the PAT and synchronous generator. The interest lies in better understanding how the balance between the mechanical power generated by the water flowing across the turbine and the electric power dictated by the Ohm laws was achieved. In order to do so, the setup from the previous experiment was upgraded via a 8-channel relay with optocoupler and an equivalent number of resistors with different power ratings. The aim was to re-create a basic MPPT device where the Arduino shield after elaborating the data from the various sensors connected (pressure, flow, speed, voltage and current) could autonomously vary the resistance of the dissipative circuit. A picture of the setup is displayed as Figure 6-19.

The rationale behind it is the knowledge that assuming an arbitrary value of hydraulic power available provided by the hydraulic force acting on the turbine according to Equation (1) on Page 44. If the frequency of the synchronous generator is allowed to vary, which in this case happens thanks to the rectifier bridge, then it has to equal the electric power dissipated by the circuit (after subtracting the generator efficiency):

$$P = VI \quad (55)$$

When the MPPT circuit changes the circuit resistance, the voltage across the generator would change and therefore induce a change in the shaft rotational speed. The basic ruling equation for the voltage output of a generator is:

$$V_{\text{gen}} = N/k_n - IR_i \quad (56)$$

where:

- V_{gen} (V) is the voltage at the terminals of the generator
- N (RPM) is the rotational speed
- k_n (RPM/V) is the speed constant of the generator
- I (A) is the current
- R_i (Ohm) is the terminal resistance of the generator

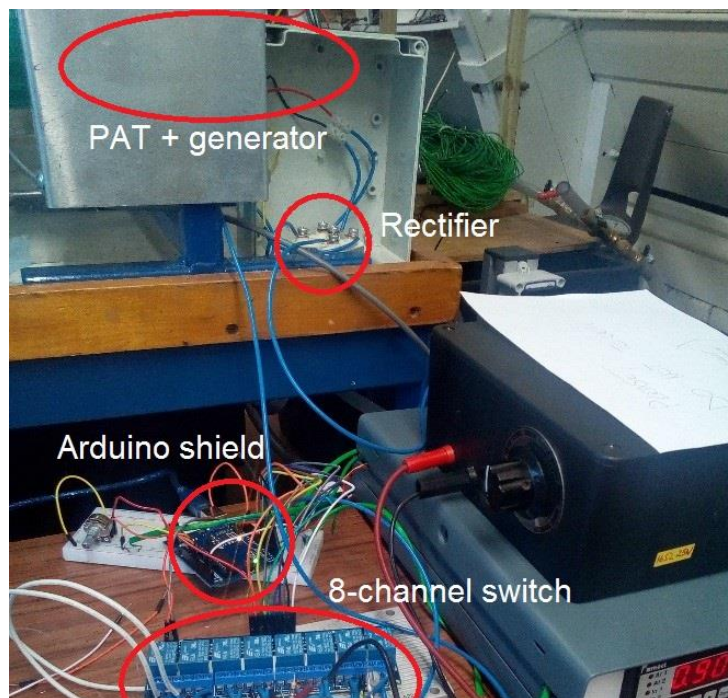


FIGURE 6-19. SETUP FOR MPPT TURBINE REGULATION WITH AN ARDUINO SHIELD.

The logic of the developed MPPT made up of the Arduino and the 8-channel switch is described in Figure 6-20.

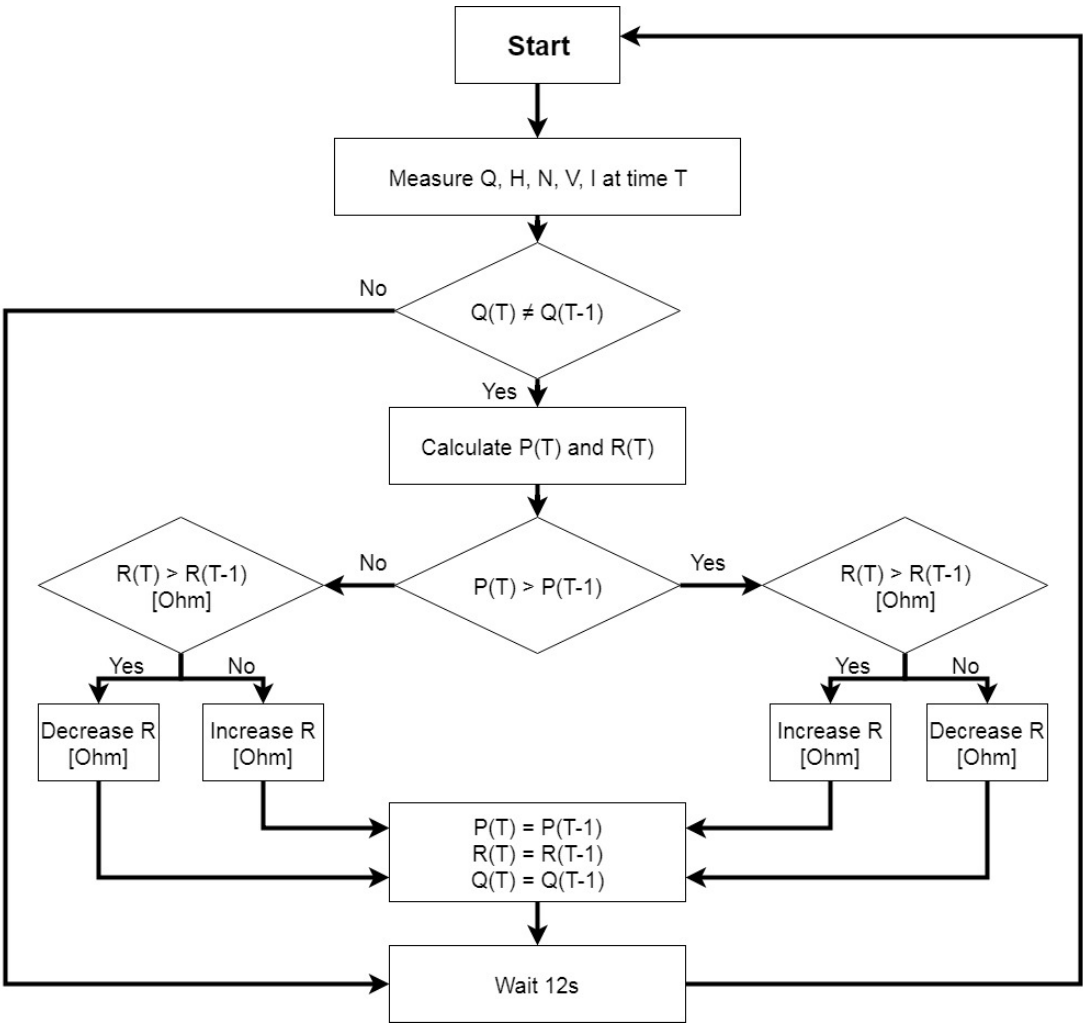


FIGURE 6-20. SCHEMATIC LOGIC OF THE MPPT ALGORITHM DEVELOPED FOR THE ARDUINO SHIELD AND 8-CHANNEL SWITCH.

The basic MPPT logic displayed in Figure 6-20 would result in a continuous oscillation of the shaft speed around the optimal point. In order to fix this, a further operating rule was introduced in order to identify the optimal point and lock the shaft speed in order to maintain it until the flow across the PAT changes.

The chosen values of resistance have been selected for each channel of the switch in order to make the voltage regulation as uniform as possible as shown in Table 6-3.

TABLE 6-3. VALUES OF THE CIRCUIT RESISTANCES OBTAINED FOR EACH CHANNEL OF THE SWITCH

Selected switch channel	Total circuit resistance (Ohm)
None (short circuit except for the resistance of the wires and rectifier)	1.12
1	1.85
2	6.63
3	11.18
4	18.97
5	31.38
6	47.92
7	76.52
8	149.02

A plot showing the functionality of the software is displayed in Figure 6-21, from which it is possible to observe that after each induced perturbation (consisting of manually regulating the bypass valve to increase or decrease the flow across the PAT) the control unit varied the shaft speed until it settled on the value which gave the highest power output under the circumstances. After having identified the relevant optimal point, the system maintained a constant speed until the flow rate across the PAT was varied manually again.

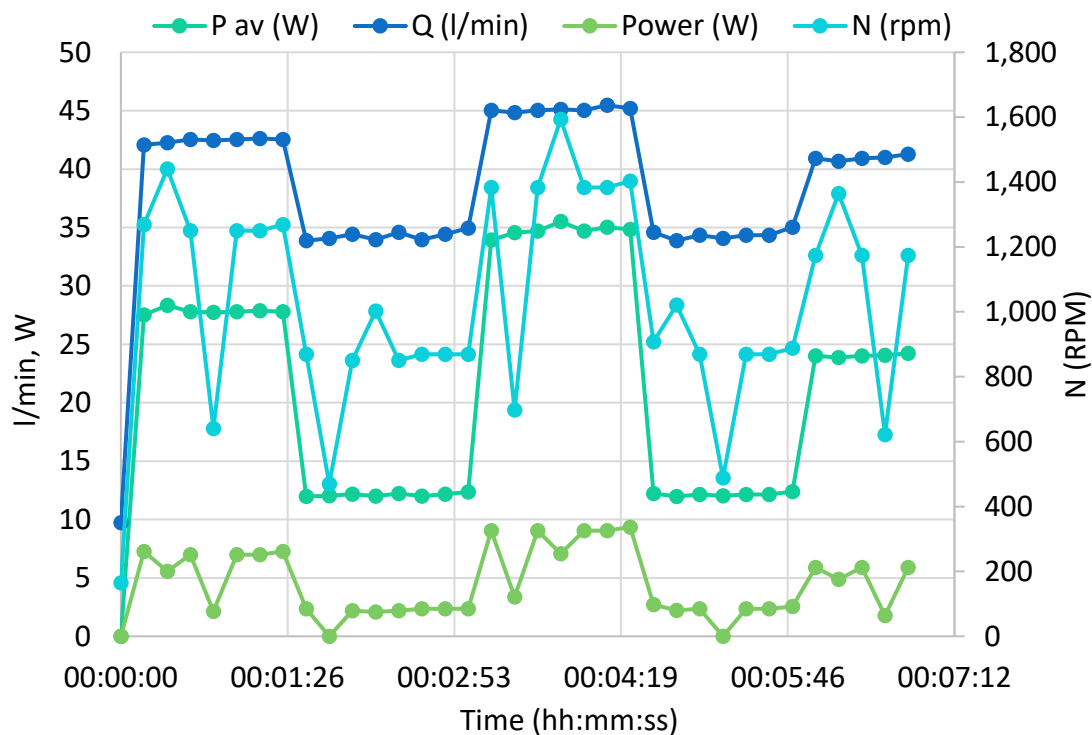


FIGURE 6-21. RESULTS OF THE MPPT PAT CONTROL SYSTEM MONITORED ALONG 7 MINUTES OF OPERATIONS.

The experimental campaign on the small IWAKI PAT can be considered positive, as it has informed both the design of the demo sites (in particular the Welsh one) and the large PAT test rig. The main findings can be summarized as:

- a full turbine characterization was performed, thoroughly assessing the head, efficiency and power generated from the unit under different loads and shaft speed values. The resulting curves closely resembled the behaviour expected from large-scale units;
- the various experimental setups involving the small IWAKI PAT have helped developing and calibrating a wide range of measuring sensors (speed, pressure, flow, current, tension...) which were eventually connected to a centralized data logging unit, providing the basis for the monitoring of the large scale PAT test rig and the demonstration sites;
- besides the characterization of the mere hydraulic behaviour of the turbine, the PAT was tested while producing electric power using two different types of generator: induction asynchronous generator and synchronous one. In fact, the two planned full-scale demo sites are expected to adopt one type of generator each;
- the principle of controlling the PAT speed by means of a synchronous generator coupled to an inverter was successfully demonstrated by building a simple voltage shifter and control unit via an Arduino shield.

6.2. LARGE SCALE PAT TEST

6.2.1. INTRODUCTION

A full-scale PAT test rig was among the key deliverables of the Dŵr Uisce project and a fundamental part of the experimental work needed to validate the PAT decision support system being developed. The planned location for installing it was the ground floor lab space at Simon Perry Building in Trinity College, Dublin. The basic design features of the experimental setup are a maximum pressure of 5 bar and maximum flow rate of 20 l/s and a power output in the range of 2-5 kW (typical of many potential MHP locations in water networks). The test rig design, equipment procurement and rig construction phases lasted in total almost 1.5 years to proceed from the first draft of the design to a fully operational setup. Despite this ending up being a much longer time span than envisaged, the whole process proved almost as informative as the final execution of the experiment.

6.2.2. AIM OF THE EXPERIMENT

The main aim of the experiment was to recreate in the laboratory the flow and pressure conditions at a PRV of a realistic water network characterized by a fluctuating daily demand. Subsequently, the developed decision support software was used to select the most appropriate PAT to be installed in that specific location. The identified PAT was then purchased and thoroughly tested at the rig under variable shaft speed when connected to the grid and generating power by means of an asynchronous induction motor used as generator. In the end, the results from the experimental campaign were compared and contrasted with the expected turbine performances.

6.2.3. DESCRIPTION OF THE EXPERIMENT

A list of the materials and equipment used is displayed in Table 6-4.

TABLE 6-4. LIST OF MATERIALS AND SOFTWARES USED FOR THE LARGE PAT TEST RIG.

Quantity	Description	Make and model	Characteristics
1	9.2 kW Supply pump	Ebara 3D 50-200/9.2 IE3 400/3/50 9.2kW	Cast Iron Pump with IE3 Motor.
1	PAT	KSB ETN 050-032-200 GGSA11GA200114B	Cast Iron bare shaft centrifugal Pump As Turbine, impeller diameter 209 mm
1	Piping	/	DN100 stainless steel
2	Control valves	2x GE 2525 100	100mm 2-piece FB Ductile Iron Ball Valves PN16
1	Shutoff valve	Uniwat VF700PGE	100mm WP Butterfly Valve, EPDM coated
1	Valve actuator	Valbia VB190M	24VDC Electric Actuator with 4-20mA Positioner
1	Water tank	Tricel	2m ² GRP Sectional Cold Water Storage tank
1	Flow meter	Omega FDT506	In-Line Ultrasonic Flow Meter with 4- 20 mA output
1	Pressure gauge	Gems Sensors 3500 Series	Absolute pressure transducer, max. pressure reading 4 bar, current output 4-20 mA
1	Pressure gauge	Gems Sensors 3000 Series	Relative pressure transducer, max. pressure reading 10 bar, current output 4-20 mA
1	Induction generator	ABB 3GBA 132 210- ADD	Standard 5.5 kW induction motor used as generator
1	Inverter / VSD	Mitsubishi Electric FR- A741-5.5K	5.5kW Regenerative Variable Speed Drive (VSD), 12A, 400V
1	Torque and speed meter	Datum Electronics M425-1D	Inline torque meter, range 0-100 Nm and 0-10,000 RPM

1	Data acquisition card	National Instruments NI USB-6009	8 Inputs, 14-bit, Multifunction I/O
1	Data logging software	LabView	/

The basic scheme of the hydraulic circuit is displayed in Figure 6-22, with the locations of the data acquisition sensors to be deployed highlighted.

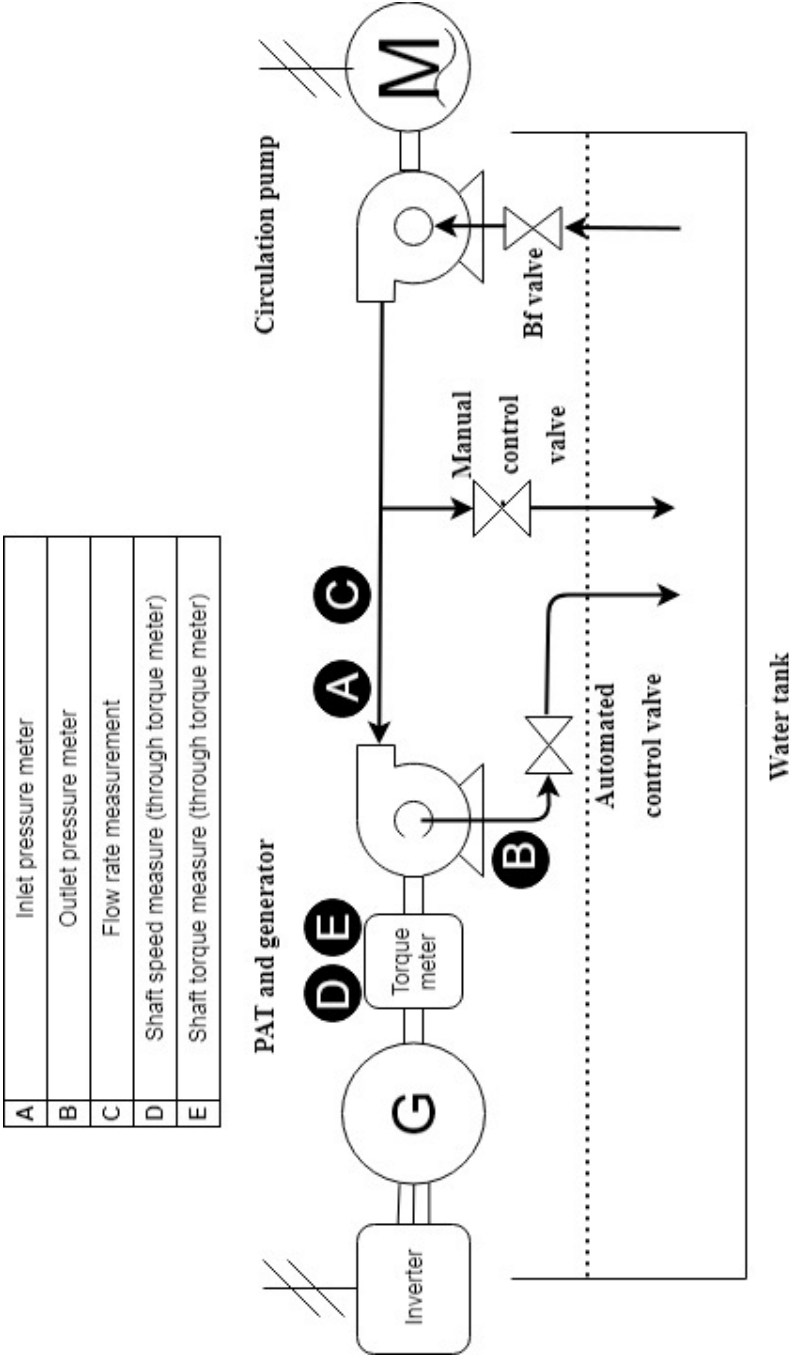


FIGURE 6-22. IDEAL SCHEMATIC OF THE LARGE PAT TEST RIG.

Over the first six months of work a number of iterations of the design were produced, initially including a pressure vessel to stabilize the flow and adapting the rig to the spaces available in the lab. An overview of these is shown in Figure 6-23. In the end it was decided to proceed with a slightly improved design eliminating the air vessel and changing the dimensions of the test rig in order to fit it in another corner of the laboratory.

Additionally, more clearance and a longer stretch of straight pipe were added before and after the turbine in order to reduce turbulence. The flanged ultrasonic flow meter was installed in the middle of the horizontal inlet pipe with 1 m of straight pipe before it and 0.5 m after in order to meet the installation requirements set by its manufacturer. The main pipe ending was capped after a tee junction by means of a blank flange in order to enable in the future the installation of a second turbine in parallel. The final configuration is displayed in Figure 6-24.

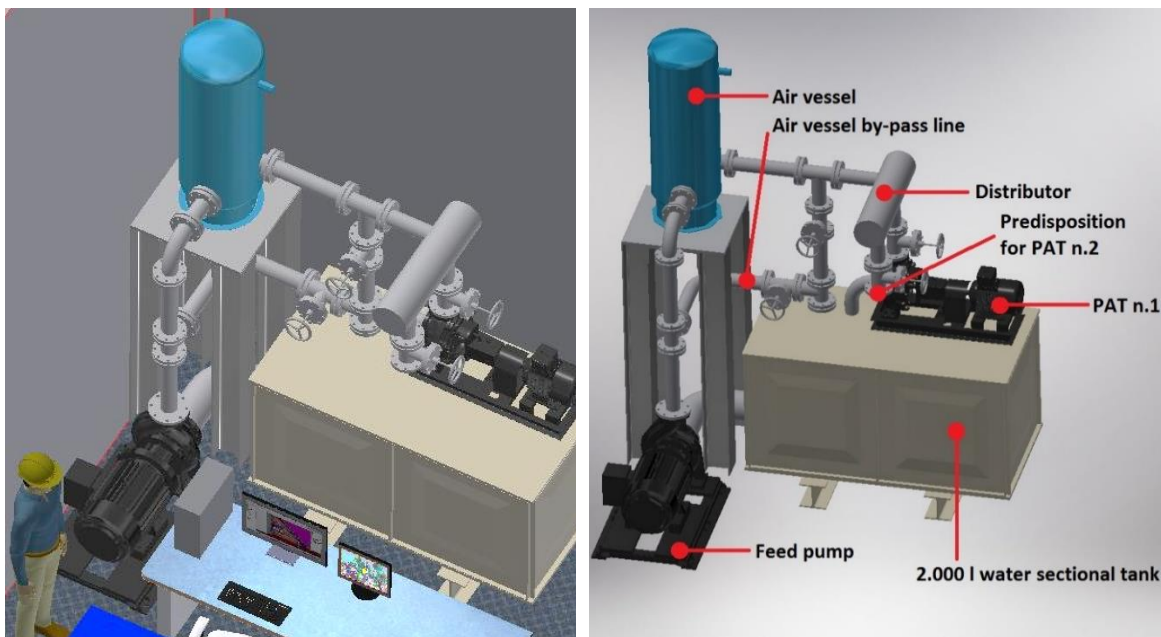


FIGURE 6-23. INITIAL ITERATIONS OF THE DESIGN OF THE PAT TEST RIG USING A 2,000 L MODULAR GRP TANK AS RESERVOIR.

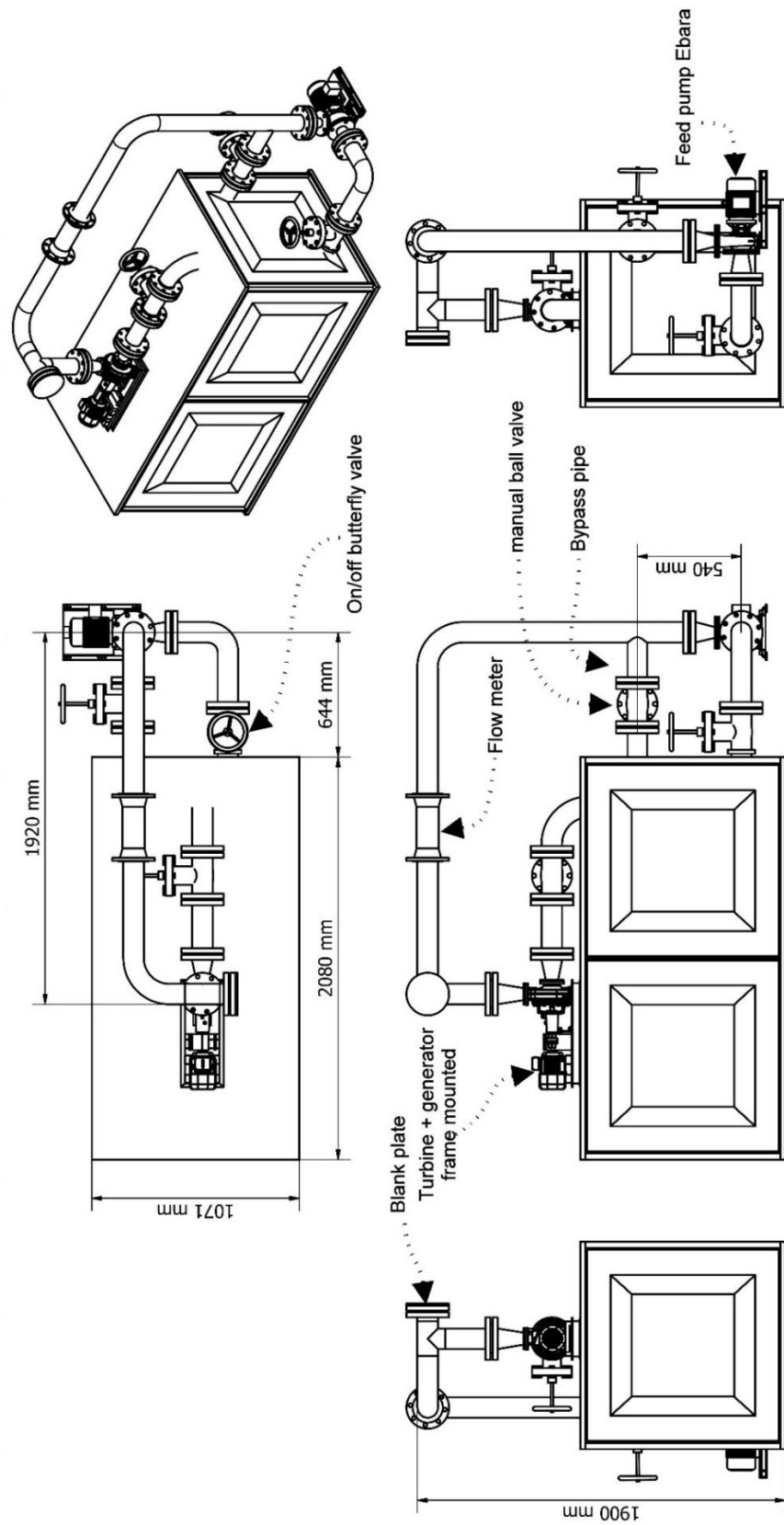


FIGURE 6-24. FINAL DESIGN OF THE PAT TEST RIG.

6.2.4. DESIGN PRINCIPLES

The primary purposes behind the construction of the lab test rig were:

- verify and validate the PAT design software being developed;
- demonstrate the concept and help the dissemination of the project.

Originally it was decided to reproduce on a small scale the demo site to be developed with Welsh Water at the outfall of the Five Fords water treatment works described in Annex 1.1. However, due to the unwillingness of the operators to pressurize the outfall pipeline, the site had to be dropped in favour of a different one and therefore it was decided to re-design the test rig in order to be as flexible as possible. This included the possibility of testing a variety of different hydraulic turbines up to 4 kW and the predisposition for testing two turbines in parallel.

In order to ensure the replicability and representativeness of the results from the test rig, it was decided to base its design on a real case-study consisting of an existing PRV located in Ireland whose data records were collected during a previous project (Hydro BPT). The PRV itself is located in Thomas Court within the network managed by Dublin City Council on behalf of Irish Water. The average pressure drop and flow through the selected PRV corresponded to 75 m and 200 l/s, with a maximum recorded flow of 500 l/s. The flow-head and flow duration curves of the PRV are displayed respectively in Figure 6-25 and Figure 6-26.

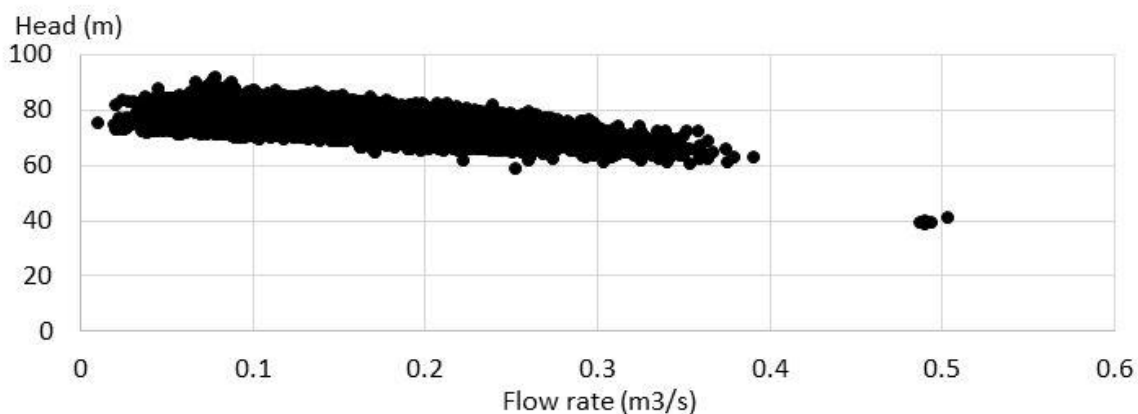


FIGURE 6-25. FLOW-HEAD RECORDED VALUES ACROSS THE PRV INSTALLED AT THOMAS COURT, IRELAND.

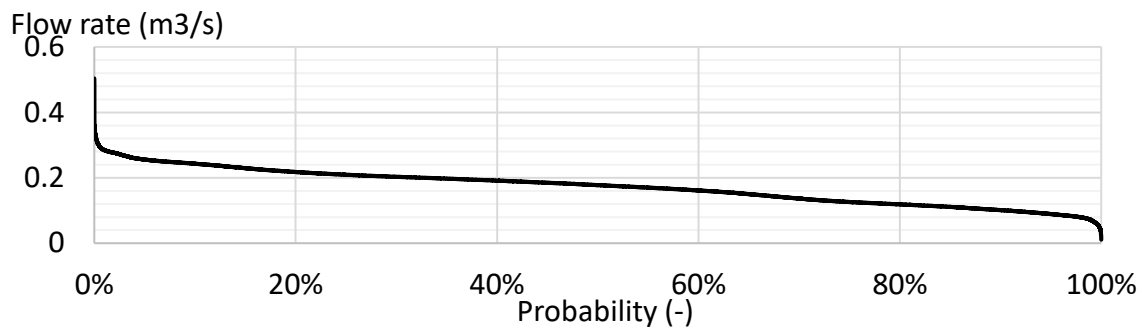


FIGURE 6-26. MEASURED FLOW DURATION CURVE AT THE PRV INSTALLED AT THOMAS COURT, IRELAND.

While the original PRV may well be suitable for a large turbine with capacity close to 200 kW, it was decided to scale down its flow and head value of a factor of 24 and 1.8 respectively in order to match the working parameters of the Ebara supply pump purchased. Besides, since the majority of PRVs in water supply networks of Ireland and Wales were found to have a potential of less than 5 kW it was decided to build a smaller and more representative experimental setup (Gallagher, et al., 2015b). The resulting Q-H and flow duration curves are described in Figure 6-27.

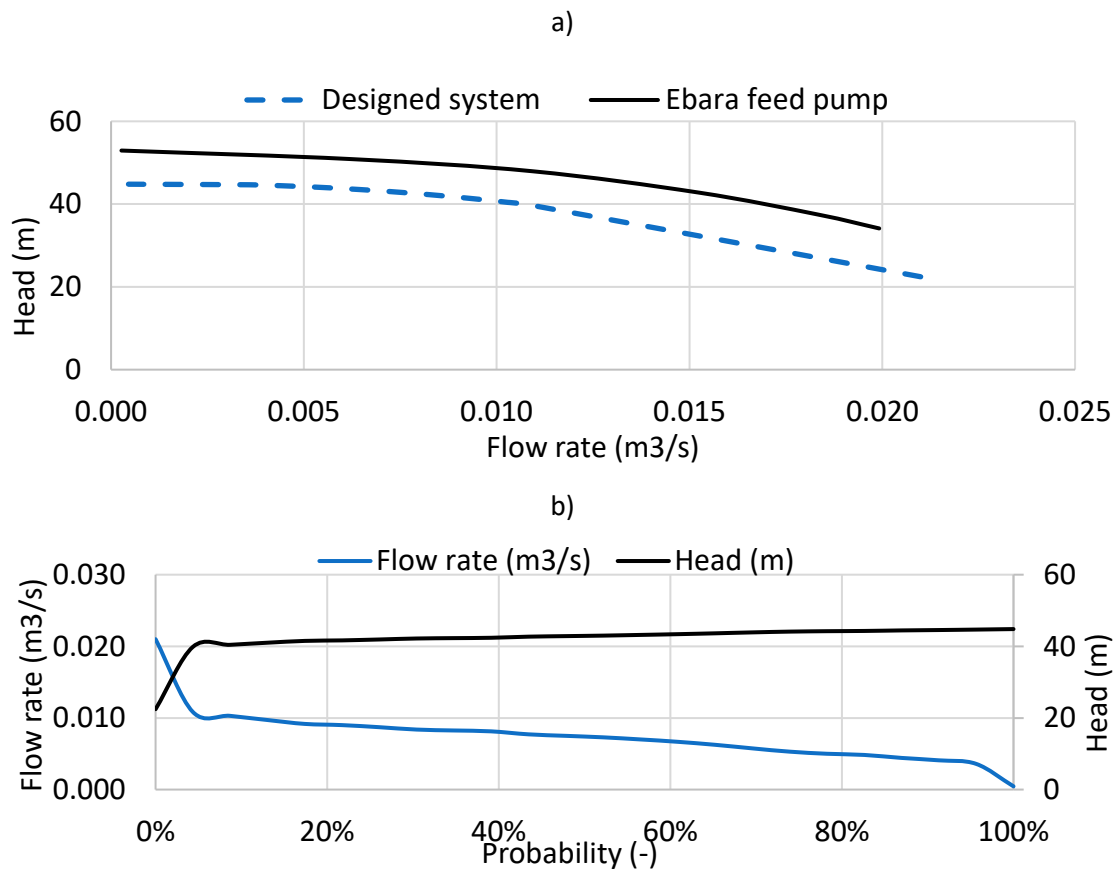


FIGURE 6-27. CHARACTERISTICS OF THE DESIGNED SYSTEM: A) FLOW-HEAD CURVE; B) FLOW DURATION CURVE.

The final PAT test rig is displayed under different stages of assembly in Figure 6-28.

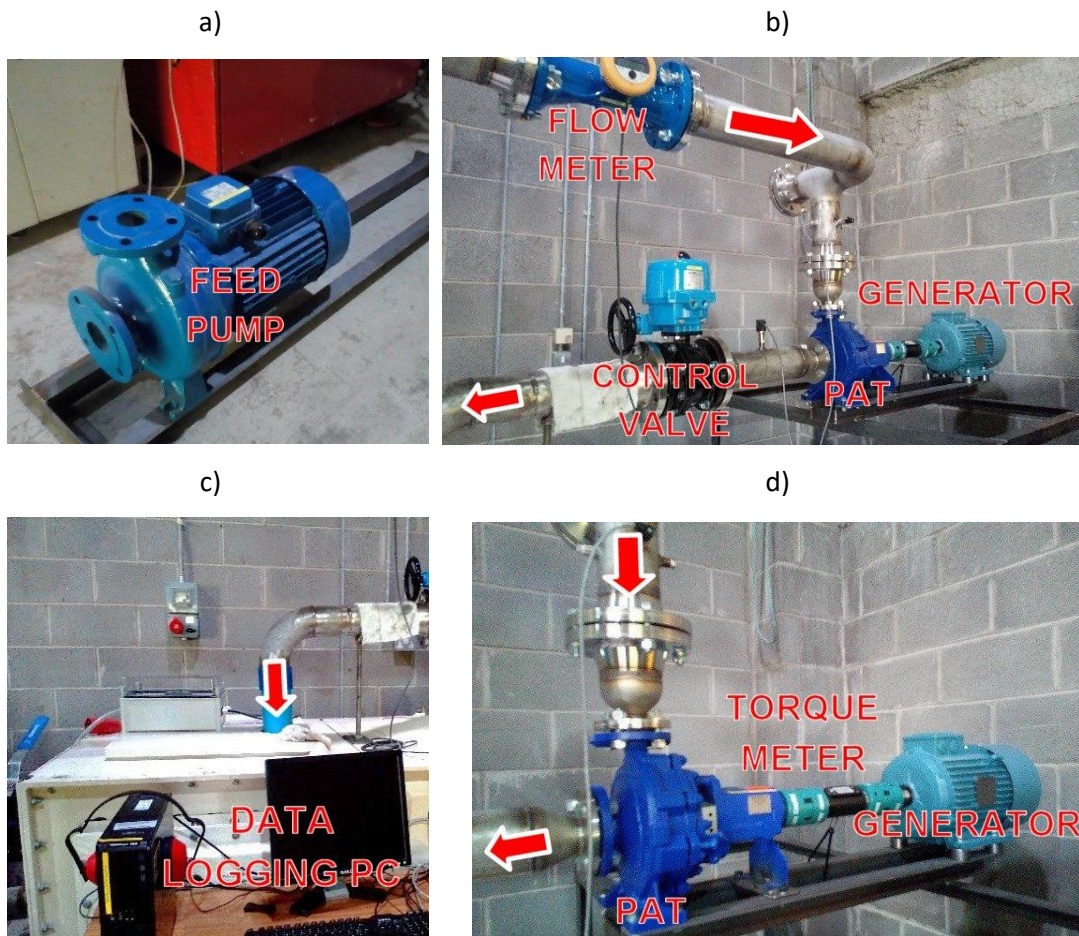


FIGURE 6-28. LAYOUT OF THE EXPERIMENTAL SETUP UNDER DIFFERENT PHASES OF CONSTRUCTION: A) INSTALLATION OF THE SUPPLY PUMP ON A CUSTOM MADE FRAME; B) VIEW OF THE PAT, FLOW METER AND CONTROL VALVE; C) VIEW OF THE DATA LOGGING APPARATUS AND WATER TANK; D) DETAILED VIEW OF THE TURBOGENERATOR.

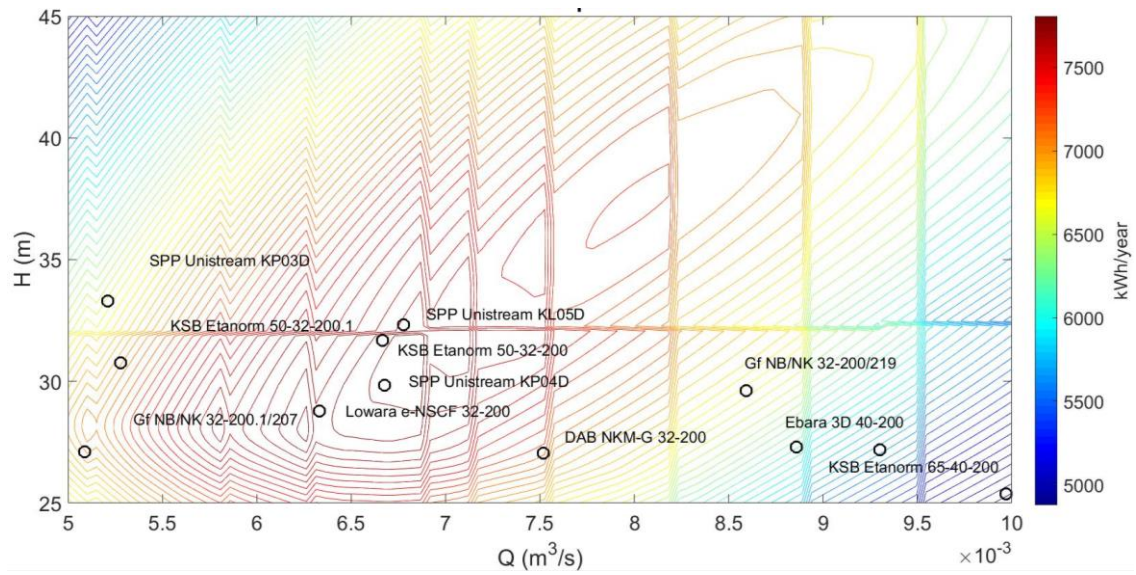
6.2.5. PAT SELECTION

The PAT decision support software was utilized in order to select the most appropriate centrifugal pump to work in the system previously described. The input conditions were the flow duration and head curves displayed in Figure 6-27, while the other specified inputs were:

- Shaft speed: 1,500 RPM
- No cap to the peak power
- VAT 20%
- Electricity cost 0.18 €/kWh

The results of the software are displayed in Figure 6-29.

a)



b)

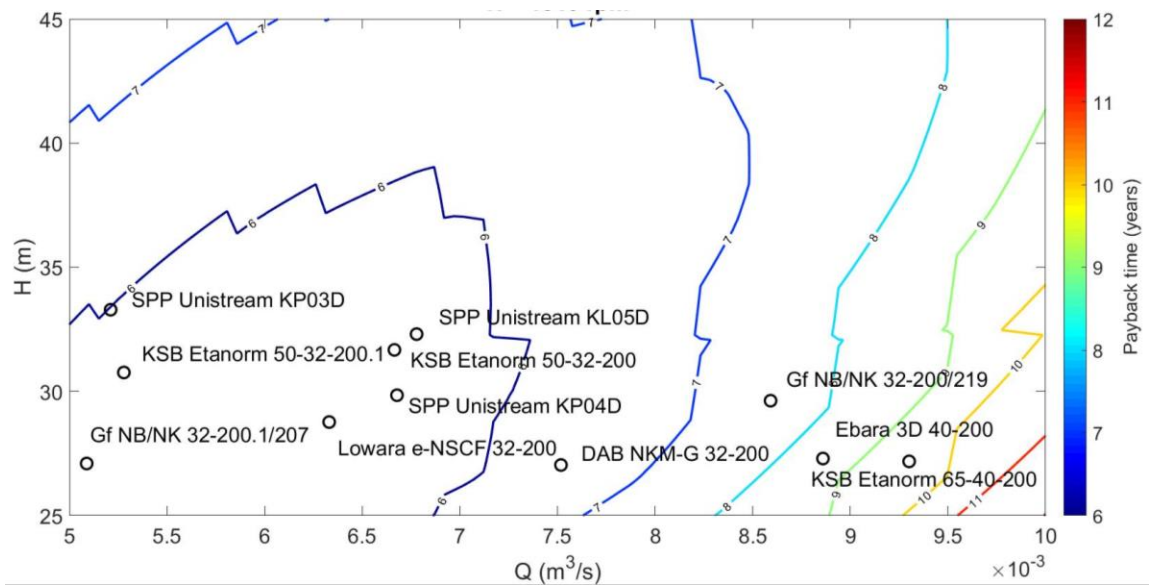


FIGURE 6-29. RESULTS FROM THE PAT SELECTION SOFTWARE: A) PAT YEARLY ENERGY OUTPUT MAXIMIZATION; B) PAT INSTALLATION OVERALL PAYBACK TIME MINIMIZATION.

As can be seen in Figure 6-29 (a) and (b), the ideal PAT according to the energy yield maximization would be a standard centrifugal unit having BEP as a turbine in correspondence of about 8 l/s and

37 m which would output some 7,800 kWh per year. However, such coordinates did not correspond exactly to a standard normalized unit available on the market. Instead, from Figure 6-29 (c) and (d) it is possible to notice how many PATs exist for which the Payback time would be lower than 6 years.

Other numerical results from the software included:

- Optimal yield: 7,837 kWh/year
- Expected PAT flow-to-wire efficiency: 45.3%
- PBT of the installation: 5.3 years
- Ns of the optimal PAT: 9.5 (-)
- Peak power of the optimal PAT: 0.9 kW
- Estimated cost of PAT + generator incl. VAT: 1,755 €
- Estimated total cost of the installation incl. VAT: 7,018 €

The choice of PAT fell therefore on a KSB Etanorm 50-32-200 with maximum impeller diameter (209 mm), expected to output around 7,600 kWh/year and have maximum efficiency with a flow of 6.6 l/s and 32 m.

Instead of purchasing an equivalent 50-32-200 normalized pump from another manufacturer the choice fell on a KSB product since it was the only fabricator able to provide on request the characteristic curves of their unit in turbine mode. Therefore, this choice would offer the opportunity not only to validate the decision support software but also to verify the accuracy of the performance charts made available by KSB.

Eventually the accuracy of the software surpassed the one of the characteristic curved provided by KSB when compared to the real performances of their unit. The datasheet of the selected KSB Etanorm 50-32-200 in turbine mode is displayed in Figure 6-30.

Baureihe-Größe Type-Size Modèle	Tipo Serie Tipo	Nennrehzahl Nom. speed Vitesse nom.	Velocità di rotazione nom. Nominaal foerental Revoluciones nom.	Laufrad-Ø Impeller diameter Diamètre de roue	Ø Girante Ø Waaler Ø Rodete	 KSB KSB LIMITED 2 Cotton Way LOUGHBOROUGH Leicestershire LE11 5TF
Etanorm 32-200 Turbine		1520 1/min		209 mm		
Projekt Project Projet	Progetto Projekt Proyecto	Angebots-Nr. Project No. No. de l'offre	Offerta-No. Offertenr. Oferta-No.	Pos.-Nr. Item No. No. de pos.	Pos.Nr. Positenr. Pos.-Nr.	
Trinity College Dublin		4003215558		PaT for Lab Test Rig		

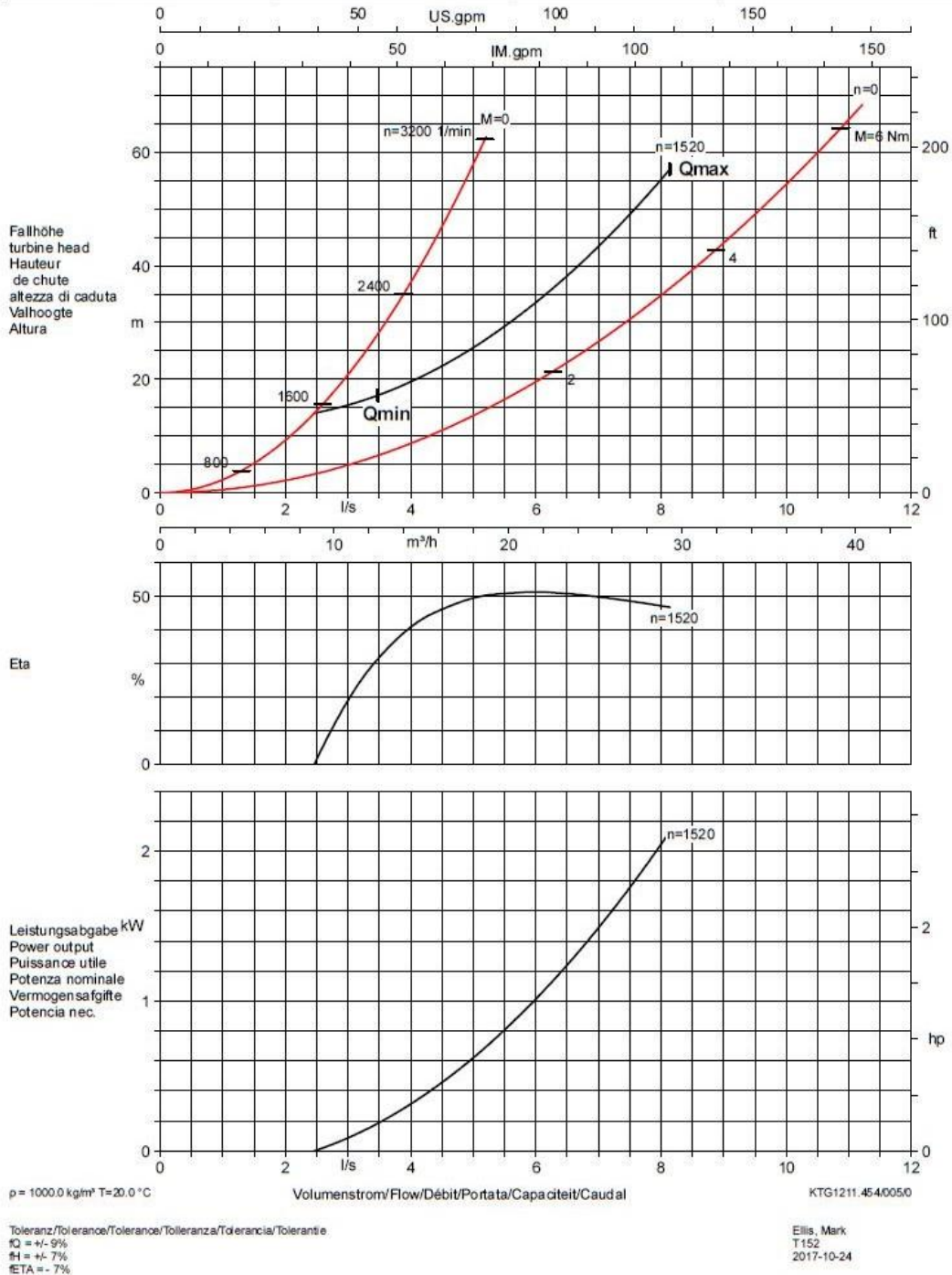


FIGURE 6-30. DATASHEET OF THE KSB ETANORM 50-32-200 PAT PROVIDED BY THE MANUFACTURER.

6.2.6. EXPERIMENTAL RESULTS

6.2.6.1. Equations used

The following equations have been used to calculate the necessary parameters:

- Q (l/s) : measured directly from the flow meter
- H (m): according to Equation (51) at Page 153
- T (Nm): measured directly
- N (RPM): measured directly
- P (kW): measured directly
- P available (kW):

$$P_{av} = 9.81 \left(\frac{m}{s^2} \right) \cdot Q \left(\frac{m^3}{s} \right) \cdot H(m) \quad (57)$$

6.2.6.2. Instrumental uncertainty

The uncertainty of the data collected from the test rig can be evaluated from the accuracy of the instrumentation used as described in Table 6-5.

TABLE 6-5. ACCURACY OF THE INSTRUMENTATION EMPLOYED IN THE LARGE TEST RIG.

Device	Uncertainty	Notes
Ultrasonic flow meter	+/- 3%	In the range 3.3-33.3 l/s, while it is 5% between 0.5-3.3 l/s
Pressure gauge HP	+/- 0.025 bar	0.25% of the scale (10 bar)
Pressure gauge LP	+/- 0.01 bar	0.25% of the scale (4 bar)
Torque meter – torque	+/- 0.1%	-
Torque meter - speed	+/- 0.1%	-

Therefore, the uncertainties for each variable can be evaluated as:

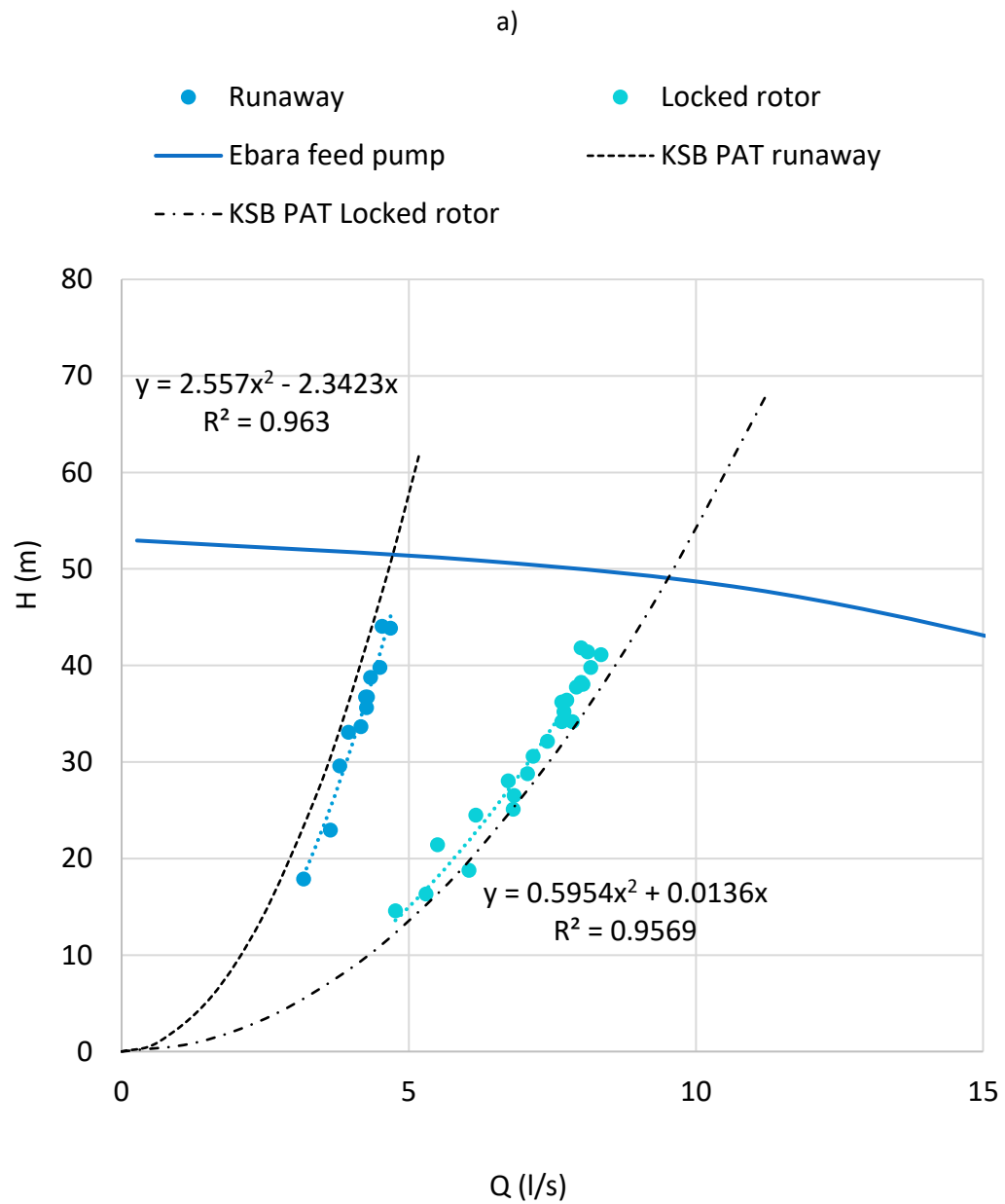
- Flow (Q): +/- 3%
- Head (H): +/- 0.5%
- Power available (P_{av}): +/- 3.5%
- Power output (P): +/- 0.2%
- Efficiency (η): +/- 3.7%

6.2.6.3. Locked rotor and runaway

Similar to what has been conducted for the small-scale test rig, at first the runaway and locked rotor curves of the turbine have been calculated following the described methodology:

- runaway - the PAT was left to spin freely leaving the generator unconnected to the VSD;
- locked rotor - a self-grip wrench was used in order to block the shaft connecting the PAT and the generator.

The results from these tests are shown in Figure 6-31.



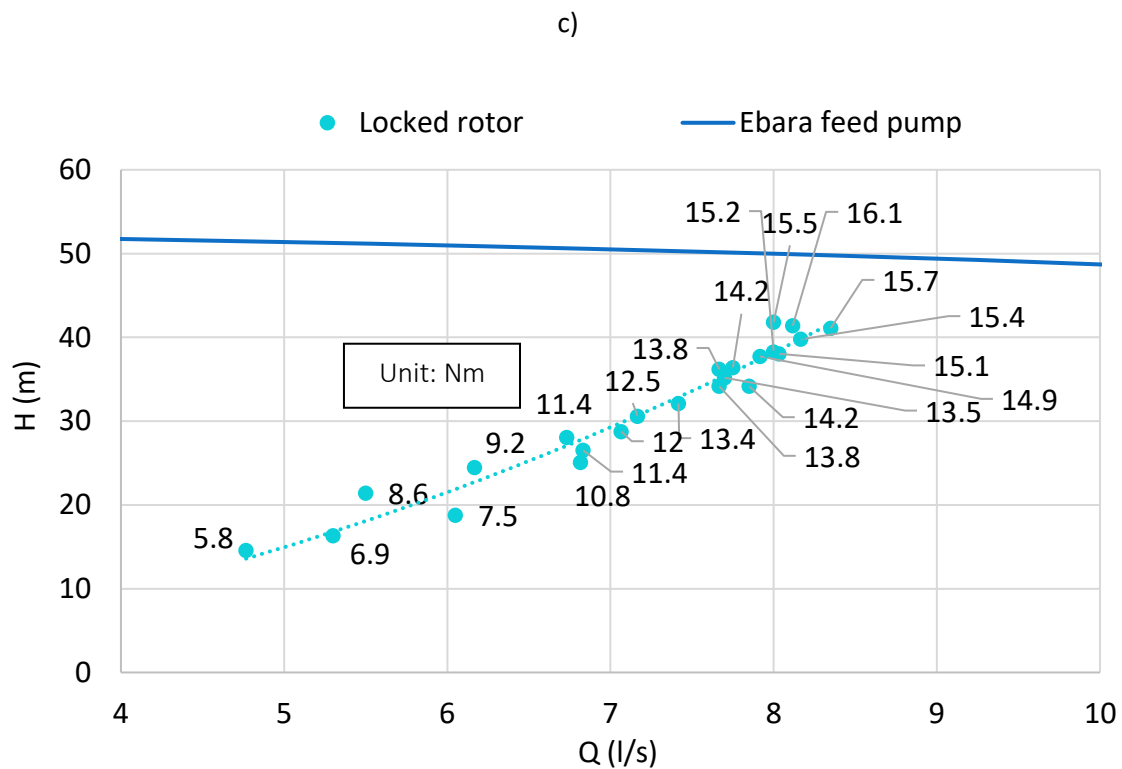
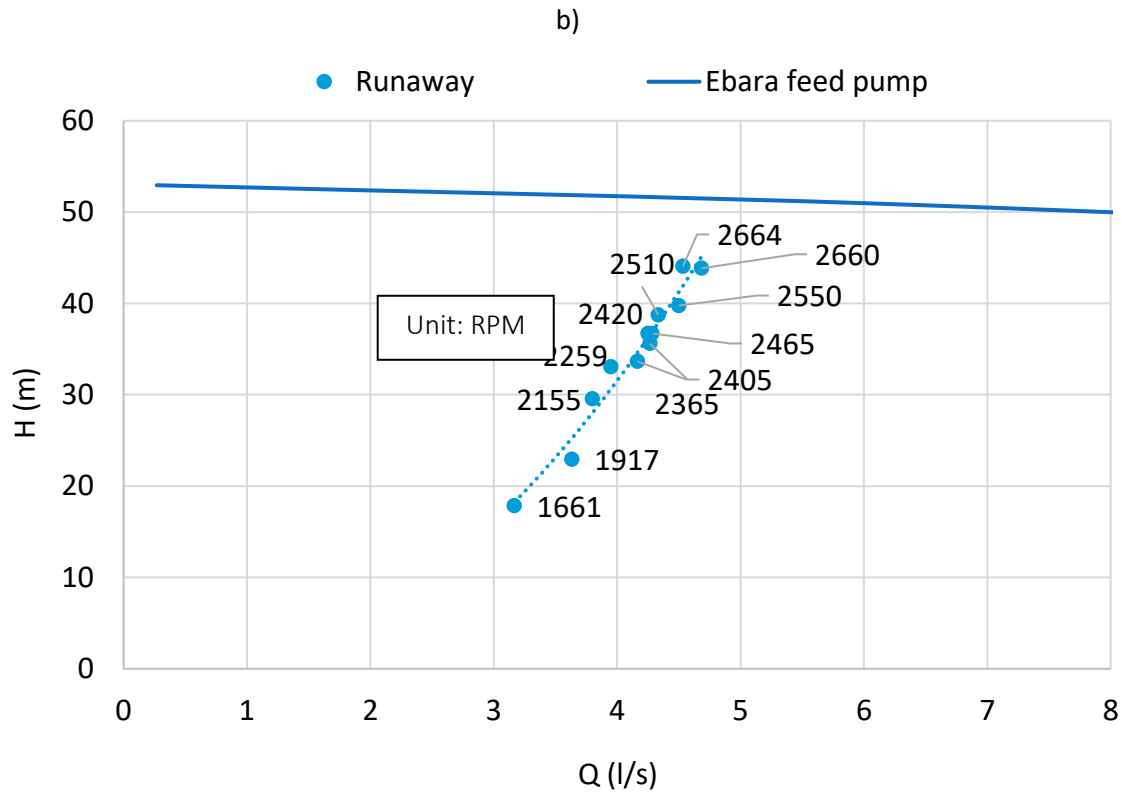


FIGURE 6-31. RESULTS FROM THE RUNAWAY AND LOCKED ROTOR CURVES TESTS: A) OVERALL RESULTS AND COMPARISON WITH THE CURVES PROVIDED BY THE PAT MANUFACTURER; B) RUNAWAY CURVE AND RELATIVE SHAFT SPEED; C) LOCKED ROTOR CURVE AND RELATIVE STATIC TORQUE.

The results from the experiment indicate that the runaway and locked rotor curves of the PAT in the flow-head chart follow closely those indicated by the manufacturer (KSB) but without a perfect correspondence.

The additional charts of Figure 6-31 (a) and (b) instead showed that with the actual configuration the speed of the PAT under runaway conditions can reach up to 2,664 RPM and the resistive torque at shaft when operated with locked rotor is up to 16.1 Nm. While the measured maximum runaway speed matched the datasheet provided by KSB, the maximum locked rotor torque was found to be nearly four times higher than what expected from Figure 6-30 (16.1 instead of 3.9 Nm). This discrepancy could have had significant implication if a braking system was installed as a safety feature to bring the turbine to standstill in case of a grid failure, as the brake may have ended up being undersized for the application.

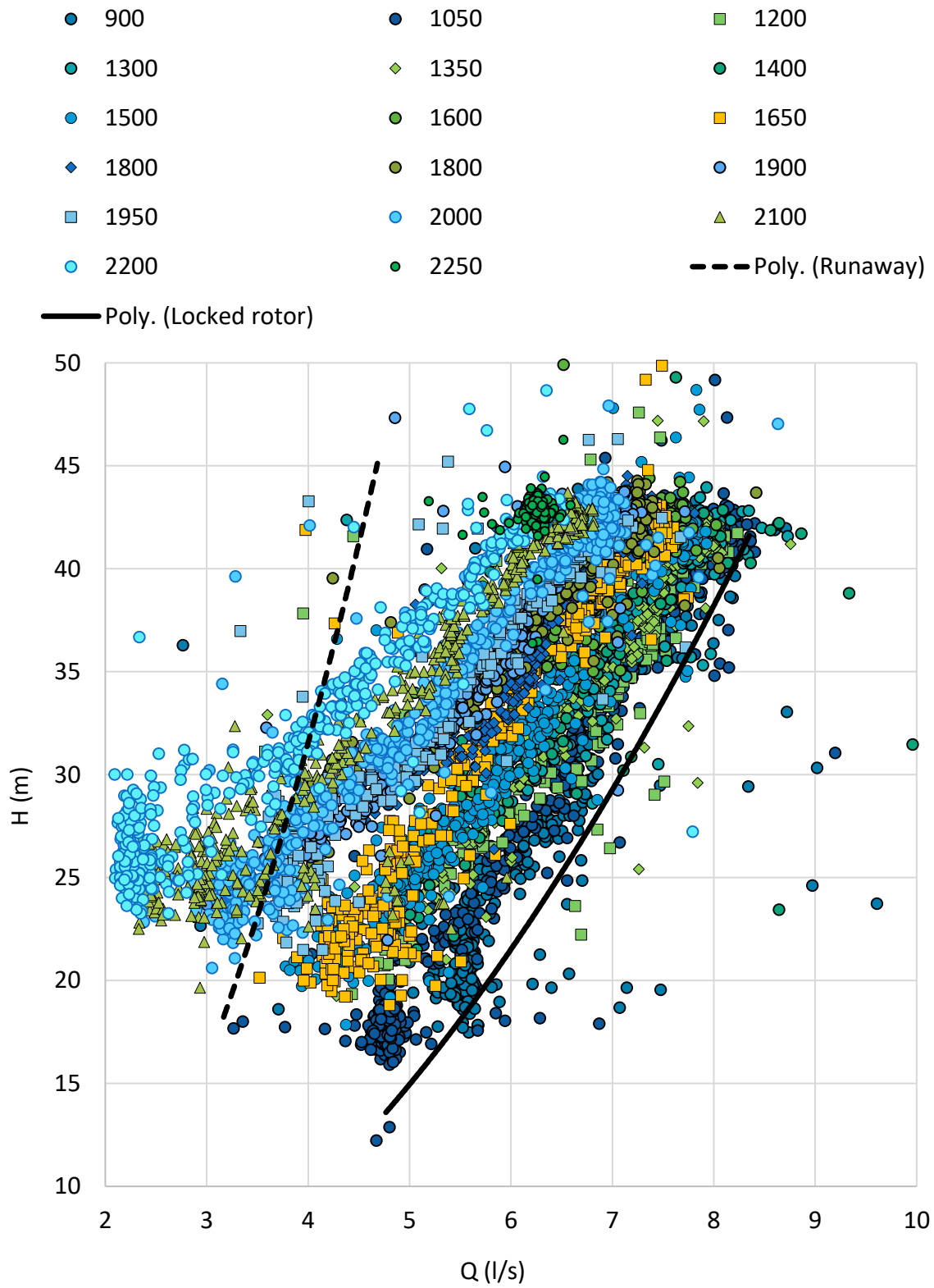
6.2.6.4. PAT characterization

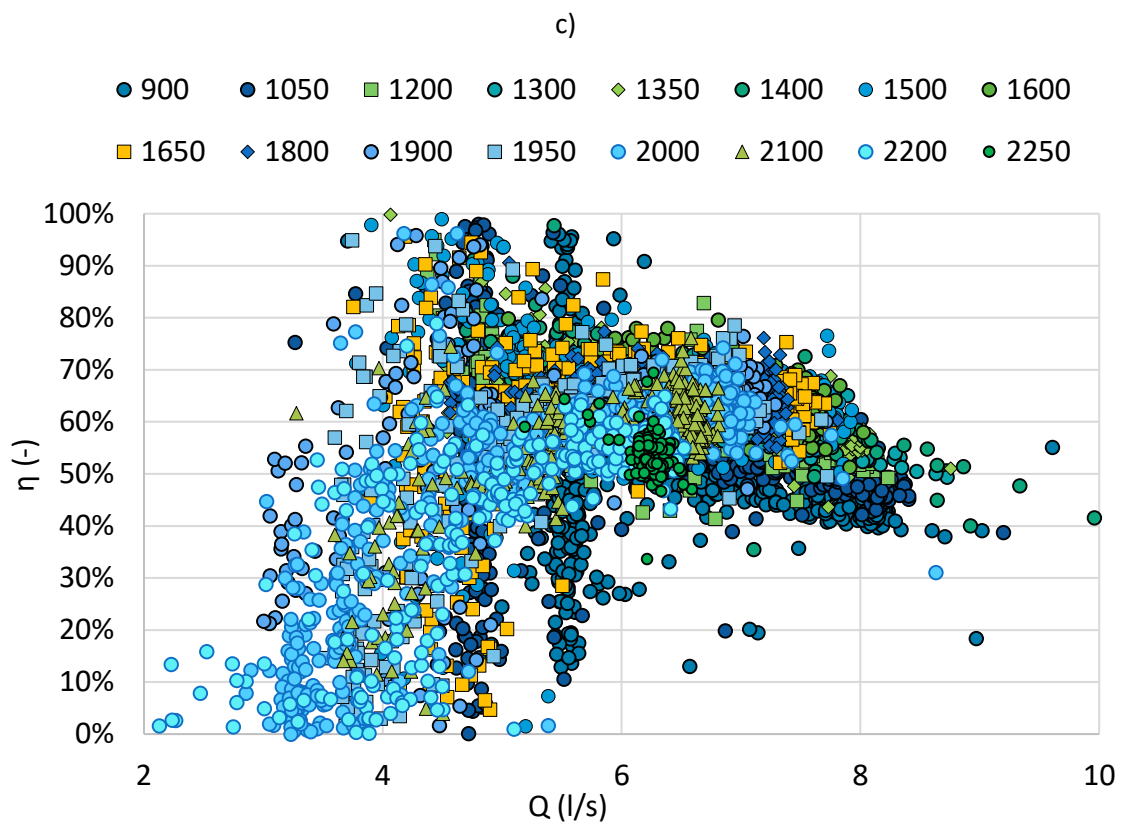
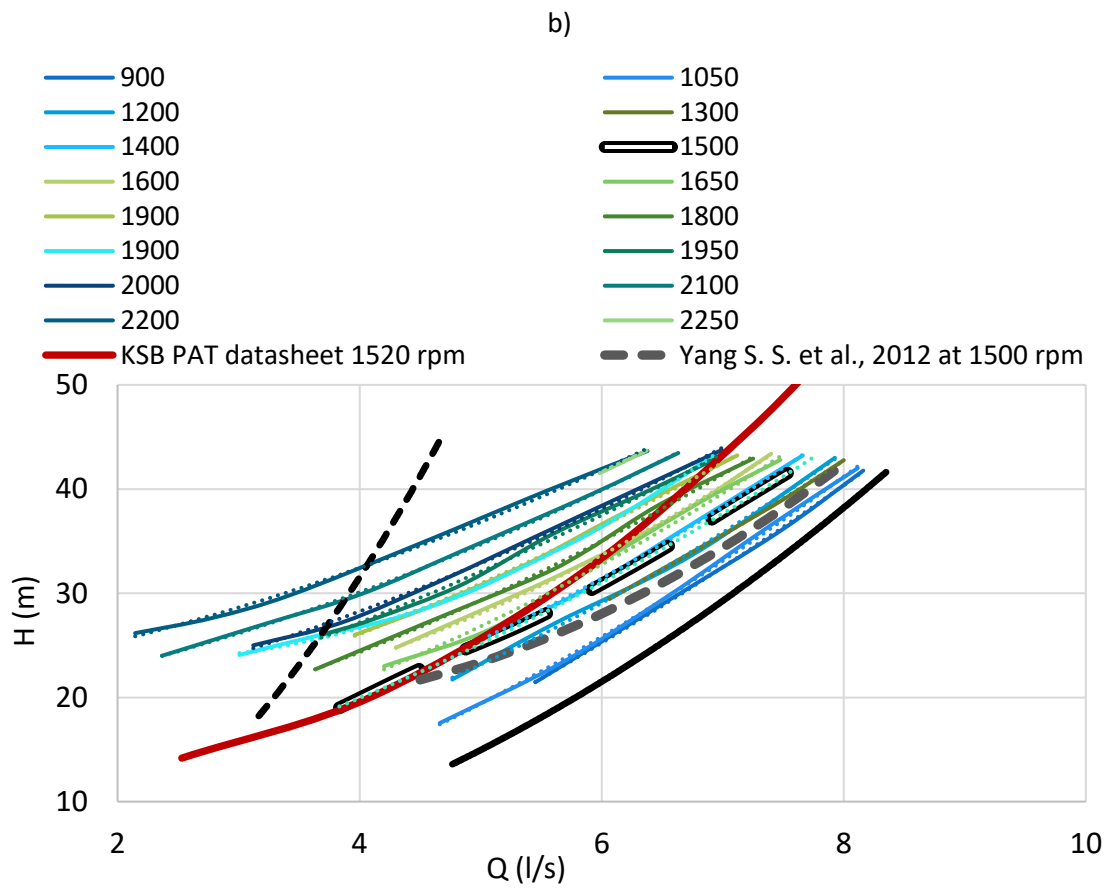
In order to carry out the complete characterization of the PAT at various speed the following procedure was followed:

- a. set the bypass valve as fully closed and the other two in-series valves (butterfly and control valve) fully open;
- b. start the supply pump at full capacity;
- c. manually open the bypass valve until the flow through the PAT is reduced to about 3 l/s;
- d. set the desired VSD frequency (e.g. 50 Hz);
- e. start the VSD in forward rotation and wait until the desired speed is reached. The VSD will dictate the induction motor/generator to either accelerate (i.e. absorbing power from the grid) or to decelerate (i.e. generating power);
- f. manually proceed to slowly closing the bypass valve in about 30s, until the entire flow produced by the supply pump is conveyed through the PAT;
- g. stop the VSD and restart from point “c”.

The results of the analysis are displayed in Figure 6-32.

a)





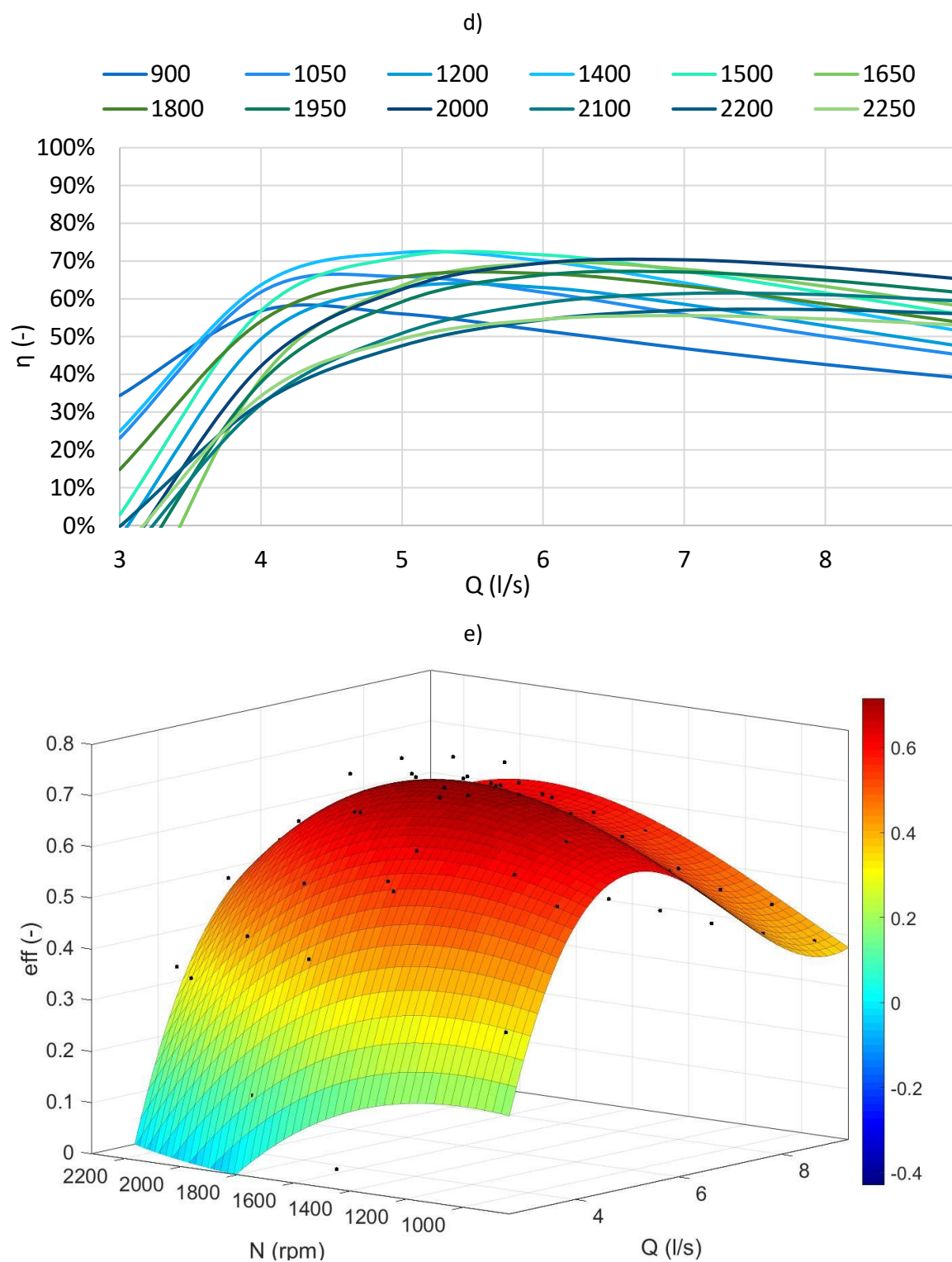


FIGURE 6-32. RESULTS FROM THE CHARACTERIZATION TESTS OF THE PAT: A) FLOW-HEAD CURVES RAW DATA AT DIFFERENT SPEED; B) FLOW-HEAD INTERPOLATED CURVES; C) EFFICIENCY CURVES RAW DATA AT DIFFERENT SPEED; D) EFFICIENCY INTERPOLATED CURVES; E) 3-D VISUALIZATION OF EFFICIENCY AS FUNCTION OF SPEED AND FLOW RATE.

The raw data shown in Figure 6-32 (a) present a significant scattering due to the inaccuracies in the instrumental readings of flow ($\pm 3\%$) and head ($\pm 0.5\%$) discussed in Chapter 6.2.6.2. However, after performing a quadratic interpolation of the measured points at each speed as shown in Figure 6-32 (b) the singular Q-H curves at the various tested speed were uniquely identified. It is interesting to notice that the experimental base head-flow curve at 1,500 RPM was closer to the one obtained through the conversion coefficients described by Yang, et al. (2012) and the curve extrapolation Equations of Chapter 5.3 (as integrated in the PAT selection software) rather than the curve provided by the manufacturer, as seen from Figure 6-32 (b).

As for efficiencies, from the results seen in Figure 6-32 (c) it can be seen that many data series present a significant dispersion at low flow values, in particular those at 900 and 1050 RPM. These are due to the erratic readings obtained from the torque meter while the VSD was struggling to maintain the set PAT speed under non-optimal conditions. Such instabilities have been disregarded in the final efficiency calculation displayed in Figure 6-32 (d) and (e).

A comparison of the PAT performances at the BEP indicated by the manufacturer with those resulting from the experimental campaign is shown in Table 6-6.

TABLE 6-6. COMPARISON OF THE PAT KEY PERFORMANCE PARAMETERS AT THE RATED SPEED.

Parameter (unit)	Experimental results	KSB PAT datasheet		(Yang, et al., 2012)	
		Values	Deviation	Values	Deviation
H (m)	27 ($\pm 0.5\%$)	33.5 ($\pm 7\%$)	19.4%	31.4	14%
Q (l/s)	5.4 ($\pm 3\%$)	6 ($\pm 9\%$)	10%	6.56	17.7%
η	72% ($\pm 3.7\%$)	51.5% ($- 7\%$)	-39.8%	55.9%	-28.8%
N (RPM)	1510	1520	0.7%	1520	0.7%

Knowing that for normalized centrifugal pumps the standard tolerances are 9% for the flow and 7% for head and efficiency, the KSB datasheet of the PAT seen in Figure 6-30 is reasonably accurate for the value of flow rate only. Instead, the stated head and efficiency are respectively overestimated and underestimated with respect to the experimental results by a large factor.

The use of the algebraic conversion method proposed by Yang, et al. (2012) which in Chapter 5.1.3 has performed better than others across a large sample of PATs would have proved more accurate than the PAT datasheet regarding the prediction of head and efficiency when applied to the BEP coordinates as pump of the same identical unit sold by KSB.

However, the most striking difference arising from the KSB of the unit as turbine and the output of the Yang method is the underestimation of the maximum mechanical efficiency by almost 20 percentage points as seen in Table 6-6.

6.2.7. DECISION SUPPORT SOFTWARE VALIDATION

The experimental head and power curves at 1500 RPM of the PAT are displayed in Figure 6-33. A fixed PAT speed of 1500 RPM was selected as it corresponded to fixed speed operations with a four-poles alternator.

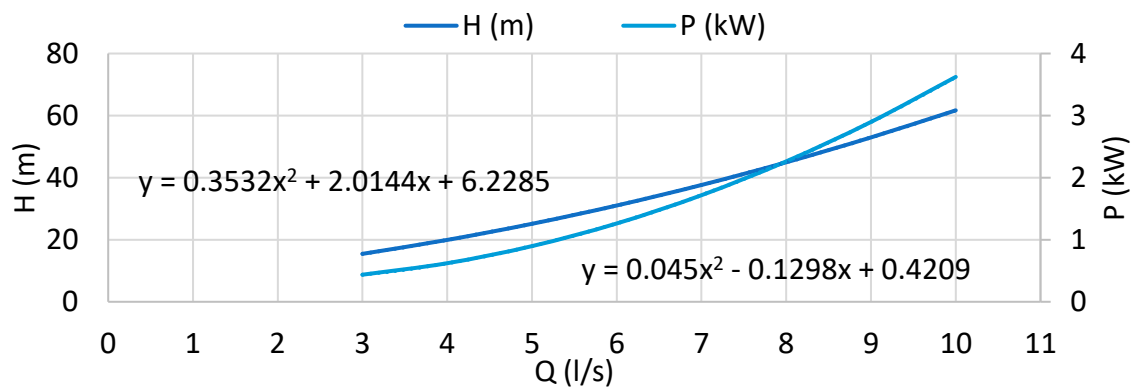
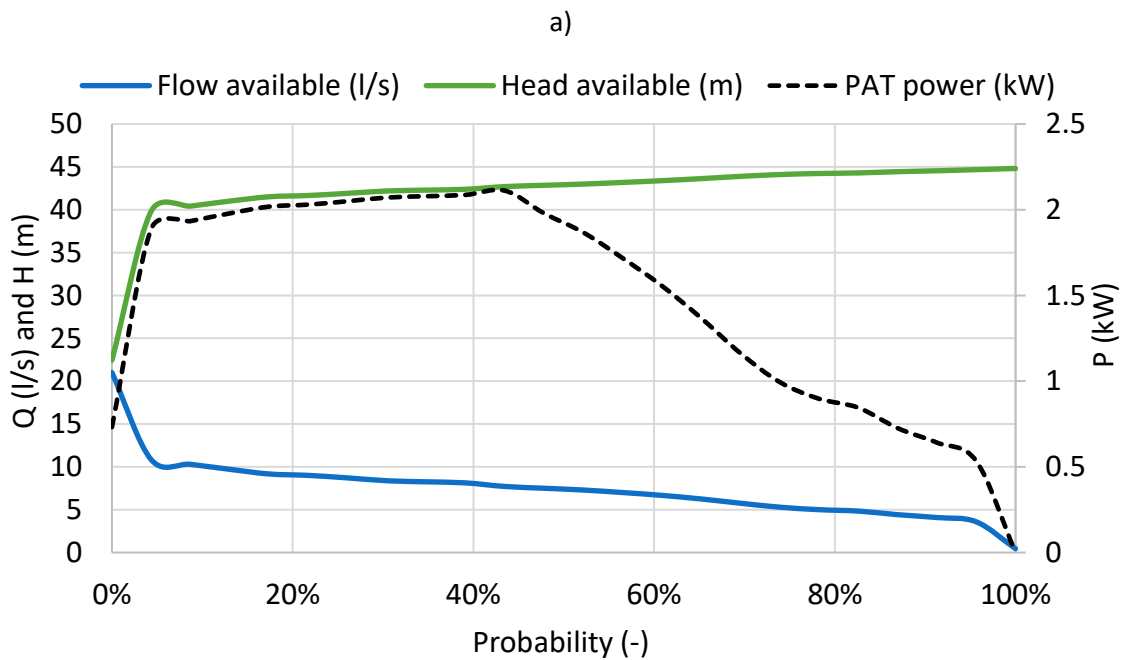


FIGURE 6-33. HEAD AND POWER CURVES OF THE PAT AT 1500 RPM.

Based on the characteristic curve displayed in Figure 6-33, the performance of the PAT are shown in Figure 6-34.



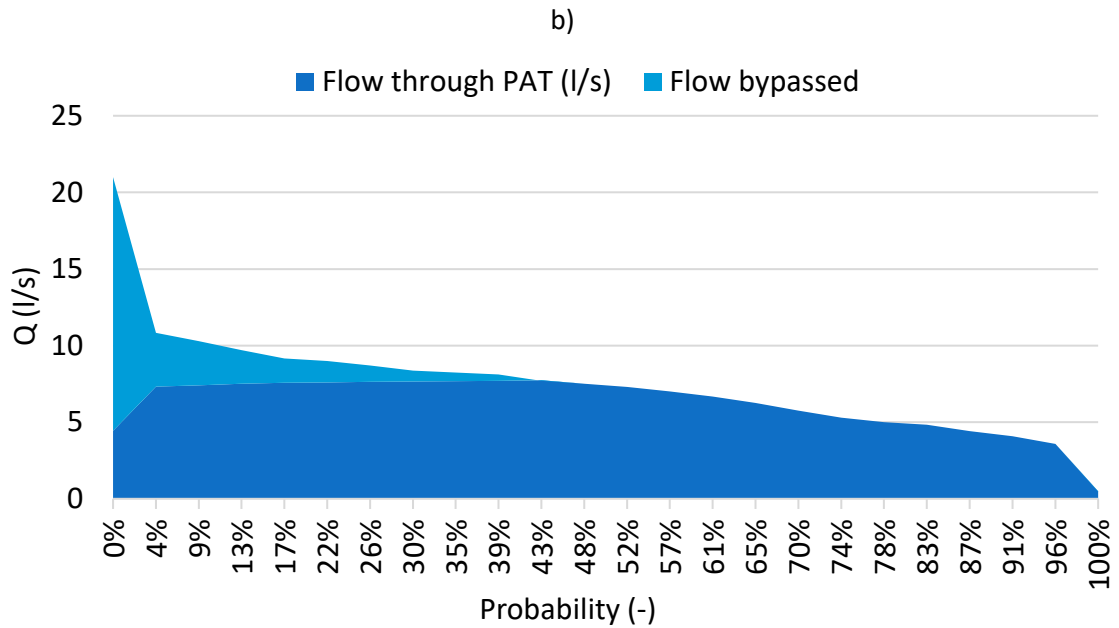


FIGURE 6-34. CALCULATED PERFORMANCES OF THE TESTED PAT WITHIN THE FICTIONAL PRV LOCATION: A) AVAILABLE HEAD AND FLOW ACROSS THE TURBINE AND RESULTING POWER OUTPUT; B) VOLUMES OF FLOW PROCESSED THROUGH THE PAT OR BYPASSED.

The results from Figure 6-34 when extended to a whole year of operations of the turbine leads to:

- 7,770 kWh per year if used 24/7 with a generator efficiency of 60%
- 10,300 kWh per year if considering a higher generator efficiency of 80 %

The software results previously depicted in Figure 6-29 (a) suggested a KSB Etanorm 50-32-200 as the preferred PAT (which was eventually purchased and tested). It also predicted for it a yearly energy output of nearly 7,500 kWh per year, which would align well with the effective projected output here when connected to a poorly efficient generator (60%). However, as a common 2.2 kW generator may commonly be as efficient as 80%, in a practical application the power output and energy yield would have been underestimated. This is mostly caused by the much higher experimental turbine efficiency (72%) with respect to the one expected from the Yang method (55.9%) and the datasheet of the manufacturer (51.5%) as seen in Table 6-6.

6.2.8. SOFTWARE ACCURACY FOR THE PAT PURCHASE PRICE PREDICTION

The final retail price at which the PAT was purchased was 1,914.2 € plus VAT for a total of 2,354.47 €. However, the KSB reseller for Ireland disclosed that such price included a +70% price increase set by KSB for the same pump if bought as a PAT. Such mentioned surcharges included:

- a special machining in order to prevent the impeller nut to come loose after prolonged operations in reverse rotation (reversed thread);
- shaft strengthening with respect to the standard model;
- booklet with performance charts of the unit as a PAT (displayed here as Figure 6-30);
- warranty.

While the first item on the list is a needed modification to guarantee the performance of the unit over time, the remaining items may be deemed as unessential by other clients who therefore may decide to purchase the same unit as a pump at a cost of 1,384.98 € incl. VAT as per its listed price. Equation (48) at Page 134 of Chapter 5.4.3 suggested a purchase price of 1,861.29 € (incl. 23% VAT) for a PAT & synchronous generator rated for 6.6 l/s and 32 m of head. From the same cost model, the cost of a four-poles 1 kW induction generator can be evaluated as 274.95 € incl. 23% of VAT. Therefore, the predicted final cost for the PAT only would have resulted in 1,586.34 € which is within 85% of accuracy when compared to the final price of buying the mentioned KSB Etanorm 50-32-200 as a pump in Ireland.

7. DEMONSTRATION SITES

Among the targets of the Dŵr Uisce Project was the construction of two demonstration sites, one in Wales and one in Ireland, which could work as pilot installations and showcase the applicability of the PAT technology in different contexts. Besides, additional aims behind the completion of the demo sites were to give an additional validation to the PAT selection software developed in Chapter 5 and to monitor and track over time the performances of a MHP installation. After developing several feasibility studies shown as Annex 1, the sites of Ty Mawr Wybrnant in Wales and Blackstairs in Ireland were selected to host the pilot installations.

7.1. TY MAWR WYBRNANT, NORTH WALES

7.1.1. OVERVIEW

The 16th century historical farmhouse of Tŷ Mawr Wybrnant is a renowned touristic site in Northern Wales managed by National Trust. It was the birthplace of Bishop William Morgan, first translator of the Holy Bible into Welsh. Pictures of the premises are shown in Figure 7-1. In an effort to reduce the energy consumption related to lighting and heating of the building a small hydropower installation has been proposed to generate about 4 kW from the water of the nearby Afon Wybrnant stream. The main purpose of the installation is to demonstrate the environmental benefits of the scheme, give a further validation of the PAT decision support software and to prove the economic attractiveness of such small-scale distributed hydro generation.



c)



d)



FIGURE 7-1. IMAGES OF THE WELSH HYDRO PILOT PLANT SITE: A) FRONT VIEW OF THE HISTORICAL FARMHOUSE; B) GROUP PICTURE AT THE CAR PARK; C) AND D) VIEW OF THE SPOT SELECTED AS INTAKE LOCATION ALONG THE STREAM.

The route of the proposed pipeline covering the 300 m of distance between the intake and the powerhouse is displayed in Figure 7-2.

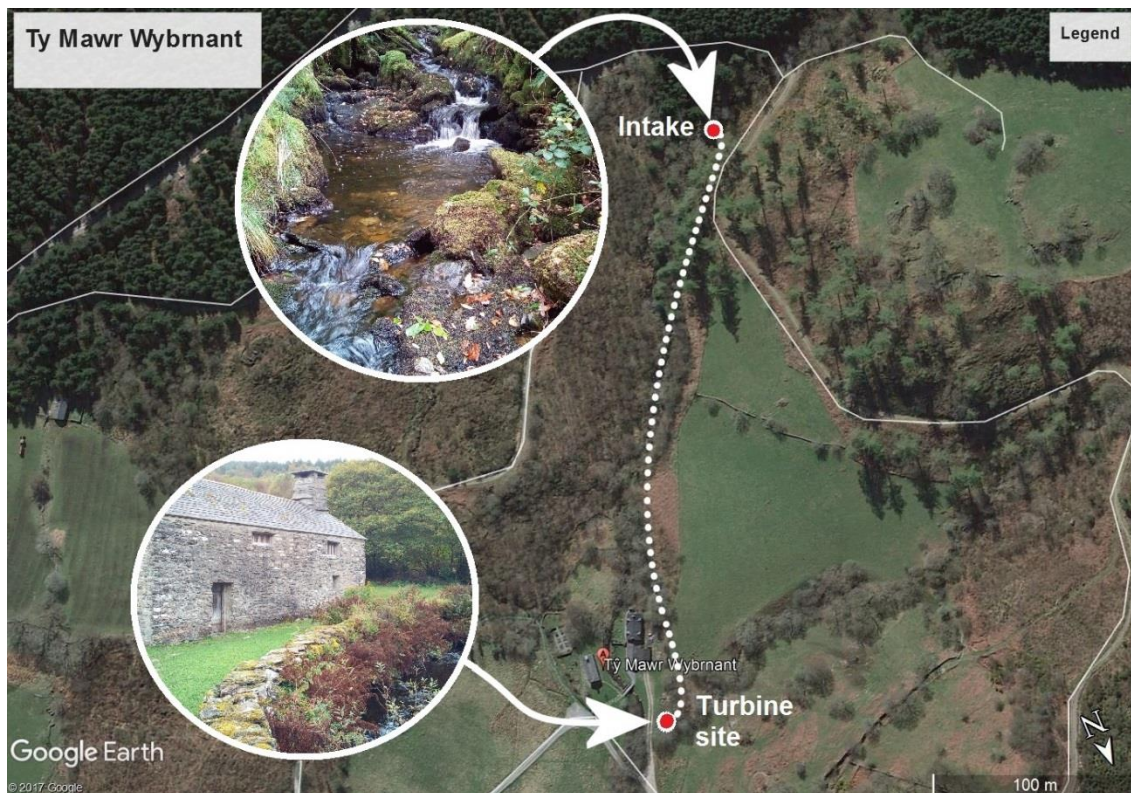


FIGURE 7-2. AERIAL VIEW OF THE PIPELINE ROUTE FROM THE CHOSEN LOCATION OF RIVER INTAKE TO THE POWERHOUSE.

7.1.2. DESIGN DATA

The measured net hydraulic head corresponds to 35 m, and the head losses associated with a 6" (DN150) pipe are displayed in Figure 7-3.

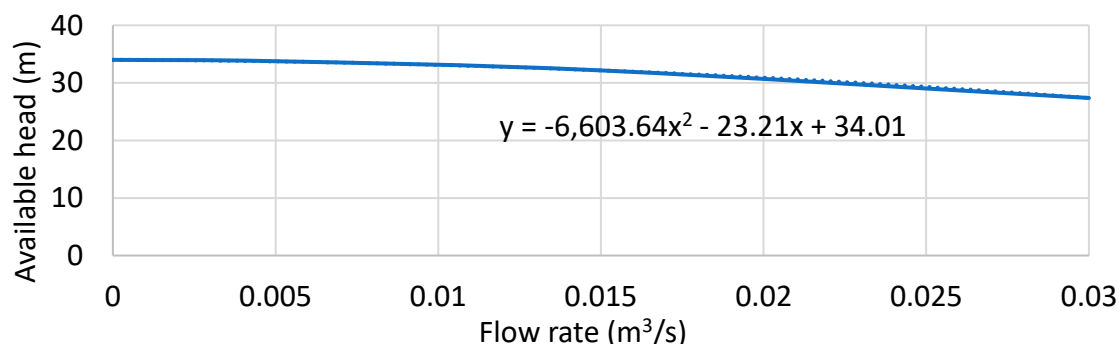


FIGURE 7-3. AVAILABLE HEAD AT THE POWERHOUSE WHEN INCLUDING THE FRICTION LOSSES GIVEN BY A DN150 PENSTOCK.

An independent consultant hired by the property (National Trust) has estimated the flow duration curve of the Wybrnant stream in correspondence to the selected intake location by means of the LowFlows software commercialized in the UK by Wallingford HydroSolutions (Wallingford HydroSolutions, 2018). According to the prescriptions of Natural Resources Wales the Hands-Off-Flow (HOF) must correspond to the Q95 value of yearly flow data, and the amount of water available for abstraction is the 70% of flow above the HOF until a maximum of Q_{mean} . From LowFlows calculations, the HOF corresponded to 7 litres per second and the Q_{mean} to 0.084 m³/s. The yearly flow regime and the flow available for abstraction are therefore plotted in Figure 7-4.

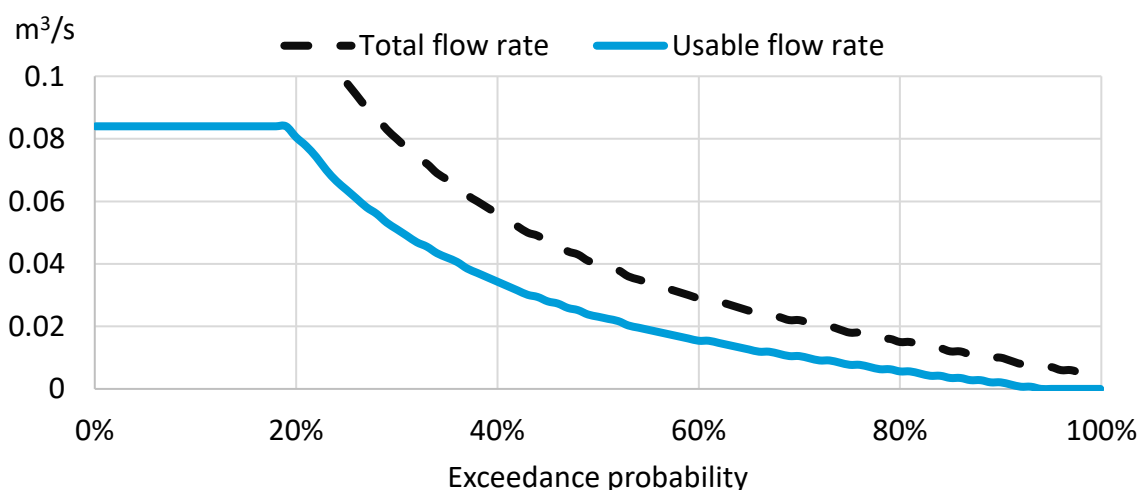


FIGURE 7-4. TOTAL YEARLY FLOW REGIME AND USABLE FLOW RATE.

7.1.3. EQUIPMENT SIZING AND DESIGN

Even though the available head and the predicted water flow allowed for abstraction may have allowed for a peak turbine power of about 20 kW, it was decided to limit the power output to 3.68 kW in order to comply with the G83 regulation in the UK (16 A per phase) by means of feeding the electricity to the grid through a solar-approved inverter. This setup would therefore replicate the configuration used for the small-scale PAT laboratory test which included a three-phase synchronous generator and a rectifier.

A fixed turbine shaft speed of 1,500 RPM was selected, corresponding to the use of a four-pole synchronous generator. A four-pole machine was preferred over a two-pole one in order to mitigate the hydraulic transients that may arise in case runaway conditions are accidentally reached. The ideal generator should be rated for 5.5 kVA.

The design software developed in Chapter 5 was applied to the selected site, and helped identify the ideal characteristics of the Pump as Turbine and generator set to be chosen. After having selected a standard shaft speed of 1,500 RPM and a power cap of 3.68 kW the selection software was run with the flow and head available at the site as inputs and the results can be seen in Figure 7-5.

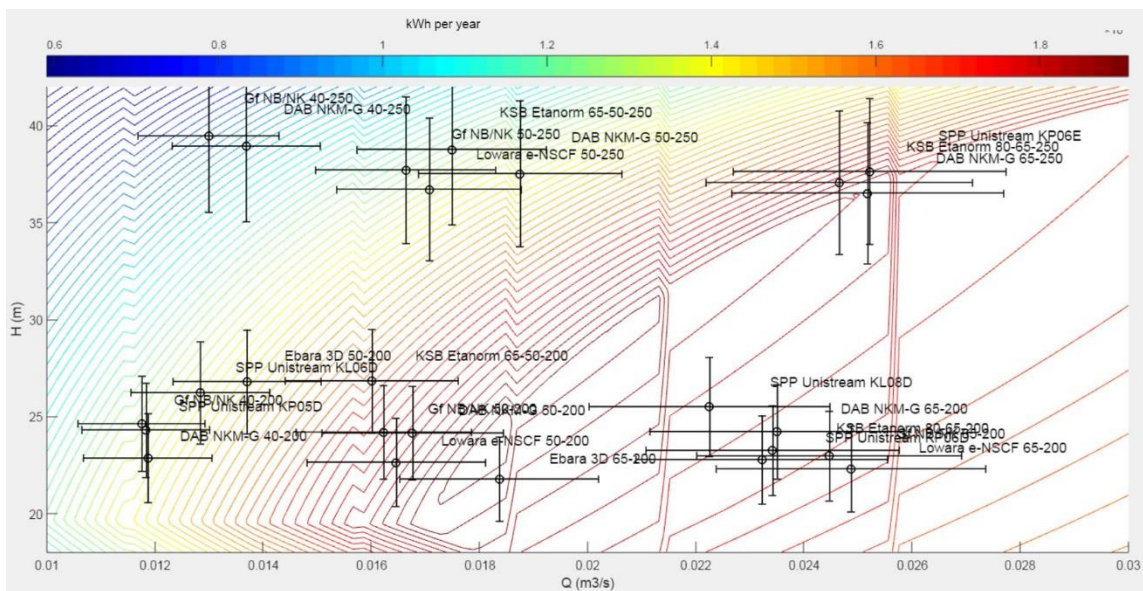


FIGURE 7-5. RESULTS FROM THE PAT SELECTION SOFTWARE.

The results from the software displayed in Figure 7-5 show that an ideal power plant would generate about 19,000 kWh of electricity per year.

Baureihe-Größe Type-Size Modèle	Tipo Serie Tipo	Nennrehzahl Nom. speed Vitesse nom.	Velocità di rotazione nom. Nominal foerental Revolutions nom.	Laufrad-Ø Impeller diameter Diamètre de roue	Ø Girante Ø Waaler Ø Rodete	
Etanorm 80-65-250 Turbine		1520 1/min		260 mm		
Projekt Project Projet	Progetto Project Proyecto	Angebots-Nr. Project No. No. de l'offre	Offerta-No. Offertenr. Offerta-No.	Pos.-Nr. Item No. No. de pos.	Pos.Nr. Positiénr. Pos.-Nr.	KSB LIMITED 2 Cotton Way LOUGHBOROUGH Leicestershire LE11 5TF
Run of River plant		4003242514		Option 4		

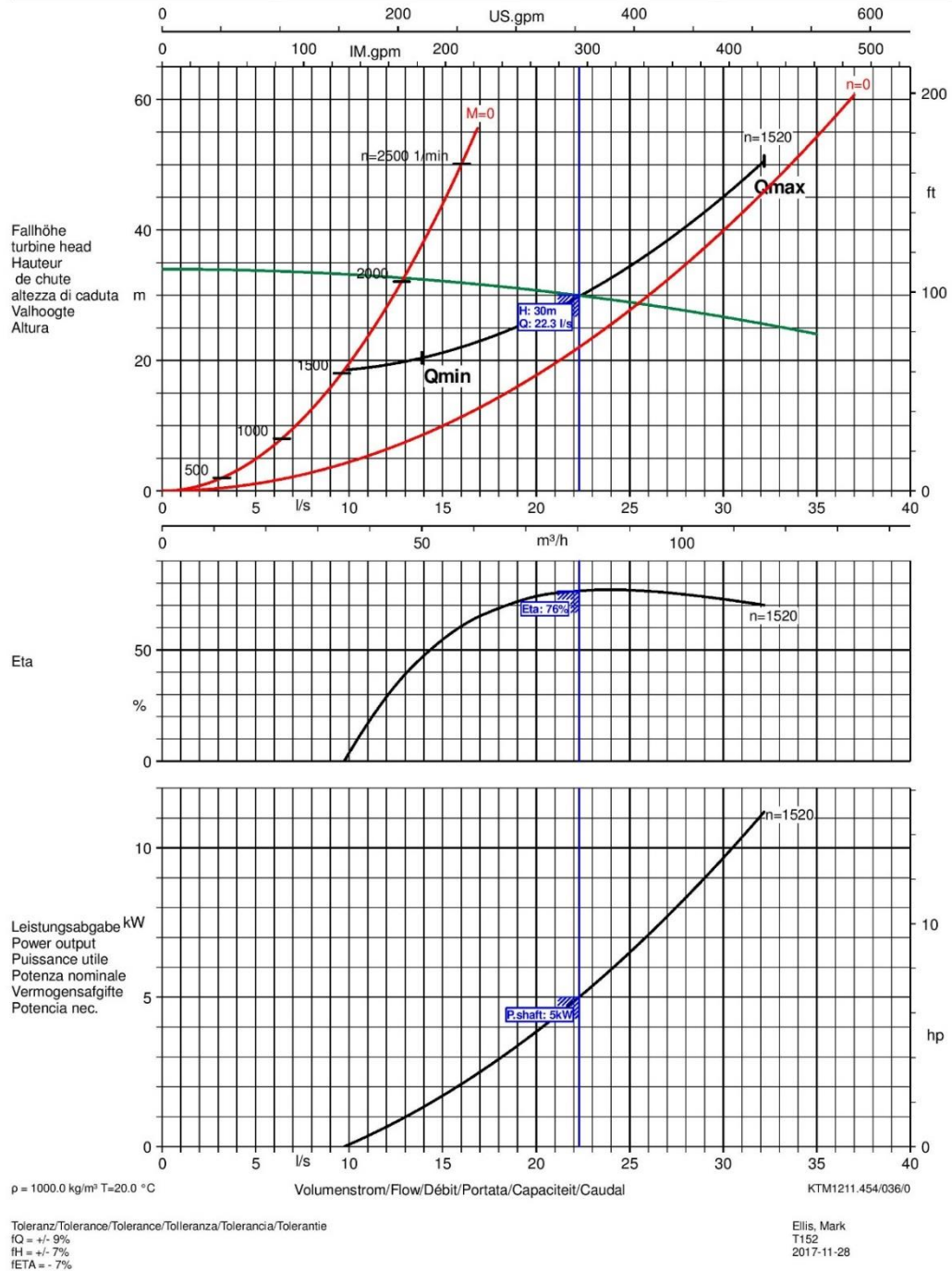


FIGURE 7-6. PERFORMANCE CHARTS OF THE KSB ETANORM 80-65-250 PREDICTED BY KSB IN TURBINE MODE.

From Figure 7-5 it can also be observed that there are two families of normalized centrifugal PATs offering the highest power output, i.e. the 80-65-200 and the 80-65-250 ones. Out of these, the latter was chosen since the performance charts of a KSB Etanorm 80-65-250 in turbine mode were available to the author thus allowing a further comparison between real turbine performances and the performances predicted by KSB besides the one done during laboratory tests.

The expected KSB performance charts for the chosen PAT are displayed in Figure 7-6, showing an expected mechanical power output of 5 kW with 22.3 l/s and 30 m. Since the rated efficiency of the selected three-phase synchronous generator, a Zanardi MeccAlte BTP3 1S/4 unit, was claimed by the manufacturer as 80% under full load, the expected electric power output at the wires of the alternator shall be 4 kW which is in line with the 3.68 kW target power output from the inverter.

A schematic view of the electric components necessary for the installation is shown in Figure 7-7, while in Figure 7-8 a detailed visual representation of the installation is presented.

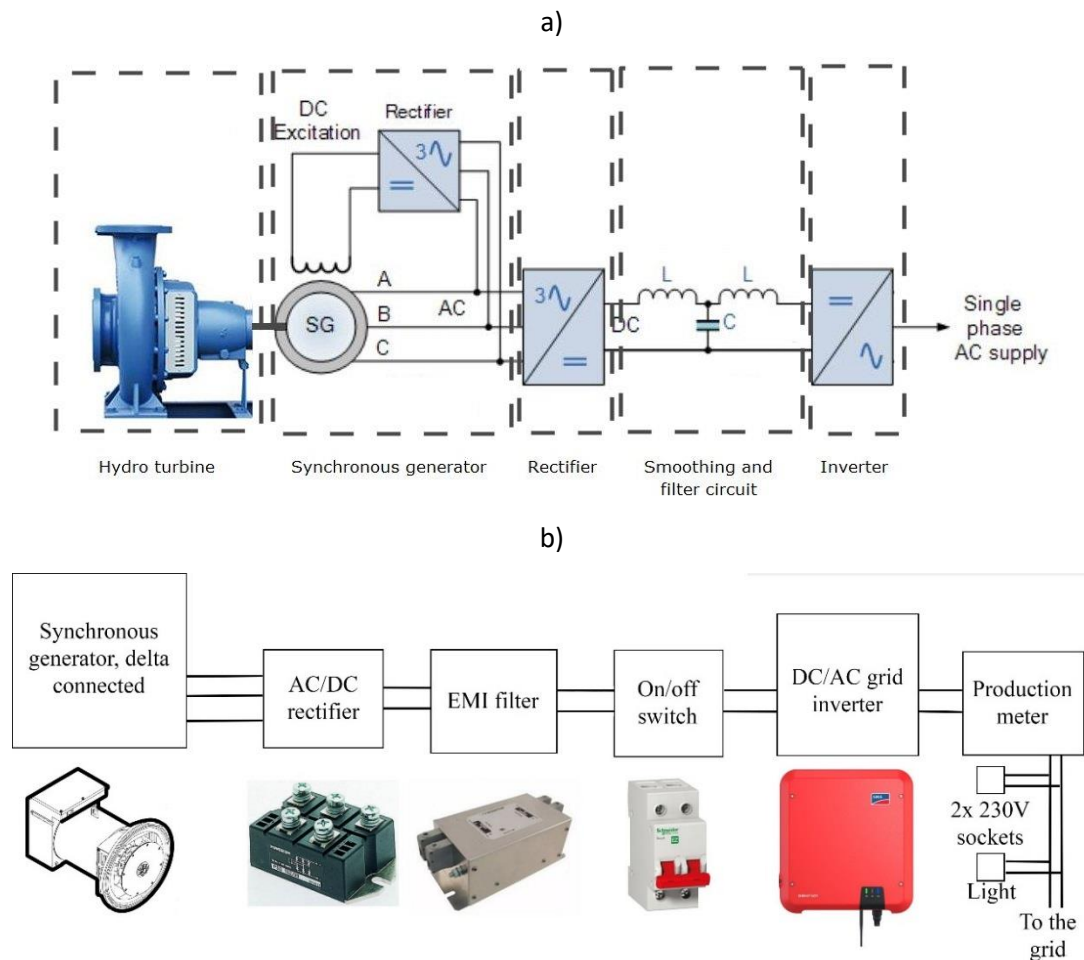


FIGURE 7-7. DESIGN OF THE ELECTRIC COMPONENTS OF THE INSTALLATION: A) ILLUSTRATIVE DIAGRAM DEPICTING THE MAIN ELEMENTS OF THE INSTALLATION; B) APPROXIMATE WIRING DIAGRAM.

The electric layout of the installation shown in Figure 7-7 was designed to be similar to the one of the small-scale PAT laboratory test with the synchronous generator described in Figure 6-13 within Section 6.1.5. The main difference between the two setups is that the small-scale laboratory test relied on a permanent magnet brushless generator while for the Welsh demo site a brushed synchronous generator was selected. Despite its greater maintenance needs and lower efficiency, the latter was chosen since no UK manufacturer was found able to supply a 4 kW permanent magnet generator.

As for the pipeline layout displayed in Figure 7-8, in agreement with National Trust experts it was decided to feed the PAT through a tee branch connected to the main pipe in order to enable the piggy-cleaning process of it if deemed necessary to remove accumulated sediments. However, this naturally introduced additional head losses due to the three added 90 degree bends in the feeder pipe.

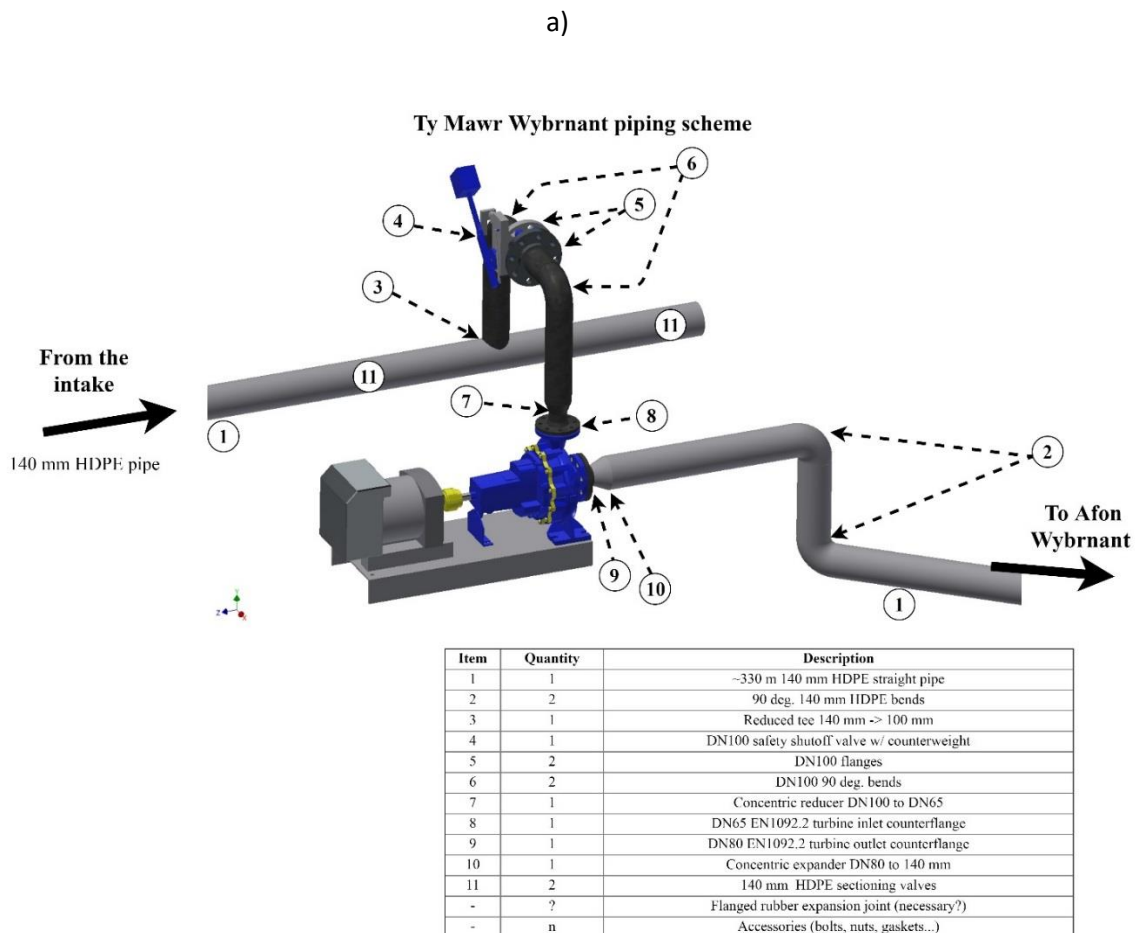




FIGURE 7-8. DETAILED INSTALLATION DRAWINGS: A) PAT CONNECTION DRAWING AND LIST OF MATERIALS; B) AND C) RENDERING OF THE POWERHOUSE.

The powerhouse itself as depicted in Figure 7-8 was designed as a 2.5x3 m wooden shed with a window on the front in order to allow visitors to see the turbine without entering the premises.

7.1.4. PROGRESS OF CONSTRUCTION PHASE

The construction of the Welsh Dŵr Uisce Demo Site took place along successive phases:

1. September-November 2018: coupling of the PAT and generator bolted on a custom-made steel frame in the Civil Engineering laboratory of Trinity College Dublin. A collection of images displaying the assembly of the turbogenerator is shown in Figure 7-9;
2. December 2018: building of the river intake and initial section of the pipeline;
3. January-February 2019: excavation and laying of the pipeline, construction of the powerhouse, delivery of the turbogenerator on site;
4. March 2019: completion of the civil works, installation of the electro-mechanical devices and of the telemetry and control unit;
5. September 2019: replacement of the inverter with a new one conforming to the new G98 regulations for connection in parallel to the UK grid, and final commissioning of the scheme. Further details on the commissioning phase are presented in Section 7.1.5.

a)



b)

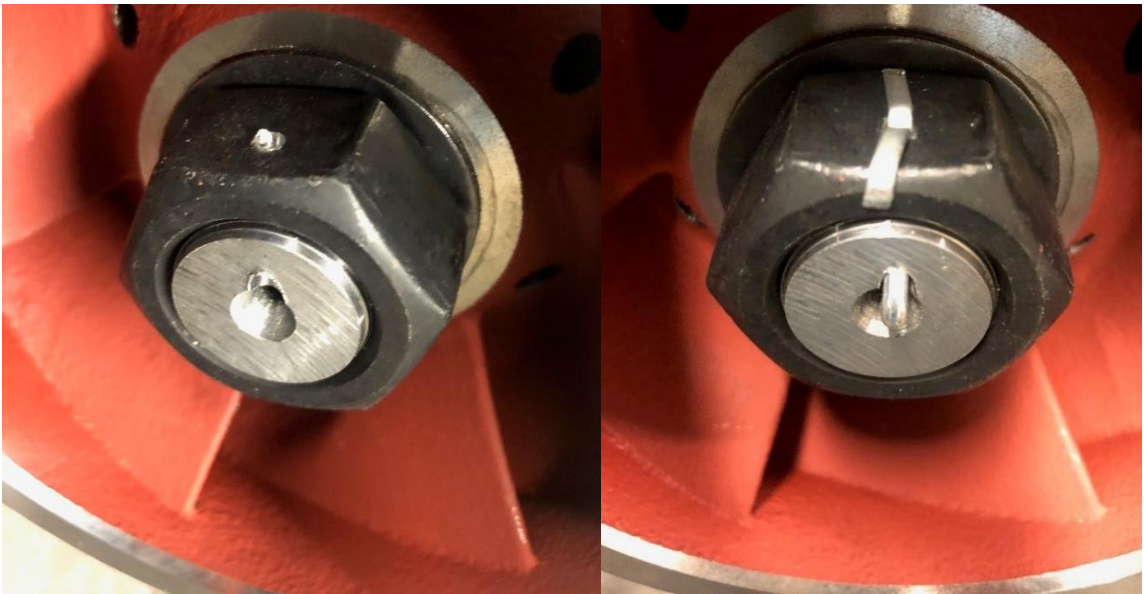


FIGURE 7-9. ASSEMBLY OF THE TURBOGENERATOR FOR THE WELSH PILOT PLANT: A) ASSEMBLY OF THE PAT AND SYNCHRONOUS GENERATOR ON A CUSTOM-MADE FRAME; B) MODIFICATION OF THE PAT TO LOCK THE IMPELLER NUT IN ORDER TO PREVENT IT FROM LOOSENING UP.

The intake was made up of a single pre-cast concrete slab which was put in place by a local construction company and whose size was approved by Natural Resources Wales during planning. A few images showing the construction of the river intake are displayed in Figure 7-10.

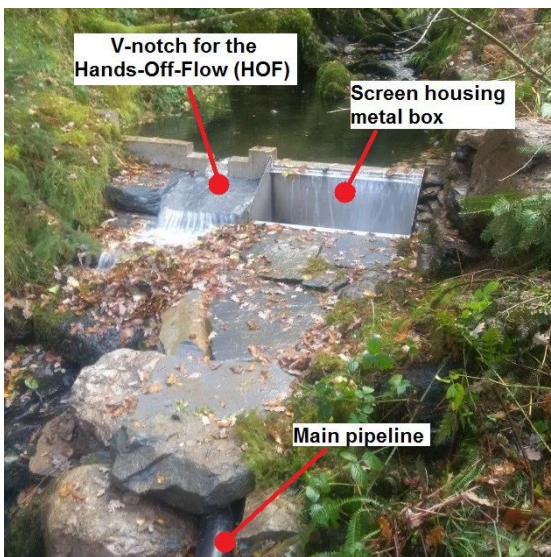
a)



b)



c)



d)



FIGURE 7-10. CONSTRUCTION OF THE WATER INTAKE STRUCTURE: A) AND B) POSITIONING OF THE PRE-CAST CONCRETE WEIR INTO THE STREAM; C) AND D) AFTER RESTORATION OF THE RIVER BED AND CONNECTION TO THE PIPELINE.

In order to lay the High-Density Polyethylene (HDPE) pipeline a trench had to be dug up partially along an unused forestry road and partially under a nearby grassy field. Figure 7-11 shows the undergoing of the field works and the equipment used to dig the trench and stretch the pipe.



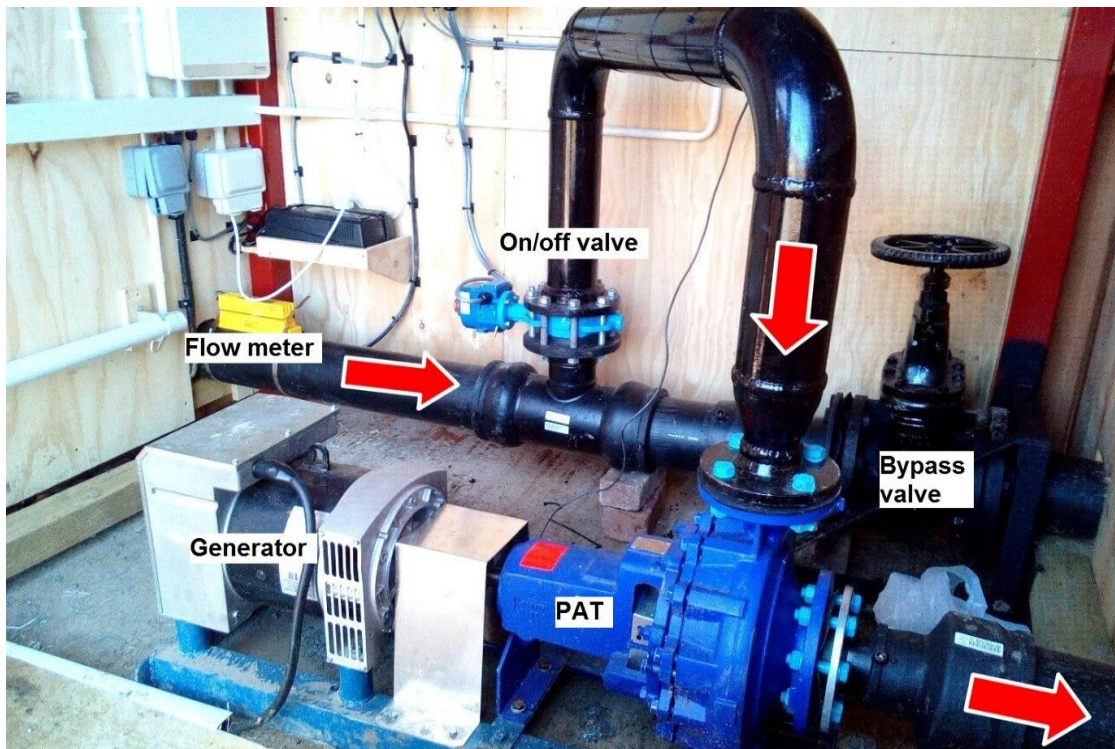
FIGURE 7-11 - LAYING OF THE 6" PIPELINE FROM THE SOURCE TO THE POWERHOUSE.

The powerhouse was erected over a concrete slab, and was fabricated by a local carpenter using larch panels over a custom-made steel frame. The final erection of the powerhouse and the completion of installation of electromechanical devices is displayed in Figure 7-12.

a)



b)



c)

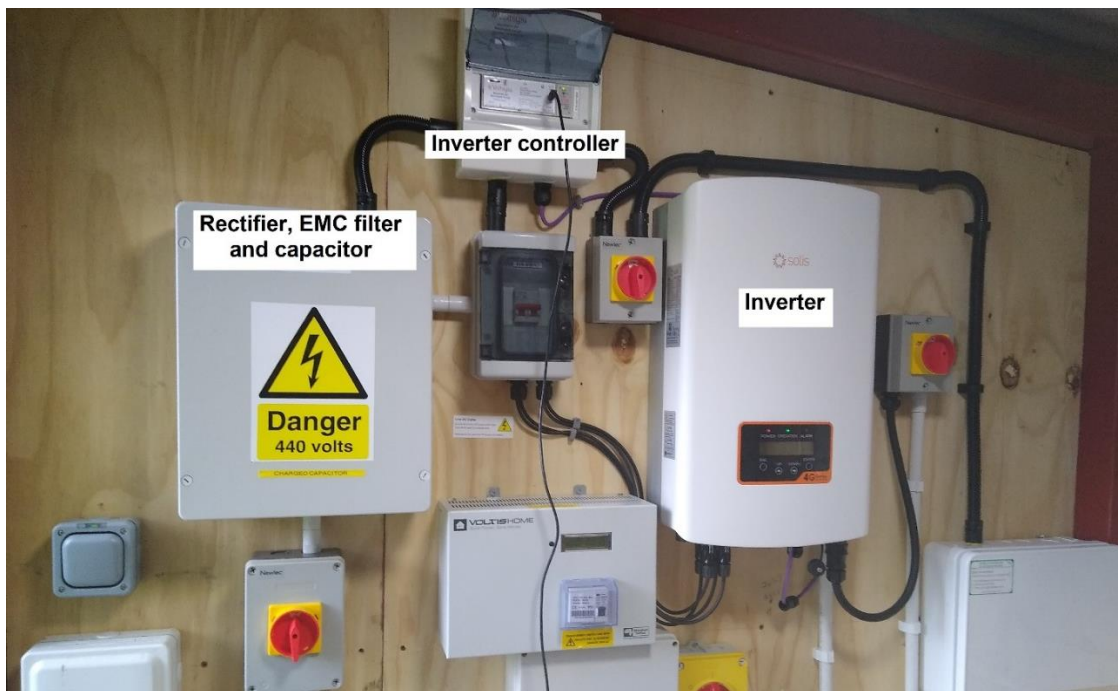


FIGURE 7-12. CONSTRUCTION OF THE POWERHOUSE: A) BUILDING OF THE SHED; B) VIEW OF THE INSTALLED TURBINE AT THE TERMINAL SECTION OF THE PIPELINE; C) ELECTRIC CABINETS AND INVERTER.

In order to remotely monitor the installation a telemetry system was installed, which also had the capability to automatically start and stop the water flow through the turbine if sufficient water is present at the intake. In order to achieve this, a submerged level sensor was installed upstream of the intake's concrete weir and a model was built and calibrated to calculate the flow rate present in the stream from the level reached by the water. The weir was modelled as a Compound Rectangular-Rectangular Sharp-Crested type following recommendations found in the literature (Chyan-Deng, et al., 2006). The relationship found between the water level at the intake and the stream flow rate is displayed in Figure 7-13.

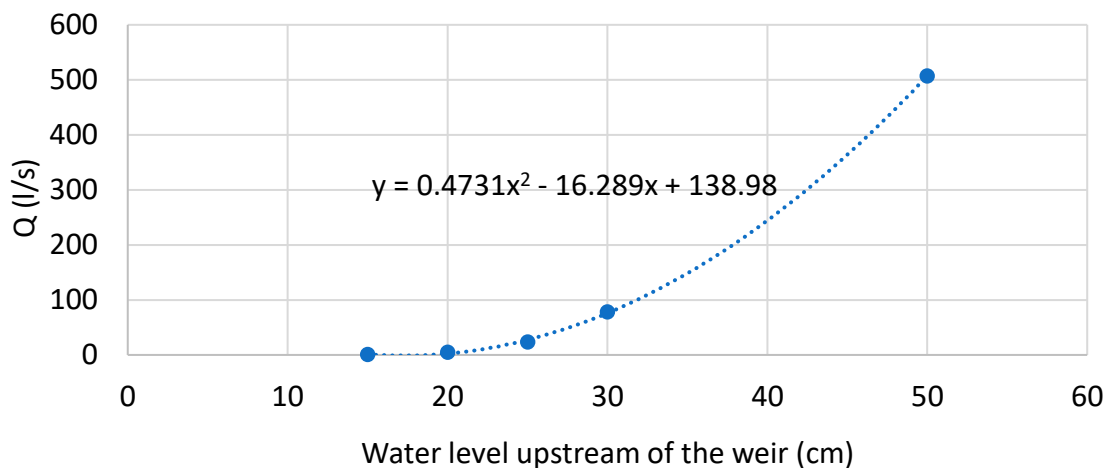


FIGURE 7-13. CALCULATED RELATIONSHIP BETWEEN THE DEPTH OF WATER AT THE WEIR AND THE FLOW RATE PRESENT IN THE STREAM BEFORE THE HYDRO INTAKE.

7.1.5. COMMISSIONING

After the completion of the civil works and the hydraulic and electric connections, the no-load open circuit DC voltage at the inverter was measured as well as the corresponding shaft speed of the turbogenerator as displayed in Figure 7-14. As it can be seen from the chart the results proved satisfactory since, even with the fully open butterfly valve, the DC voltage (410 V) was much lower than the maximum accepted by the inverter (600 V) and the runaway speed (1820 RPM) was lower than the maximum one recommended by the manufacturer of the generator (2,200 RPM). As a consequence, the system would be able to run via a manual on-off valve without the need of any safety control device as in case of grid loss the runaway conditions would not damage any of the equipment. However, as mentioned in Chapter 7.1.4 the telemetry unit installed for this scheme also offered the capability of operating a control valve automatically according to the flow available in the stream and at the same time isolating the turbine in case of electric failure.

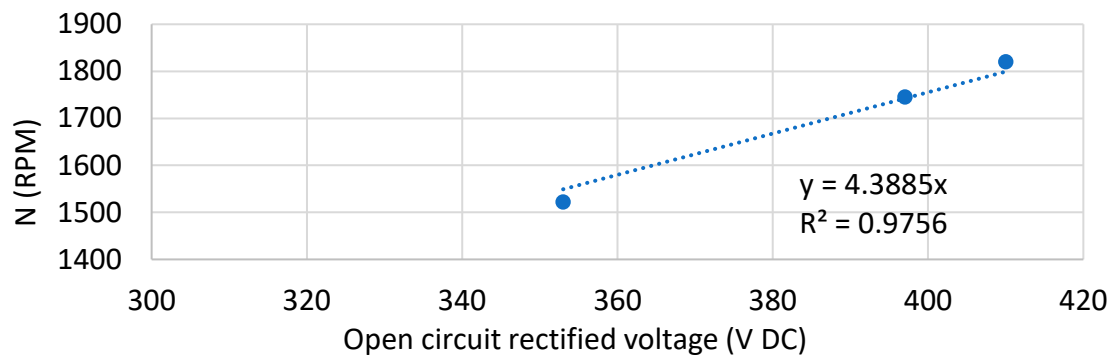


FIGURE 7-14. PRELIMINARY TEST TO VERIFY THE RELATIONSHIP BETWEEN NO-LOAD DC VOLTAGE AND GENERATOR SPEED.

Despite such promising early tests, the PV inverter was not able to generate power from the turbine DC output even if the latter was well in line with the operating interval of its MPPT algorithm. At first this was attributed to an inverter malfunctioning, and the manufacturer sent a replacement unit which was swapped and tested in May 2019 without success. After consulting with several experts two hypotheses were raised to explain the malfunctioning of the PV inverter:

1. the measured AC grid voltage was found to be relatively high (249-256 V) which may have been triggering the G83 relay of the inverter. At first the Distribution Network Operator (DNO) was contacted asking for the line transformer serving the property to be tuned down, but since the action was scheduled for after the summer an alternative was investigated. Eventually it was decided to install a transformer (voltage optimizer) on the property side able to reduce the AC voltage to 230 V which unfortunately didn't enable power generation from the inverter;
2. the quality of the DC voltage fed into the inverter was investigated, and it was decided to install a heavy-duty 5,600 μ F capacitor in parallel in order to smoothen the ripple generated by the rectifier. However, this showed no influence on the inverter's inability to generate power.

As the above strategies didn't lead to the desired outcome, the responsibility for the problem was attributed to the PV inverter purchased for the scheme. In order to verify this third hypothesis a different second-hand wind grid tie inverter was brought to the powerhouse and tested on 8/7/2019. The test proved successful, as the scheme with the new inverter generated up to 2.9 kW with an almost completely dried-up riverbed. However, the new inverter did not comply with the new G98 standard which from 1/5/2019 replaced the previous G83 for all new generators connected in parallel to the UK grid with less than 16 A per phase. Therefore, an alternative G98-compliant wind inverter was purchased consisting of a combined PV inverter and an external

controller which could be programmed with a customized power curve. The new setup which can be seen in Figure 7-12 (C) was installed and commissioned on 9/9/2019.

Once the final set of equipment was installed, the unit was able to generate a maximum of 4,020 W of electric power which confirmed the validity of the PAT selection process through the developed software. Besides, a working point of the turbine was found in correspondence of 19.3 l/s and 32 m while running at a speed of 1,660 RPM. By means of the affinity laws described in Equation (6) and (7) at Page 46 the recorded working point was correlated to its equivalent at the rated speed of 1,520 RPM and compared against the PAT characteristic curve predicted by the software (through the model developed by Yang et al., 2012) and the one provided by the manufacturer and displayed in Figure 7-6. The results of the comparison are shown in Figure 7-15.

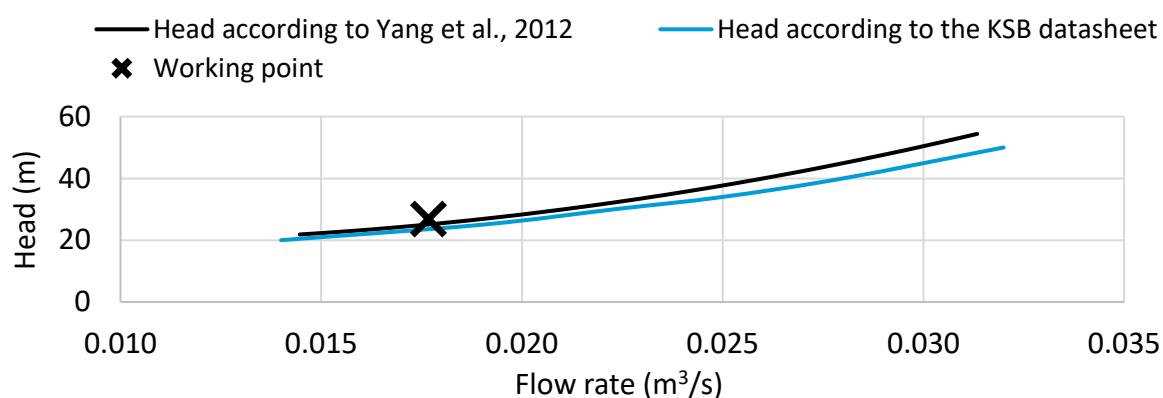


FIGURE 7-15. COMPARISON OF THE RECORDED WORKING POINT AGAINST THE TURBINE CHARACTERISTIC CURVE AS PREDICTED BY THE SOFTWARE (THROUGH THE MODEL DEVELOPED BY YANG ET AL., 2012) AND THE ONE FROM THE KSB PAT DATASHEET.

The findings from Figure 7-15 suggest that the PAT performances predicted by the software were slightly more accurate than those claimed by the manufacturer.

7.1.6. PERFORMANCE ANALYSIS

7.1.6.1. Comparative analysis with other turbine technologies

Prior to getting in contact with the Dŵr Uisce project the site owners (National Trust) had taken a decision to develop the hydro scheme independently and powering it by means of either a conventional Pelton turbine or an array of up to three pico Turgo units. At the same time, they had evaluated the pros and cons of the two grid connection standards available to the public for micro-generation on the UK grid (G59/83, later upgraded to G99/98).

After the initial contact with the Dŵr Uisce project the use of a PAT was introduced in the comparative analysis under four different configurations:

- *three pico Turgo turbines in parallel via the G83 standard*
- *one PAT via either the G59 or G83 standard*
- *one pico Turgo and one PAT in series via the G59 standard*
- *one Pelton turbine via the G59 standard*

The economic analysis and cost breakdown of the electro-mechanical equipment under such scenarios is displayed in Figure 7-16, while a multi-dimensional comparison including installation benefits, life cycle, environmental and safety aspects was carried out as shown in Figure 7-17.

		3x Powerspout Turgo HP	Pump as Turbine		Pump as Turbine + 1x Powerspout Turgo HP	IREM Pelton turbine
		 3 x				
Grid connection standard		G83	G83	G59	G59	G59
Cost of components (GBP)	Turbine(s)	4,570	1,600	1,990	6,560	12,400
	Generator(s)	Included	675	Included	Included	Included
	Inverter	427	427	-	427	-
	3-ph AC-DC bridge rectifier	Included	10	-	Included	-
	EMC filter	Included	76	-	Included	-
	Load protection breaker?	?	?	-	?	-
	G59 relay	-	-	526	526	4,400
	Siemens PLC	-	-	160	160	
	Electro-actuated valve	-	-	300	300	
	24 V battery	-	-	30	30	-
Partial total (GBP)		4,997	2,788	3,006	8,003	16,800
Partial total with 20% VAT (GBP)		5,996	3,346	3,607	9,604	20,160
G59 inspection and approval cost (GBP)		-	-	1,500	1,500	1,500
Total (GBP)		5,996	3,346	5,107	11,104	21,660

FIGURE 7-16. CROSS-COMPARISON OF THE COST FOR ELECTRO-MECHANICAL DEVICES FOR THE FIVE SCENARIOS ANALYSED.

A striking results presented in Figure 7-16, in which all amounts were based on actual quotes rather than estimates, is that for approximately the same amount of power a PAT solution through either G83 or G59 standards would cost significantly less, not only with respect to a conventional

Pelton turbine, but also to a set of pico turbines. From the same figure it is also evident that the G83 grid connection standard despite needing a few additional elements (rectifier, type-approved inverter...) removed the need for G59 inspection and approval procedure which can be lengthy and expensive. When comparing the final cost of the PAT solution through G83 standard to the one of the Pelton turbine in Figure 7-16 it turned out that the electro-mechanical devices in the case of the conventional turbine were roughly 6.5 times more expensive than the equivalent with a PAT which is in line with the findings from Figure 5-20 at Page 132.

		3x Powerspout Turgo HP	Pump as Turbine		Pump as Turbine + 1x Powerspout Turgo HP	IREM Pelton turbine
Installation	Grid connection standard	G83	G83	G59	G59	G59
	Nominal power output (kW)	3.68 ⁽¹⁾	3.68 ⁽¹⁾	≈ 4 ⁽²⁾	≈ 5.2	4 ⁽²⁾
	Equipment cost incl. VAT (GBP)	6,000	3,350	5,100	11,100	21,600
Life cycle	Operability	Fully manual operations, need to constantly operate the 6 feeding valves	Fully manual operations, but only one valve needs to be operated	Automatic on/off operations, manual adjustment of the throttling valve still needed	- See single machines -	Fully manual operations, need to constantly operate the feeding valves
	Estimated yearly energy yield (kWh)	19,300	19,400	22,300	- See single machines -	24,300
	Estimated yearly energy savings (GBP)	3,070 ⁽³⁾	3,080 ⁽³⁾	3,540 ⁽³⁾	- See single machines -	3,870 ⁽³⁾
	Ease of maintenance	Manual greasing of bearings every 3 months, spare parts available from New Zealand	Manual greasing of bearings every 2 years, spare parts available from Welsh resellers		- See single machines -	Lube-for-life bearings, spare parts available from Italy
	Durability	Up to 10 years (guaranteed for 2)	Up to 20 years, inverter guaranteed for 10	Up to 20 years	- See single machines -	Up to 20 years
Env. & safety	Noise	High noise ≈ 83 dB at 1 m ⁽⁴⁾	Low noise ≈ 61 dB at 1 m ⁽⁵⁾		- See single machines -	?
	Safety	High nominal speed (1,930 RPM) and potentially high runaway speed	Medium nominal speed (1,500 RPM) and runaway speed (≈ 1,950 RPM)		- See single machines -	Low nominal speed (1,000 RPM)

1. Power output limited by the grid connection standard (G83, single phase)
2. Power output arbitrarily set as 4 kW to match the average load at site
3. Based on a standard electricity price of 15.88 p per kWh (standard Scottish Power domestic tariff according to www.uswitch.com/gas-electricity). All of the generated electricity is supposed to be consumed on-site.
4. Test carried out by a Powerspout engineering team on a Pelton HP, very similar to the proposed Turgo HP
5. Standard noise from a 4 kW pump (KSB Etanorm series), measured at a distance of 1 m from the unit outline as per ISO 3744.

FIGURE 7-17. MULTI-DIMENSIONAL COMPARISON REGARDING THE ELECTRO-MECHANICAL EQUIPMENT FOR TY MAWR INCLUDING INSTALLATION BENEFITS, LIFE CYCLE, ENVIRONMENTAL AND SAFETY ASPECTS.

The results from Figure 7-17 provided even more insights on the comparison of turbine technology for Ty Mawr as it highlighted that a PAT would have comparable noise emissions, maintenance needs, durability and ease of operations than a conventional Pelton turbine at only a fraction of the cost. Instead, the adoption of three pico Turgo units would bring about many shortcomings on day-to-day operations such as requirements for manual operation, frequent maintenance and high noise emissions.

7.1.6.2. Projected economic and environmental benefits

As seen in Figure 7-5 the projected energy generation from the site with the adopted PAT (KSB Etanorm 80-65-250) was calculated as approximately 19,000 kWh/year. The expected nominal working conditions of the turbine correspond to a flow rate of 22 l/s, a head of 28 m and a power output of 3.68 kW as per Figure 7-6. With respect to the flow duration curve displayed in Figure 7-4 the PAT will derive only a small portion of the licenced flow as clearly visualized in Figure 7-18.

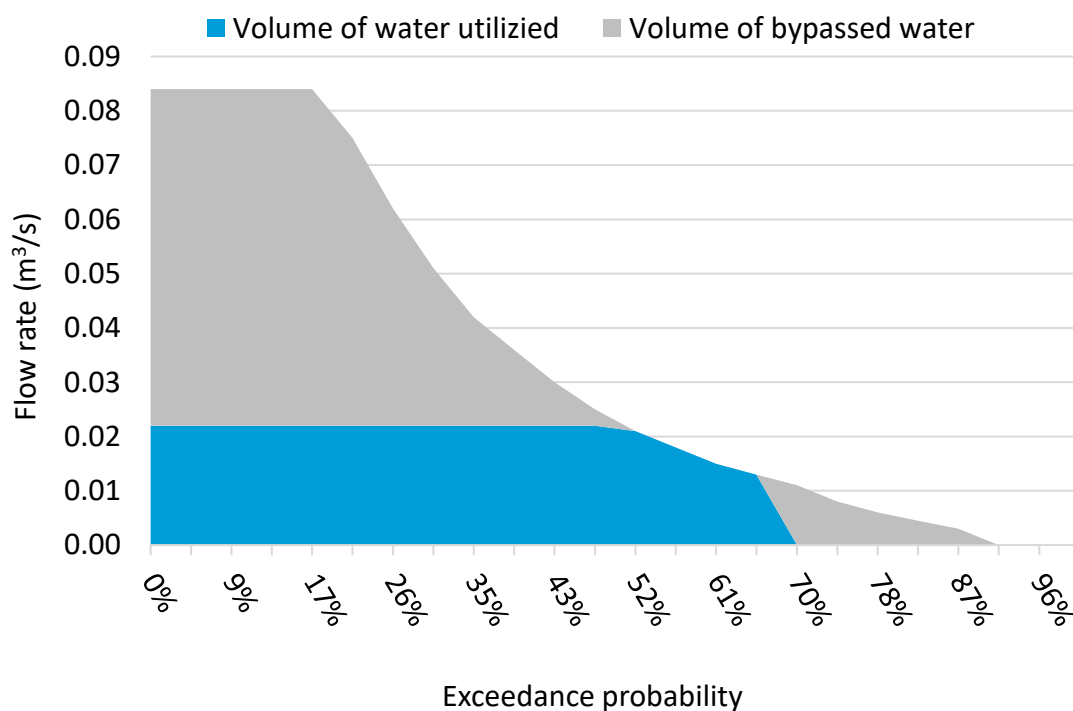


FIGURE 7-18. VOLUME OF WATER UTILIZED VS. BYPASSED.

Along a typical year the power output from the inverter is expected to equal the nominal value of 3.68 kW for about the 50% of time as seen in Figure 7-19. Additionally, for another 20% of time the scheme will be generating a lower power due to the presence of partial flow rates. Eventually, it is expected that the power plant will remain shut down for about 30% of the year.

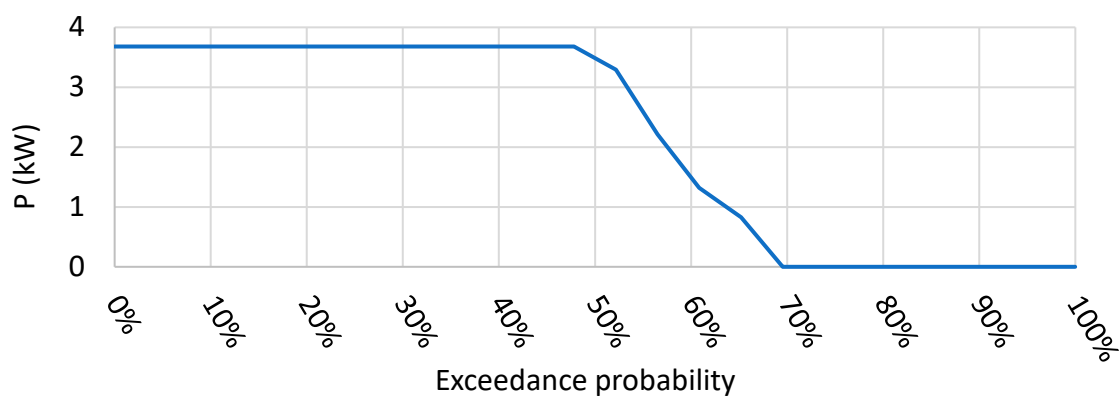


FIGURE 7-19. EXPECTED POWER OUTPUT ALONG A TYPICAL YEAR.

The total cost of the Ty Mawr hydropower scheme, split between the Dŵr Uisce project and the National Trust, was equal to nearly 29,000€ as detailed in Table 7-1.

TABLE 7-1. COST BREAKDOWN OF THE TY MAWR HYDROPOWER SCHEME.

Element		Final cost (GBP)	Final cost (€)	Percentage of overall cost
Planning		6,000	6,720	23%
Equipment	PAT	1,925	2,156	7%
	Generator	907	1,016	4%
	Rectifier	31	35	<1%
	Smoothing capacitor	62	69	<1%
	Inverter and controller	795	893	3%
	Total equipment	3,720	4,166	14%
Civil works	HDPE pipeline	5,000	5,600	19%
	Powerhouse	2,000	2,240	8%
	Intake construction	4,000	4,480	16%
	Other	2,000	2,240	8%
	Total civil works	13,000	14,560	51%
Commissioning		3,000	3,360	12%
Total		25,720	28,806	100%

The cost of the monitoring and control unit and its relative sensors, corresponding to nearly 8,000 €, has been excluded from Table 7-1 since it originated from a requirement of the Dŵr Uisce project but is not necessary for similar future installations. Given the final price tag of the scheme being 28,806€, the cost per installed power was equal to 7,830 €/kW which compares favourably with the figures for pico hydropower reviewed in Chapter 3.5.6.

Supposing that the entire electricity generation will be used as self-consumption, and with an electricity price of 15.88 pence as per Figure 7-17, the expected yearly savings equal roughly 3,400€. With such figure, the simple return on investment is forecast to happen in 8 years and half. It is worth noticing that had the Pelton turbine described in Figure 7-16 been selected the final cost of the whole power plant would have been close to 49,000 € with a payback time of almost 14 years and half, which highlights the relevance of PAT use for MHP installations.

The Levelized Cost Of Electricity (LCOE) with a plant life span of 20 years, an interest rate of 2% and 2.5% of the equipment cost to be spent on maintenance each year corresponded to 0.148 €/kWh.

The scheme operations are expected to offset a significant quota of greenhouse gases each year, equivalent to over 5.3 tons of CO_{2,eq} avoided per year considering an emission factor of 0.28307 kg CO_{2,eq} per kWh of electricity reported for the UK in 2018 (UK Government - Department for Business, Energy & Industrial Strategy, 2018).

Even if the power plant was not operational between March and May 2019 due to the electric issues described in Section 7.1.5, during that time the remote monitoring unit was already collecting data from the flow sensor at the intake which correlates to the flow rate present in the stream through the relationship shown in Figure 7-13. The measured flow duration curve for those three months of 2019 is presented in Figure 7-20, together with the one predicted by the LowFlows software (upon which the hydro scheme was sized) for the same period of the year.

The comparison between the two curves shown in Figure 7-20 suggests that March, April and May 2019 have been extraordinarily dry when compared to the expected runoff from the Wybrnant watershed. However, it may also be possible that the results from the LowFlows software were overestimating the flow rate of the Wybrnant river as they were not based on actual field measurements from a river gauge.

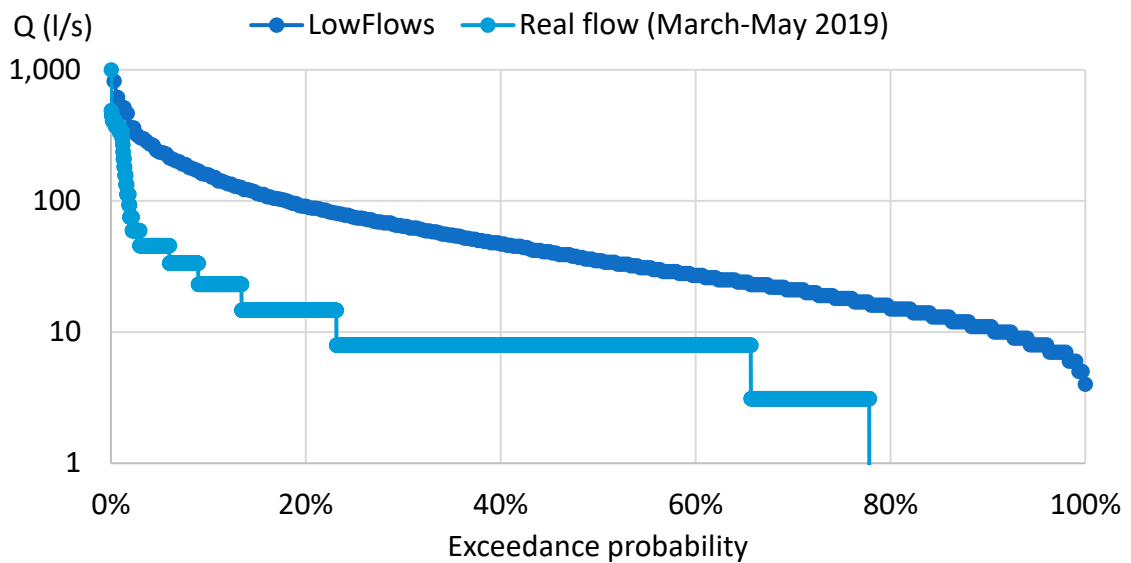


FIGURE 7-20. COMPARISON BETWEEN THE EXPECTED STREAM FLOW RATE AND THE MEASURED ONE BETWEEN MARCH AND MAY 2019.

7.1.6.3. Performance analysis of the power plant

As mentioned earlier in Section 7.1.5 the turbine and inverter were eventually commissioned in the beginning of September 2019 and the scheme was started on 7/10/2019. An initial issue with the valve actuator meant that the turbine was offline for most part of October, but it resumed operations in the beginning of November after a fix. The power generation from the beginning of October 2019 and the end of January 2020 is shown in Figure 7-21.

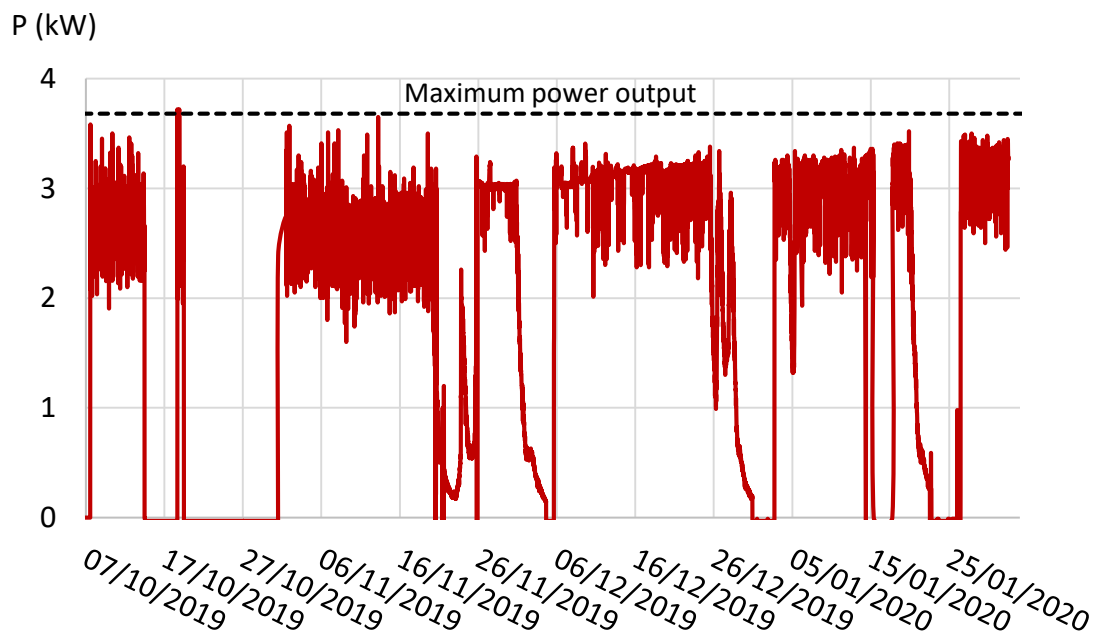


FIGURE 7-21. POWER GENERATION OF THE TY MAWR WYBRNANT HYDRO SCHEME BETWEEN OCTOBER 2019 AND JANUARY 2020.

As seen from Figure 7-21, the power output from the inverter has remained for most of the time between 2 and 3.5 kW when enough water was available at the intake. On 21/11/2019 the inverter power curve was edited resulting in a more stable power output. The monthly energy generation across the first four months of operations is displayed in Figure 7-22.

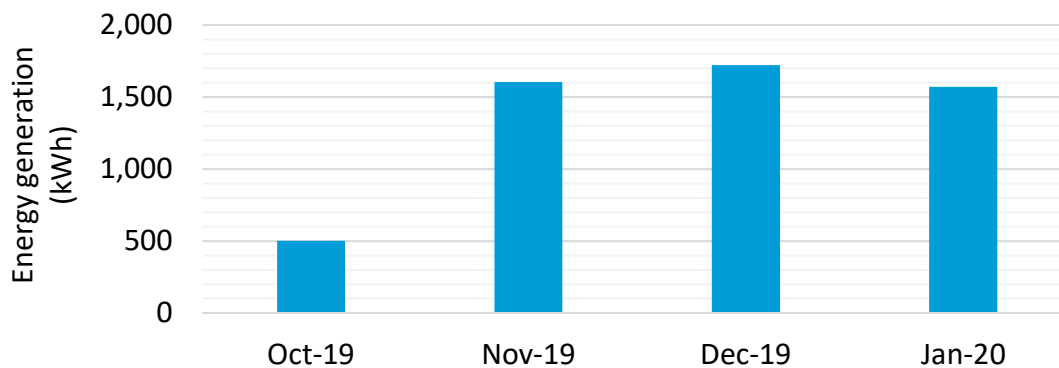


FIGURE 7-22. MONTHLY ENERGY GENERATION FROM THE TY MAWR WYBRNANT TURBINE BETWEEN OCTOBER 2019 AND JANUARY 2020.

From Figure 7-22 it can be seen that the average monthly energy generation between November 2019 and January 2020 corresponded to 1,650 kWh/month. If this trend is confirmed, the scheme can realistically generate the target 19,000 kWh/year predicted by the PAT selection software as per Figure 7-5.

7.2. BLACKSTAIRS WATER TREATMENT WORKS, IRELAND

7.2.1. OVERVIEW

As the main focus of the Dŵr Uisce project was to investigate and demonstrate the feasibility of energy recovery in the water industry, selecting a demo site located in a water network was of paramount importance. Thanks to the collaboration with the National Federation of Group Water Schemes (NFGWS), the energy recovery potential of a series of rural Irish group water networks was investigated. After a first screening phase, a significant potential for energy recovery was identified at the site of the EPS-managed Blackstairs water treatment plant in Co. Wexford.

The water feeding the Blackstairs treatment works is abstracted from a stream on the side of the Slievebaun peak and led to the plant until entering a raw water storage tank. An overall view of the plant is shown in Figure 7-23, with the large raw water storage tank visible in the background. Four separate intake lines are present of which three are gravity-fed and one is pumped and serves as a backup in case of prolonged draughts.



FIGURE 7-23. OVERALL VIEW OF THE BLACKSTAIRS WATER TREATMENT PLANT.

The main intake line draws water from the Connells intake through a 2-km long DN150 pipeline until the raw water storage tank at the treatment work. A schematic of the mentioned pipeline is shown in Figure 7-24, while Figure 7-25 depicts a drawing of the terminal part of the Connells intake line on entering the treatment works.

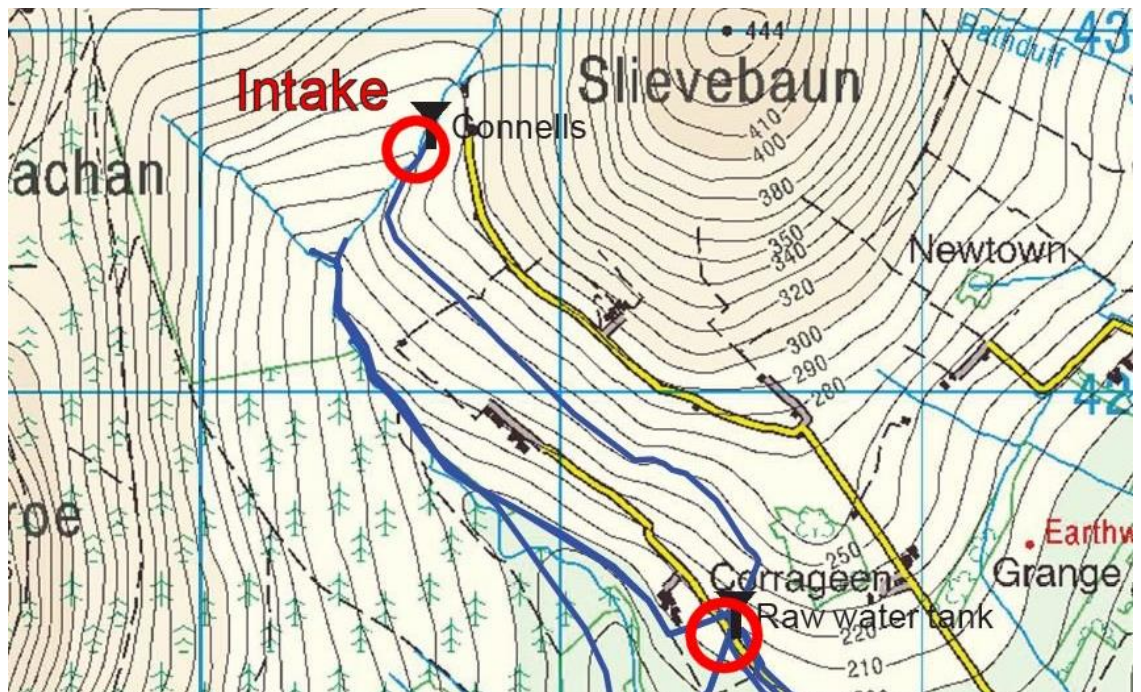


FIGURE 7-24. NETWORK MAP OF THE WATER PIPELINE FROM THE CONNELLS INTAKE TO THE RAW WATER TANK.

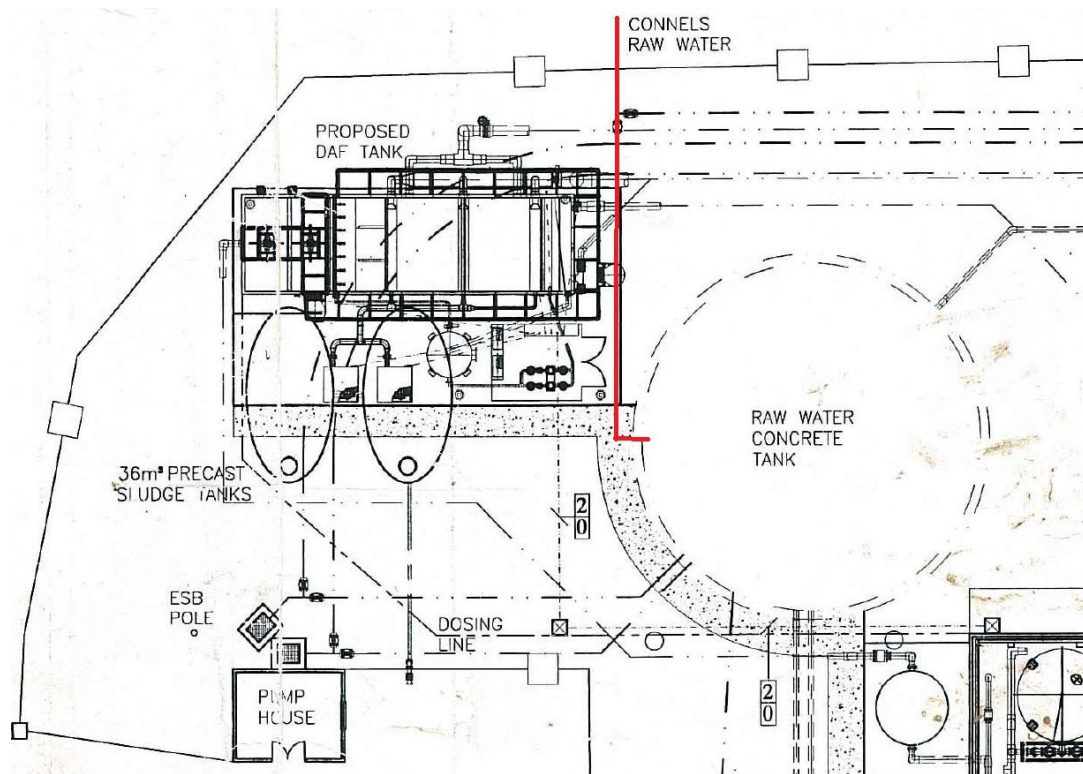
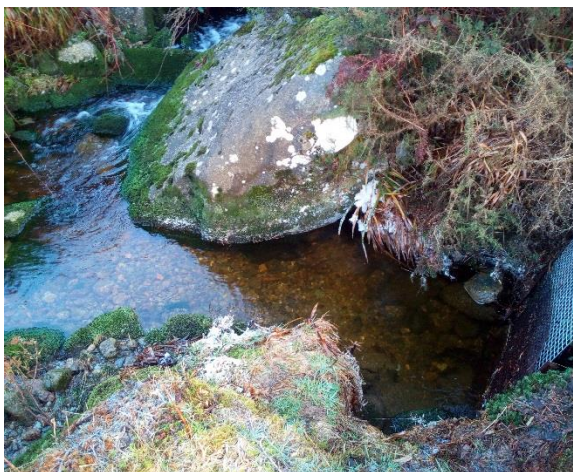


FIGURE 7-25. DRAWING OF THE TREATMENT WORKS WITH HIGHLIGHTED IN RED THE TERMINAL PATH OF THE CONNELLS RAW WATER SUPPLY LINE (COURTESY OF EPS).

The path of water from the river bed to the storage tank consists of several steps shown in Figure 7-26: a water intake with screening mesh, a settling tank from where the buried pipeline departs, a rising branch at the base of the raw water storage tank within the premises of the treatment works which eventually leads water on top of the tank itself.

a)



b)



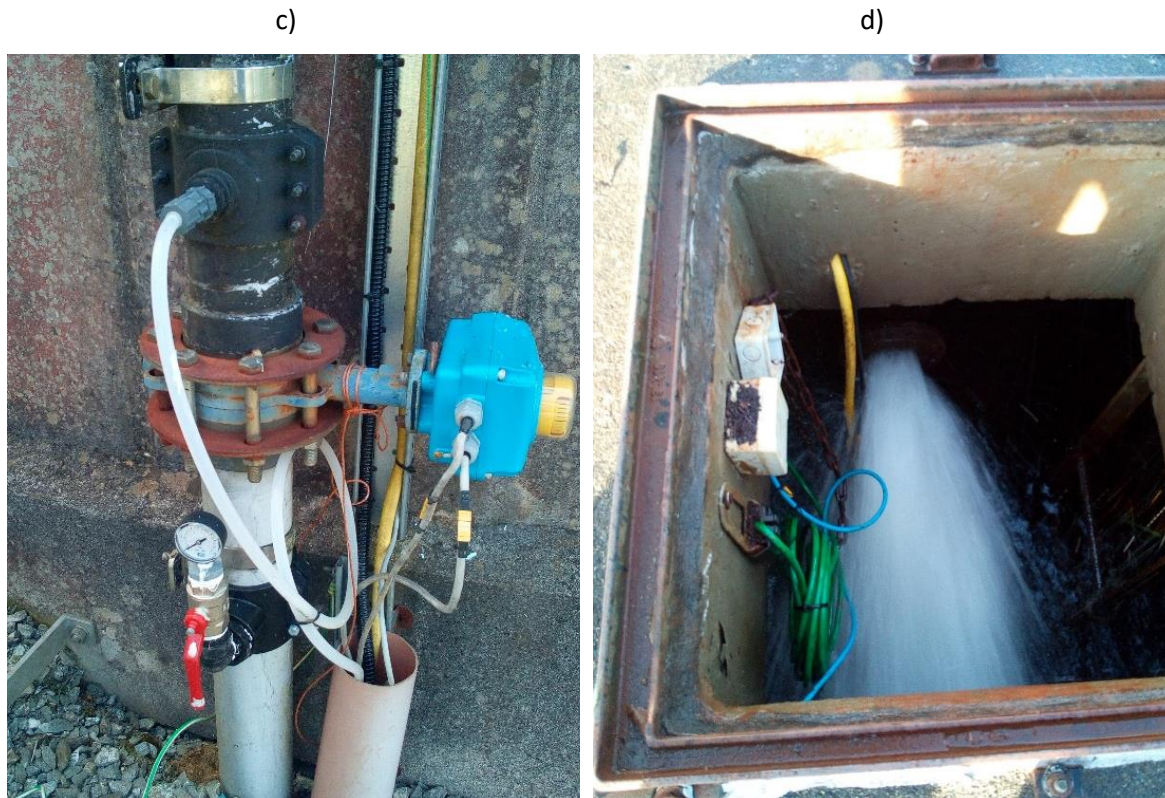


FIGURE 7-26. PORTIONS OF THE CONNELLS SUPPLY LINE: A) RIVER INTAKE WITH SCREEN MESH; B) SETTLING TANK AT THE START OF THE PENSTOCK; C) THE BASIS OF THE RAW WATER STORAGE TANK, WITH VISIBLE ON-OFF AUTOMATED VALVE REGULATING THE INFLOW; D) WATER FLOW ENTERING THE TANK.

7.2.2. DESIGN DATA

The gross hydraulic head calculated through the Google Earth commercial software corresponds to approximately 63 m as an elevation difference between the Connells intake settling tank location (278 m a.s.l.) and the top of the raw water tank (215.8 m a.s.l.).

At the beginning of March 2018 a flow meter and a pressure gauge were installed on the terminal part of the pipeline at the basis of the raw water tank in the location shown in Figure 7-26 (c). The flow records were collected manually between 11/3/2018 and 8/6/2018 by the plant operator at times when the Connells supply line was in use and showed a flow rate variability in the range 8-32 l/s during times when the Connells intake was operated.

Since the flow rate through the Connells supply line depends strictly on the availability of water in the source river, and in the absence of existing hydrological gauges on it, it was decided to compare the recorded flows to other proxy measures which could explain the variability found. The first indicator chosen was the amount of daily precipitation from the nearest weather stations

managed by Met Éireann which were identified as the one in Carlow (35.4 km away from the intake) and the so-called “Johnstown II” station close to the Johnstown Castle in Co. Wexford (34.2 km away). All three data series are plotted in Figure 7-27, from where it appears that the prolonged scarcity of precipitation which occurred from the beginning of May 2018 until the beginning of June may explain the generally negative trend of recorded flow rates (Met Éireann, 2018).

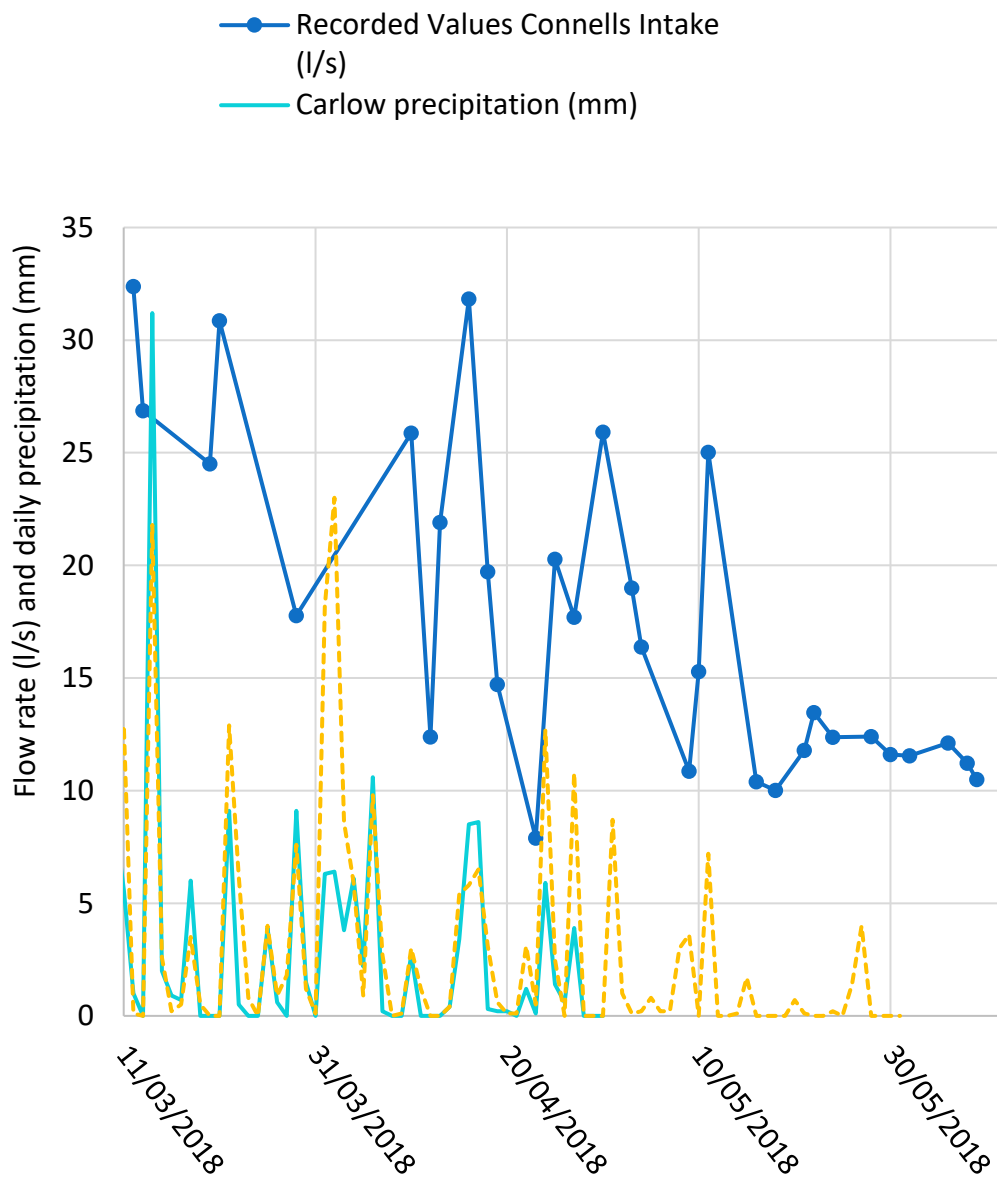


FIGURE 7-27. RECORDED FLOW RATES FROM THE CONNELLS INTAKE COMPARED TO THE PRECIPITATION RECORDED BY TWO NEARBY WEATHER STATIONS (MET ÉIREANN, 2018).

As a second step, the Connells recorded flows have been compared to the flow values measured by two nearby river gauging stations managed by the Office of Public Works (Office of Public Works, 2018):

- St. Johns Bridge in Enniscorthy located at 16.8 km from the Connells water source on the river Urrin, and which receives the water of all the springs from the east side of the Blackstairs mountains;
- Graiguenamanagh hydrometric station on the river Barrow, located on the western side of the Blackstairs mountains at 9.7 km.

The position of the two river level gauging stations as well as the two weather stations previously described is shown in Figure 7-28 with respect to the location of the Connells intake.

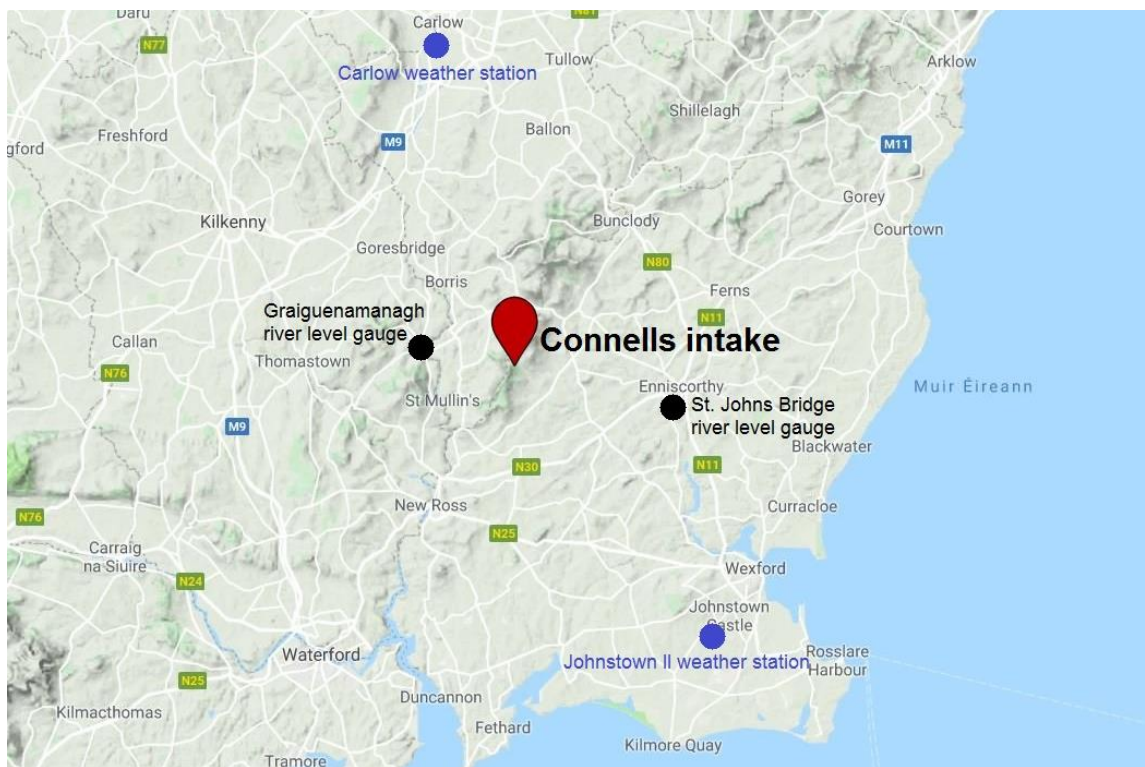


FIGURE 7-28. LOCATION OF THE ANALYSED RIVER LEVEL GAUGES AND WEATHER STATIONS WITH RESPECT TO THE CONNELLS INTAKE.

The results of the comparison between the flows at Connells intake and the records from the two river gauging stations considered is plotted in Figure 7-29.

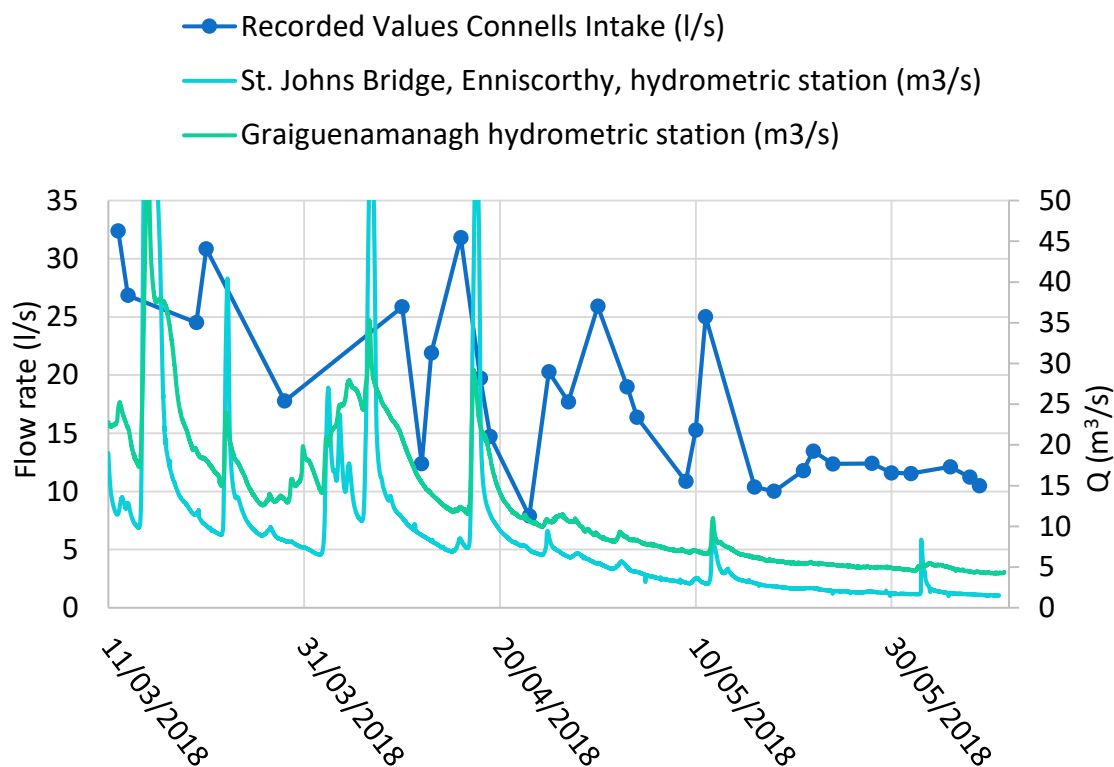


FIGURE 7-29. FLOW RECORDS FROM CONNELLS INTAKE

While the results from Figure 7-27 already indicated a plausible correlation between the general decrease in flow available at Connells intake and the prolonged scarcity of precipitations, the similarity between the downwards trends of recorded flow rates from the two nearby rivers with the measured flow in Figure 7-29 is striking. However, the nature of flow records from Connells intake which were taken manually only on days where an operator was present on site did not allow capturing the full profile of water availability at the source.

The maximum flow rate from the Connells intake occurring with the fully open inlet valve was found to correspond to 32 l/s, while the two consecutive low flow records of the 11th and 23rd of April seen in Figure 7-29 were caused by the accumulation of leaves against the intake mesh screen.

An EPANET computer model of the whole Connells intake line whose layout is shown in the map of Figure 7-24 was generated and calibrated based on the recorded pressure and flow values. The resulting net available hydraulic head available for energy recovery is shown in Figure 7-30 together with the resulting power output assuming a fixed flow-to-wire turbine efficiency of 65%.

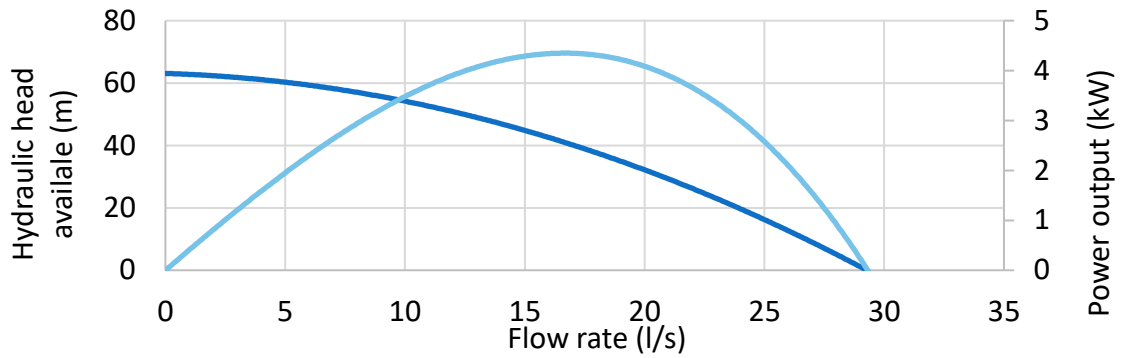


FIGURE 7-30. NET HYDRAULIC HEAD AND POWER OUTPUT AS FUNCTION OF FLOW RATE IN THE EPANET MODEL

7.2.3. EQUIPMENT SIZING AND DESIGN

The potential energy available for recovery from the Connells intake line could most conveniently be exploited by a centrifugal PAT to be installed in the terminal section of the pipeline before entering the raw water storage tank.

Based on the results of Figure 7-30 selecting a design flow rate and head of 18 l/s and 40 m respectively would lead to the maximum power output of around 4 kW. The design software developed in Chapter 5 was applied on the selected site and helped identifying the ideal characteristics of the main PAT and generator set to be chosen. The results of the analysis are displayed in Figure 7-31.

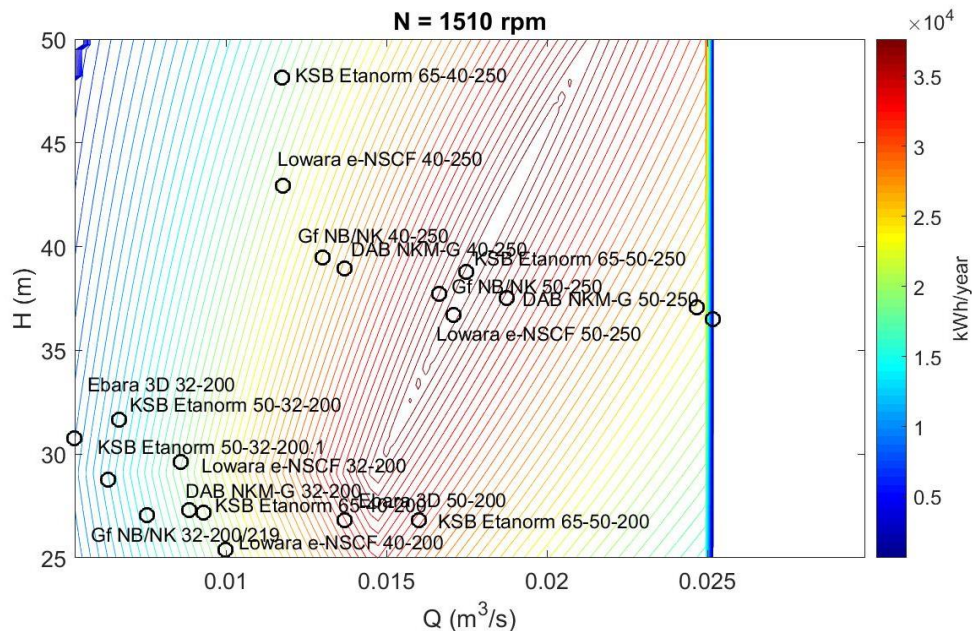


FIGURE 7-31. RESULTS OF THE PAT SELECTION SOFTWARE FOR THE BLACKSTAIRS SITE

The results from Figure 7-31 suggest that a 50-250 standardized centrifugal PAT rotating at 1520 RPM (corresponding to the use of a four-pole asynchronous motor as generator) would produce a net power output of 4.3 kW.

If operated 24-h all year round this would lead to 37,000 kWh of generated electricity worth 6,070€ when considering an average industrial electricity price for SMEs in Ireland of 0.164€ per kWh (Costs of Doing Business in Ireland, National Competitiveness Council, April 2016).

The expected characteristic flow-head curve of the selected KSB Etanorm 65-50-250 PAT obtained through Equation (36) at Page 115 is plotted in Figure 7-32 against the available system head.

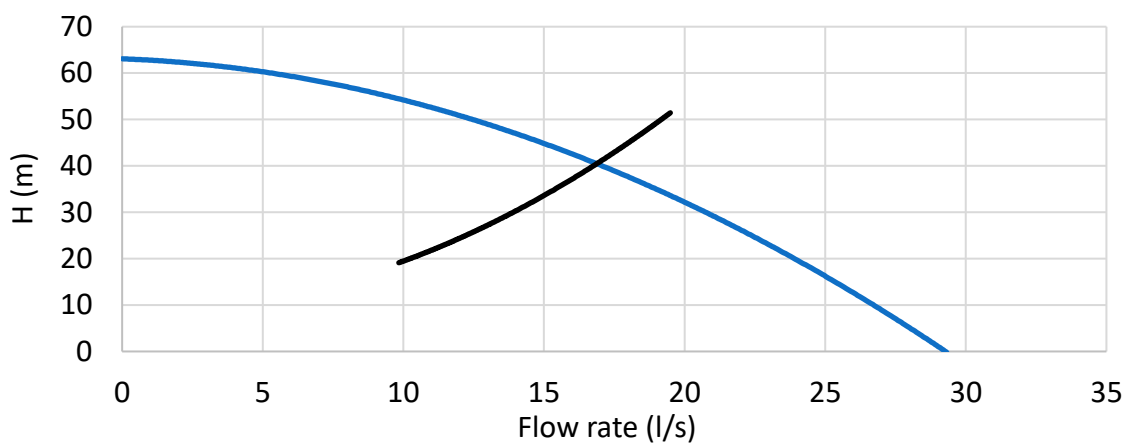


FIGURE 7-32. CHARACTERISTIC CURVE OF THE KSB ETANORM 65-50-250 PAT (BLACK) PLOTTED AGAINST THE AVAILABLE HEAD AT THE END OF THE CONNELLS INTAKE LINE (BLUE)

The expected purchase price and features of the proposed PAT are described in Table 7-2.

TABLE 7-2. CHARACTERISTICS OF THE PROPOSED PUMP AS TURBINE (PAT) FOR BLACKSTAIRS SITE

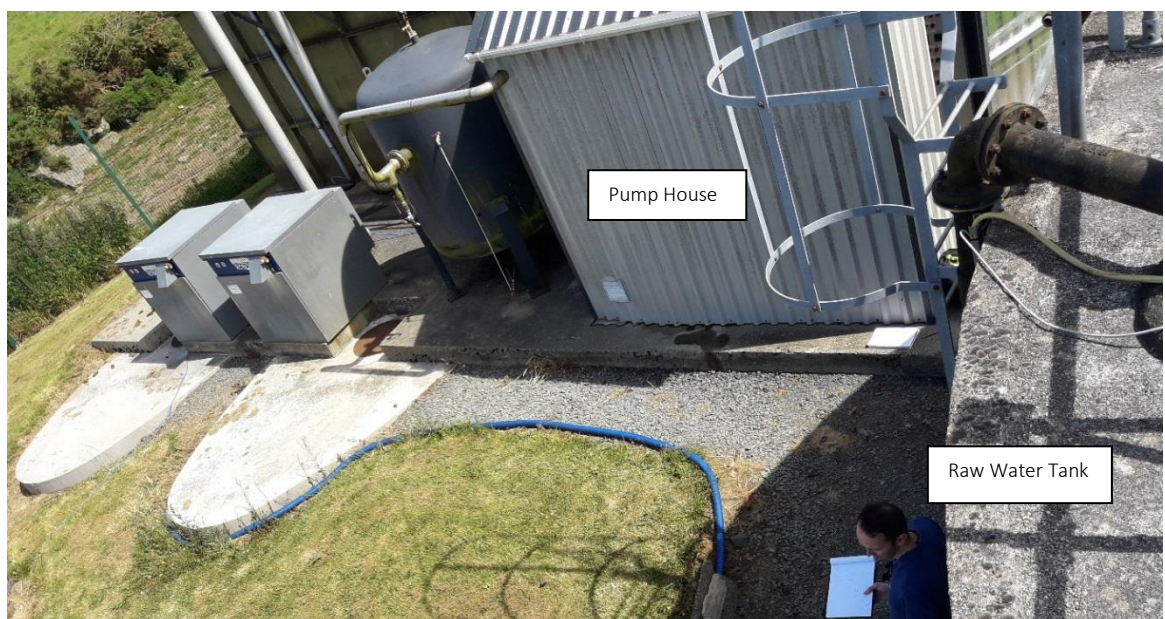
Type	Centrifugal standardized hydraulic pump
Model	65-50-250
Shaft speed (RPM)	1,520
Indicative price of PAT excl. VAT (€)	2,400 (± 20%)
Indicative price of generator excl. VAT (€)	420 (± 20%)
Nominal turbine head (m)	38.5
Nominal turbine flow rate (l/s)	17.6
Indicative hydraulic efficiency (-)	65%
Indicative impeller diameter (mm)	250

A fixed turbine shaft speed of 1,520 RPM was selected, corresponding to the use of a four-pole asynchronous motor as generator. The proposed design consists of installing a PAT in a bypass of the Connells supply line in the proximity of an existing pumphouse adjacent to the raw water tank. The pre-existing facilities are depicted in Figure 7-33 while a schematic drawing of the same is shown in Figure 7-34.

a)



b)



c)

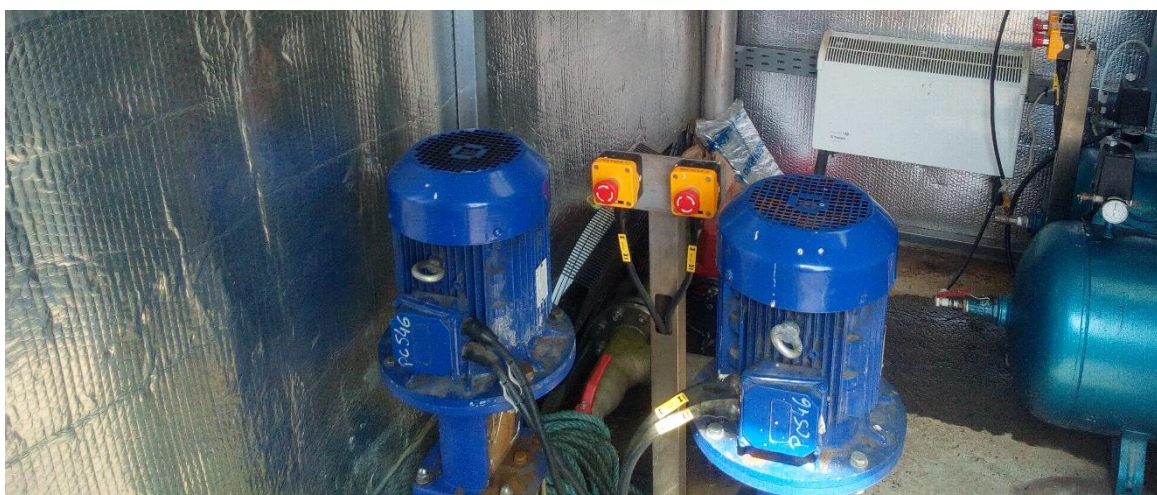


FIGURE 7-33. VIEWS OF THE PRE-EXISTING INSTALLATION: A) SIDE VIEWS OF THE TREATMENT WORKS WITH THE RAW WATER TANK IN THE FOREGROUND; B) AERIAL VIEW FROM THE RAW WATER STORAGE TANK; C) INSIDE VIEW OF THE EXISTING PUMPHOUSE.

As it can be seen in Figure 7-33 (c) although the inside of the pump house is currently occupied by a set of hydraulic and electric ancillary devices since it is built on a metal frame the whole shed could be extended laterally.

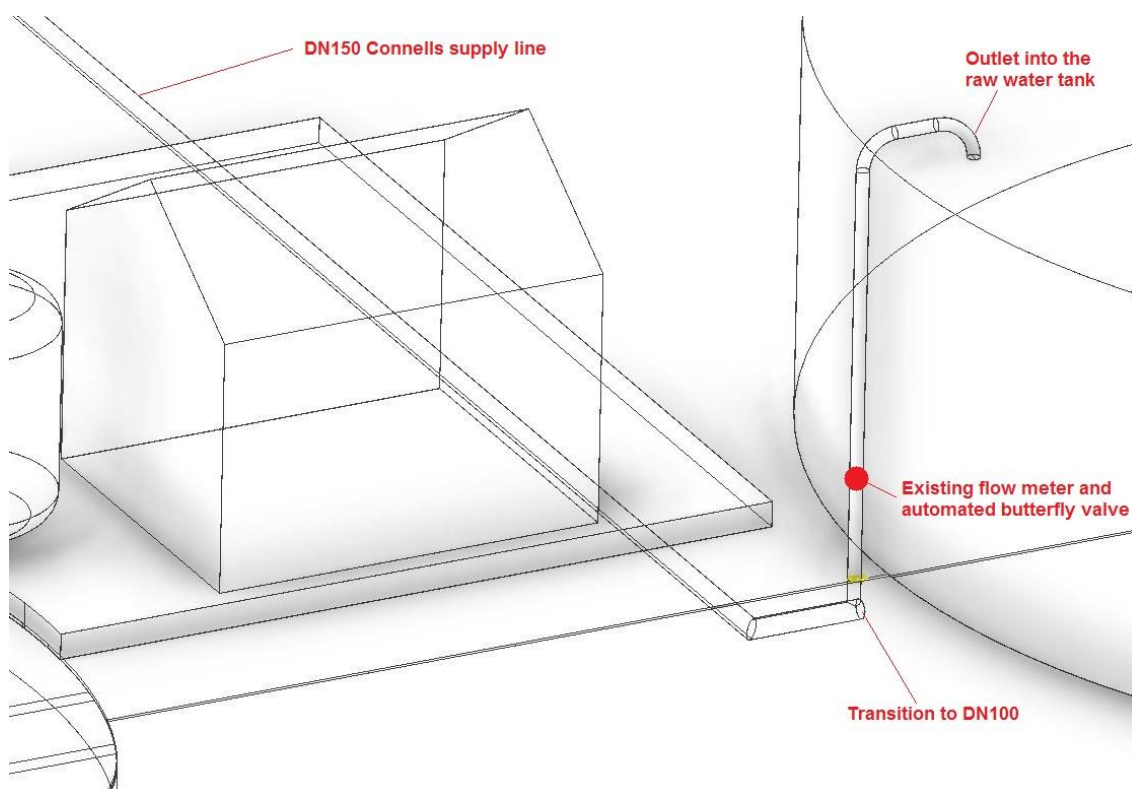


FIGURE 7-34. 3D RENDERING OF THE EXISTING FACILITIES

In order to house the new set of equipment it was suggested to cast a 1x4 m extension of the existing concrete slab and expand the existing metallic shed/pumphouse as shown in Figure 7-35, while the proposed plant layout is highlighted in Figure 7-36.

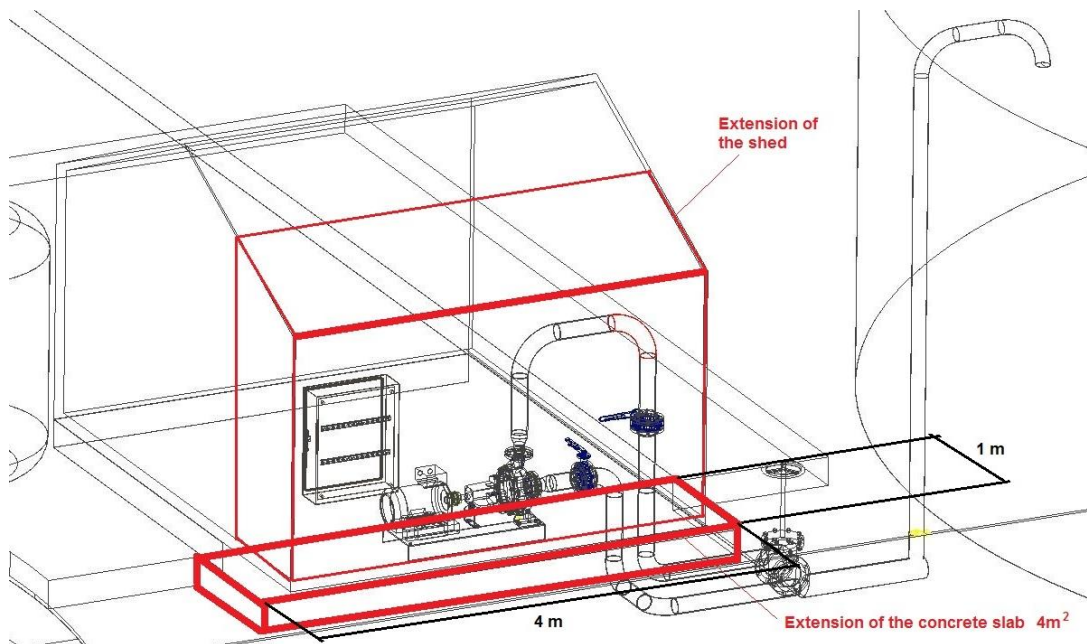


FIGURE 7-35. PROPOSED INSTALLATION WITH HIGHLIGHTED THE MAJOR CIVIL WORKS.

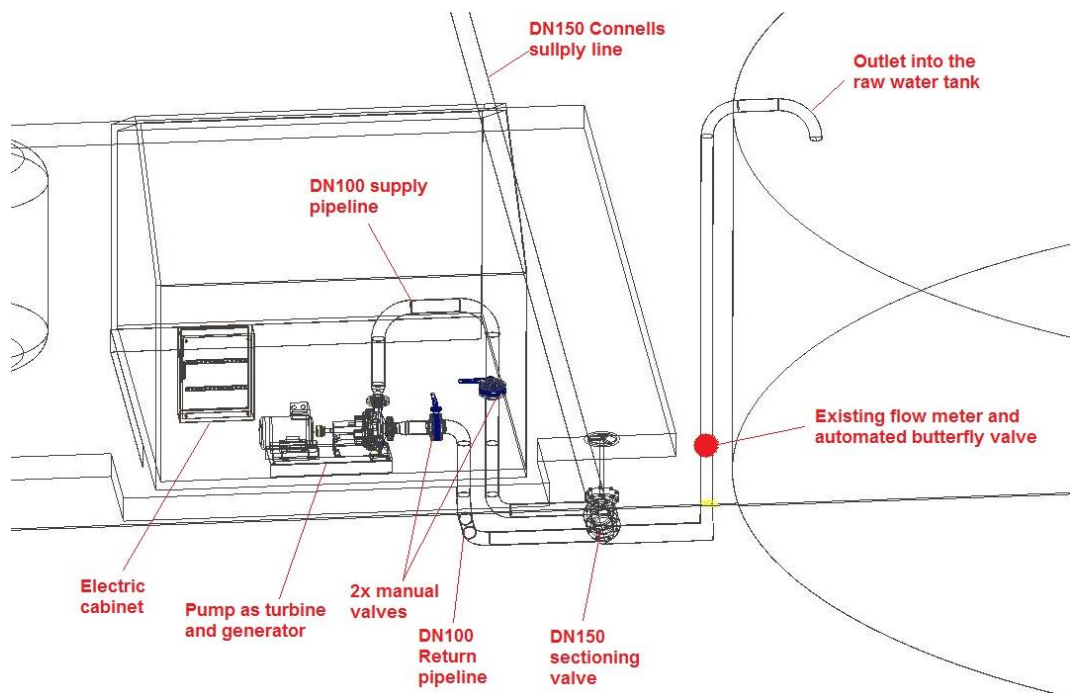


FIGURE 7-36. PROPOSED PLANT LAYOUT WITH NOVEL ELEMENTS HIGHLIGHTED.

As seen in Figure 7-35 and Figure 7-36 the needed civil works consist mainly of the installation of the pipe bypass and the extension of the concrete slab to support the turbine, while other existing elements would be maintained in their actual state.

Given the self-synchronising nature of induction generators, the electric circuit needed will be extremely simple and include a soft starter, an over load protection and a grid interface relay certified for the EN50438 standard as required by the Irish legislation. An indicative electric circuit wiring diagram is displayed in Figure 7-37.

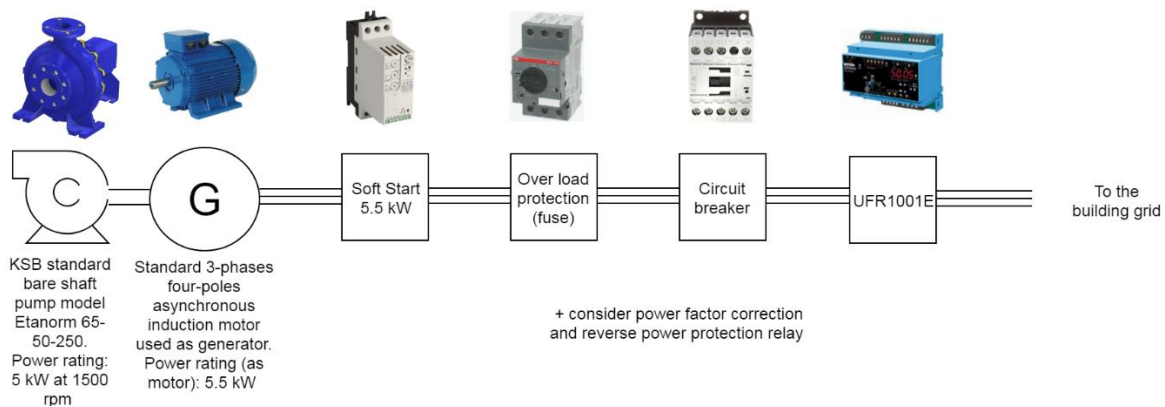


FIGURE 7-37. INDICATIVE WIRING DIAGRAM OF THE TURBINE.

7.2.4. PROGRESS OF CONSTRUCTION PHASE

The construction of the Blackstairs hydro demo site took place along different phases:

1. October 2018: after a competitive selection process EPS was appointed to carry out the works
2. December 2018 – February 2019: the civil works were carried out while the PAT and generator were being assembled on a frame in a mechanical workshop
3. March-April 2019: the electro-mechanical installation was completed, with first startup on 18/4/2019
4. May 2019: the official opening day of the demo site took place on 2/5/2019
5. June-September 2019: a raw water tank level sensor and a new PLC was installed by EPS to automatically operate the bypass valve and exclude the PAT in the event of low water level.

A gallery showing the progress of the field works is displayed in Figure 7-38.

a) Initial situation



b) Excavations to expose the pipeline



c) Installation of a rising tee



d) Casting of the concrete slab extension



e) *Turbine installation*



f) *Final pipe layout*



FIGURE 7-38. PROGRESS OF FIELD WORK: A) INITIAL SITUATION; B) EXCAVATIONS TO EXPOSE THE PIPELINE; C) INSTALLATION OF A RISING TEE; D) CASTING OF THE CONCRETE SLAB EXTENSION; E) TURBINE INSTALLATION; F) FINAL PIPE LAYOUT.

7.2.5. COMMISSIONING

After the completion of the installation in April 2019 the turbine bypass created by the contractor included a number of additional fittings and items likely to cause higher head losses with respect to provided design previously described in Figure 7-35 Figure 7-36. In particular, the final pipe layout shown in Figure 7-38 (f) included:

- twelve 90° bends instead of the 7 planned;
- 5 metres of DN100 PVC pipeline PVC;
- One sudden diameter enlargement from DN50 to DN100 after the turbine outlet.

Such additional head losses with respect to the ones accounted for in the initial design phase are expected to limit the power output to 4 kW instead of the 4.3 kW predicted by the PAT selection software.

The runaway flow rate and head of the PAT were measured in April 2019 before connecting the generator to the electric grid and were equal to 9.7 l/s and 45.5 m respectively. Instead, the first turbine start-up was performed on 18/4/2019 and on that day the unit managed to generate a

stable output of 2.93 kW with a flow of 15 l/s and head of 37.3. Instead, on the opening day which took place on 2/5/2019 the power output was 2.15 kW at a flow rate of 13.3 l/s and head of 32 m. Eventually, the turbine started automatic operations on 7/10/2019 and on the day it generated 3.35 kW at a flow rate of 16.1 l/s and a head of 42 m. The recorded working points and runaway condition are plotted in Figure 7-39 together with the expected PAT characteristic curve previously displayed in Figure 7-32.

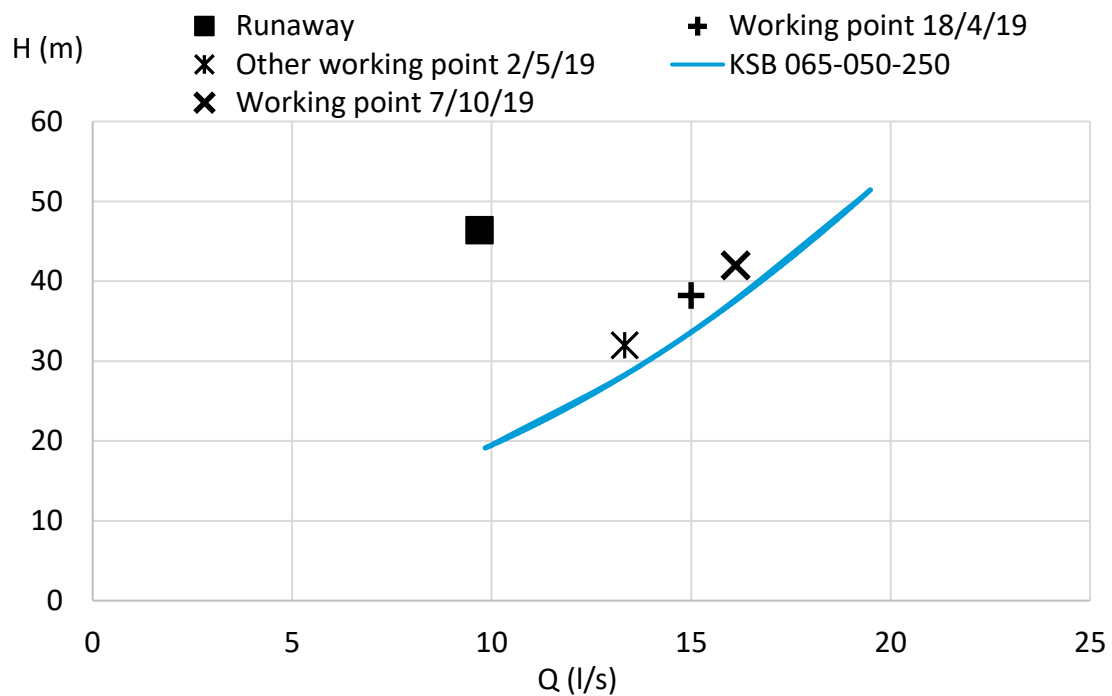


FIGURE 7-39. COMPARISON BETWEEN THE EXPECTED AND MEASURED PAT PERFORMANCE.

The key findings of Figure 7-39 are that the two measured working points are remarkably close to the PAT curve predicted by the Yang method and therefore by the selection software presented in Chapter 5. Also, the position of the runaway point is coherent with the available literature on PATs. After considering the electric conversion losses of the generator, the hydraulic efficiency of the PAT corresponded to 61% at 13.3 l/s, 63% at 15 l/s and 60% at 16.1 l/s. Such data compare favourably to the expected efficiency of 65% at 17.6 l/s previously stated in Table 7-2.

An investigation was launched to identify the causes of the low turbine output power, as the maximum value ever recorded was a mere 3.35 kW with respect to the expected 4 kW. After a high-precision geographical survey carried out on 3/12/2019 it was eventually discovered that the gross head between the river intake and the raw water tank ceiling was only 55.65 m as opposed to the 62 m calculated during the design phase from both online and offline maps.

7.2.6. PERFORMANCE ANALYSIS

7.2.6.1. Projected economic and environmental benefits

Provided that a sufficient water flow is available at the source, and that an air relief valve is installed along the pipeline, the PAT at Blackstairs is expected to operate at a constant working point of 17.6 l/s and 38.5 m with a power output of 4 kW.

The total project cost corresponded to 29,360 €, which included remote monitoring and associated flow and power meters that are non-essential costs for future installations. In relation to the power output, the Blackstairs scheme had a cost of 7,340 €/kW which is coherent with the reported unit cost of other PATs installed in water networks as reported in Table 5-8. In a second stage, the plant operator EPS demanded additional works worth 5,000 €+VAT for a complete hands-off automation of the scheme to ensure no interference with their operations would occur. However, these costs have been excluded from the total since they originated from a specific EPS requirement.

The projected energy generation predicted by the software is equal to 37,000 kWh of electricity per year which would correspond to 16.2 tons of avoided CO_{2eq} emissions by using the 2018 carbon intensity of 437 g CO_{2eq} /kWh (SEAI, 2018).

As discussed in Section 7.2.3 the expected energy savings per year are equal to 6,070 € which would lead to a return of the investment within 5 years and an LCOE of 0.079 €/kWh.

In conclusion, it is worth remarking that if the Connells supply line at Blackstairs had to be rebuilt as new with larger nominal diameter (at least DN250 instead of DN150) in order to accommodate a larger PAT than the one installed, indicatively generating 10 kW with a flow of 30 l/s and 50 m of head. Instead, a large portion of the head is currently being dissipated as friction.

7.2.6.2. Performance analysis of the power plant

From the opening day on 2/5/2019 up to September 2019 no more data was collected on the turbine since the plant operator decided to bypass the PAT and leave it unused. However, between June and September 2019 a new PLC was installed to automatically operate the bypass and thus allow for hands-off turbine operations. The demo site was eventually put back online after the completion of such works on 7/10/2019 and the power output from then until the end of January 2020 is displayed in Figure 7-40.

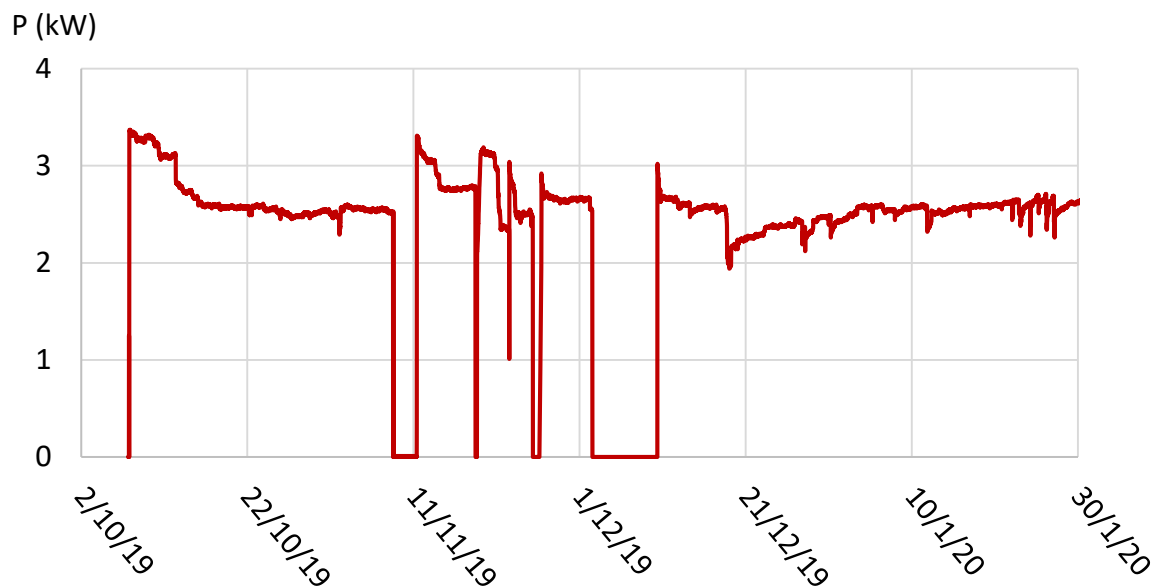


FIGURE 7-40. POWER OUTPUT FROM THE BLACKSTAIRS PILOT TURBINE BETWEEN OCTOBER 2019 AND JANUARY 2020.

As seen from Figure 7-40, the power generation of the turbine during the first four months of operations has remained stable in the 2-3 kW range according to the water availability at the intake. The scheme was operational for most of the time, except for a few dry days during which the turbine was bypassed and a whole week in December when the turbine was stopped for a shaft alignment. The monthly energy generation during the same period is displayed in Figure 7-41.

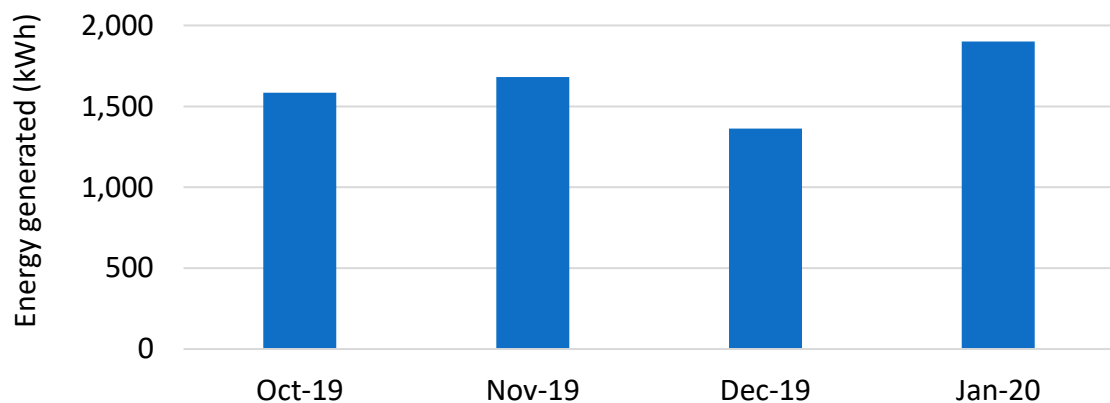


FIGURE 7-41. MONTHLY ENERGY GENERATION FROM THE BLACKSTAIRS PILOT PLANT BETWEEN OCTOBER 2019 AND JANUARY 2020.

As seen in Figure 7-41, the average monthly energy generation during the first four months of operations corresponded to 1,630 kWh/month. If the trend was confirmed, the expected yearly generation would be just short of 20,000 kWh/year which is less than the 37,000 kWh/year predicted by the software in Section 7.2.3. However, the difference can be ascribed to the lower gross head (55.65 as opposed to 62 m) and higher pipe head losses with respect to the ones expected during the design phase.

8. DISCUSSION

The main findings from Chapters 5, 6 and 7 are summarized in the following pages, outlining the relevance and impact which the work is expected to have in the field.

CHAPTER 5

Comparison of empirical methods for PAT performance prediction

Many methods to predict the behaviour of a PAT from its performances in pump mode already present in the literature were reviewed, as most of them were developed between 1957 (Stepanoff) and 2012 (Yang). Each of those methods had been developed upon a sample of PATs whose performances were known to the original authors. In many cases the sample was limited to less than 5 machines while two authors based their methods upon data from tens of units (e.g. Yang 2012 and Alatorre-Frenk, 1994). Even if a certain method proves reasonably accurate for a single machine, another one may give overall better results on a larger sample of PATs. Therefore, it was decided to compare the available methods in a statistically relevant way against a large database gathered from academic and industrial sources. This is believed to be the largest such database compiled in literature to date.

The results showed that in some cases the error on the flow or head coefficient determination arising from using nondimensional methods was close to 200%, while in others was close to zero. However, averaging out the errors for low and high specific speed PATs it became evident how some of those proved to be consistently better when tested on more than 100 machines. In particular, the best two methods were the Yang and Alatorre-Frenk models which, not by chance, were the ones built upon a larger PAT performance database than other methods.

Eventually, it was suggested to adopt the Yang method for all PATs to be sized through the selection software. Not only did it perform slightly better than the Alatorre-Frenk with PATs of different specific speed, but it was unbiased since it utilized data from a Chinese manufacturer. Instead, the database included in this PhD thesis had almost 60 PATs which were already used by Alatorre-Frenk to build his own model.

As seen in Figure 5-2 and Figure 5-3 the Yang method on average has generated an average error on the flow and head determination of 16.2% for PATs with $N_s < 60$ and 18.8% for units with $N_s > 60$ which would be considered acceptable for most practical applications.

Ultimately, the relevance of the findings described in Section 5.1 lays in the fact that to the knowledge of the author no such cross-comparison of PAT performance prediction methods had been carried out to date, with other respective authors all claiming the superiority of their own method developed upon a limited sample of machines. Furthermore no comprehensive benchmarking against a very large database of known PAT curves has been conducted either. Eventually, knowing which one among the available methods performs better on a large sample of pumps and which average uncertainty is to be expected by applying it is a key information towards a robust PAT selection methodology.

This work was therefore an important step towards removing barriers against and improving confidence in the use of PATs by hydropower designers, where clear guidance on the conversion method to select is now given (from the large list of methods available). In addition a clear expectation on the accuracy of results obtained has been quantified by this research.

Development of a model for PAT efficiency estimation

While the maximum efficiency of PATs was found to vary greatly in the literature, many authors ascribed it uniquely to scale effects for which a larger turbine would have higher peak efficiency. However, a formulation was found in the literature correlating the efficiency of centrifugal pumps to their nominal flow Q and specific speed N_s . While Q relates to the above mentioned scale effect, N_s is a composite parameter which includes information on the PAT flow, head and speed.

Eventually, a new set of coefficients was derived to work with the pre-existing equation and estimate the rated efficiency of PATs from Q and N_s . This was achieved by means of a least squares optimization against a large database of 280 pumps whose PAT performances were predicted by means of the Yang method as recommended in Section 5.1.

The development of such PAT efficiency prediction method is another key improvement to the present state of the knowledge, as it allows for a more precise estimate of the performance and benefits that a PAT can achieve at any determined site once its main operating parameters are known (Q , H and N). Without such knowledge, many authors until now have considered an arbitrary PAT efficiency (e.g. 65%) when calculating the potential for energy production at an early design stage (Gallagher, et al., 2015b). However, as the effective peak mechanical efficiency of a PAT was found to vary between 40% and 90%, the use of a fixed arbitrary value is likely to have led to significant under/overestimation of the energy output achievable in practice.

Prediction of PAT performance away from BEP

While Sections 5.1 and 5.2 focused on the problem of identifying the BEP of a pump working as a PAT (flow, head and efficiency), Chapter 5.3 revolved around the issue of predicting how a PAT will perform when operating under conditions other than their rated BEP.

Several polynomials have been proposed in the literature by various authors in order to predict the full shape of the characteristic curves of a PAT knowing its BEP. Besides, in the literature it has been proven that PATs with higher N_s have steeper head and power characteristic curves than those with lower N_s . However, the mentioned polynomials all relied on fixed coefficients and therefore did not cater for the above described phenomenon.

Based on analysis performed on 112 PATs of which the full characteristic curve was known, a new set of polynomials was derived whose coefficients vary according to the N_s . When compared to the other pre-existing methods, the new model gave a considerable improvement in accuracy when predicting the full head and power curves of a PAT reaching an overall R^2 coefficient of 97%. This is a remarkable improvement over the previously existing methodologies, and it can provide a hydro designer with a tool to reliably predict the shape of the full characteristic curve of a PAT designed for a determined site and therefore evaluate more accurately its behaviour over the full range of expected flow and head operating conditions.

Finally, the partial efficiency of PATs was compared to that from conventional turbine classes demonstrating visually that while the efficiency itself is comparable to that of a Francis or propeller turbine the overall energy output from a PAT is lower due to the impossibility of keeping a constant head drop under variable flow.

This step in the research followed on from the previous 1-D conversion method analysis, and efficiency predictions, in that it allows the identified BEPs to be converted into full efficiency performance curves. Again this is a key requirement to enable PATs to be used in practice with confidence. As the accuracy of the curve extrapolation achieved here also outperformed previous attempts at this process by other investigators, it also builds confidence in the PAT design process for practitioners.

Prediction of PAT cost

While in the literature there's an ample consensus on the fact that PATs are much cheaper than conventional turbines, different authors have proposed contradicting figures to estimate the cost

of a PAT and a generator. In general these previous cost estimates have been sometimes optimistic, and have often been subsequently extrapolated well beyond their safe interval in follow-on studies. Hence, a very large database of PATs and asynchronous generators with associated price list has been gathered in order to quantify a reliable PAT market price according to the key operating parameters at the BEP of the turbine.

The unit cost of a PAT + generator has been found to vary from as low as 130 €/kW for larger units (≈ 280 kW) up to 5,400 €/kW for small machines (≈ 100 W) according to realistic economies of scale. A set of linear equations was proposed to predict the purchase cost of PAT-based turbogenerators according to the product of nominal flow rate Q and the square root of H .

In order to better evaluate the overall projects costs involved in building a PAT energy recovery installation in a water network information was gathered from 9 designed or installed schemes. An interesting finding from comparing the cost breakdown from those projects is that on average the portion of total cost allocated to the purchase of PAT and generator was equal to 26%, ranging from a minimum of 14 to a maximum value of 48%. Besides, no identifiable trend was found with the percentages of cost items with varying nominal power of the installation.

The robust PAT cost estimation model developed here is an important advance in the field, since it enables realistic cost estimates of PAT installations to be conducted, thus removing a further barrier to the implementation of the technology. In the absence of it, most previous authors who had carried out studies to evaluate the economic feasibility of PAT installations had used cost figures which significantly underestimated what a realistic cost would have been, and/or were only valid in a narrow interval of turbine nominal power.

PAT Decision support and selection software

The selection of the ideal PAT and generator for a given site is a matter of non-trivial trade-offs between technic-economic parameters and operational constraints, for instance maximum achievable energy yield vs. shorter payback time or maximization of CO₂ savings vs. maximization of the project NPV. Ultimately, the final decision on which exact model of PAT should be installed at a specific site largely depends on the arbitrary weight that the project lead is willing to assign to the mentioned parameters.

Hence, in order to provide hydro designers with a robust decision support system for PAT selection amongst contrasting goals a software was created which integrated the main achievements of the

previous sections 5.1 to 5.4 in terms of prediction of PAT performance and cost. The software only requires as input basic hydraulic data of flow and head available at the selected location, as well as any eventual site-specific constraints. As a result it calculates the performances and costs of an array of imaginary PATs if they were to be deployed under those conditions. Eventually, the user is able to navigate various tabs each representing a different design goal (e.g. yearly energy yield or payback time) and visually identify the existing off-the-shelf available pump that best fits his/her requirements (i.e. not imaginary).

The software was tested against two case-studies found in the literature and showed interesting insights with respect to the machinery selection done by the authors of the original papers. It was also utilised successfully in the design of the large scale PAT testing unit and two demonstration sites in Ireland and Wales. All three exercises confirmed the accuracy of its predictions.

Ultimately, the developed PAT decision support software by combining the models developed in Sections 5.1 - 5.4 turned out to be a novel and robust design tool with the potential of unlocking significant portions of the untapped hydropower potential which could be best harnessed by means of PAT technology as discussed in Chapter 4.

CHAPTER 6

Small scale PAT test

In order to gain confidence in the hydraulic and electric aspects of PAT operations, as well as informing the design of the larger test rig and the demo sites, an extensive experimental campaign was performed on a small IWAKI magnetic-sealed radial pump.

At first the mechanical behaviour of the unit in turbine mode was determined, by deriving the head and power curves for varying flow rates across it. The shape of the obtained characteristic curves closely resembled the one expected from full-scale PATs, but given the very small scale of the IWAKI pump the maximum efficiency obtained at 0.71 l/s and 3.3 m was just 32.5% which is lower than what expected by a standard centrifugal PAT. This is however in accordance with the findings of Chapter 5.2 since by applying Equation (27) at Page 108 the expected peak efficiency from the IWAKI PAT would have been 33%.

Secondly, the IWAKI PAT was connected to its original two-pole motor acting as generator and tested again changing its synchronous speed by means of a small single-phase inverter with braking resistor. This experiment successfully demonstrated that for the available head and flow generated

by the feed pump the IWAKI PAT was not capable of generating power through the induction generator at its rated speed (3020 RPM). Instead, by lowering the supply frequency and thus the synchronous speed, the system working point moved between the locked rotor and runaway curves of the PAT as shown in Figure 6-11. After the occurrence of such an event, a positive power output was recorded with a maximum output of 20W as per Figure 6-12 with fully closed bypass valve. This experiment was very insightful as it informed the design of the large laboratory test rig and the Irish pilot plant which both relied on asynchronous induction generators to produce power in parallel to the grid, respectively with and without an inverter to vary the excitation frequency.

Thirdly, the same IWAKI PAT was connected to a synchronous permanent magnet motor whose output was converted to DC through a rectifier and dissipated through a variable set of resistors. The reason for this setup was to reproduce the layout planned for the Welsh demo site where a solar PV inverter was due to absorb the produced DC power. The results were recorded by means of an Arduino-based automatic data logging system which represented a big improvement over the earlier manual reading of the various sensors.

As seen in Figure 6-17 the electric power obtained through the earlier induction generator and the one obtained through the later synchronous permanent magnet generator were closely comparable, and both were lower than the mechanical power that the PAT gave as output under the same flow rate. This indicated that the efficiency of both generators was comparable and in the range 35-70% according to the shaft speed as reviewed in Figure 6-18.

As a final test, the same Arduino shield utilized as data logger was configured to act as a rheostat and vary the resistance of the DC dissipation circuit according to a custom-written Maximum Power Point Tracker (MPPT) algorithm. The code displayed as logic blocks in Figure 6-20 was tested by varying continuously the flow rate supplied by the circulation pump and allowing the control unit to vary the circuit resistance in order to provide the maximum possible power output under each conditions.

Large scale PAT test

In order to evaluate the accuracy of the designed PAT selection software presented in Chapter 5, a large scale PAT test rig was established with the capability of accurately mapping the full characteristic curves of a PAT selected through the software. At first, a fictitious albeit realistic PRV working environment was calculated by scaling down the recorded values of flow and dissipated

head through an existing PRV inserted in a water network. Subsequently, a KSB centrifugal PAT was selected after running the software with the chosen design parameters.

The full characterization of the hydraulic characteristic curves of the PAT was carried out by manually varying the opening degree of the hydraulic bypass and the supply frequency of the regenerative inverter.

The results showed a significant discrepancy between the PAT characteristic curves provided by the manufacturer of the machine and the actual curves. In particular, the measured peak mechanical efficiency of the turbine was found to be 72% which is much higher than the 51.5 % in turbine mode and 55.9 % in pump mode stated by KSB. As a consequence, the chosen PAT would have yielded 33% more energy over a year of operations with respect to the 7,770 kWh estimated through the software whose calculations were based on the reported pump peak efficiency given by the manufacturer.

This experience, although cannot be deemed as conclusive nor representative of the behaviour of all centrifugal PATs, may suggest to exert caution when relying on the PAT datasheets that a very limited number of pump manufacturers may provide. Instead, the PAT selection software developed in Chapter 5 proved to be more accurate than the very same manufacturer datasheet in predicting the operating curve of the PAT as shown in Figure 6-32 (b).

CHAPTER 7

One of the unique features of the Dŵr Uisce Project is the translation of research into practice by building a number of demonstration sites to prove the viability of the technologies developed. However this step poses significant challenges, as we have witness with the two designed hydropower energy recovery installations in Wales and Ireland. Even if the two pilot plants deploy the same PAT technology, their context is significantly different and therefore the inherent challenges to bring them to completion were quite different.

Ty Mawr Wybrnant, North Wales

The culmination of the research project consisted of the design and construction of two demonstration sites located in Ireland and Wales. The location of the Welsh demo site was selected with our project partner National Trust to show the potential of PAT utilization for run-off-river rural electrification projects. The philosophy agreed with National Trust was to build a power plant using as much local materials and expertise as possible, with easy to maintain equipment that

is replicable by Welsh farmers and landowners. The use of a PAT device as turbogenerator was particularly well matched with the requirements, since it could be procured by local resellers and repaired with local knowledge.

The design of the Ty Mawr Wybrnant demo site included a PAT coupled to a synchronous generator feeding its power to a solar type-approved inverter connected in parallel to the grid and offsetting the energy demand of the historical farmhouse.

As the turbine was going to be installed on a newly built pipeline deriving water from a river in the Snowdonia Natural Park, the planning and authorization processes ended up being rather lengthy and expensive. Thankfully our project partner National Trust took care of the process across a whole year. Secondly, an additional difficulty was established by UK legislators who from the 27th of April 2019 changed the legislative framework to connect a renewable energy generator in parallel to the grid to the new G98 standards thus making it difficult to source appropriate electrical equipment for the nearly built site.

The commissioning of the scheme offered several challenges, since the local workers hired for the civil works did not have experience working of hydro schemes. Besides, there were some technical issues with the inverter connection to the grid as the power line reaching the property was subject to voltage spikes above 253 V AC and the first inverter purchased (SMA Sunny Boy) was not working properly and had to be replaced with another unit complying with the latest G98 UK regulations.

Eventually, the scheme was commissioned on 9/9/2019 and it proved capable of generating up to 4,020 W of power out of the designed 3.68 kW which validated the PAT performances predicted by the selection software. Besides, as seen in Figure 7-15, the PAT selection software developed for this PhD dissertation proved slightly more accurate in predicting the characteristic curve of the turbine with respect to the actual datasheet provided by its manufacturer. The reasons for this are unclear and may be that the manufacturer predicts rather than directly measures the device as a PAT. However we can only speculate on this. This occurrence additionally highlights the suggestion not to rely excessively on the PAT performance charts given by manufacturers as already stated in Section 6.2.

When compared to other alternative turbine technologies in Figure 7-16 and Figure 7-17, the use of a PAT stood out in terms of cost reduction, maintenance simplicity, durability and ease of operations. When compared against a set of three pico Turgo turbines and one conventional

Pelton unit the PAT offered most of the benefits and strengths of the Pelton turbine without its price tag, since its acquisition cost was lower by 6.5 times when including the rest of electro-mechanical devices.

The scheme is expected to generate around 19,000 kWh per year thus displacing the emissions of 5.3 tons of CO_{2eq}. The total cost of the installation corresponded to approximately 29,000 € out of which the majority was necessary for the civil works (51%) and planning process (23%). Only the 11% of the total cost figure went into purchasing the PAT and generator, but if a conventional Pelton turbine had been selected in lieu of the PAT the same cost element would have accounted for the 49.5 % of a total figure close to 49,000 €. This highlights the relevance of PAT utilization for micro and pico hydropower in obtaining a significant reduction of the installation cost.

The cost per installed power was equal to 7,830 €/kW with a LCOE of 0.148 €/kWh, and the expected payback time corresponded approximately to eight and half years. However the last two parameters depend on the amount of electricity that the scheme will actually generate over the years to come, which in turn is influenced by several external factors among which the amount of precipitation falling over the Wybrnant watershed is the most relevant. The stream flow rate monitoring during March, April and May 2019 shown in Figure 7-20 suggested that either these three months have been extraordinarily dry for the season, or that the LowFlows software used to predict the hydrological regime of the river lead to an overestimation of the flow rate. However, a definitive answer will only be possible after a constant monitoring of the river flow along a sufficient period of time.

Blackstairs Water Treatment Works, Ireland

The Irish demo site was meant as a prototype demonstrating the concept of energy recovery from water networks relying on elevated water sources.

A first challenge consisted of the identification of the ideal location for it. As a first step, a preliminary screening was performed assessing multiple rural water schemes across the south-east of the country. Hence, once the water scheme with the highest potential was identified (Blackstairs, Co. Wexford) a more detailed analysis and measuring campaign had to be carried out to come up with the final plant design. After this, there was the need to have a discussion between the different institutions and companies involved (Trinity College Dublin, EPS Ireland, Blackstairs Group Water Scheme) on how to draft a satisfactory agreement before proceeding with the construction phase.

A flow meter and a pressure gauge were installed at the inlet of the raw water tank within the premises of the treatment works, and the water inflow was recorded for months during spring 2018. Such an experimental campaign greatly helped to inform the design of the power plant, which required a minimum level of modifications of the existing infrastructure.

The initial turbine startup took place on 18/4/2019 and the recorded data showed that the working curve of the PAT closely followed the one predicted by the software. However, the maximum achieved power output was equal to 2.93 kW instead of the expected 4 kW. The cause for this was mainly ascribed to the presence of two air pockets along the 2 km of pipeline which may be eliminated in the future by installing an air valve.

The experimental characteristic curve of the PAT installed at Blackstairs was remarkably close to the one predicted by the selection software as seen in Figure 7-39.

The final plant cost was 29,360 € including VAT, and the works were carried out by the same company operating the treatment works (EPS). However, this figure includes the cost of the remote monitoring system and its connected flow and pressure sensors which were non-essential to PAT operations. Besides, the use of two subcontractors has likely contributed to raising the final cost above the strictly necessary. The cost per unit of installed power ended up being very similar to the one of the Ty Mawr Wybrnant demo site (7,340 €/kW), but since the design flow is guaranteed 24/7 except than in the event of exceptional droughts the expected yearly energy yield is nearly double (37,000 kWh/year). Accordingly, the projected LCOE is much lower than the previous demo site and corresponds to 0.079 €/kWh while the CO_{2eq} savings are equal to 16.2 tons per year.

9. CONCLUSIONS

9.1. ADDRESSING THE RESEARCH QUESTIONS

The research questions posed in Chapter 1 were as follows:

- a) **Which are the major bottlenecks to hydraulic energy recovery schemes from water networks?** Strategies for energy recovery from the water industry must be formulated in order to make hydropower more attractive even after the general ceasing or gradual decrease of Feed-in-Tariffs (FIT) incentives for renewable generation in Europe.
- b) **Which are the main barriers to the adoption of Pump as Turbines (PAT) as energy recovery devices?** Why aren't Pumps as Turbines more used on the market, either as energy recovery devices or as micro hydro generators? It's been almost a whole century since the first team of technicians determined that pumps can operate fairly efficiently as turbines when the flow direction through them is reversed. Still, their market presence is risible around the world.

Right from the onset of the research work a comprehensive literature review was carried out to better identify the current status of the sector by consulting both research and industry documents. The resulting picture showed that in the context of a global effort towards the decarbonisation of human activities the water sector has a significant role to play being among the highest energy users in industrialized countries.

At the same time, hydropower technology is a mature one which has been widely implemented around the world especially on the Small, Medium and Large scale ($P > 100$ kW). However, in the last decades the interest towards Micro and even Pico hydro installations ($P < 100$ kW) has been steadily growing due to the minor impact of the required civil works and the large existing untapped potential for schemes of this size.

Amongst the possible strategies to reduce the energy requirements and associated emissions from the water sector is the implementation of Micro and Pico hydropower within water networks, which has the potential to offset part of the energy needs by converting excess pressure into electricity. While many types of hydropower converters exist, a particularly suitable one for the

task is PAT technology whose main strengths are the low cost, easy maintenance and possibility to accommodate a residual pressure in the pipeline. Even if a small number of PAT installations in water networks have been built in the past as reviewed in Section 3.6.3, the penetration of the technology in the sector is minimal and most of those installations have been built as pilot plants and have since not been replicated on a large scale.

In order to get a better understanding of the bottlenecks to hydraulic energy recovery from water networks and to PAT adoption on a wide scale a survey among workers in the water sector, hydropower designers, pump manufacturers and academic researchers was conducted as described in Chapter 4. The main findings from the survey can be summed up in three points:

- I. Pump manufacturers: not interested in investigating PAT use or promoting sales due to the marginality of the sector respect to their core market.
- II. Hydro design firms and consultants: unaware of the PAT potential, inexperienced, not willing to take on project liability, concerned by the lack of design guidelines and existing field applications.
- III. Potential PAT users (water utilities): unaware of the extent of energy recovery potential, not informed about PAT, adverse to the risk of service disruptions in water mains.

As a consequence, the research work hereby presented was oriented towards two main goals:

- a) **Development of a robust and comprehensive PAT decision support and selection software, enabling hydro designers to select the most appropriate set of PAT and generator for any given site (either within a water networks or as a conventional hydropower scheme).** The software development described in Chapter 5 incorporated several statistical methods to predict the PAT performances and cost which was developed upon large datasets of machines whose information has been gathered from the available literature. The resulting software has been applied and tested with favourable outcomes in the following experimental PAT testing campaign and in the construction of the two demo sites.
- b) **Construction of two pilot plants and dissemination of the results in order to promote further adoptions of technologies for the recovery of hydraulic energy.** The two sites selected to host the pilot PAT installations were a water treatment works in southern Ireland and a run-of-river scheme in northern Wales. Both schemes had a nominal power

output of around 4 kW and were successfully designed, built and commissioned between autumn 2018 and summer 2019. The final cost was lower than 30,000 € in both installations, with an envisaged return of investment lower than 9 years and a combined projected yearly CO₂ savings of 21.5 tons of CO_{2eq}.

The present body of work has demonstrated that PAT technology if designed appropriately can be a valid alternative to conventional hydropower turbines, especially in the Micro-Hydropower scale. The insights gained through the development of the selection software and the construction of the demo sites have been widely disseminated by attending scientific conferences and through the Dŵr Uisce cluster.

9.2. SCOPE FOR FUTURE WORK

The scope for future work includes:

- improvements of the PAT performance and cost prediction methods described in Sections 5.1 to 5.4 by increasing the size of the data sets upon which each analysis was carried out, ideally including different types of PATs which were under-represented in the present collection (e.g. double-entry split-flow and multistage PATs);
- improvement of the Graphic User Interface of the PAT selection software, in order to further help hydro designers to use it as a tool to plan future installations;
- in-depth analysis of the adoption of more than one PAT designed to operate in parallel, developing guidelines on how to select the appropriate machinery for a modular PAT installation with multiple units;
- long-term monitoring of the performances of the two built demo sites, which eventually could suggest improvements to the design strategies and PAT selection software;
- trialling the software on the design of a PAT with pressure control;
- trialling the application of the software on larger scale PAT applications ($P > 4$ kW).

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ANNEX 1: CASE STUDIES

1. FIVE FORDS WWTP (WELSH WATER)

Site

The Five Fords Waste Water Energy Park is located in the municipality of Wrexham, North Wales and serves a population of 96,000 households.

Background

Treated waste water is discharged to the nearby river Dee through a 5.6 km long outfall pipe. Since a significant potential head difference is present between the works and discharge point (equal to 28.6 m), the construction of a micro-hydropower station has been investigated in the past. The pipe system and the power house have already been built, albeit the turbine itself has not being installed due to economic barriers. The aim of the present document is to assess the viability of using the existing infrastructure, installing a micro-hydropower station adopting Pumps as Turbines (PAT) as the generation technology.

Existing infrastructure

The outfall pipeline is made up of four distinct sections of concrete and GRP pipe. The elevation profile of the existing pipe system is shown in Figure 0-1.

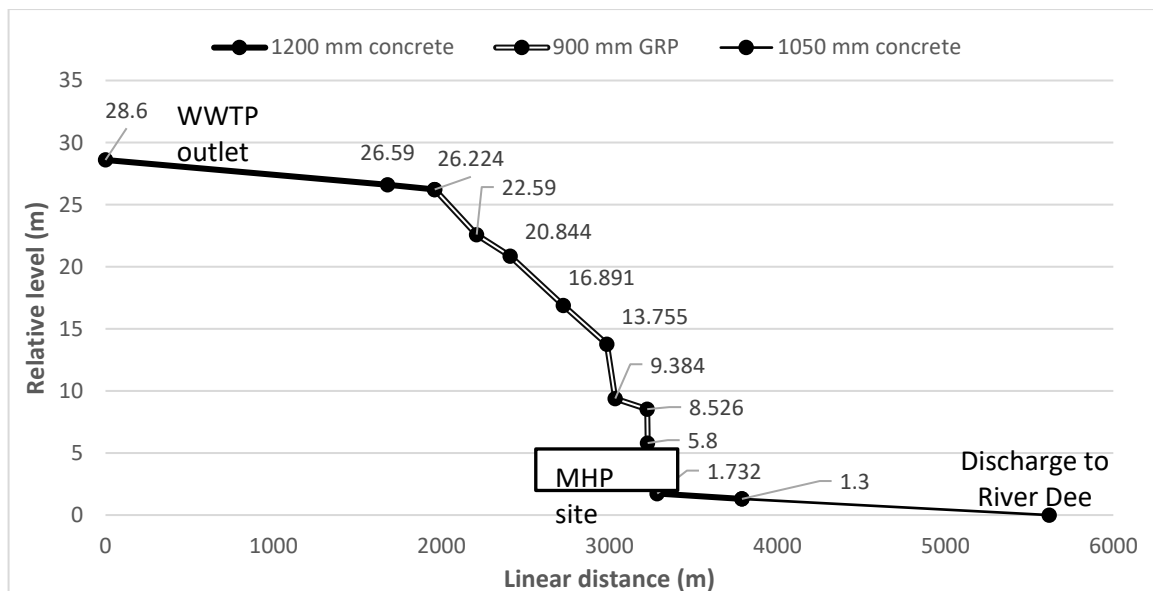


FIGURE 0-1. ELEVATION PROFILE OF EXISTING PIPE SYSTEM FROM FIVE FORDS PARK OUTLET TO RIVER DEE

All discharged water currently flows through the existing turbine bypass pipe until it reaches the river. The existing turbine housing structure can be seen in Figure 0-2.



FIGURE 0-2. TURBINE HOUSE BASE

Hydropower potential assessment

Available data on water flow rate from Five Fords site include two instantaneous readings recorded every day over a timeframe of one year (from 7/6/2011 to 5/6/2012). The flow duration cumulative curve and daily flow variability are shown in Figure 0-3 and Figure 0-4 respectively.

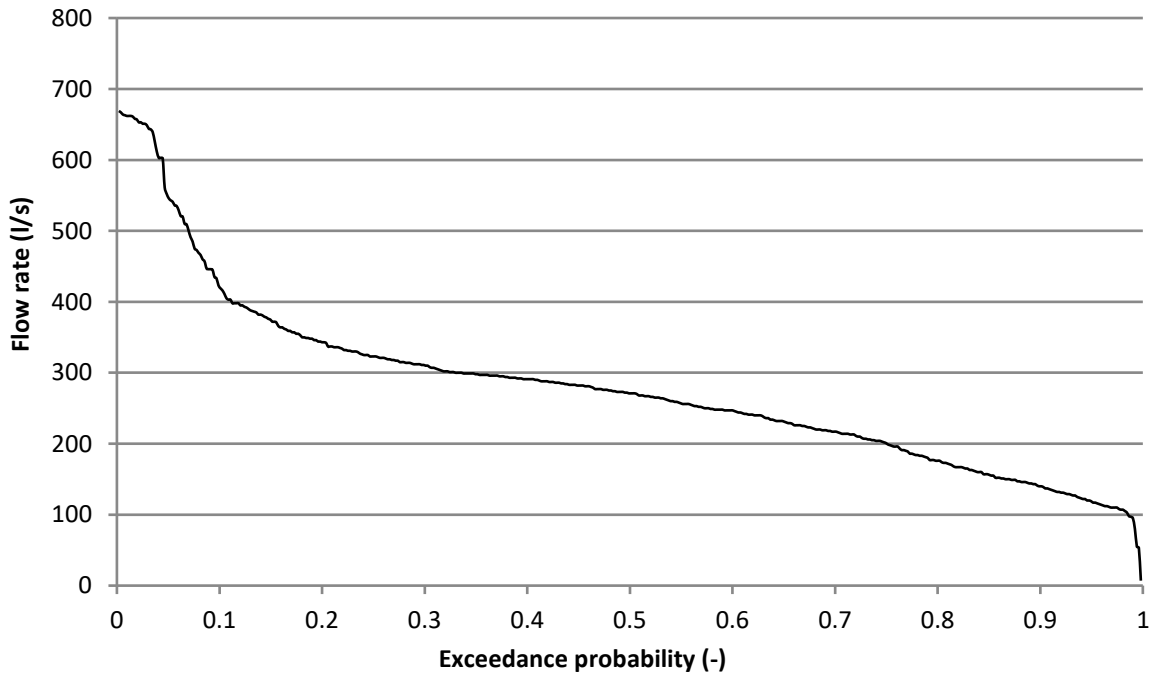


FIGURE 0-3. FLOW DURATION CUMULATIVE CHART

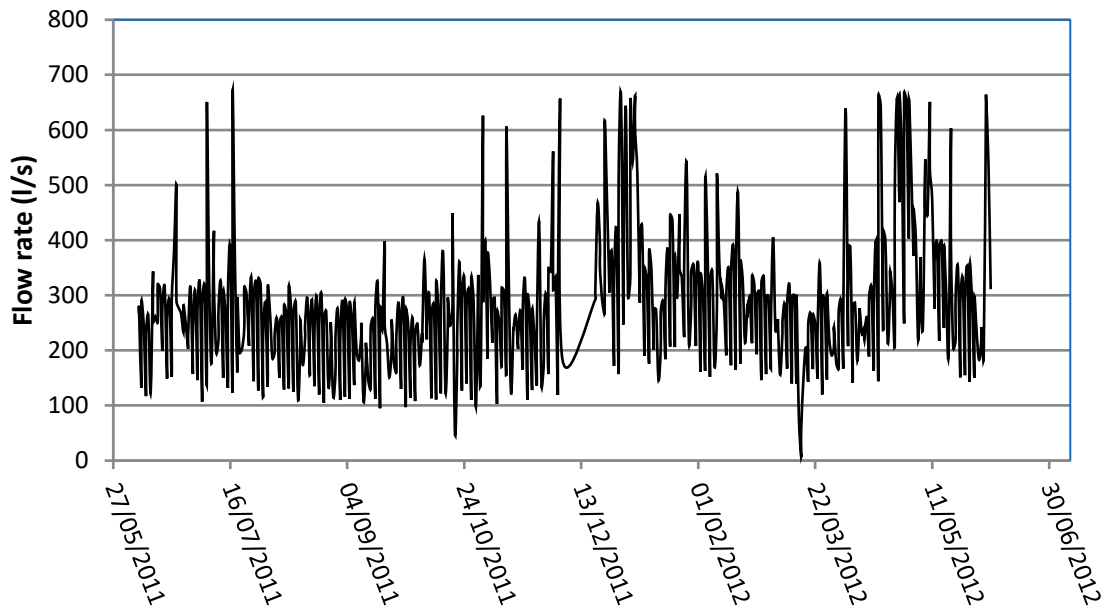


FIGURE 0-4. FLOW RATE VARIABILITY OVER THE MEASURED TIME

Considering the opportunity of flooding the 900 mm GRP pipe along its entire length (excluding the concrete section), the gross available hydraulic head would correspond to 24.49 m equal to the height difference between pipe intake and the outlet of existing turbine discharge chamber. Additionally, friction head losses and local head losses along the pipe system have been carried out for different flow rates, resulting in a net head profile as shown in Figure 0-5.

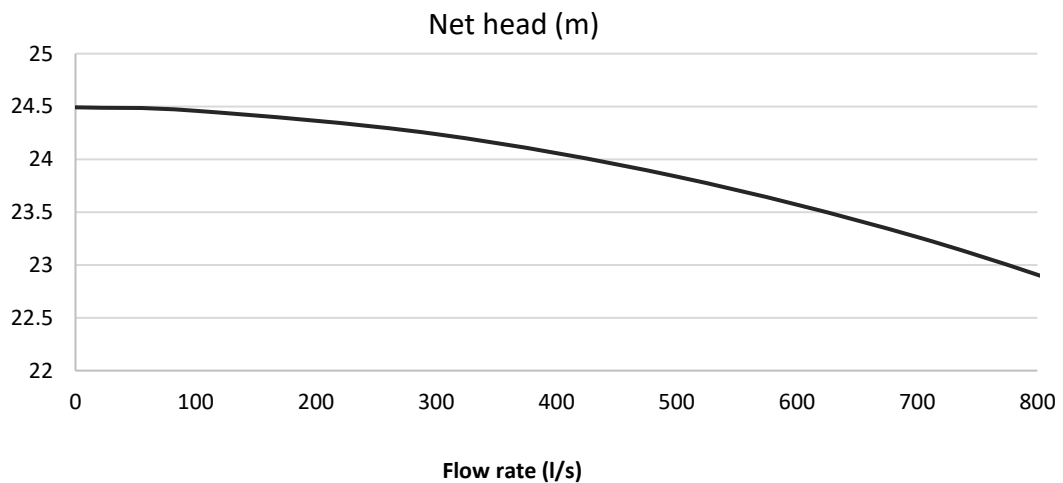


FIGURE 0-5. PROFILE OF NET AVAILABLE HEAD AS FUNCTION OF FLOW RATE

Such values are based on the assumption that the 900 mm GRP pipe is robust enough to be pressurized on a long-term basis.

Proposed technology

Given the location of the ideal design point (net head equal to 23.3 m and flow between 100 and 700 l/s), from the selection chart presented in Figure 0-6 it can be concluded that a radial, double flow or mixed flow PAT would best fit in this scenario.

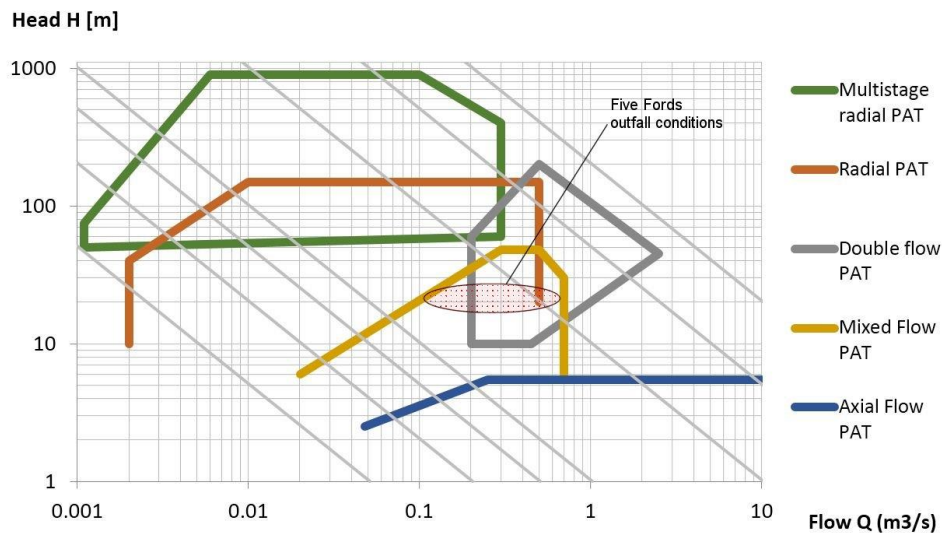


FIGURE 0-6. PUMPS AS TURBINES SELECTION CHART (ADAPTED FROM (CHAPALLAZ, ET AL., 1992))

Option 1

As first option, the installation of a single PAT-generator unit (PAT A) has been investigated. The optimal machine would operate 80% of the time and process a maximum flow rate of 270 l/s as shown in Figure 0-7. The remainder of the flow would be by-passed where this is too high or too low for the turbines operating range.

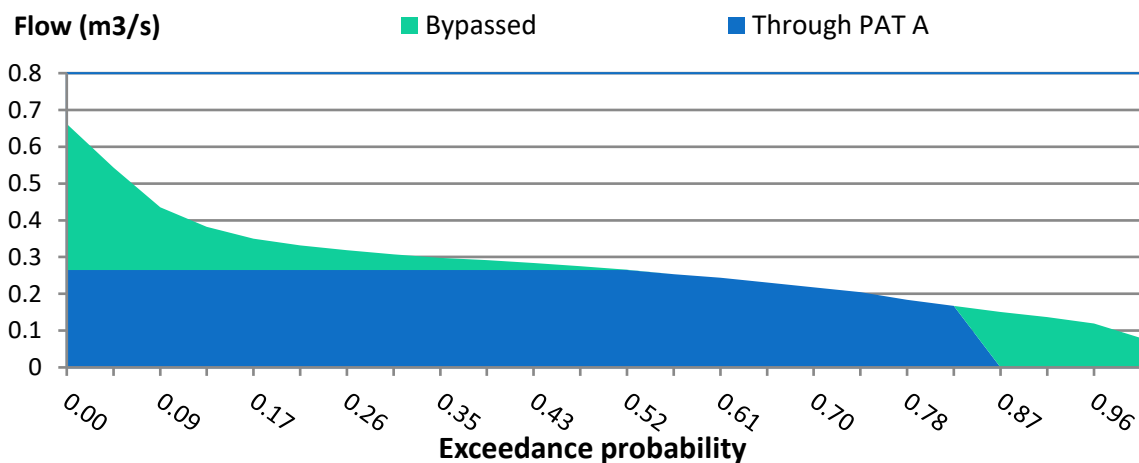


FIGURE 0-7. SCENARIO WITH ONE INSTALLED PAT UNIT

Option 2

The installation of a second smaller PAT unit (PAT B) in parallel to PAT A, would allow for processing water flow under low and peak conditions as seen in Figure 0-8. This second PAT would enable the capture of the energy potential of a wider range of the low and high flow conditions at the site

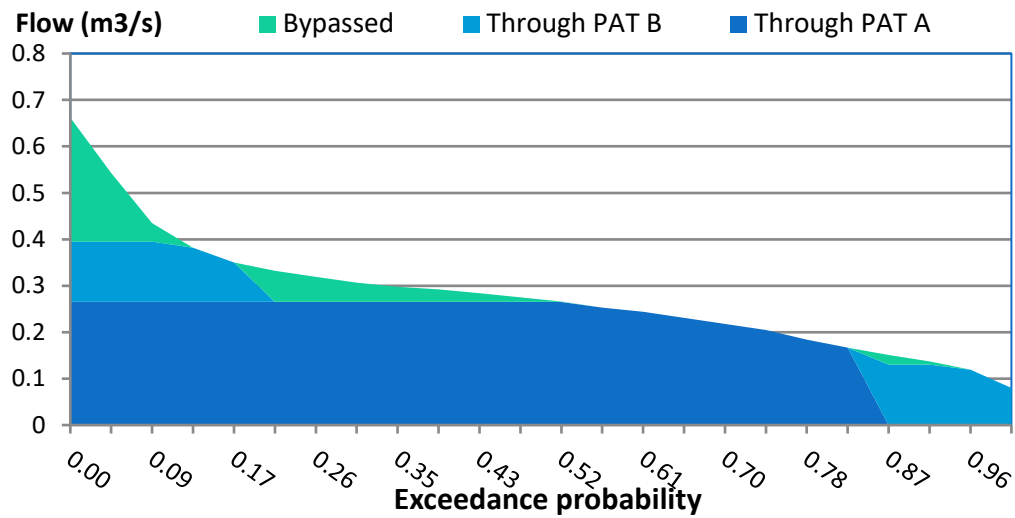


FIGURE 0-8. SCENARIO WITH TWO INSTALLED PAT UNITS

Option 3

The additional installation of a third smaller PAT (PAT C) in parallel to the first two was also considered as way to further minimize the amount of bypassed flow during peak events as seen in Figure 0-9.

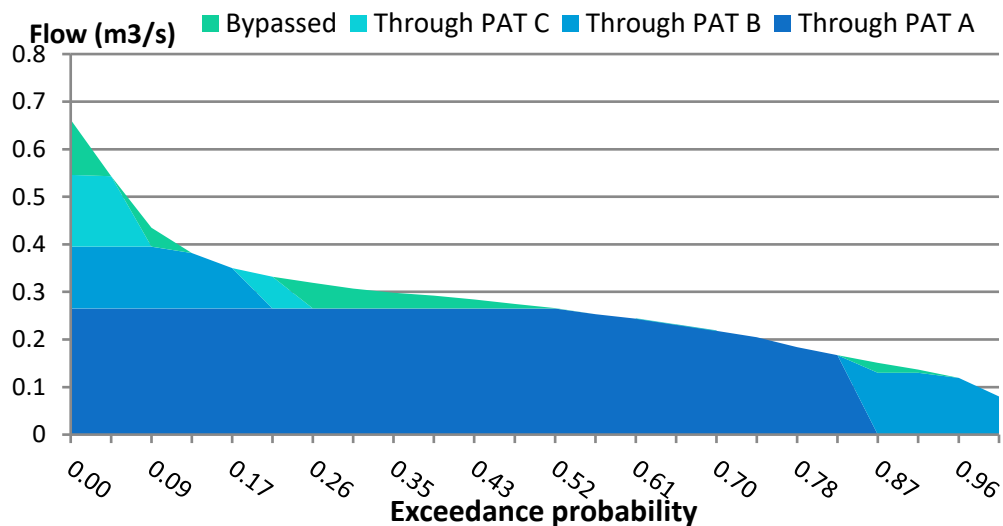


FIGURE 0-9. SCENARIO WITH THREE INSTALLED PAT UNITS

Summary

The characteristics of PATs A, B and C are shown in Table 0-1. Their respective head and efficiency curves are presented in Figure 0-10.

TABLE 0-1. CHARACTERISTICS OF SELECTED PAT A, B AND C

PAT A		PAT B		PAT C	
H_{BEP} (m)	26.6	H_{BEP} (m)	24.66	H_{BEP} (m)	13.95
Q_{BEP} (m ³ /s)	0.2946	Q_{BEP} (m ³ /s)	0.1364	Q_{BEP} (m ³ /s)	0.1119
P_{BEP} (kW)	66.59	P_{BEP} (kW)	27.24	P_{BEP} (kW)	12.98
Q_{max} (m ³ /s)	0.4481	Q_{max} (m ³ /s)	0.1829	Q_{max} (m ³ /s)	0.17014
Q_{min} (m ³ /s)	0.1534	Q_{min} (m ³ /s)	0.0788	Q_{min} (m ³ /s)	0.0583
N (RPM)	1520	N (RPM)	1520	N (RPM)	1520
Ns,t (kW, m)	205.3	Ns,t (kW, m)	144.4	Ns,t (kW, m)	203.1
Ns,t (m ³ /s, m)	70.4	Ns,t (m ³ /s, m)	50.7	Ns,t (m ³ /s, m)	70.4
$P_{max\ gen}$ (kW)	51.9	$P_{max\ gen}$ (kW)	24.1	$P_{max\ gen}$ (kW)	25.5
D (m)	0.29	D (m)	0.27	D (m)	0.21
Cost (€)	19,207	Cost (€)	9,127	Cost (€)	6,015
Energy (kWh/y)	312,664	Energy (kWh/y)	61,309	Energy (kWh/y)	19,354

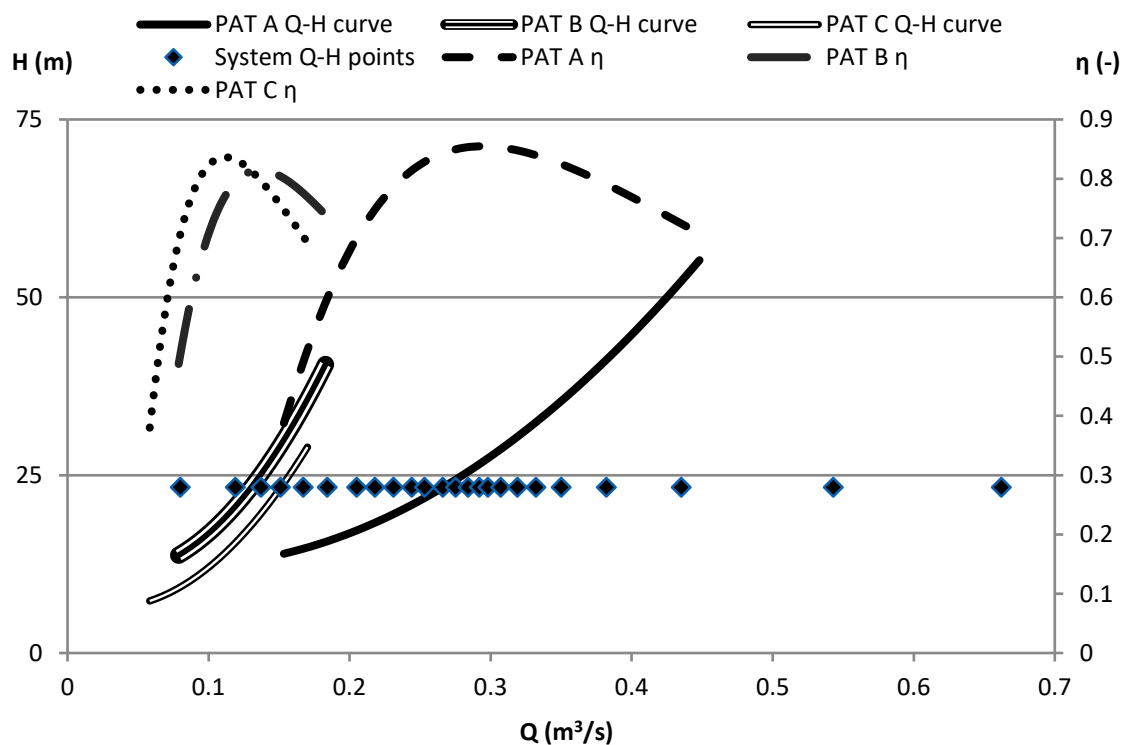


FIGURE 0-10. CHARACTERISTIC CURVES OF PATS A, B AND C

Financial assessment

The 2014 Technical Feasibility Study by Allied Hydropower (Allied Hydropower, 2014) contained a cost estimation of a micro-hydropower scheme to be constructed in Five Fords. The cost breakdown excluding the generating set is shown in Table 0-2.

TABLE 0-2. ESTIMATED BUDGET FOR FIVE FORDS MICRO-HYDROPOWER PLANT (GENERATING SET EXCLUDED)

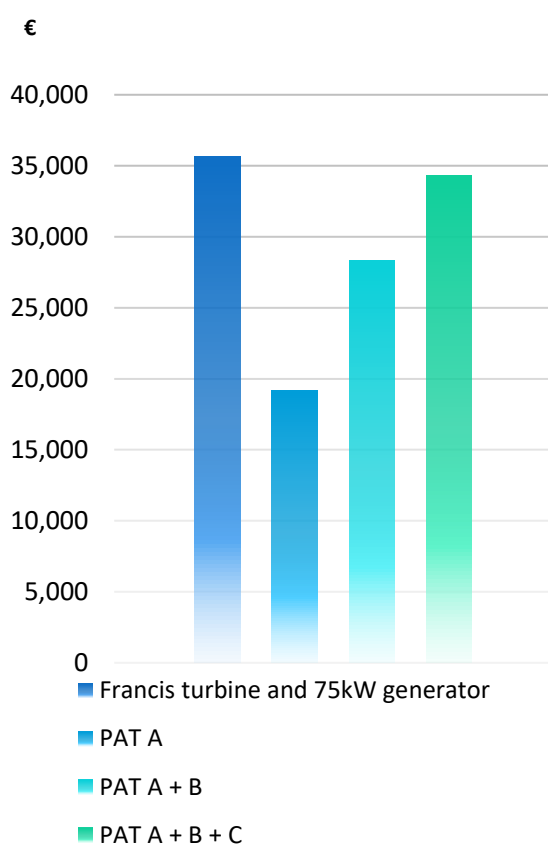
Item	£	€
Feasibility and Design	10000	12729
Planning and Authorisation	10000	12729
Control System	10000	12729
Automated bypass valve	5000	6365
Lifting Beam Structure	2000	2546
Metering and Control cubicle	8000	10183
Grid connection	20000	25458
Installation	10000	12729
Commissioning	3000	3818.7
Total	78000	99286

The quoted Feasibility Study proposed the installation of a second-hand overhauled Francis turbine and 75 kW asynchronous generator whose cost is compared with the proposed PATs in Table 0-3 and Figure 0-11.

TABLE 0-3. COMPARISON OF GENERATING SET COST AND ENERGY YIELD

	Francis turbine and 75kW generator	PAT A	PAT A + B	PAT A + B + C
Cost (£)	28,000	15,089	22,259	26,985
Cost (€)	35,641	19,207	28,334	34,349
Energy (kWh/year)	309,432	312,664	373,973	393,327

a) Cost comparison for different turbine
+ generator units



b) Energy yield comparison for different
hydro-mechanical equipment

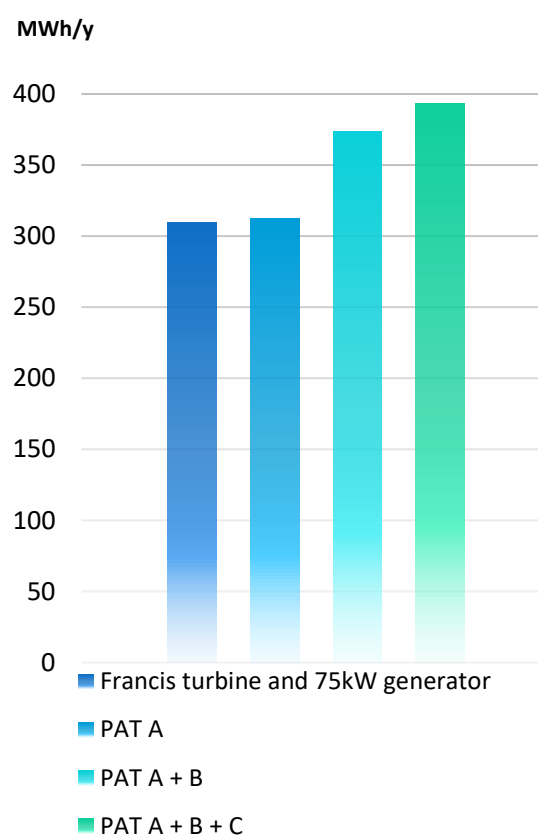


FIGURE 0-11. COMPARISON OF GENERATING SET COST (A) AND ENERGY YIELD (B)

The Simple Payback time of the whole project budget including different proposed hydro-mechanical equipment, excluding capital remuneration, is evaluated in Table 0-4.

TABLE 0-4. COMPARISON OF TOTAL SIMPLE PAYBACK TIME FOR THE DIFFERENT GENERATING SET CONFIGURATION

	Francis turbine and 75kW generator	PAT A	PAT A + B	PAT A + B + C
Simple Payback time (years)	8	6	6	6

2. DROGHEDA WWTP (IRISH WATER/EPS)

Overview

The basic known characteristic of the potential hydro sites at Drogheda Wastewater Treatment Plant (WWTP) are:

- Gross hydraulic head:
 - 4 m after the preliminary treatment building
 - 8 m in correspondence of the final clarifiers
- Average daily flow rate of 23,661 m³
- Water flowing through a DN 900 pipeline

Based on data from a similar sized WWTP in Italy (Asti), the daily profile of flow rate has been assumed as shown in Figure 0-12.

Choice of technology

Both sites at Drogheda WWTP would be ideally suitable for a Kaplan or propeller turbine. As an alternative an axial flow PAT could be utilised, but little information is known on these type of PATs concerning their performances and costs.

Another option could be an Archimedean screw converter, but this choice would require an open air channel instead of a pressurized system.

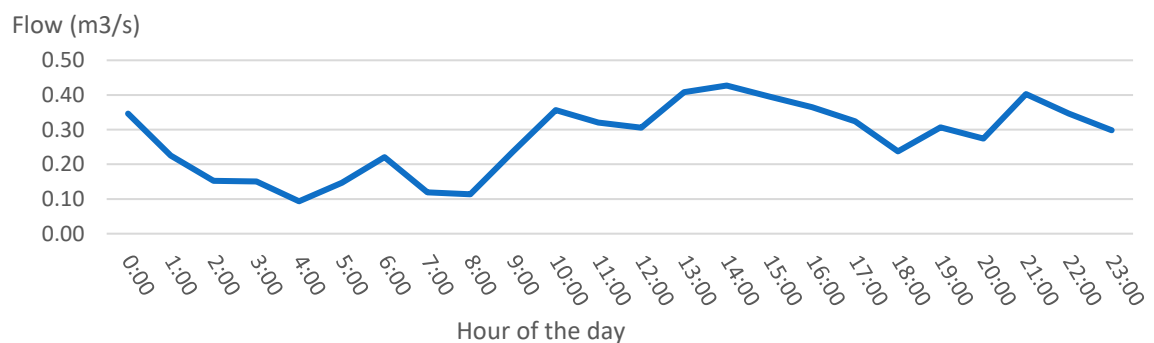


FIGURE 0-12. DAILY PROFILE OF FLOW RATE THROUGH THE TREATMENT PLANT

Turbine and generator sizing

Assuming a flow-to-wire efficiency of 70%, the power output from the considered sites is shown in Figure 0-13. Mean values across a typical day correspond to 7 kW for a 4 m head and 14 kW for 8.

According to (Ogayar & Vidal, 2009), the indicative cost per kW installed of a Kaplan turbine and connected generator may be calculated through the equation (1):

$$19,498 \cdot P^{-0.583380} \cdot H^{-0.113901} \quad (1)$$

Where P refers to the nominal power in kW and H to the available head in m.

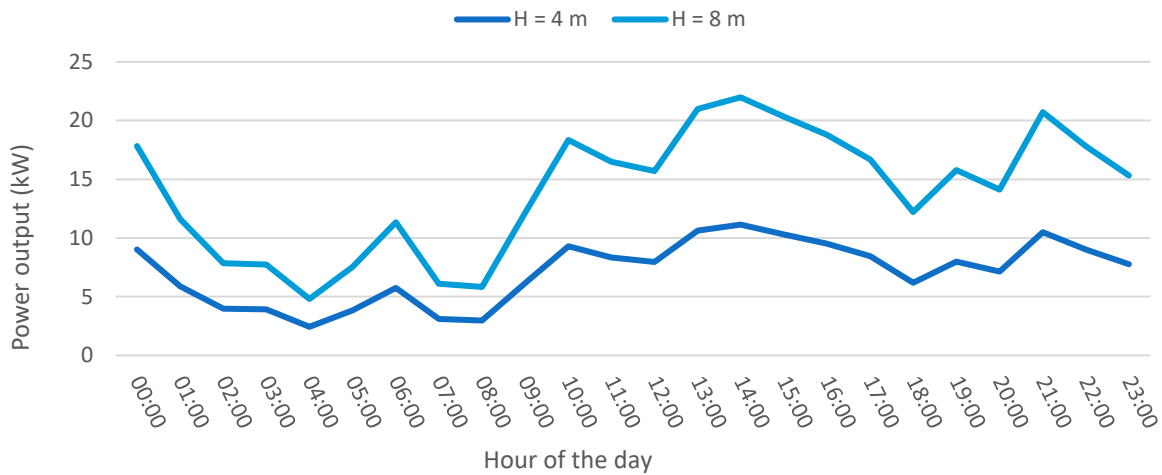


FIGURE 0-13. DAILY PROFILES OF POWER OUTPUT ACCORDING TO AVAILABLE HYDRAULIC HEAD

The indicative cost for the purchase of turbines and generators was calculated through Equation 1, while as first approximation the budget for civil and electrical works has been estimated as equal to the cost of turbine and generator. The generated electricity would be preferably used in the adjacent WWTP, thus saving electricity worth 0.164€ per kWh (Costs of Doing Business in Ireland 2016, National Competitiveness Council, April 2016). A summary of estimated project costs, benefits and turbine specifications is shown in Table 0-5.

TABLE 0-5. DESIGN PARAMETERS FOR DROGHEDA SITES

Type	Kaplan turbine	
Nominal turbine head (m)	4	8
Nominal turbine flow rate (l/s)	420	420
Indicative flow-to-wire efficiency (-)	70%	70%
Peak output power (kW)	11	22
Yearly energy yield (kWh/year)	62,500	123,400
Indicative price of turbine + generator excl. VAT (€)	45,500 (± 20%)	55,800 (± 20%)
Total estimated project costs (€)	91.000	112.000

Economic analysis

Neglecting the finance costs and capital remuneration (inflation, cost of credit, etc.) the investment payback time was evaluated to occur within **6 years** for the site with $H = 8$ m and within **10 years** for the one with $H = 4$ m. The Net Present Value according to the turbine choice is shown in Figure 0-14.

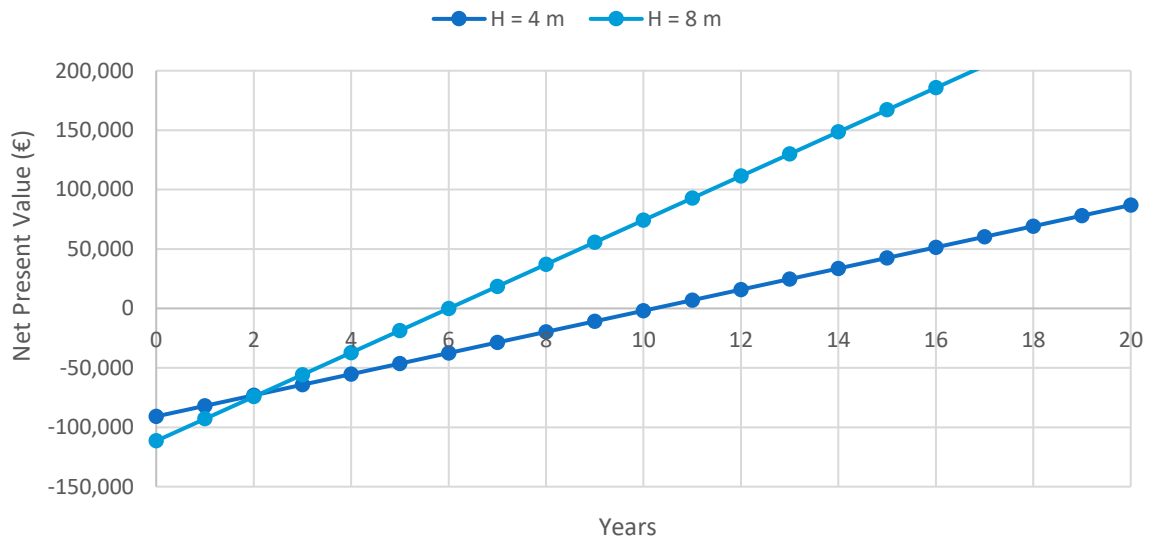


FIGURE 0-14. NET PRESENT VALUE OF THE HYDRO SCHEME ACCORDING TO AVAILABLE HEAD (VAT INCLUDED)

Conclusions

The evaluated micro hydro schemes to be placed at different sites of the Drogheda WWTP proved to be technically feasible and showed an interesting economic potential, especially the one with a gross head of 8 m as could be expected. For a more comprehensive evaluation of the site a number of additional information are required, including most importantly high resolution records of flow rate (these we assumed here and may greatly influence feasibility). Also, a meticulous discussion with the management of the WWTP would be necessary in order to validate the assumptions upon which the above calculations were based.

3. CORK LOWER HARBOUR (IRISH WATER/EPS)

Overview

The newly built Cork Lower Harbour Waste Water Treatment Plant (WWTP) discharges treated water via a 580 meter long DN900 pipeline into a pre-existing foul outfall. The elevation difference between the WWTP outlet chamber and the pipe outlet is approximately 10 m, and at regular intervals are placed several inspection manholes. The profile of discharge pipeline and the position of the manholes are shown in Figure 0-15.

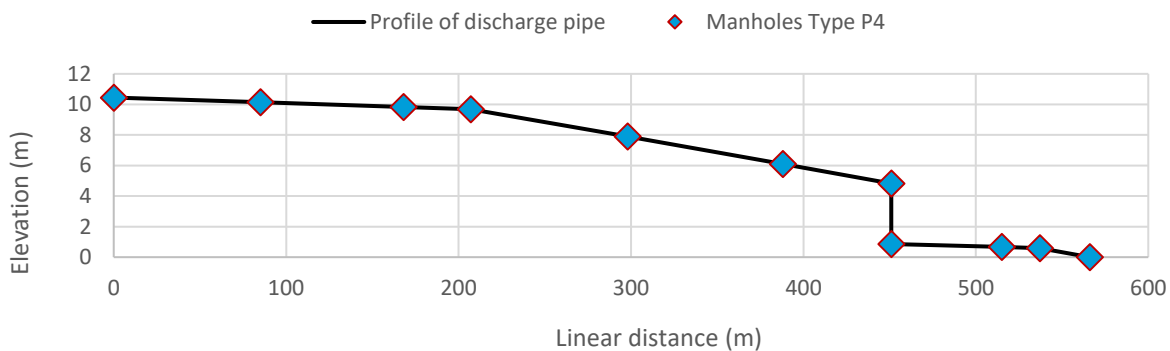


FIGURE 0-15. ELEVATION PROFILE OF EXISTING DISCHARGE PIPE AND LOCATION OF MANHOLES

Values of daily water volumes at WWTP outfall between June 2nd and July 1st 2017 show an average flow rate of 6,050 m³/day, with a minimum of 3,400 and a maximum of 12,450 m³/day. However, within a few years the WWTP is expected to reach its design load and discharge an average of 11,700 m³/day. Similarly to what has been done for Drogheda site, the daily profiles of flow rate at WWTP outlet have been assumed both for the present and the future situation as shown in Figure 0-16.

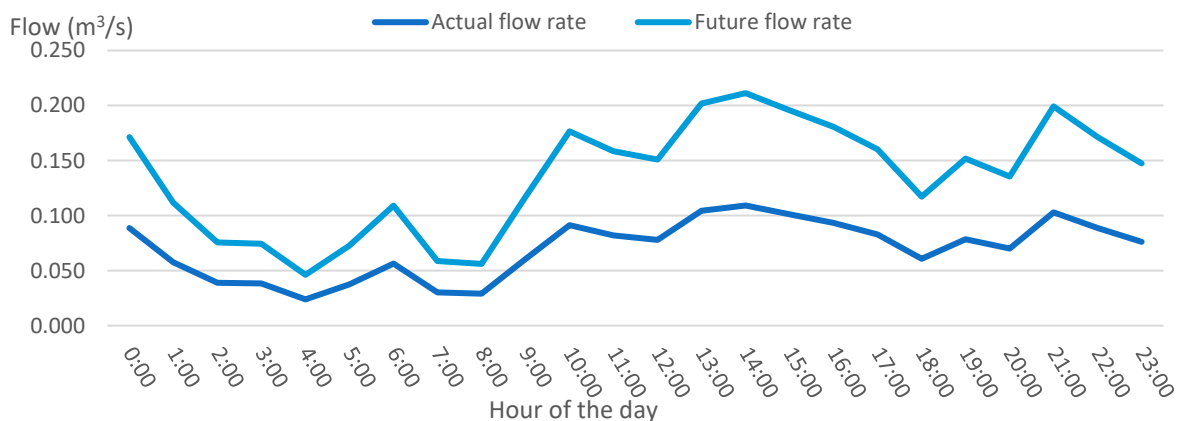


FIGURE 0-16. ESTIMATED HOURLY PROFILES OF FLOW RATE AT ACTUAL AND PREDICTED REGIMES

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Design data

The gross hydraulic head difference corresponds to 10.5 m, and the net head has been calculated subtracting the losses occurring in a 580 m long DN500 GRP pipeline. The resulting net head is shown in Figure 0-18. The design flow rate should correspond to the scenario at which the WWTP works at full load as described in the “Future flow rate” series of Figure 0-16.

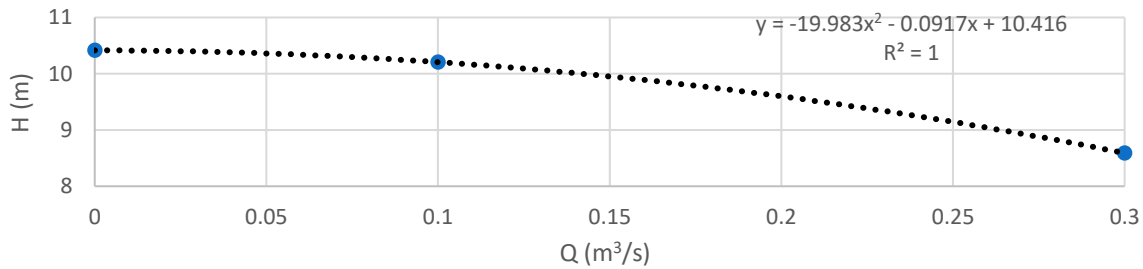


FIGURE 0-18. NET HYDRAULIC HEAD AS FUNCTION OF FLOW RATE

Choice of technology

The optimal turbine for the Lower Cork Harbour site would be either:

- I. a radial or mixed flow PAT
- II. a Banki/Crossflow turbine

The efficiency of such units on varying flow conditions is compared in Figure 0-19, from which it is evident that a Crossflow turbine would maintain a higher efficiency over a wide set of flow rates while instead the efficiency of a PAT lowers significantly away from its design point.

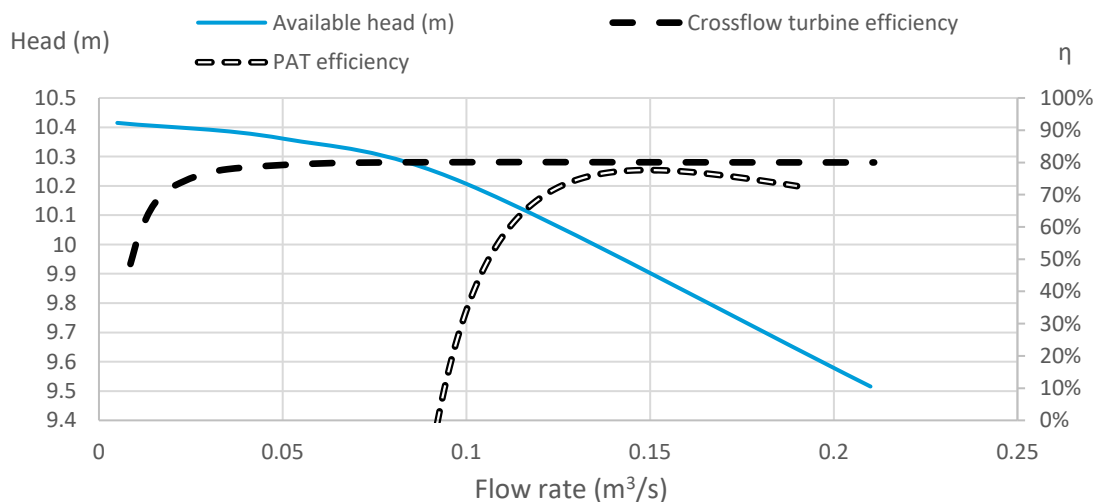


FIGURE 0-19. TURBINE EFFICIENCY VERSUS THE AVAILABLE HYDRAULIC HEAD

The hourly profile of output power during a typical day is shown in Figure 0-20 according to the choice of turbine technology.

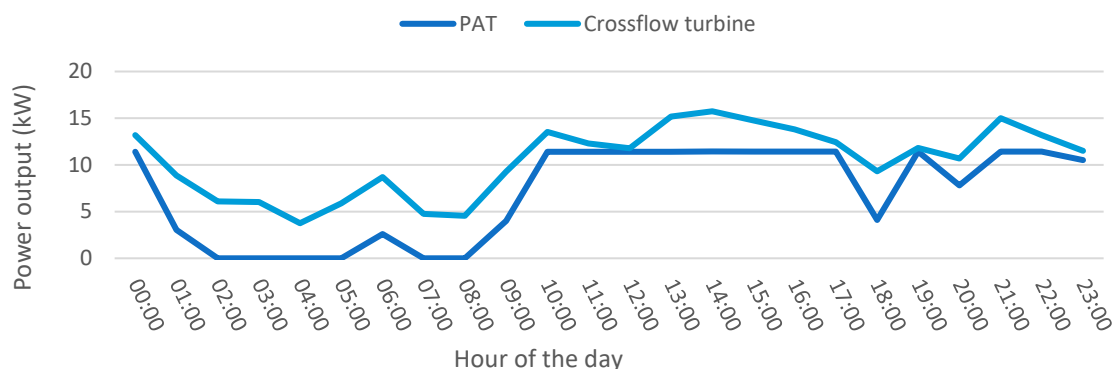


FIGURE 0-20. HOURLY PROFILES OF POWER OUTPUT FROM THE PROPOSED INSTALLATION

Economic analysis

The key parameters of the possible turbines suitable for the considered site are listed in Table 0-6, together with a few significant cost figures. The adoption of a Crossflow turbine would lead to a +50% of generated energy with respect to the use of a PAT, but the cost of equipment would be multiplied by a factor of 10. In both cases, the cost of accessory works (e.g. grid connection, civil works, pipeline laying and manpower necessary to it) has been estimated at 65,000 €.

TABLE 0-6. CHOICE OF TURBINE FOR CORK LOWER HARBOUR SITE

Type	PAT	Crossflow turbine
Nominal turbine head (m)	10.1	10.4
Nominal turbine flow rate (l/s)	152	211
Indicative flow-to-wire efficiency (-)	77.6%	80%
Peak output power (kW)	11.4	15.7
Yearly energy yield (kWh/year)	60,000	92,000
Indicative price of turbine + generator excl. VAT (€)	6,400 (± 20%)	65,000 (± 20%)
Other estimated project costs excl. VAT (€)	65,000	

The generated electricity would be preferably used in the adjacent WWTP, thus saving electricity worth 0.164€ per kWh (Costs of Doing Business in Ireland 2016, National Competitiveness Council,

April 2016). In Figure 0-21 are shown the total cost of the hydro scheme as well as the yearly revenues due to the electricity savings according to turbine choice.

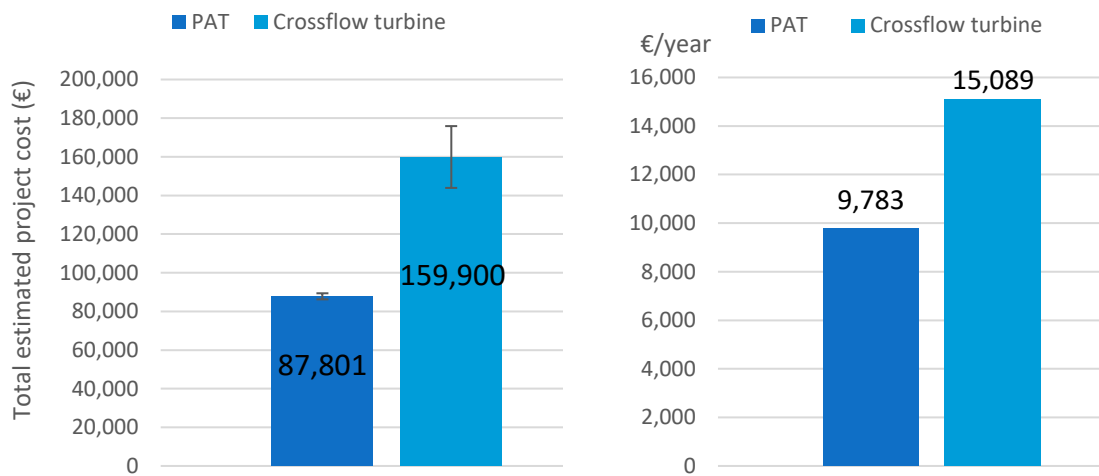


FIGURE 0-21. PROJECT COST AND YEARLY REVENUES FROM ENERGY SALES ACCORDING TO TURBINE CHOICE

Neglecting the finance costs and capital remuneration, the investment payback time was evaluated to occur within **9 and 12 years** since the hydro plant starts operations with the WWTP working at full regime. The Net Present Value according to the turbine choice is shown in Figure 0-22.

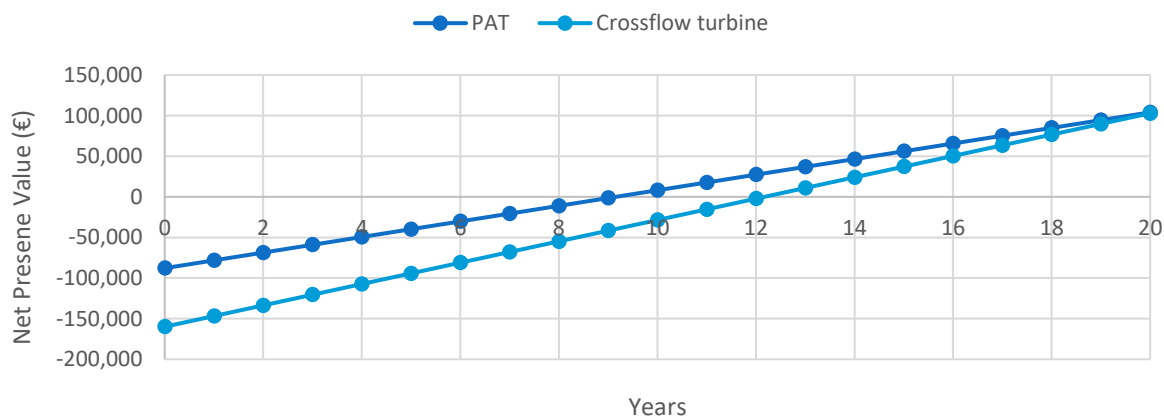


FIGURE 0-22. NET PRESENT VALUE OF THE HYDRO SCHEME ACCORDING TO TURBINE CHOICE (VAT INCLUDED)

Conclusions

The evaluated micro hydro scheme for the outfall of Cork Lower Harbour WWTP proved to be technically feasible and showed an interesting economic potential. However, significant civil works should be carried out for its realization including the laying of a new pipeline running parallel to the existing one.

4. RUN-OF-RIVER SITES IN WALES (BLAENAU GWENT CITY COUNCIL)

Overview

The Dŵr Uisce team has been contacted by representatives of the Blaenau Gwent City Council in Wales to evaluate the feasibility of a series of potential hydro sites located on communal land.

Power output estimation

The power output of a hydropower scheme can be computed applying the following equation:

$$P = Q\rho gH\eta$$

Where

P = power output (kW);

Q = flow rate through the turbine (m^3/s);

ρ = fluid density (kg/m^3);

g = gravity acceleration (m/s^2);

H = hydraulic head available at the turbine (m);

η = efficiency of the power plant.

The power output from the selected sites was assessed for two scenarios, optimistic and pessimistic, relying respectively on the highest and lowest flow rate estimates suggested by the study of Dr. Thorsten Stoesser & Stefan Runge from Cardiff University. The overall efficiency of the schemes η was considered equal to 0.65 in order to account for mechanical, hydraulic and transmission losses.

Project cost estimation

Turbine and generator cost

The turbine and generator are typically the most expensive components of a Micro-Hydropower installation. Their ultimate cost for a determined site may vary due to site-specific constraints and market availability, but as a first approximation some general equations can be used to predict it.

As for PATs, the cost of the turbine and generator has been estimated using the chart from Figure 0-23 for which the expression corresponding to a generator having two magnetic pole pairs ($pp = 2$) has been selected.

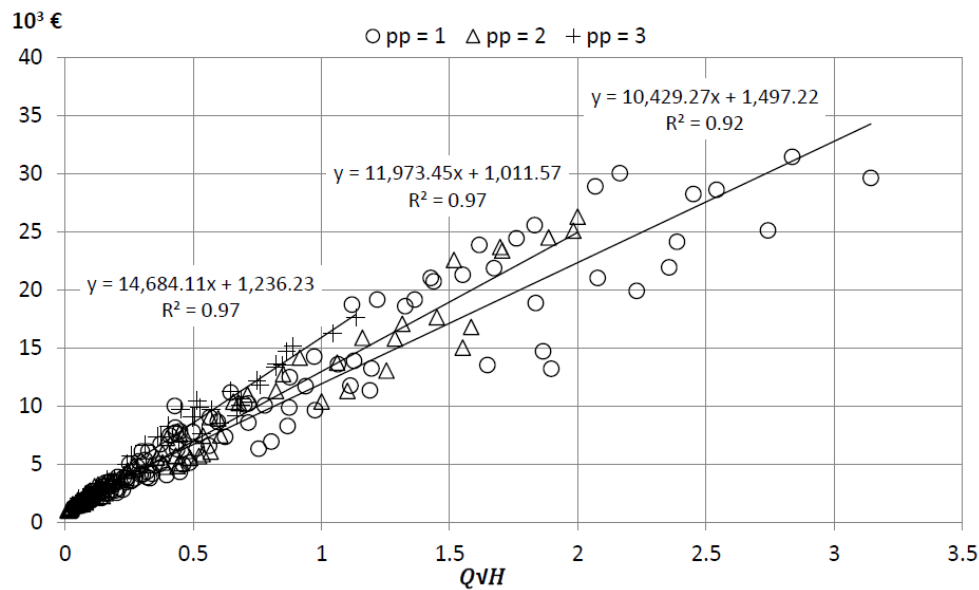


FIGURE 0-23. PAT COST MODEL.

Hence, the cost of the pump-generator unit is estimated as:

$$C_{PAT+generator}(\text{€}) = 11,973 Q\sqrt{H} + 1012$$

(2)

Where

Q = flow rate through the turbine (m^3/s);

H = hydraulic head available at the turbine (m).

Instead, for the low head turbine suggested for the site n. 5 (Waun Pond) a cost figure of 2,300 €/kW was used as confidentially suggested by a representative of a hydro turbine manufacturer (Turbiwatt).

Accessory costs

Apart from the cost of the turbine and generator, other important factors to be considered are the design, civil works, installation, financial and commissioning costs. Since they are highly variable due to site-specific conditions, two scenarios have been considered based on available case-studies:

- optimistic scenario, in which the accessory costs are equal to the cost of the turbine and generator;
- pessimistic scenario, in which the accessory costs total three times the cost of the turbine and generator.

Total estimated investment cost (C)

The total estimated project cost has been computed as the sum of the components relative to the turbine and generator and the accessory costs:

$$C(GBP) = C_{turbine+generator}(GBP) + C_{accessory}(GBP)$$

Yearly energy yield (E)

Given the uncertainties involved in the flow and head estimates, two scenarios have been assessed for each site:

- optimistic scenario, in which the maximum estimated flow rate is available for 70% of the time during a typical year;
- pessimistic scenario, in which the maximum estimated flow rate is only available for 30% of the time during a typical year.

Economic payback

Finally, the economic payback period has been calculated as:

$$PB = \frac{C(GBP)}{E(kWh/year) i(GBP/kWh)}$$

Where

PB = economic payback of the scheme (years)

C = total cost of the MHP facility (GBP)

E = total power output over one year (kWh/year)

i = income from electricity sale.

The factor i has been set equal to 7.8 pence per kWh as of September 2017 (source: Ofgem).

Overview of the selected sites

The results shown in Table 0-7 sum up the main findings regarding the eleven proposed hydropower sites. Highlighted are the rows relative to the sites n. 1, 2, 3 and 11 which even under

the most pessimistic scenario (lowest flow availability and highest percentage of accessory costs) present a payback period lower than ten years.

TABLE 0-7. OVERVIEW OF THE POTENTIAL OF THE SELECTED SITES

	Location	Power output (kW)	Technically feasible?	Turbine type	Total estimated investment (GBP)	Payback (years)
1	Anvil Court Culvert	8 - 16	✓	PAT	12 - 25,000	2 - 8
2	Sirhowy tributary (b)	3 - 38	✓	2 PATs	32 - 64,000	2 - 8
3	Rassau Industrial Estate culvert (a,b)	3 - 8	✓	PAT	6 - 12,000	2 - 8
4	Cwm Tillery lake (b)	1 - 5	✓	2 PATs	8 - 15,000	4 - 16
5	Waun Pond	3 - 6	✓	Low head	20 - 41,000	8 - 34
6	Waun y pound		X			
7	Tanglewood stream	6	✓	2 PATs	10 - 21,000	4 - 16
8	Culvert off Brynbach Park	3 - 5	✓	1 or 2 PATs	10 - 20,000	4 - 18
9	Whitworth Terrace Stream		X			
10	Carno Reservoir Rassau runoff	2	✓	PAT	3 - 6,000	4 - 15
11	Carno reservoir	10	✓	PAT	8 - 15,000	2 - 8

Next steps

As a next step, it would be recommendable to put in place for an adequate period of time some type of flow rate measuring devices in order to gain quantitative flow data from one or more of the shortlisted sites. Such knowledge will allow for a more precise calculation of the technical and economic potential, as well as environmental aspects.

5. DODDER WATER SUPPLY SCHEME (IRISH WATER)

Overview

The Dodder Water Supply Scheme was built in the early 1880's to regulate the flow regime of the Dodder river and to supply the townships of Rathmines and Rathgar with clean drinking water. In 1932 the townships were dissolved and the Dublin Corporation (later Dublin City Council) took over the management of the infrastructure.

A delegation from the Dŵr Uisce project visited the Dodder Water Supply Scheme on 13/06/2018 to assess the potential for hydraulic energy recovery within the existing infrastructure.

The scheme consists of:

- Two reservoirs created by means of earth filled embankments along the Dodder river:
 - An upper reservoir used for water supply to the population
 - A lower reservoir used for flood control
- A 2.7 km-long 600 mm pipeline renovated in the late 2000's conveying water from the upper reservoir until the car park where flow control valves are located
- Two parallel 12" and 15" pipelines, 4.8 km-long, bringing water from the mentioned car park to the Ballyboden Water Treatment Plant (WTP). These cast iron pipelines are still the original elements dating back from the 1880's.
- The Ballyboden WTP, with a capacity of 16,000 m³ per day.

An overview of the whole infrastructure is shown in Figure 0-24, while Figure 0-25 displays the terminal part of the supply lines entering the treatment works. A picture of the actual terminal part of the two 12" and 15" lines at the WTP is shown in Figure 0-26.

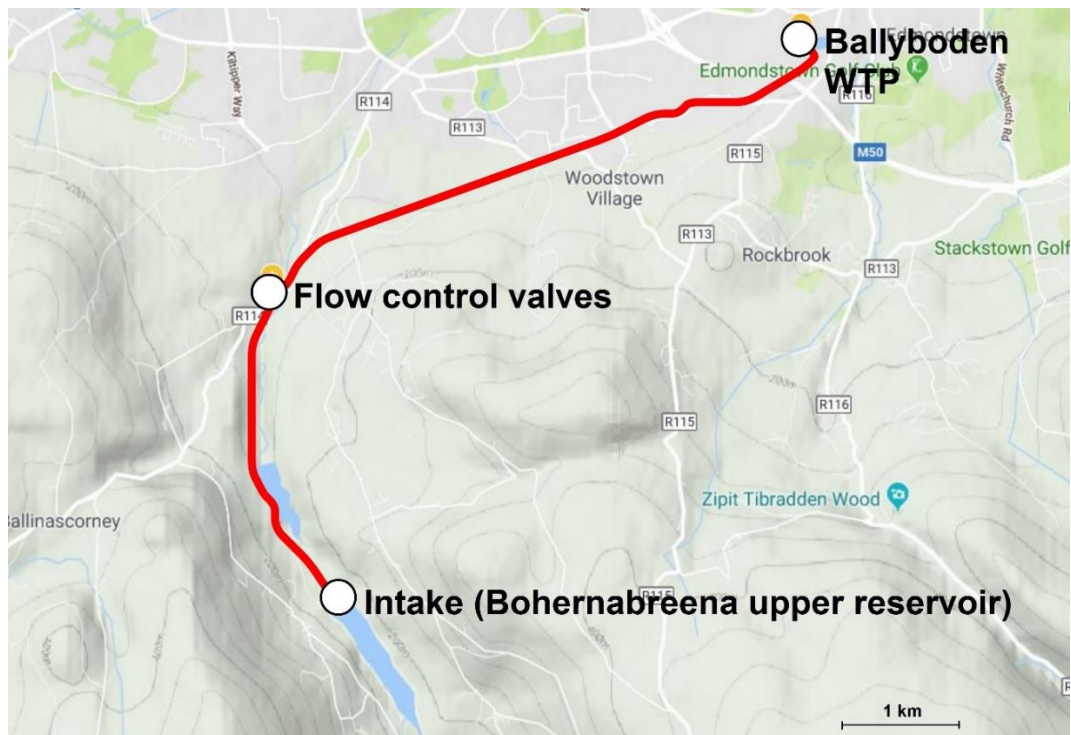


FIGURE 0-24. NETWORK MAP (SOURCE: GOOGLE MAPS)

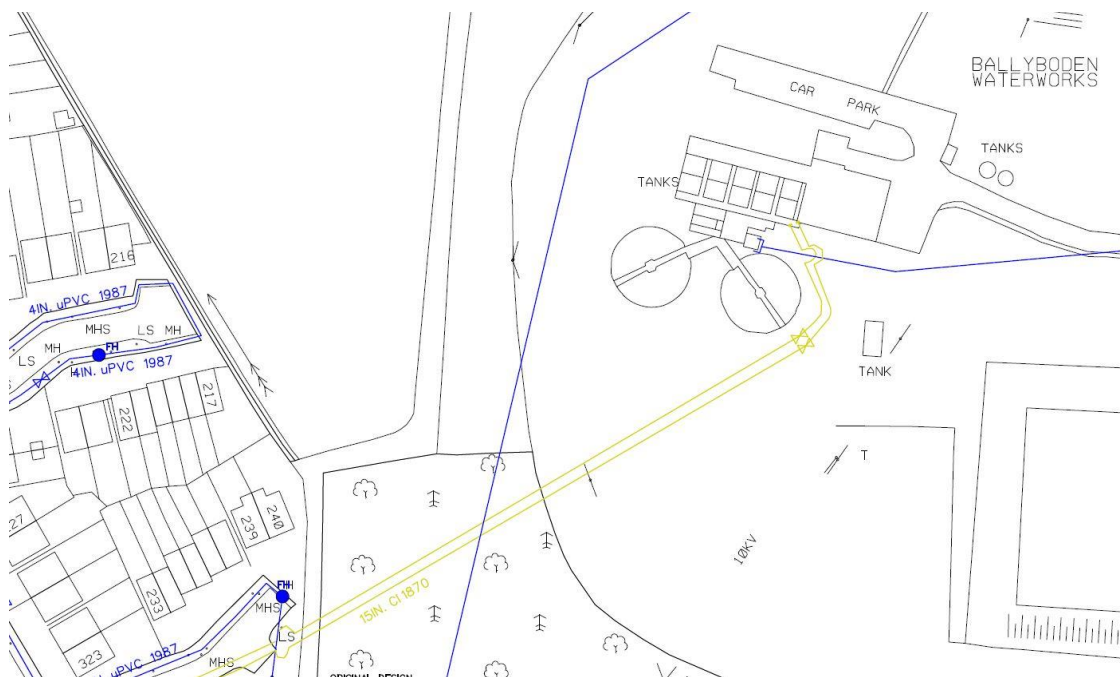


FIGURE 0-25. TERMINAL SECTION OF THE 12" AND 15" PIPELINES (IN YELLOW) ON ENTERING THE BALLYBODEN WTP



FIGURE 0-26. THE WATER INLET AT BALLYBODEN WTP

The flow rate entering the treatment works is kept within the range 500-600 m³/h by adjusting the opening of two control valves located below the ground level of the Bohernabreena reservoir car park (see Figure 0-28), immediately after the main 600 mm pipe splits into the 12" and 15" parallel lines. On the day of the Dŵr Uisce team site visit the flow rate corresponded to 652 m³/h as seen from Figure 0-27.



FIGURE 0-27. RAW WATER FLOW METER AT BALLYBODEN WTP AS SEEN ON 13/06/2018



FIGURE 0-28. THE LOCATION OF FLOW CONTROL VALVES AT BOHERNABREENA RESERVOIR CAR PARK AS SEEN ON 13/06/2018

The potential for hydroelectric energy recovery derives from the difference in elevation between the reservoir and the Ballyboden WTP, summed up in Table 0-8.

TABLE 0-8. ELEVATION ABOVE SEA LEVEL OF KEY POINTS ALONG THE DODDER WATER SUPPLY SCHEME

Location	Approx. elevation (Metres above sea level)
Free surface of Bohernabreena upper reservoir	173
Flow control valves at the car park	119
Ballyboden WTP	99

As seen from Table 0-8 a 74 metres gross hydraulic head exists between the level of the upper reservoir and the inlet of the treatment works, which under current operations is partially dissipated as friction along the pipelines and partially by means of the two flow control valves at the car park.

Hydraulic model of the pipework

The whole water network supplying the Ballyboden WTP was re-created by means of the EPANET hydraulic modelling software, and the chart of Figure 0-29 shows the hydraulic head/flow characteristic of the pipe system.

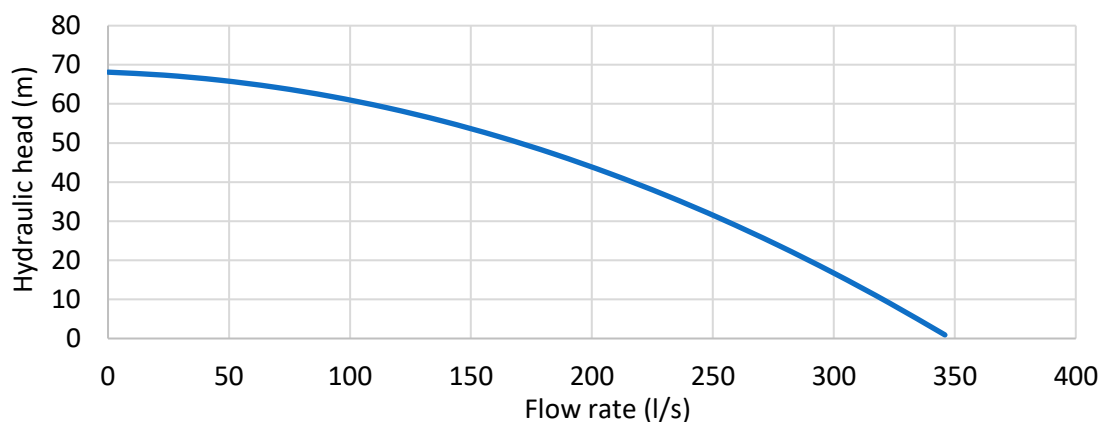


FIGURE 0-29. NET HYDRAULIC HEAD AS FUNCTION OF FLOW RATE

Preliminary design of the hydropower installation

The small size and the characteristics of the site make it ideal for the installation of a Pump as Turbine energy recovery device, able to generate significant amounts of renewable electricity by exploiting the potential being currently dissipated by the flow control valves. With a mean flow of 180 l/s (650 m³/h) and 48 metres of hydraulic head a single centrifugal PAT would be an

appropriate choice. Considering a fixed turbine efficiency of 75% the potential for energy recovery from the existing pipeline as function of the flow rate is displayed in Figure 0-30.

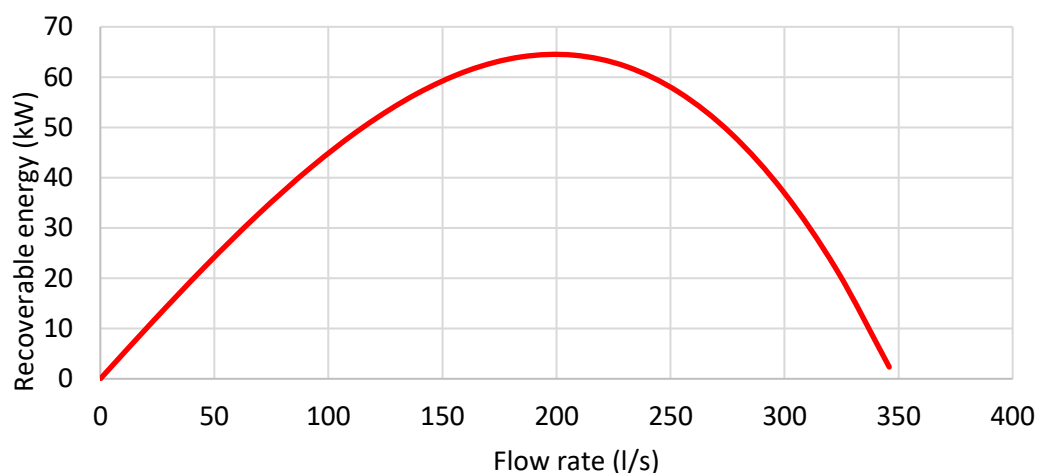


FIGURE 0-30. POTENTIAL ENERGY RECOVERY FOR A TURBINE WITH 75% OVERALL EFFICIENCY

As seen from Figure 0-30, with the typical 24-h flow rate of 180 l/s a hydraulic turbine installed along the water supply line would generate approximately 62-64 kW of renewable energy.

As a further step, the Pump As Turbine selection software developed at Trinity College Dublin as part of D  r Uisce research project has been applied to the selected site and helped identifying the ideal PAT and generator to be chosen. A fixed turbine shaft speed of 1520 RPM was selected, corresponding to the use of a four-pole standard asynchronous motor as generator. The results from the PAT selection software are displayed in Figure 0-31.

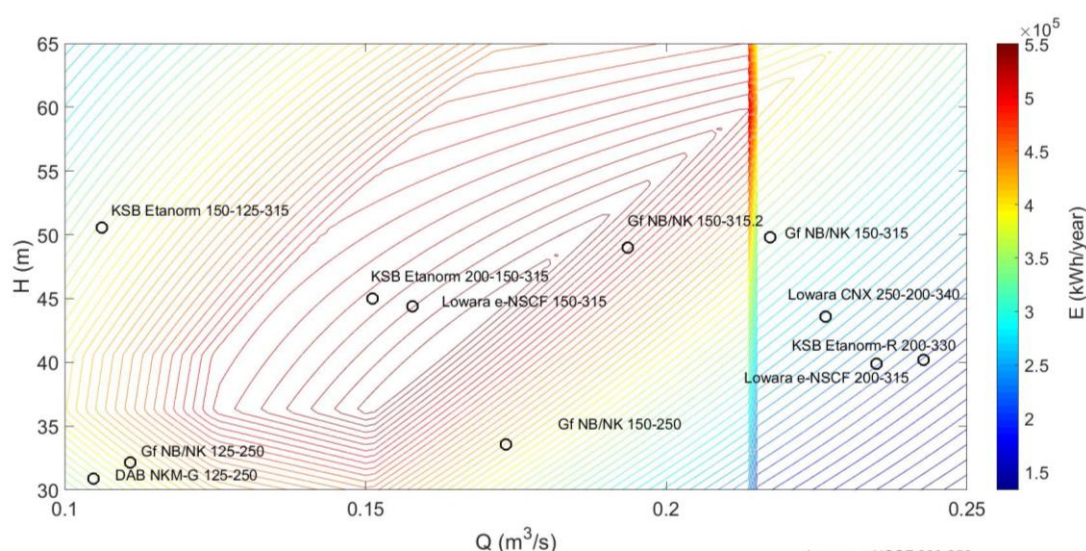


FIGURE 0-31. RESULTS OF THE PAT SELECTION SOFTWARE DEVELOPED AT TRINITY COLLEGE DUBLIN

The results of Figure 0-31 indicate that a PAT installed along the supply line to the Ballyboden WTP would generate up to 540,000 kWh per year of renewable electricity.

The elevation difference between the Ballyboden WTP and the flow control valves at the Bohernabreena car park matches approximately the amount of friction dissipated at a flow rate of 650 m³/h along the 12" and 15" pipelines running between those same nodal points. Therefore, the proposed energy recovery turbine could be installed alternatively at the inlet of the treatment works or at the Bohernabreena car park, provided that the mentioned valves remain fully open.

The installation of the hydropower equipment at the Ballyboden WTP would be more economically beneficial since it could directly offset the energy consumption of the plant itself and may share parts of the existing infrastructure. However, in order to avoid further solicitations on the 140 years-old 12" and 15" pipelines it would be appropriate to install it at the car park location instead or as part of an upgrade of this section of pipeline.

Economic analysis

The cost forecast for the proposed 64 kW hydropower installation is displayed in Table 0-9. The required investment may range from less than 80,000 to about 140,000 € according to several factors such as powerhouse location and selection of equipment manufacturer.

TABLE 0-9. LIST OF PREDICTED PROJECT COSTS

Item	Indicative cost (€)
64 kW Pump As Turbine & induction generator	13,500 – 20,500 €
Other project costs, including:	
1. Control system	
2. Commissioning	
3. Civil works and other hydraulic equipment	
4. Grid connection	
5. Project costs	65,000 - 120,000 €
TOTAL	78,500 – 140,500 €

The main advantages of installing a Pump As Turbine rather than a conventional hydro turbine (e.g. Francis or Crossflow) for this particular application are twofold:

1. It can work submerged, thus not interrupting the hydraulic supply line
2. A conventional turbine plus generator having identical 64 kW power output would be priced between 66,000 and 100,000 € with respect to the 13,500 – 20,500 € cost forecast of a PAT with generator as seen from Table 0-9.

In case the turbine is installed at the treatment works the produced electricity is expected to be wholly self-consumed within the plant which at the moment relies on a 150 kVA grid supply. Considering an average industrial electricity price for SMEs in Ireland equal to 0.164€ per kWh (Costs of Doing Business in Ireland 2016, National Competitiveness Council, April 2016), the yearly revenues due to the electricity savings granted by the scheme would be equal to 88,500 €. This would represent **a return on investment of less than two years** even under the least favorable costs indicated in Table 2.

Instead, if the turbine was to be installed at the car park location the generated electricity could be sold to the grid at a rate subject to negotiation. As an example, the 90 kW hydropower installation at Vartry treatment works in 2008 was awarded a feed-in tariff of 0.08 €/kWh for the excess energy exported to the grid.

Conclusions

The potential for energy recovery within the existing infrastructure of the Dodder Supply Scheme is equal to 64 kW, and could conveniently be exploited by installing a Pump As Turbine (PAT) along the water supply line. From an economic point of view the project has an interesting potential, since the initial investment is expected to be offset between **one and two years** since the beginning of operations.

6. MULLINS MILL, IRELAND

Hydropower potential assessment



FIGURE 0-32. VIEW OF THE MULLINS MILL (LEFT) AND THE RIVER WEIR (CREDITS GOOGLE MAPS)

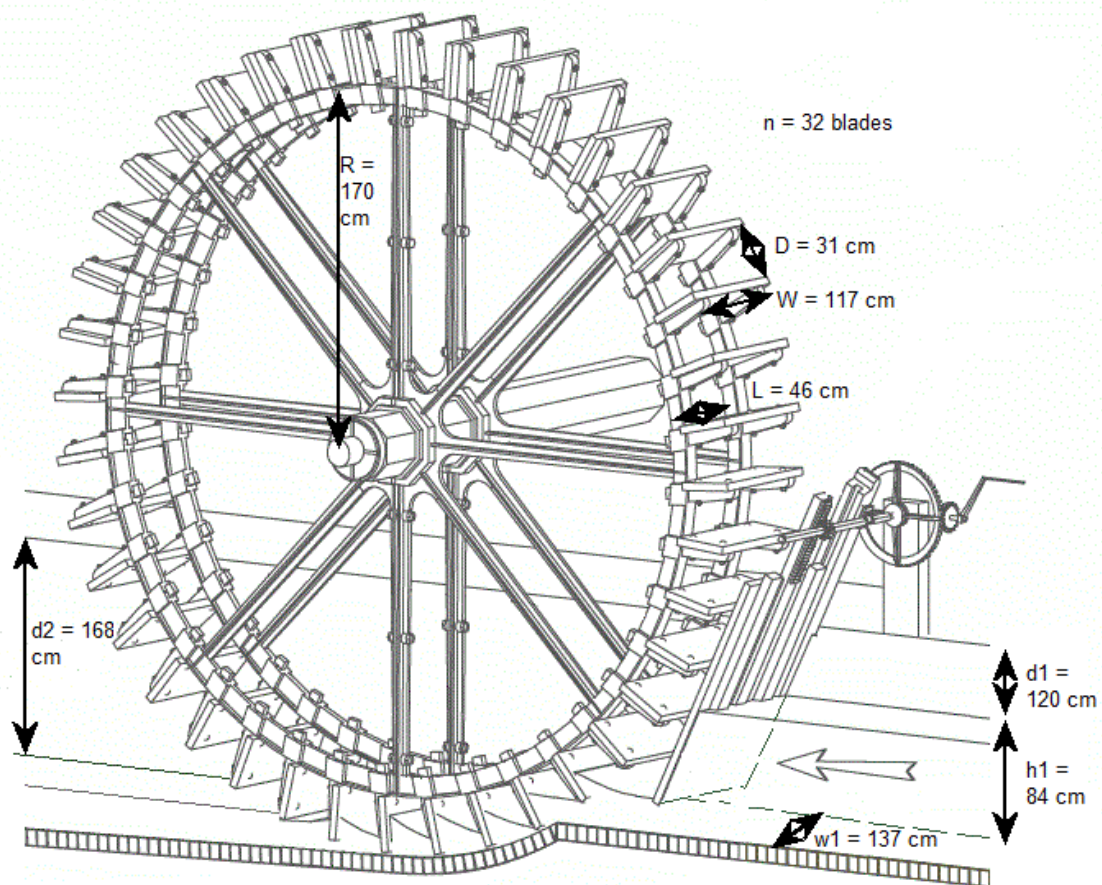


FIGURE 0-33. DIMENSIONS OF THE WHEEL (ORIGINAL PICTURE:
[HTTPS://MILLSARCHIVE.ORG/FILES/IMAGES/ILLUSTRATED-GLOSSARY/UNDERSHOT_WATERWHEEL.GIF](https://millsarchive.org/files/images/illustrated-glossary/undershot_waterwheel.gif))

On the day of the visit (3/9/2018) the estimated hydraulic parameters were:

- Flow rate Q : $0.54 \text{ m}^3/\text{s}$
- Wheel angular speed: 5 RPM
- Driving head $H = 0.48 \text{ m}$ (equal to $d_2 - d_1$)

The theoretical hydraulic potential was therefore equal to:

$$P_{th}(kW) = 9.81 \cdot H(m) \cdot Q \left(\frac{\text{m}^3}{\text{s}} \right) = 2.54 \text{ kW}$$

Considering that the reported efficiency of similar undershot waterwheels are in the range 30-70%, the effective power output was approximately **0.8-1.8 kW** and was consumed as friction by the connected gears. There is a possibility to recover a little amount of electricity with little modifications to the existing setup, by installing the following elements:

1. A combination of transmission belt and fixed gear with an overall 1:100 ratio
2. A small permanent magnet generator with rectified input, rated for 1 kW at 500 RPM
3. A direct DC load (e.g. a ceramic heater) or a small power inverter to utilize the electricity within the building

The mechanic to electric energy conversion efficiency of the setup can be estimated as 50%, therefore the expected power output will correspond to **0.4-0.9 kW**. The power output can be furtherly enhanced if one or both the following conditions are met:

- a) A rise of the upstream river level, which is likely to occur under wet periods
- b) An increase of the wheel speed by a greater opening degree of the inclined sluice gate, at a cost of greater structural stresses on the machine



FIGURE 0-34. EXTERIOR AND INTERIOR VIEW OF THE MULLINS MILL

7. RATHNURE TANK (BLACKSTAIRS GROUP WATER SCHEME)

Overview

The Blackstairs group water network is a gravity flow system supplied by a main reservoir located at the eastern face of Blackstairs Mountain, Co. Wexford. With an average demand of 1,500 m³/day, 1,037 households are supplied. The piping system has an approximate length of L = 117 km and the diameters range from D = 50 mm to D = 150 mm. A previous study from the Dŵr Uisce team (Dwr Uisce, 2017) estimated a potential of 0.75 kW for a PAT installation at the Rathnure water break pressure tank inlet, based on a model of the distribution network. The location of the Rathnure tank is circled in red in Figure 0-35.

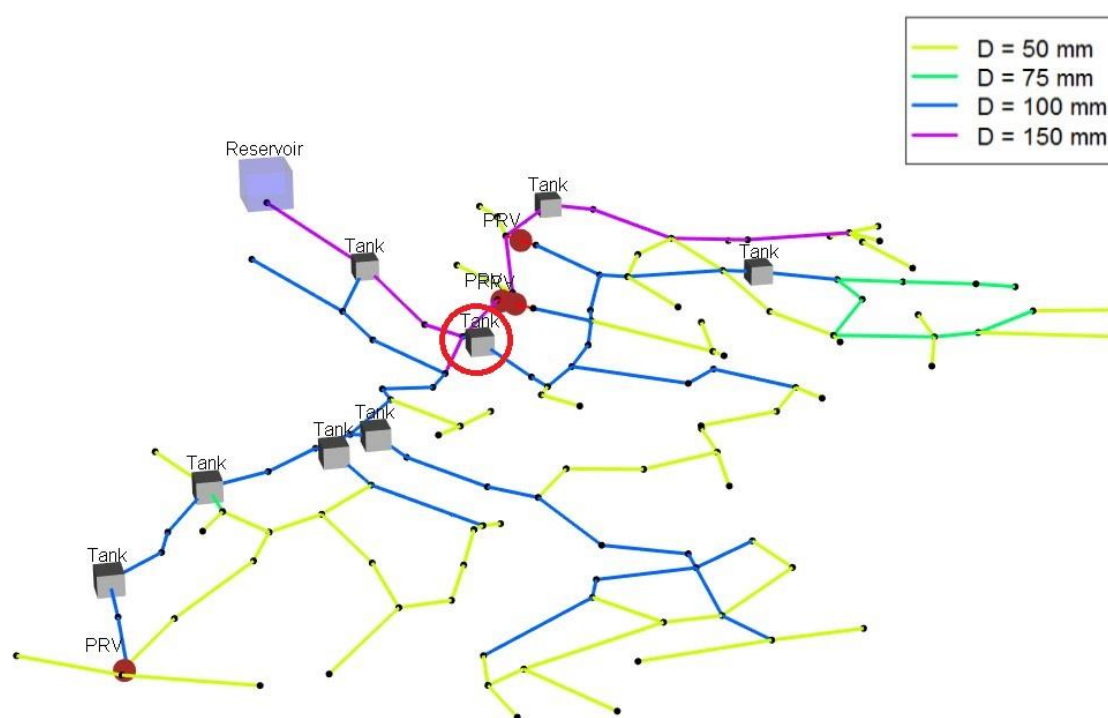


FIGURE 0-35. OVERVIEW OF BLACKSTAIRS NETWORK LAYOUT, WITH THE LOCATION OF RATHNURE TANK CIRCLED IN RED

Measured data from Rathnure storage tank

Knowing the network map and the population pattern, as well as having flow data records at the water treatment plant, a model of the system was created on EPANET software to simulate the steady-state flow and pressure at any nodal point (Dwr Uisce, 2017). After having selected the Rathnure tank inlet as the most promising site among the analysed sites, in-field measurements of pressure upstream of the tank and outflow records have been acquired and compared with the data

obtained from the numerical model. The records relative to flow rate, tank level and upstream pressure are graphically shown in Figure 0-36 and Figure 0-37.

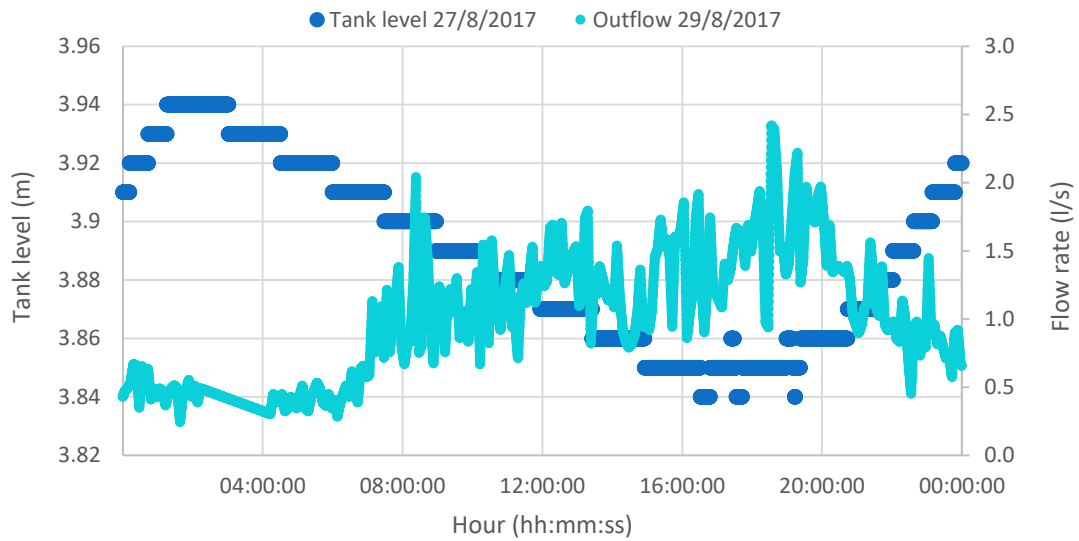


FIGURE 0-36. RECORDED OUTFLOW AND WATER DEPTH AT RATHNURE TANK

Since the daily fluctuation in water tank level is minimal, we must assume that the operational rule governing the opening of the inlet valve consists in maintaining the water level as constant as possible by granting an inflow pattern almost equal to the outflow one.

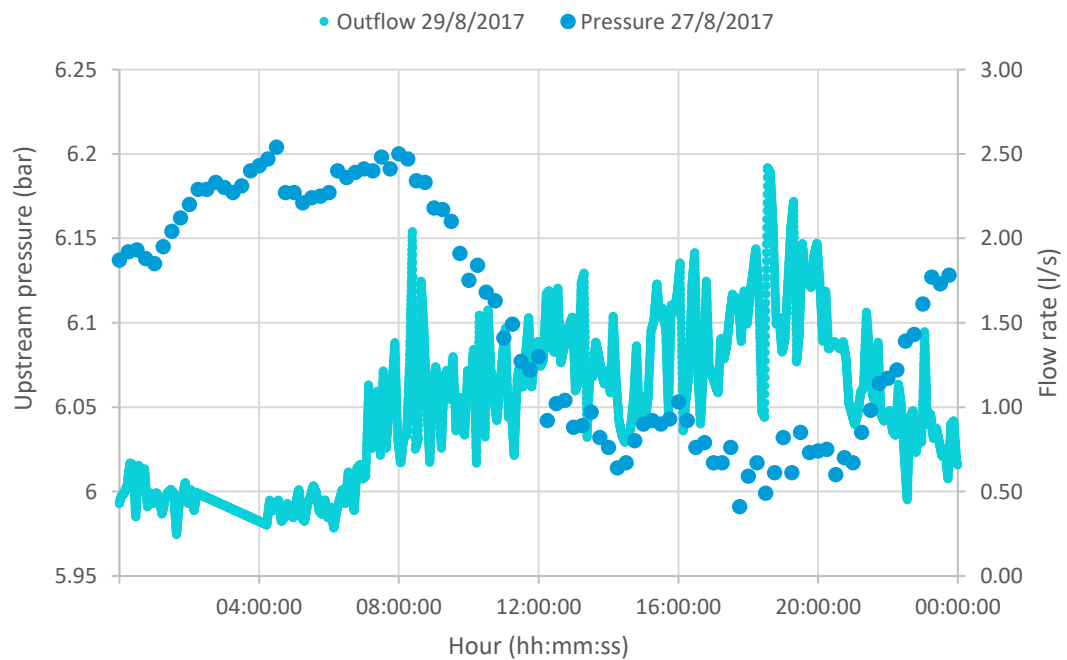


FIGURE 0-37. RECORDED OUTFLOW AND UPSTREAM PRESSURE AT RATHNURE TANK

Given the “mirrored” shape of upstream pressure and outflow displayed in Figure 0-37, we can corroborate the assumption that the inflow daily pattern is equal to the outflow one at Rathnure tank. Besides, the recorded upstream pressure is nearly constant along the 24 hours. The mean water consumption from Rathnure tank was found to be equal to 80 – 90 m³ per day.

With respect to the constructed EPANET model, both the effective upstream pressure and flow rate from Rathnure tank are lower than predicted. The predicted average values of upstream pressure and flow rate were respectively equal to 7.36 bar and 1.52 l/s, +17% and +34% of the recorded values respectively.

The flow – pressure curve shown in Figure 0-38 is largely flat, meaning that water conveyance pipes present a low level of head losses due to their large diameter in relation to the flow rate.

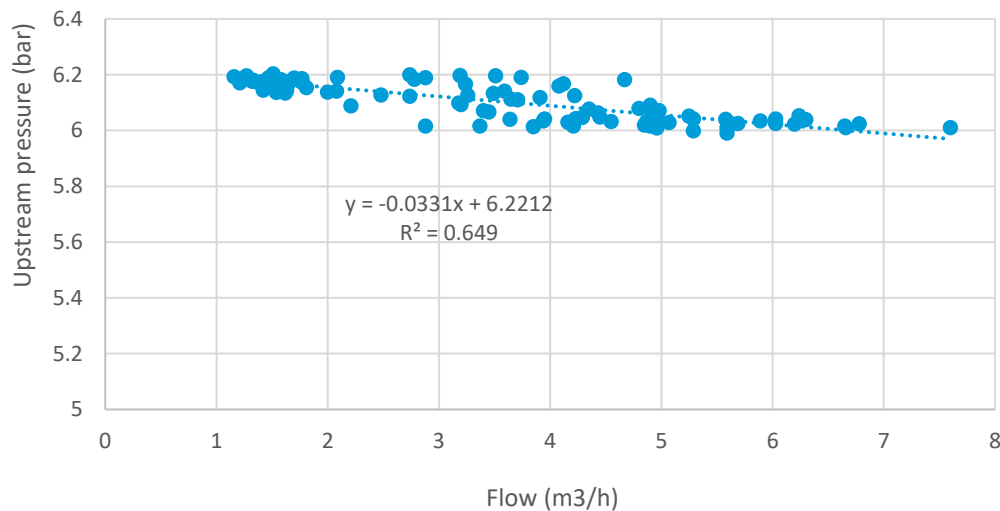


FIGURE 0-38. UPSTREAM PRESSURE VS. FLOW RATE UPSTREAM OF RATHNURE TANK

Figure 0-39 displays the theoretically available power for energy recovery along a typical day with the current tank regulation system.

This is the amount of power that a 100% efficient hydropower plant could potentially harvest. In reality, a turbine inserted in this context could realistically achieve a maximum power of 1-200 W during peak hours and a minimum of 0-40 W during the night.

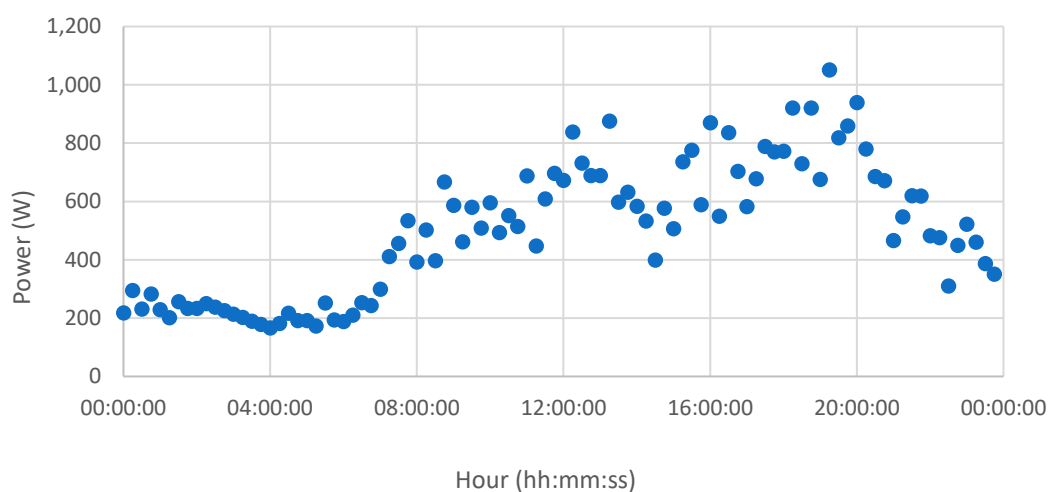


FIGURE 0-39. POWER AVAILABLE FOR ENERGY RECOVERY ALONG A TYPICAL DAY

Electricity demand at Rathnure tank

The on-site energy consumption at Rathnure tank at the moment is very low, as the only equipment installed is a water level transducer and a flow meter. The meter readings during 16 months are displayed in Table 0-10.

TABLE 0-10. METER READINGS AT RATHNURE TANK

Date of reading	kWh	Consumption (kWh)
30/03/2016	391	
07/07/2016	448	57
20/09/2016	491	43
23/11/2016	527	36
19/01/2017	564	37
11/05/2017	633	69
20/07/2017	673	40

The mean daily consumption of energy over the monitored 16 months was found to be equal to 0.591 kWh per day, with a maximum value of 0.649 kWh/day and a minimum of 0.563 kWh/day. This is consistent with the nominal power demand of the installed flow meter and level transducer, which corresponds to approximately 25 W.

The Blackstairs group water scheme management has expressed an interest in adding a chlorine analyser at the site, which could potentially bring the onsite consumed power to 33-35 W for a daily electricity consumption of around 0.84 kWh/day.

Rathnure tank pictures

Figure 0-40 and Figure 0-41 show the status of Rathnure tank inlet duct and electric cabinet as of 25/8/2017.

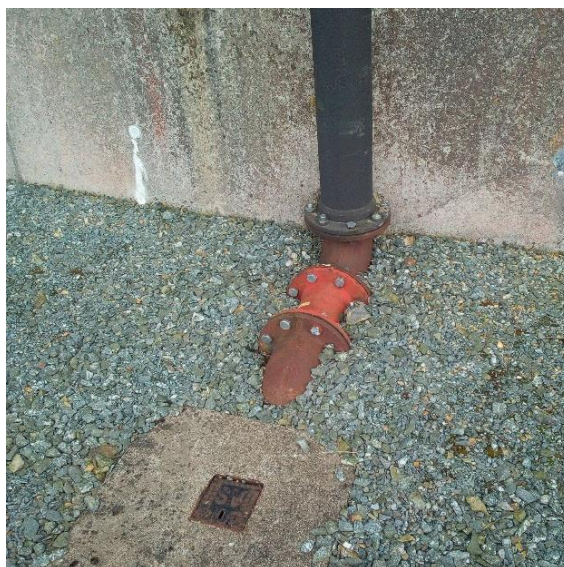


FIGURE 0-40. PICTURES OF THE WATER INTAKE INTO RATHNURE TANK AS OF 25/8/2017



FIGURE 0-41. ELECTRIC CABINET AT RATHNURE TANK WITH VISIBLE FLOW METER AND LEVEL TRANSDUCER AS OF 25/8/2017

Design alternatives

Two alternatives are possible: maintaining the current tank inflow regulation, aimed at keeping an almost constant water level, or adopting an on/off tank feeding procedure. An illustration of the scheme layout for the current situation and the two design options is shown below in Figure 0-42.

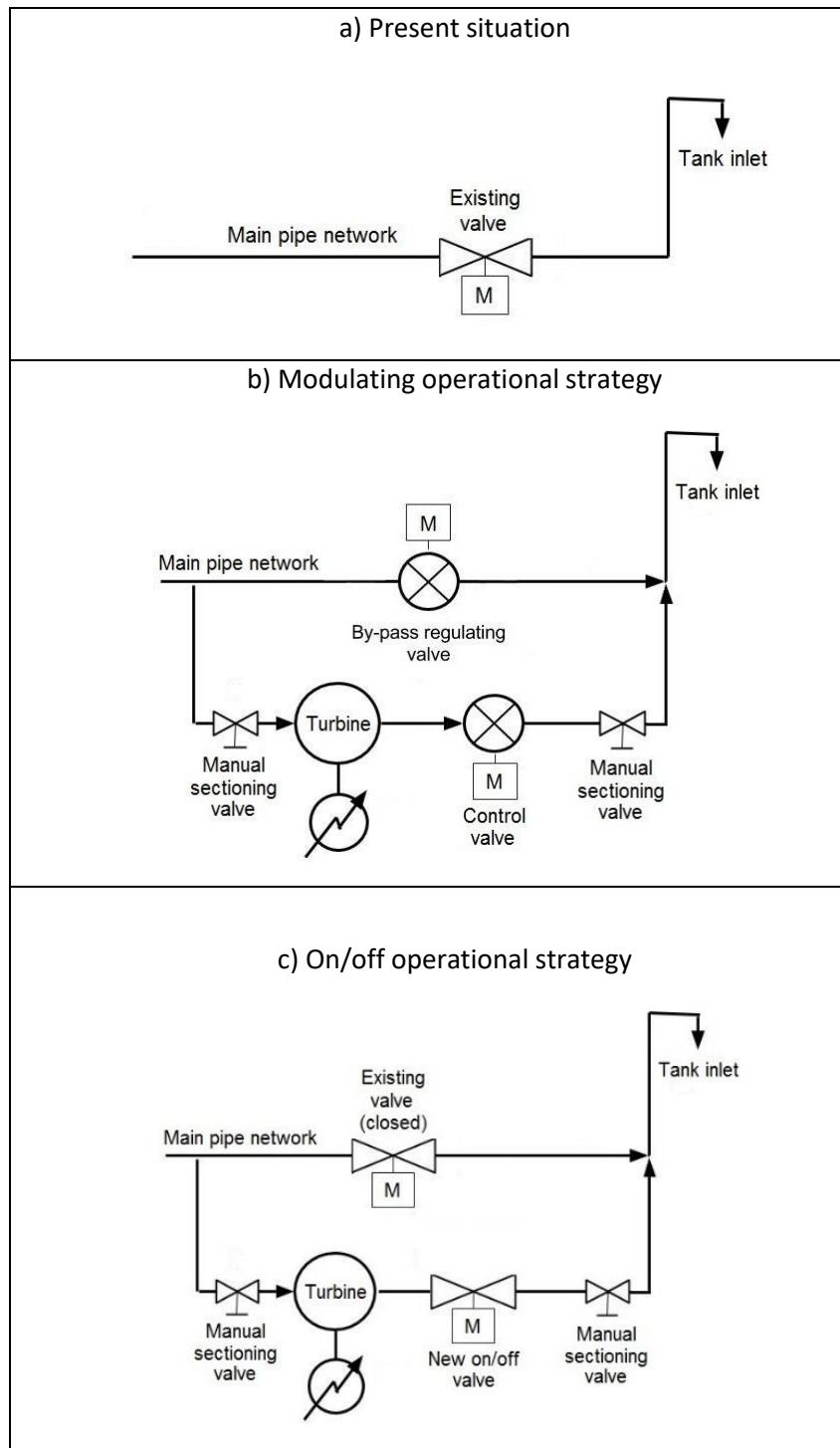


FIGURE 0-42. SIMPLIFIED LAYOUT OF TANK INLET ACCORDING TO DIFFERENT DESIGN ALTERNATIVES

Modulating operational strategy

The first option, corresponding to the pipe layout of Figure 0-42b, aims at maintaining the present day modular inflow regulation with a flow rate ranging from 0.4 l/s during the night up to 2.4 l/s during peak demand hours.

The advantages of such strategy would be:

- the need for a smaller, inexpensive turbine;
- the building of an energy recovery scheme powerful enough to power the local instrumentation (flow meter, level transducer and chlorine analyser) without generating a significant surplus energy.

Conversely, the disadvantages of such design strategy would consist in:

- the requirement for two expensive modulating actuated control valves, worth around 2,000€ each;
- much of the available potential would be wasted because such a small turbine would have extremely low efficiency and some water will need to be diverted through a bypass;
- a lack of confidence in the turbine design for such small sizes ($P \approx 150$ W).

A few examples of pico-turbines for tank inflow regulation either already being sold on the market or under development are shown in Figure 0-43. All of these units are meant to operate with a battery pack in order to supply power to nearby instrumentation.

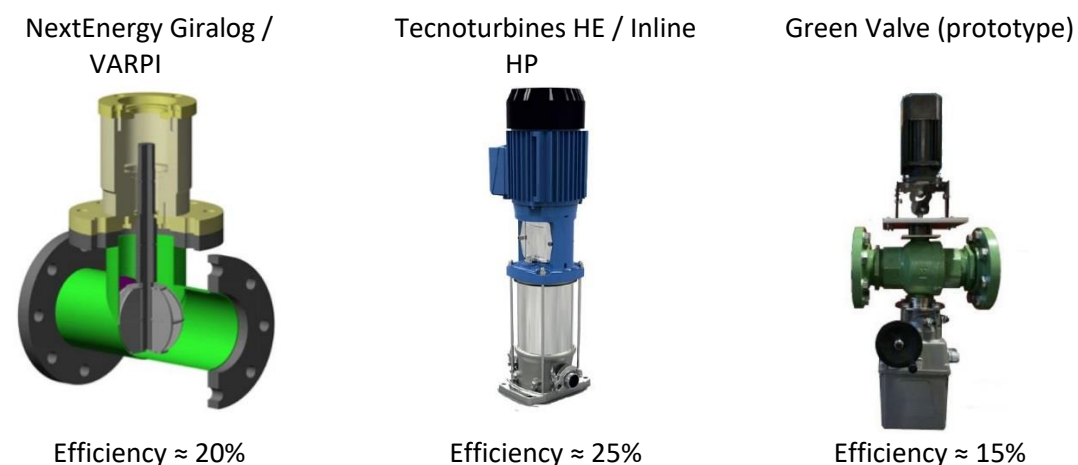


FIGURE 0-43. EXAMPLES OF TANK INFLOW REGULATING PICO-TURBINES EXISTING OR UNDER DEVELOPMENT

All of the above units have a minimum flow rate threshold, equal to approximately 1 l/s, and possess a fairly low efficiency. The Tecnoturbines HE unit is indeed a PAT, although sold directly as turbine. An off-the-shelf available pump could also be selected to operate as PAT on the Rathnure site, but the PAT design algorithm developed at TCD is based on larger machines ($P > 1$ kW) and therefore its applicability on such a small turbine ($P \approx 150$ W) is very questionable.

On/off operational strategy

Instead, adopting an on/off inflow regulation strategy as schematically shown in Figure 0-42c would potentially bring about several advantages:

- the larger scale of the turbine could allow for selecting a more efficient unit, with an efficiency of $\approx 50\%$ instead of the previous 15 – 25%;
- an increased confidence in the design procedure would allow a cheap and reliable PAT to be used as turbine;
- the ability to generate up to 6 kWh per day as opposed to 2-3 kWh/day when in modulating operational strategy;
- only a single cheap and reliable on/off valve would need to be installed;
- there would be no energy losses due to bypassed water;
- it may even be economically feasible to connect the installation to the electric grid.

Instead, the few disadvantages of an on/off regulating strategy could be identified as:

- the requirement for a larger storage battery if no grid connection is established;
- it would induce oscillations in the tank level during a typical day up to 0.5 m.

For instance, a reasonable on/off inflow control strategy could correspond to allowing into the tank a constant flow rate of 4 l/s for 6 hours per day in order to match the 86 m³ of daily water downstream consumption. Under such flow rate, a net head of 50 m could be exploited by a turbine installed at the tank's feet. A multistage PAT could be an appropriate choice for the given flow and head conditions.

As an example, a KSB Multitec 32-2.1 multistage PAT whose performance charts and dimensions are shown in the Annex could fit well in the context. In case such a turbine is adopted, and

supposing to set the on/off inlet valve to stay open for two consecutive hours three times per day, the generated electric power and the resulting Rathnure tank level are shown in Figure 0-44 and Figure 0-45.

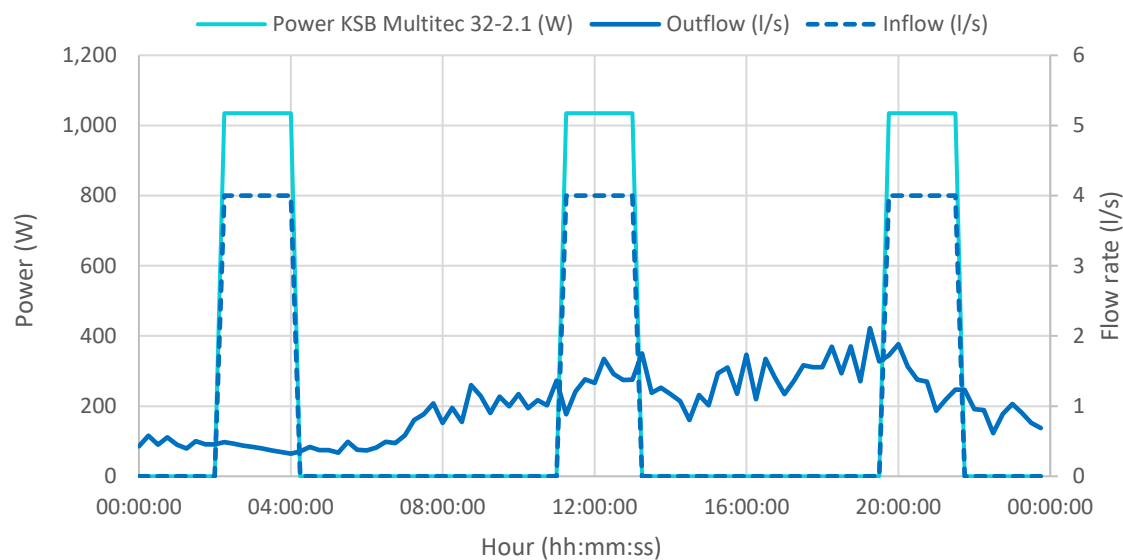


FIGURE 0-44. GENERATED ELECTRIC POWER AND WATER FLOWS INTO AND FROM THE RATHNURE TANK ALONG 24H

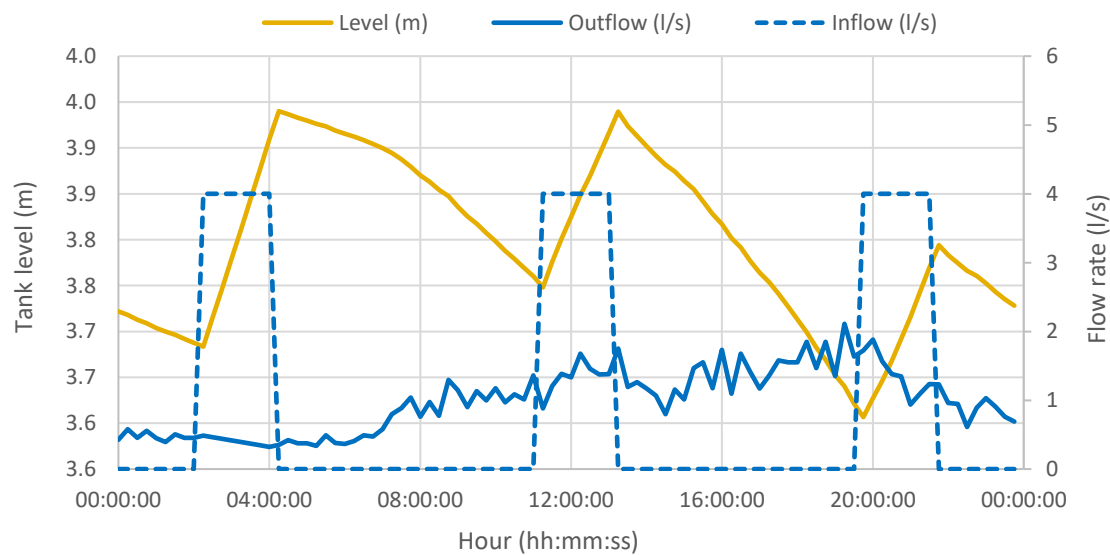


FIGURE 0-45. WATER DEPTH AND FLOWS INTO AND FROM THE RATHNURE TANK ALONG 24H

The duration of the on/off cycles should be set as the longest possible that doesn’t pose limitations in terms of water pressure delivered to users and leaves a sufficient level of water in the tank to assure supply reliability.

Energy usage opportunities at Rathnure tank

Independently from the chosen tank operational strategy, the proposed turbine system (option 2) would generate an excess of energy with respect to the in-situ equipment consumption (0.6 – 0.84 kWh/day). Therefore, it could be convenient to identify other potential consumption points located in the proximity of Rathnure tank. An example could be an LED street light (rated for 100 W) able to increase the night time visibility of the nearby crossroad as shown in Figure 0-46.



FIGURE 0-46. AERIAL VIEW OF THE RATHNURE TANK AND THE NEARBY CROSSROAD

In case that an on/off operational strategy is chosen, its significant daily energy yield (≈ 6 kWh) would allow for powering a set of up to 6 such LED light that may be placed along the nearby road.

Conclusions

The potential for energy recovery observed at Rathnure tank inlet is in the range of pico (0 – 5 kW) rather than micro (5 – 100 kW) hydropower. As a rule of thumb, this generally restricts the generated energy to on-site use rather than feeding it into the national grid.

Two possible operational rules controlling the inflow into Rathnure tank have been analysed, among which the on/off strategy proved to offer significant advantages under technical, economic and operational terms. Within such scenario, a particular multistage PAT manufactured by KSB would allow for an energy yield of around 6 kWh/day. Since the electricity consumption requirements of

the locally deployed monitoring equipment is limited, new opportunities for local use of the generated energy as feeding new LED street lamps could be considered.

Cost estimate for on/off operation strategy

A preliminary cost estimate for turbine installation at Rathnure tank inlet following an on/off tank feeding procedure is shown in Table 0-11. The item list was compiled assuming that the electric and control equipment may be housed in the existing electric cabinet, while the turbine and the bypass shall be housed in a 1x1x1 m inspection chamber at the basis of the tank.

TABLE 0-11. PRELIMINARY COST ESTIMATE

Item	Indicative cost (€, excl. VAT)
Y-strainer at turbine inlet	
To avoid accidental damage to the turbine due to gravel or small stones	150
Actuated on/off valve	
In series to the turbine	250
Actuated on/off valve for bypass opening	
During turbine maintenance	780
Manual butterfly valves (2x)	
For turbine maintenance	130
Pipework	400
Power meter	80
Control System (Siemens PLC)	
Logo8 series, with analog input extension	250
Turbine and generator	3,000
Concrete manhole and inspection chamber	
Pre-cast concrete, 1x1x1 m, housing the turbine and the bypass	400
Manpower	1,000
1 kWh lithium battery	
For powering the equipment along 24h	570
Dump load	40
Charge controller	150
Total (€)	7,150
Total with VAT (€)	8,800

The amount is exclusive of the cost for purchasing a chlorine analyser (?) and six street lamps and respective poles (3,000 € each). Unless the turbine is connected to the grid, and the tank feeding routines are uniformly spaced along a 24-h period, a large 6 kWh lithium battery (5,000 €) will be

required to power the lamps. On the other hand, the connection to the grid could cost up to 2,000 € between procedural expenses and equipment.

An alternative to connecting to the grid or to buy a large 6kWh battery could be to schedule the tank feeding routine for setting the valve on during the night only, in order to feed directly the LED lamps without the need for storing energy. The drawback of this strategy would be causing a drop in the tank level of approximately 1 m between early morning (full tank) and late evening (lowest water depth).

ANNEX 2: MPPT CODE FOR THE ARDUINO SHIELD

```
//MPPT controller 12/10/2018

//Daniele Novara

//Variables declaration

float v_q;

float v_p;

float q;

float p;

float t;

float v;

unsigned long duration;

float f;

float RPM;

unsigned long time;

float a;

float d0, d1, d2, d3, d4, d5, d6, d7, d8, d9;

float r[9] = {1.12, 1.85, 6.63, 11.18, 18.97, 31.38, 47.92, 76.52, 149.02};

float power;

float power_1 = -20;

int n = 2;

int n_1 = 1;

int n_2 = 0;

int n_3 = -1;

int n_4 = 0;

float q_1 = 100;

float q_2 = 2000;

float q_3 = 30000;

float qratio;
```

```

float qratio_1;

float qratio_2;

int r_n;

int r_n_1;

int x;

float rx;

int k = 0;


void setup() {

  pinMode(A4,INPUT);

  pinMode(A5,INPUT);

  //pinMode(1, OUTPUT);

  pinMode(2, OUTPUT);

  pinMode(3, OUTPUT);

  pinMode(4, OUTPUT);

  pinMode(5, OUTPUT);

  pinMode(6, OUTPUT);

  pinMode(7, OUTPUT);

  pinMode(8, OUTPUT);

  pinMode(9, OUTPUT);

  Serial.begin(9600);

}

void loop() {

  digitalWrite(2, HIGH); //Turn off relay

  digitalWrite(3, HIGH); //Turn off relay

  digitalWrite(4, HIGH); //Turn off relay

  digitalWrite(5, HIGH); //Turn off relay

  digitalWrite(6, HIGH); //Turn off relay

```

```
digitalWrite(7, HIGH); //Turn off relay
digitalWrite(8, HIGH); //Turn off relay
digitalWrite(9, HIGH); //Turn off relay
digitalWrite(n, LOW); //Turn on relay
```

```
v_q = analogRead(A5);
q = (0.0891*v_q+9.5603);
v_p = analogRead(A4);
p = ((v_p*4)/1023);
```

```
r_n = (n - 1);
r_n_1 = (n_1 - 1);
```

```
d0 = pulseIn(0, HIGH);
d1 = pulseIn(0, HIGH);
d2 = pulseIn(0, HIGH);
d3 = pulseIn(0, HIGH);
d4 = pulseIn(0, HIGH);
d5 = pulseIn(0, HIGH);
d6 = pulseIn(0, HIGH);
d7 = pulseIn(0, HIGH);
d8 = pulseIn(0, HIGH);
d9 = pulseIn(0, HIGH);
duration = (d0+d1+d2+d3+d4+d5+d6+d7+d8+d9)/10;
time = millis();
t = (time/1000);
if (duration == 0)
{
```

```

f = 0;

RPM = 0;

v = 0;

power = 0;
}
else
{
f = (500000/duration);

RPM = (19.05*f+11.983);

if (f > 25.7)
{
v = (-0.0002*(pow(f, 2))+0.1555*f-3.8654);
}
else
{
v = 0;
}

a = (v/r[r_n]);

power = ((pow(v, 2))/r[r_n])+(2.623*a);
}

x = n;

rx = r[r_n];

qratio = abs((q-q_1)/q);

qratio_1 = abs((q_2-q)/q);

qratio_2 = abs((q-q_3)/q);

if (qratio <= 0.1 & qratio_1 <= 0.1 & qratio_2 <= 0.1 & n == n_2)
{

```

```

    k = 1;
}
else
{
    k = 0;
}

digitalWrite(n, HIGH);
if (power > power_1 & k != 1)
{
    if (n > n_1)
    {
        n_4 = n_3;
        n_3 = n_2;
        n_2 = n_1;
        n_1 = n;
        q_3 = q_2;
        q_2 = q_1;
        q_1 = q;
        n = n_1+1;    //OR n = ++n;
    }
    else
    {

        n_4 = n_3;
        n_3 = n_2;
        n_2 = n_1;
        n_1 = n;

```

```

q_3 = q_2;
q_2 = q_1;
q_1 = q;
n = n_1-1;    //OR n = --n;
}
}
if (power < power_1 & k != 1)
{
    if (n > n_1)
    {
        n_4 = n_3;
        n_3 = n_2;
        n_2 = n_1;
        n_1 = n;
        q_3 = q_2;
        q_2 = q_1;
        q_1 = q;
        n = n_1-1;    //OR n = --n;
    }
    else
    {
        n_4 = n_3;
        n_3 = n_2;
        n_2 = n_1;
        n_1 = n;
        q_3 = q_2;
        q_2 = q_1;
        q_1 = q;

```

```
n = n_1+1; //OR n = ++n;  
}  
}
```

```
power_1 = power;  
digitalWrite(n, LOW);  
Serial.print("Q = ");  
Serial.print(q);  
Serial.print(" l/min ");  
Serial.print(',');  
Serial.print("P = ");  
Serial.print(p);  
Serial.print(" bar ");  
Serial.print(',');  
Serial.print("N = ");  
Serial.print(RPM);  
Serial.print(" RPM ");  
Serial.print(',');  
Serial.print("V = ");  
Serial.print(v);  
Serial.print(" Volts ");  
Serial.print(',');  
Serial.print("P = ");  
Serial.print(power);  
Serial.print(" W ");  
Serial.print(',');  
Serial.print("R = ");  
Serial.print(rx);
```



```
Serial.print(" Ohm ");  
Serial.print(',');  
Serial.print("n = ");  
Serial.print(x);  
Serial.print(',');  
Serial.print("k = ");  
Serial.print(k);  
Serial.println(" / ");  
  
delay(12000);  
  
}
```

ANNEX 3: PUBLICATIONS

3.1. ESTIMATION OF UNIT COST AND MAXIMUM EFFICIENCY FOR PUMPS AS TURBINES

Estimation of unit cost and maximum efficiency for Pumps as Turbines

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Abstract

Energy recovery through Micro-Hydropower schemes installed along water supply systems has the potential to generate clean renewable electricity and lower the energy costs faced by utilities. Since in typical MHP projects the acquisition cost of turbine and generator represents a large share of the total project cost, the development of low-cost energy converters is a key step towards wider diffusion of such schemes. Pumps as Turbines (PATs) may constitute an effective and economic alternative to conventional water turbines for MHP projects, however the extent of their convenience over other turbine types and the maximum achievable efficiency for a given site is not clear. To address this, the cost per unit of nominal power of 280 radial end-suction PATs was calculated and compared with previously published cost figures. When compared to conventional Francis, Pelton or Kaplan turbines, generating sets relying on PAT technology showed a cost per nominal power reduced up to 15 times. Secondly, an equation was developed to estimate the maximum efficiency of a PAT connected with asynchronous generator as function of nominal flow rate and specific speed based on literature studies about hydraulic pumps.

Together, the two methods predicting the equipment cost and efficiency may enable the assessment of an MHP project adopting PAT technology at the design stage, with higher accuracy and convenience.

Keywords

Cost Analysis, Efficiency, Energy Recovery, Hydropower, Pump as Turbine (PAT)

Nomenclature

C: cost (€)
c: cost per unit of nominal power (€/kW)
H: hydraulic head (m)
N: rotational speed (RPM)
Ns: specific speed (-)
P: power (kW)
Q: flow rate (m³/s or l/s)

Greek symbols

η : efficiency (-)
 ω : rotational speed (rad/s)

Subscripts

p: pump
t: turbine

Abbreviations

BEP: Best Efficiency Point
CFD: Computational Fluid Dynamics
MHP: Micro-Hydropower
PAT: Pump as Turbine
VAT: Value Added Tax

INTRODUCTION

Micro-Hydropower (MHP) plants having a capacity lower than 100kW can be effectively implemented to exploit the existing potential along water supply systems producing renewable energy and lowering the energy costs faced by water utilities (Gaius-obaseki, 2010; McNabola et al., 2014; Wallace, 1996; Zakkour et al., 2002). The investment cost of such small-size hydropower plants is heavily influenced by the cost of the generating set, consisting of the turbine and generator units, which commonly account for a share between 30% and 40% of the total cost (Ogayar et al., 2009). However, it has been suggested that in MHP schemes having nominal capacity in the range of a few kW the turbine/generator assembly may account for up to 70% of the total project cost (Gallagher et al., 2015). Therefore, development of new types of hydro turbines characterized by lower purchase price may enable the exploitation of large portions of untapped hydro potential. An effective and low-cost technology for MHP are Pumps as Turbines (PAT), consisting of standard water pumps running in reverse mode with the connected induction motor acting as a generator (De Marchis et al., 2014; Gaius-obaseki, 2010; Giugni et al., 2009; Ramos et al., 2005, 2010).

Despite the general consensus on their reduced purchase price with respect to conventional water turbines, in literature there is a lack of agreement on PAT cost per unit of rated power with several authors proposing figures between 220 and 2,000 €/kW (Carravetta et al., 2013; De Marchis et al., 2014; Ramos et al., 2009). Besides, the extent of the cost advantage of PATs over Francis, Pelton or Cross-flow turbines is uncertain. For instance, some suggested that PATs are five times less expensive than the equivalent conventional hydro turbines (Alatorre-Frenk, 1994). While others pointed out that for $P > 40$ kW the price difference between these lowers below 35% (Ramos et al., 2009).

Secondly, data in the literature shows that the maximum efficiency of PATs may range between values as low as 40% up to nearly 90%, but no practical mathematical tool or chart exists to readily estimate the rated efficiency of a PAT under given boundary conditions.

Research Target

In order to assess the financial and technical viability of a hydro project adopting PAT technology it is fundamental to have access to realistic estimates of both equipment cost and turbine rated efficiency. Therefore, the aims of the present research are to:

1. provide an estimate of PAT+generator unit purchase price and compare this with the prices of conventional hydropower generating sets as well as previously published PAT cost estimates;
2. develop a mathematical function estimating the maximum efficiency of generating sets adopting centrifugal PAT technology.

ESTIMATION OF PAT AND GENERATOR UNIT COST

PAT + generator unit cost

The estimation of the cost per kW of centrifugal PATs coupled with induction generators having 2, 4 or 6 magnetic poles has been carried out based on commercially available data in 2016. In particular, the cost (excluding VAT) of 280 radial end-suction pumps coupled to IE3 class generators has been related to their predicted power output when operated as turbines at their Best Efficiency Point (BEP).

However, one of the major limitations of the application of PATs is that manufacturers do not commonly provide the characteristic curves of pumps running in reverse mode, thus their BEP must

be estimated. Several methods exist to characterize the BEP of a PAT: full laboratory tests, Computational Fluid Dynamics (CFD) analysis, and mathematical semi-empirical correlations. Since laboratory testing campaigns are time and resource consuming and CFD data generally needs empirical confirmation, the most straightforward approach is to rely on non-dimensional 1D coefficients relating the head and flow rate values at the BEP of a PAT with those of the same machine operating as pump. Even if a large number of methods can be found in literature to determine such coefficients from one or more machine parameters (most often N_s and/or η_{BEP}), it has been shown that none of them can give an accurate estimate of BEP location for any pump (Singh, 2005; Lydon et al., 2017). One of the latest methods available in literature has been shown to provide higher accuracy with respect to previously published ones - namely the Sharma and Stepanoff method considering a large database of PATs (Yang et al., 2012):

$$h = \frac{H_{t,BEP}}{H_{p,BEP}} = \frac{1.2}{(\eta_{p,BEP})^{1.1}} \quad (1)$$

$$q = \frac{Q_{t,BEP}}{Q_{p,BEP}} = \frac{1.2}{(\eta_{p,BEP})^{0.55}} \quad (2)$$

Equations 1-2 have been used within the present research to estimate the values of $H_{t,BEP}$ and $Q_{t,BEP}$ of the chosen set of pumps when running in turbine mode. Even though such 1-D mathematical methods may occasionally result in gross under/overestimation of the BEP location for a determined machine, it was supposed that their application to a large number of machines (280) should dampen such scattered errors. The nominal power of PATs was subsequently calculated as:

$$P \text{ (kW)} = 9.8 \cdot \eta_t \cdot Q_{t,BEP} \left(\frac{m^3}{s} \right) \cdot H_{t,BEP} \text{ (m)} \quad (3)$$

where η_t stands for the maximum efficiency of a PAT and was considered equal to the maximum efficiency of the same machine operated as a pump, in agreement with the available literature (Derakhshan et al., 2008; Garay, 1990). The PAT + generator unit cost was then computed as:

$$c \left(\frac{\text{€}}{\text{kW}} \right) = \frac{C_{PAT+gen}(\text{€})}{P \text{ (kW)}} \quad (4)$$

Comparison of PAT + generator unit cost

The resulting figures of PAT specific cost have been compared with the cost per rated power of conventional hydro turbines - Francis, Pelton and Kaplan - as compiled by Ogayar et al. (2009). The cost figures proposed by Ogayar et al. (2009) have been updated to account for the inflation which has occurred in Spain between 2008 (year of original data collection) and the end of 2016. Also, the specific cost of PAT generating sets has been compared with other previously published cost figures. The resulting information is displayed in Figure 1.

The PAT and generator assembly unit cost decreases with the increase of nominal power output, showing realistic economies of scale. However, the previously published cost estimates can be considered accurate only within narrow intervals of nominal power. Generating sets relying on PAT technology proved to be less expensive per unit of installed power than conventional hydraulic turbines by a factor of 5 for $P > 20\text{kW}$ and up to 15 for smaller power outputs.

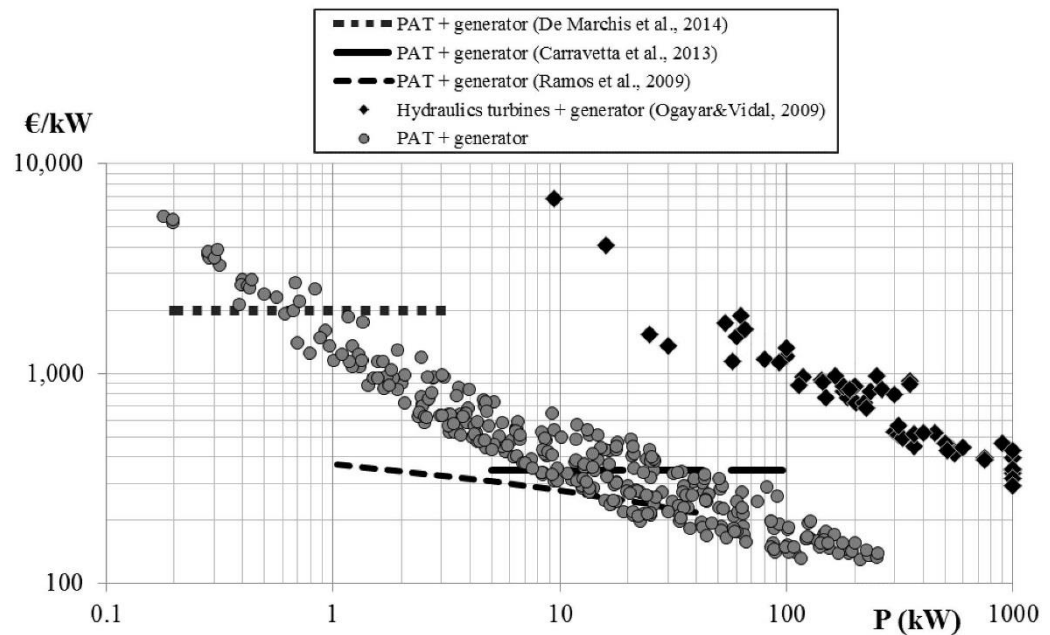


Figure 1. Unit cost per rated power of the evaluated PATs coupled to IE3 asynchronous generators, as compared with previously published PAT cost figures and unit cost of conventional hydro turbine + generator units.

ESTIMATION OF PAT MAXIMUM EFFICIENCY

Based on the previously cited database of 280 pumps coupled with asynchronous generators and on the characteristics of 59 experimentally tested PATs found in literature (Alatorre-Frenk, 1994; Derakhshan et al., 2008; Singh, 2005) a 3D function has been developed relating the maximum efficiency of a PAT + generator unit to its nominal flow rate and specific speed. All PATs have been considered as coupled to IE3 efficiency class asynchronous generators whose efficiency requirements are shown in Figure 2.

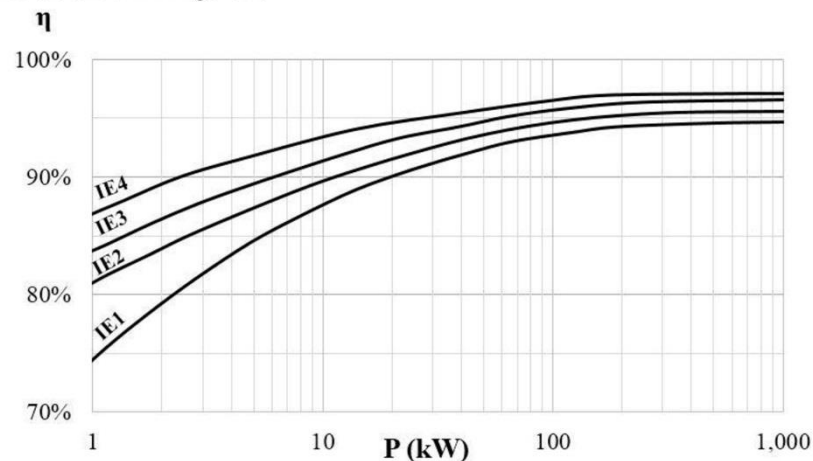


Figure 2. Performance of asynchronous motors according to rated power and efficiency class (ABB, 2014)

The dependence of maximum efficiency of pumps on specific speed and flow rate has been widely acknowledged, and often represented on 2D charts (Chapallaz et al., 1992; Łazarkiewicz et al., 1965). Another author proposed a mathematical correlation to estimate the rated efficiency of a hydraulic pump (Anderson, 1977):

$$\eta_{MAX} = a - \frac{b}{Q_{BEP}^c} - d \left(e + \ln \frac{N_s}{52.933} \right)^2 \quad (5)$$

The term N_s refers to the specific speed defined as:

$$N_s = N \frac{\sqrt[3]{Q}}{H^{0.75}} \quad (6)$$

where N (RPM) is the shaft speed, Q (m³/s) and H (m) the flow rate and head respectively. The value 52.933 in Equation (5) corresponds to the conversion factor between N_s and the so-called “non-dimensional specific speed” Ω defined as:

$$\Omega = \omega \frac{\sqrt[3]{Q}}{(gH)^{0.75}} \quad (7)$$

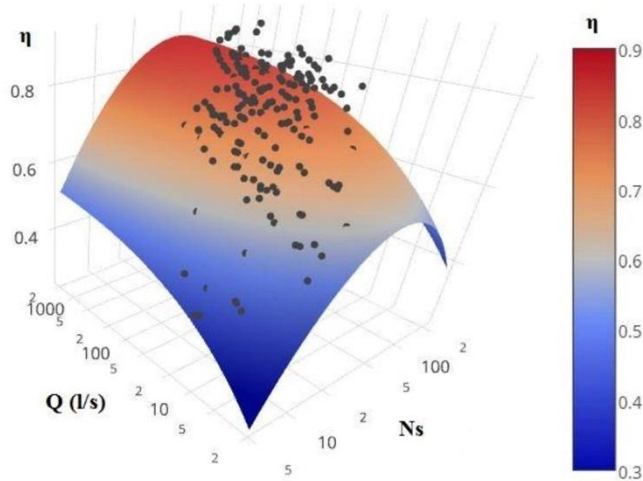
where ω (rad/s) is the shaft speed and g (m/s²) the gravitational acceleration.

While Anderson suggested coefficients for Equation 5 adequate for pump performance prediction, new coefficients have been calculated here for PATs coupled to IE3-class induction generators by means of least squares method as reported in Table 1.

Table 1. Coefficients for Equation (5)

	a	b	c	d	e
Pumps (Anderson, 1977)	0.94	0.048	0.32	0.055	0.18
PAT + IE3 generator	0.89	0.024	0.41	0.076	0.22

The resulting equation for PAT + generator units is shown in Figure 3, while a 2-D contour chart similar to the one proposed for pumps by (Chapallaz et al., 1992) is presented in Figure 4.



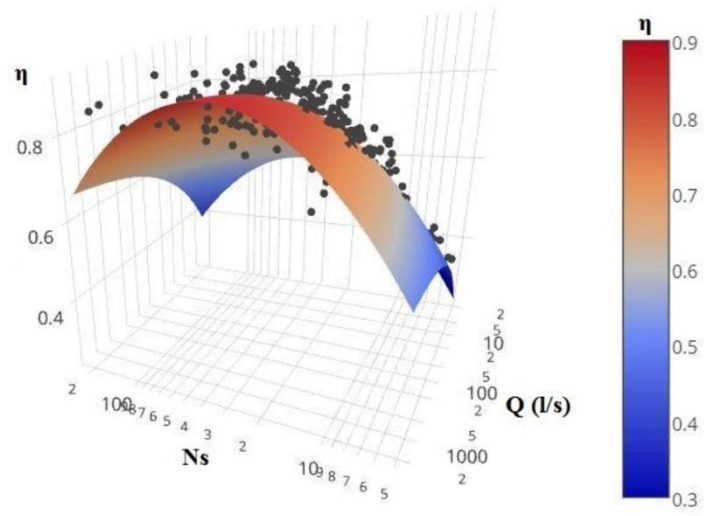


Figure 3. Proposed 3D function for PAT efficiency estimation plotted against the actual efficiency of selected machines

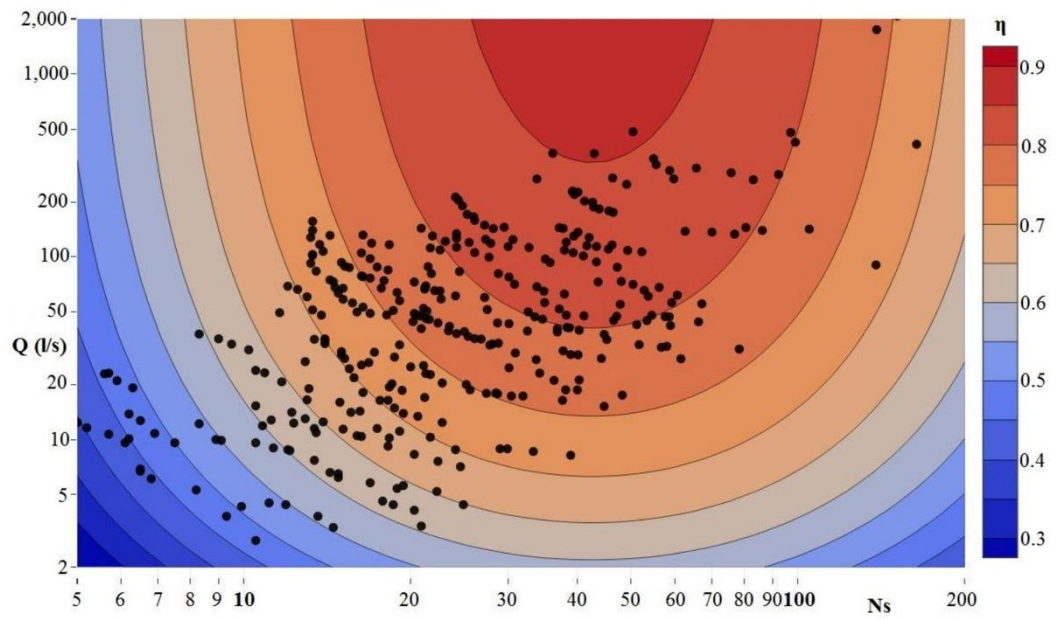


Figure 4. 2D contours of the proposed function for PAT efficiency estimation plotted against the nominal flow rate and specific speed of selected machines

Finally, Figure 5 shows the distribution of errors associated with the proposed function with respect to the actual efficiency of the analysed PAT + generator units. The mean error corresponded to 5.2%, with 86% of samples falling in the $\pm 10\%$ error interval and only four units showing a deviation from estimated efficiency higher than 20%.

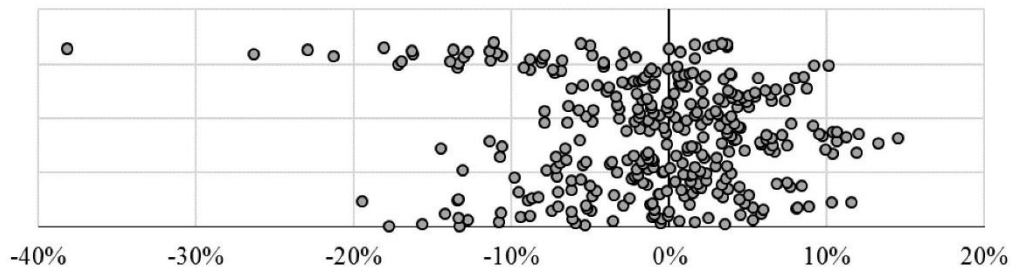


Figure 5. Percentage of error on PAT η estimation through the proposed equation

CONCLUSIONS

As for cost considerations, the results shown in Figure 1 prove that previously published PAT + generator cost figures can be accurate only in a fraction of the whole application range and the unit cost may vary from 130 to 5,600 €/kW according to the machine rated power. In all cases, PAT + generator unit cost proved to be between 5 and 15 times lower than generating sets based on conventional hydro turbines.

Conversely, the turbine peak efficiency may vary greatly according to PAT specific speed and nominal flow rate as seen in Figures 3-4 which explains the wide range of values found in literature. A function was developed to predict the efficiency of a PAT generating set given its nominal flow rate and specific speed.

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CONFLICT OF INTEREST

The authors declare no conflict of interest.

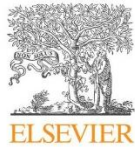
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3.2. A MODEL FOR THE EXTRAPOLATION OF THE CHARACTERISTIC CURVES OF PUMPS AS TURBINES FROM A DATUM BEST EFFICIENCY POINT

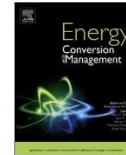
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A model for the extrapolation of the characteristic curves of Pumps as Turbines from a datum Best Efficiency Point

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ABSTRACT

Pumps used as Turbines (PATs) are a viable and low-cost technology of energy converters suitable for small-scale and in-pipe energy generation from water resources. However, amongst the main barriers to PAT technology diffusion is the unavailability of the characteristic curves of most hydraulic pumps on the market when used as turbines. Several methods exist in the literature to derive the shape of head and power characteristic curves of a PAT knowing its Best Efficiency Point (BEP). Such methods based on fixed-coefficient polynomials have been assessed here in a cross-comparison and a new method was developed based on a database of 113 experimentally tested PAT curves, which proved to reproduce the behaviour of the sampled machines more accurately by improving the overall goodness of fit up to 60%, 37% and 5% according to the different selected indicators. Finally, the mechanical efficiency and system efficiency of a PAT under variable flow rates has been compared and contrasted with that of conventional hydro turbines.

1. Introduction

1.1. Generalities and applications

Pumps As Turbines (PATs) consist of standard water pumps utilized as turbines by inverting the direction of the flow across them [1]. The hydraulic behaviour of a PAT is similar to that of a Francis turbine, but in comparison a PAT usually features an impeller of larger diameter and backwards-swept blades [2]. Among the many commercially available pump types which can be used as turbines are the centrifugal, mixed flows or axial units as well as multistage and double-flow pumps [2,3].

Since hydraulic pumps are mass produced and easily available off-the-shelf in a variety of sizes and materials in most countries worldwide, their use as turbines allows a significant cost reduction with respect to conventional hydro turbines [4]. Other advantages include their compact dimensions, easy maintenance and long life span [3,5,6]. Being robust, easily sourced and reliable machines, PATs have been indicated as ideal hydraulic converters for rural electrification projects in developing countries [7]. Besides, their wide application range and their capability to work with a downstream residual pressure makes them the ideal device for in-pipe energy recovery in existing water infrastructures [3,8–11].

1.2. PAT characteristic curves prediction

A major barrier to PAT implementation on a wide scale is the unavailability of the characteristic curves of pumps when operated as turbines since most manufacturers do not provide clients with such information. The most error-free method available to verify the performance of pumps in reverse running mode is their physical testing in a dedicated lab facility, which is an extremely resource-demanding process. Alternatively, some authors have proposed the use of Computational Fluid Dynamics (CFD) which in turn is prone to errors, requires considerable computational resources and needs an extensive knowledge of the internal geometry of each unit [12].

Therefore, researchers and practitioners need to cope with the lack of knowledge involving the vast majority of the off-the-shelf available PATs through numerical models based on empiric data which are capable of estimating with reasonable accuracy the performance of any unit under turbine operations based on its known pump behaviour. For instance, several mathematical correlations have been proposed to correlate the location of the Best Efficiency Point (BEP) of a unit between pump and turbine mode, which is defined as the coordinates of head H (m) and flow rate Q (m^3/s) under which the machine works at maximum efficiency [12–14]. However, since most applications of PATs for in-pipe energy recovery require the units to accommodate large variations of flow and head [15–16] it is of vital importance to

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Nomenclature		Ψ	head coefficient (–)
H	hydraulic head (m)	<i>Subscripts</i>	
N	rotational speed (RPM)	rel	relative
n	rotational speed (rad/s)	<i>Abbreviations</i>	
Ns	specific speed (–)	BEP	Best Efficiency Point
P	power (kW)	PAT	Pump as Turbine
Q	flow rate (m ³ /s)	3D	three-dimensional
R ²	coefficient of determination	CFD	computational fluid dynamics
<i>Greek symbols</i>			
η	efficiency (–)		

derive not only the location of the BEP in turbine mode but also the complete characteristic curve of a unit in order to predict its performances within its whole application range. This becomes very relevant when considering that the absence of in-built regulation devices in PATs commonly leads to poor part-load performances when compared to classic hydro turbines.

A key parameter for assessing the performance of a PAT is the specific speed N_s defined as:

$$N_s = N((\sqrt{Q_{BEP}})/(H_{BEP}^{0.75})) \quad (1)$$

where N represents the shaft speed (RPM) and Q (m³/s) and H (m) respectively the nominal flow rate and head across the turbine. Chapallaz et al. [5] suggested that the complete Q-H characteristic curves of PATs having different N_s tend to have variable shapes according to the respective value of N_s as shown in Fig. 1. However, they did not propose any mathematical correlation to approximate those curves. It was assumed that a properly designed PAT scheme would operate exclusively at the machine's BEP, and therefore knowing the full shape of the characteristic curves was unimportant. Such an assumption may arguably be valid in the context of traditional hydro-power plants associated with a reservoir. However this is not true in cases in which a PAT would need to cope with recurring variations in available flow rate and pressure head (e.g. in water-pipe networks).

Acknowledging the importance of estimating the full head and efficiency characteristic curves within the whole PAT range, Derakhshan and Nourbakhsh [17] suggested to approximate the relative head and power curves over the processed flow rate for any given machine using a second and third order polynomial respectively. The authors then suggested a set of empirical coefficients based on experimental testing of eleven PATs having N_s in pump mode in the range 14.6–55.6 [17], resulting in:

$$H_t/H_{t,BEP} = 1.0283(Q_t/Q_{t,BEP})^2 - 0.5468(Q_t/Q_{t,BEP}) + 0.5314 \quad (2)$$

$$P_t/P_{t,BEP} = -0.3092(Q_t/Q_{t,BEP})^3 + 2.1472(Q_t/Q_{t,BEP})^2 - 0.8865(Q_t/Q_{t,BEP}) + 0.0452 \quad (3)$$

Pugliese et al. [18] review the above equations based on laboratory testing of two radial PATs, one of which was a multistage, having N_s of 28.7 and 40.7. The authors confirmed the validity of Eq. (2) relative to the head curve, but instead suggested that Eq. (3) shall be considered valid only for values of non-dimensional flow rates lower than 0.4 [18]. Such non-dimensional flow rate is defined as:

$$\Phi = Q/(nD^3) \quad (4)$$

where Q (m³/s) is the nominal flow rate, n the shaft speed (rad/s) and D (m) the impeller diameter. The updated third-degree polynomial approximating the power curves of PATs proposed as a replacement of Eq. (3) is [18]:

$$P_t/P_{t,BEP} = 4.000 \cdot 10^{-3}(Q_t/Q_{t,BEP})^3 + 1.386(Q_t/Q_{t,BEP})^2 - 0.390((Q_t/Q_{t,BEP})) \quad (5)$$

Within the same paper the authors pointed out that the accuracy of such equations was questionable when dealing with multistage units, since the combined use of Eqs. (2) and (5) “under-estimated the experimental results by around 20–30%” regarding the tested two-stage pump. Subsequently, they proposed further experiments on multistage units to be performed in order to evaluate whether dedicated equations are needed or rather the measured inaccuracies have no statistical relevance in the context of larger experimental data [18].

Barbarelli et al. [19] also suggested alternative coefficients for head and power polynomials based on the formulation proposed by Derakhshan and Nourbakhsh [17]. Based on experimental data from 27

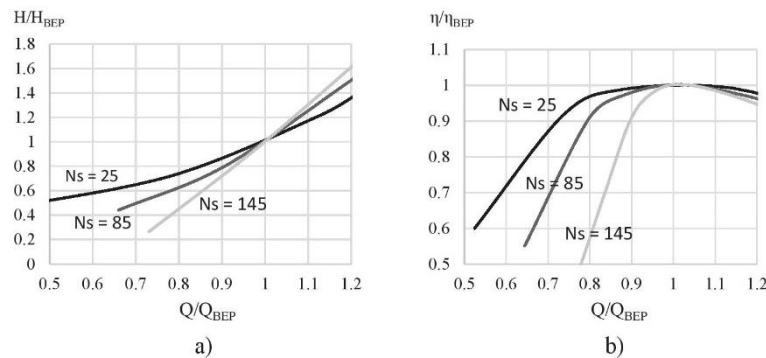


Fig. 1. Comparison of three PATs having different N_s : (a) relative head versus relative flow rate, (b) relative efficiency versus relative flow rate. Adapted from: [5]

pumps tested as turbines the authors generated coefficients fitting the relative head and power curves for each machine and subsequently calculated their mean values to replace the original coefficients of Eqs. (2) and (3), resulting into:

$$H_t/H_{t,BEP} = 0.922(Q_t/Q_{t,BEP})^2 - 0.406(Q_t/Q_{t,BEP}) + 0.483 \quad (6)$$

$$P_t/P_{t,BEP} = 0.040(Q_t/Q_{t,BEP})^3 + 1.185(Q_t/Q_{t,BEP})^2 - 0.043(Q_t/Q_{t,BEP}) - 0.183 \quad (7)$$

Finally, Fecarotta et al. [20] proposed the approximation of the head and power curves of a PAT by means of two second-degree polynomials. The coefficients of such equations have been derived by the experimental characterization of five PATs of different type: centrifugal, semi-axial and submersible. The resulting equations are [20]:

$$H_t/H_{t,BEP} = 1.61(Q_t/Q_{t,BEP})^2 - 1.41(Q_t/Q_{t,BEP}) + 0.805 \quad (8)$$

$$P_t/P_{t,BEP} = 1.85(Q_t/Q_{t,BEP})^2 - 0.858(Q_t/Q_{t,BEP}) - 0.00567 \quad (9)$$

1.3. Research targets

Given the unavailability of reasonable alternatives, researchers and practitioners who deal with PATs are in need of a set of mathematical correlations enabling them to select the most appropriate equipment based solely on the known machine characteristics in pump mode. Several methods exist to predict the values of flow rate and head across a PAT leading to maximum efficiency (i.e. the Best Efficiency Point). However, in most cases this information is not sufficient since practically most PATs will need to work under variable flow and head conditions. Therefore, as a further step it is necessary to develop a numerical model based on empiric data to predict the full shape of the characteristic curves of any pump in turbine mode once that its BEP has been calculated.

Four such mathematical methods for PAT curves prediction have been proposed in the literature, but their accuracy is limited by the reduced sample of empirical PAT curves upon which the underlying calculations were based on. Besides, the use of fixed-coefficients polynomials for each of the four models does not capture the real behaviour of PATs having different N_s as shown in Fig. 1.

The scope of the present research is to derive a new set of polynomials with variable coefficients for the extrapolation of the full characteristic curves of PATs based on a large dataset of experimental PAT performance curves (113). The new model obtained will be compared to the previously existing ones in order to assess its accuracy. Secondly, thanks to the existing PAT curves database the part-load efficiency of PATs will be compared to that of conventional hydro turbines.

2. Method and calculations

The characteristic curves of 113 PATs of different sizes and from

different manufacturers, tested in turbine mode have been compiled [2,17,21–25]. The sample consisted of radial end-suction, semi-axial and axial PATs. As a means of further characterising the data of 113 PATs, the relationship between head coefficient (psi) and specific speed of a subset of them is shown in Fig. A1 of the annexed Appendix. For each machine the experimental characteristic head and power curves were approximated as best fits of two second degree polynomials as shown in Eqs. (10) and (11). The relative power curves were treated as second degree polynomials as suggested by Fecarotta et al. [20] rather than third degree as proposed by other authors since in both cases the resulting R^2 coefficient was found to be higher than 95%. Therefore, such choice allowed the reduction of the number of variable coefficients from a total of seven to six without an impact on the model accuracy.

$$H_t/H_{t,BEP} = a(Q_t/Q_{t,BEP})^2 + b(Q_t/Q_{t,BEP}) + c \quad (10)$$

$$P_t/P_{t,BEP} = d(Q_t/Q_{t,BEP})^2 + e(Q_t/Q_{t,BEP}) + f \quad (11)$$

For each machine, the above mentioned coefficients have been found by means of least squares linear regression from the tabulated points of characteristic curve. The non-dimensional Flow-Head (Q-H) and Flow-Power (Q-P) characteristic curves of all the considered PATs are shown respectively in Figs. 2 and 3.

Looking at Figs. 2 and 3 it can be noticed that there is a significant dispersion of the sample curves away from the BEP, which corresponds to the point of coordinates (1,1). Hence, approximating the head and flow curves of a PAT with fixed polynomials may lead to errors becoming more and more significant in the regions away from the BEP. For each individual non-dimensional head and power curve of Figs. 2 and 3 the derivatives at the BEP have been calculated through the same coefficients previously introduced for Eqs. (10) and (11):

$$dH_{rel}/dQ_{rel} = 2a + b \quad (12)$$

$$dP_{rel}/dQ_{rel} = 2d + e \quad (13)$$

As a further step, it has been investigated whether the steepness of the non-dimensional head and power curves of the 113 analyzed PATs increased on increasing the value of N_s as suggested by Chapallaz et al. [5]. In Figs. 4 and 5 the derivatives of the head and power relative curves at the BEP for each PAT calculated according to Eqs. (12) and (13) have been plotted against the N_s . The values of derivatives at the BEP of the non-dimensional PAT curves with fixed coefficients proposed by other authors and presented as Eqs. (2), (3) and (5)–(9) have also been included in Figs. 6 and 7 for comparison.

From the analysis of Fig. 4 and Fig. 5 it can be concluded that despite some dispersion of the samples a significant linear trend exists which correlates high values of N_s with a greater steepness of the non-dimensional head and power curves of a PAT. This behaviour in turn cannot be captured by the four fixed-coefficient polynomials existing in the literature and presented in the Introduction, thus a new set of coefficients for Eqs. (10) and (11) based on the linear trend line displayed in Figs. 4 and 5 are proposed as shown in Table 1. The values for

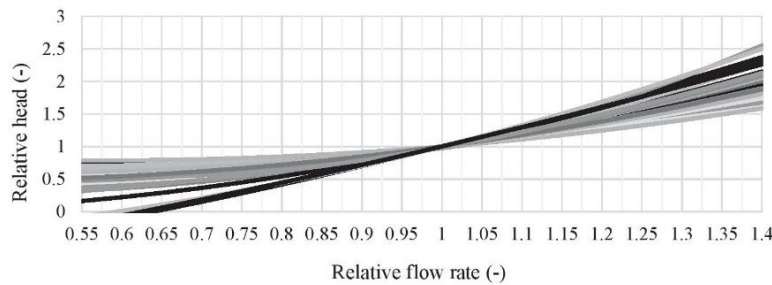


Fig. 2. Relative flow and head characteristic curves of all the 113 analyzed PATs.

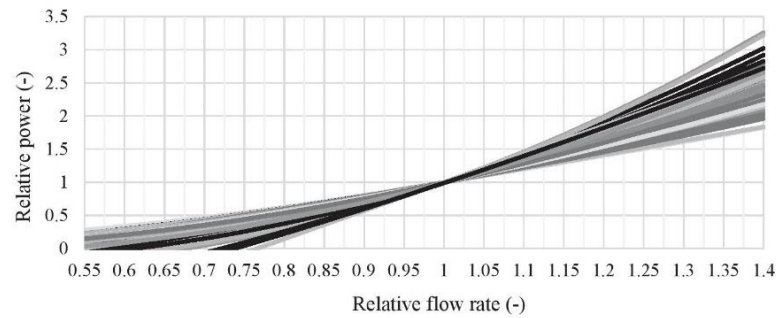


Fig. 3. Relative flow and power characteristic curves of all the 113 analyzed PATs.

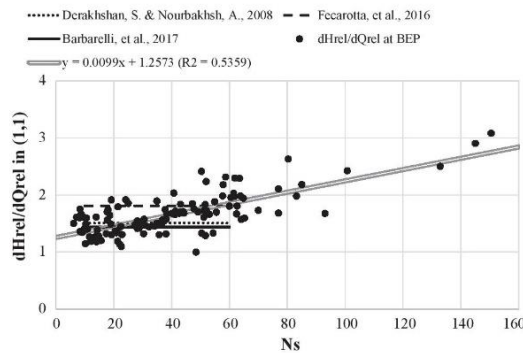


Fig. 4. Derivatives at the BEP (1,1) of the non-dimensional head curves of all 113 sample PATs.

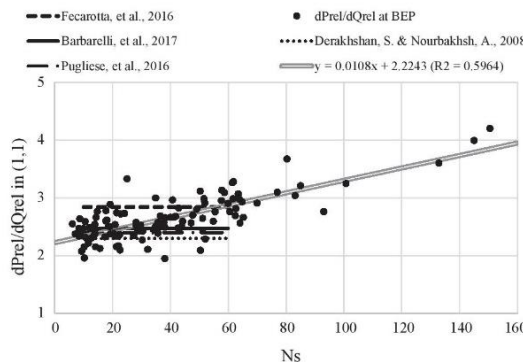


Fig. 5. Derivatives at the BEP (1,1) of the non-dimensional power curves of all 113 sample PATs.

the coefficient “a” and “d” in Table 1 have been calculated as mean values of the 113 derived PAT curves.

3. Model accuracy verification

The accuracy of the proposed model for the extrapolation of PAT full characteristic curves knowing the BEP has been assessed against the PAT database and compared to other methods described in the literature. Three widely accepted indicators showing the goodness of the fit were adopted as benchmark: the Sum of Squares Error (SSE), the coefficient of determination R^2 and the Root Mean Square Error

(RMSE). The SSE is the most basic indicator among the possible ones, R^2 is conveniently scaled between 0 and 1 and therefore easier to interpret while the RMSE is not scaled to any particular values and gives a more explicit measure of how much our predictions deviate, on average, from the actual values in the dataset. The results for the non-dimensional head curve extrapolation are displayed in Table 2, while those for the non-dimensional power curve are shown in Table 3.

The results from both Tables 2 and 3 indicate that the proposed method, which considers the coefficients of the non-dimensional curves as depending from N_s , performs consistently better than all other methods available in the literature which adopt fixed coefficients. In particular, it enabled a reduction of the SSE indicator by over 60% and 55% with respect to the worst performing model regarding respectively the relative head and power curve prediction. The RMSE indicator accordingly saw its value reduced with respect to the adoption of the four pre-existing models between a 24% and a 37%. Lastly, the overall R^2 coefficient was raised to over 97% from values in the range 92–93% regarding the relative head curve prediction and from a range of 94–95% as for the relative power curve prediction as displayed in Tables 2 and 3. The graph of fittings of the proposed numerical model from which results from Tables 2 and 3 were derived from the 113 analysed PATs can be seen in Figs. 6 and 7.

4. Comparison of PAT efficiency at partial load

Using the same dataset of performance curves from 113 machines the mean mechanical efficiency of PATs under variable flow rates has been assessed and compared to that of conventional hydro turbines. For each PAT the relative efficiency was calculated as ratio between the efficiency at a certain relative flow rate and the value of peak mechanical efficiency of the turbine defined as:

$$\eta = P/(9.81QH) \quad (14)$$

where P is the output power in kW, Q the flow rate in m^3/s and H the head in m across the PAT.

Besides, the BEP of a PAT was fixed to the 75% of the maximum admissible flow rate through it which is consistent with the behaviour of the gathered PAT curves. The relative PAT efficiency with varying flow rate compared to that of other classes of hydro turbines is shown in Fig. 8.

The results of Fig. 8 indicate that the relative mechanical efficiency of a PAT is comparable to that of a Francis turbine while at the same time Pelton, Crossflow or Kaplan turbines indicatively present a more flat efficiency curve over the whole operating range. However, such a comparison is not entirely fair because as opposed to a conventional turbine a PAT does not have an in-built regulation device but rather features a rigid volute and impeller geometry. This means that when a higher flow rate than the BEP is available in the system a portion of it must be bypassed, while if less flow rate than the BEP is available a

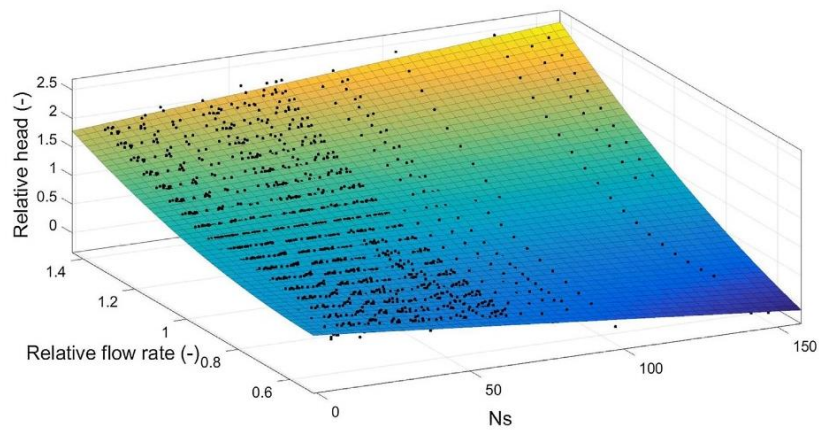


Fig. 6. Relative flow and head characteristic curves according to the N_s of all the 113 analyzed PATs, displayed together with the proposed interpolating 3D function.

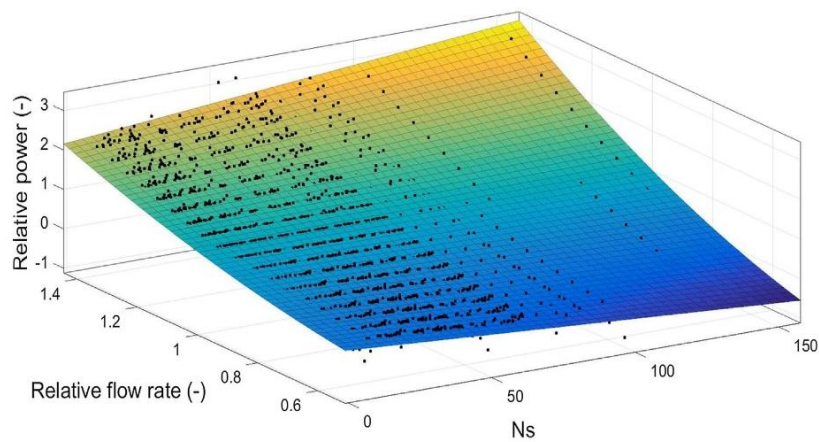


Fig. 7. Relative flow and power characteristic curves according to the N_s of all the 113 analyzed PATs, displayed together with the proposed interpolating 3D function.

Table 1
Tabulated coefficients for Eqs. (10) and (11) depending on N_s .

Non-dimensional head curve		Non-dimensional power curve	
a	1.160	d	1.248
b	$(0.0099 N_s + 1.2573) - 2a$	e	$(0.0108 N_s + 2.2243) - 2d$
c	$1 - a - b$	f	$1 - d - e$

Table 2
Goodness of fit of the proposed correlation compared to other proposed methods for the approximation of the relative head curve of a PAT.

Parameter	Proposed correlation	Derakhshan and Nourbakhsh [17]	Fecarotta et al. [20]	Barbarelli et al. [19]
SSE	11.76	25.87	27.75	30.06
R^2	0.9701	0.9343	0.9295	0.9236
RMSE	0.0751	0.1113	0.1153	0.1200

portion of the fluid pressure must be dissipated by an in-series control valve. Ultimately this phenomenon reduces the energy output from a PAT system. This can be evaluated by introducing the concept of system efficiency as the ratio of the power output from a turbine to the

Table 3
Goodness of fit of the proposed correlation compared to other proposed methods for the approximation of the relative power curve of a PAT.

Parameter	Proposed correlation	Derakhshan and Nourbakhsh [17]	Pugliese et al. [18]	Fecarotta et al. [20]	Barbarelli et al. [19]
SSE	22.88	46.39	51.91	39.72	45.82
R^2	0.9776	0.9545	0.9491	0.9611	0.9551
RMSE	0.1057	0.1505	0.1592	0.1393	0.1496

available system power:

$$\text{System efficiency (-)} = P / (9.81 Q_{av} H_{av}) \quad (15)$$

where P is the output power from the turbine in kW, Q_{av} the flow rate in m^3/s and H the head in m available in the system. Fig. 9 shows the indicative variation of system efficiency of different turbine technologies under different flow rates. The head curve of the system was considered constant, while representative values of turbine efficiency have been selected as 90% for Kaplan and Pelton units, 82% for Crossflows, 90% for Francis machines and 75% for PATs.

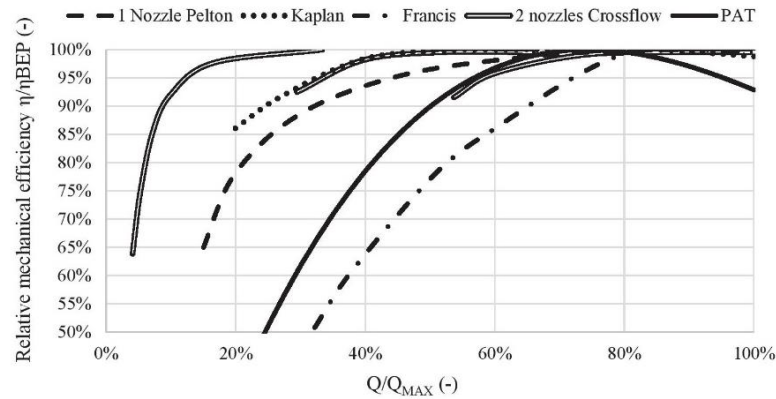


Fig. 8. Relative mechanical efficiency vs. relative flow rate of PATs and other types of conventional hydro turbines. Adapted from: [26]

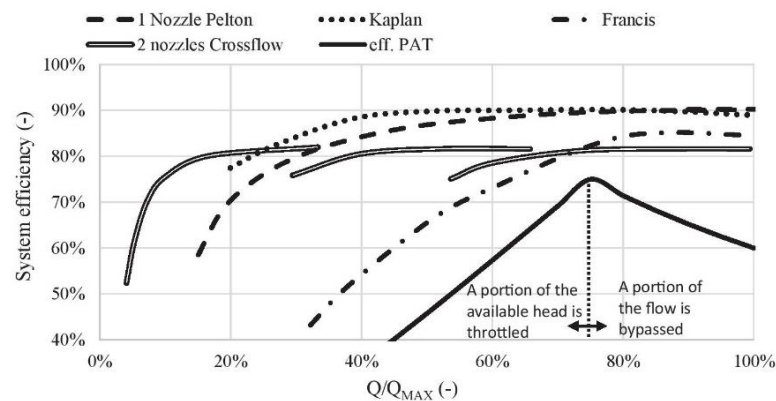


Fig. 9. Global system efficiency vs. relative flow rate of PATs and other types of conventional hydro turbines. Adapted from: [26]

From Fig. 9 it can be concluded that in terms of system efficiency a PAT has a clear peak at the BEP around which its effectiveness declines steeply due to its inability to adapt to different flow and head conditions.

5. Conclusions

The extrapolation of full head and power characteristic curves of pumps as turbines is a fundamental step in order to correctly select the most effective machine working under variable load conditions for a determined site. Four existing methods in the literature based on fixed-coefficients and second and third degree polynomials have been evaluated against a large dataset of 113 PAT characteristic curves. Since the steepness of the head and power non-dimensional curves gathered in the database has been demonstrated to vary according to the specific speed of each machine, a new method for the full characteristic curves extrapolation was proposed in which the coefficients of two second-degree polynomials are variable as functions of N_s . This proposed method was found to be statistically more accurate in reproducing the 113 PAT curves of the dataset with respect to the other four methods available in the literature. In particular, the accuracy regarding the prediction of relative head curves has been improved up to 60%, 37% and 5% when considering the SSE, RMSE and R^2 indicators respectively. Due to the limited number of PAT characteristic curves in the

database with N_s higher than 100 the authors recommend caution with adopting the proposed method for head and power relative curves extrapolation in the mentioned range.

Secondly, based on the same database the average mechanical efficiency of PATs with varying flow rate has been shown to compare favourably to the behaviour of other classes of conventional hydro turbines (Francis). However, when considering the global system efficiency the average performance of PATs are inferior to that of other turbine classes with a clearly identifiable peak preceded and followed by a steep decline in efficiency. This suggests that the process of selecting a hydraulic pump to work as a turbine for a given site is a highly critical operation, since even a small assessment mistake could have significant repercussions on the system effectiveness.

As a scope for future research it is suggested to experimentally test PATs having N_s in the range 60–150 and to integrate their derived characteristic curves into the database utilized for the present calculations. Indeed, the vast majority of PATs whose performance curves are known to the authors present a N_s lower than 60.

Acknowledgements

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Conflict of interest

The authors declare no conflict of interest.

Appendix A

As a further characterisation of the available PAT performance curve database, the non-dimensional head coefficient has been introduced defined as:

$$\Psi = gH/(D\omega)^2 \quad (16)$$

where g is the gravitational constant (9.81 m/s^2), H the head in m , D the impeller outer diameter and ω the shaft speed of the PAT in rad/s . Among the 113 investigated PATs, the outer impeller diameter of 58 of them was known to the authors. Therefore it was possible to calculate the head coefficient parameter (Ψ) only for those. In Fig. A1 the values of Ψ versus N_s for the subset of 58 PATs are displayed, showing a significant correlation between the two. However, due to the existence of this correlation the inclusion of Ψ in the equations for the determination of the relative power and head curves shown in Table 1 would be redundant given the presence of N_s already.

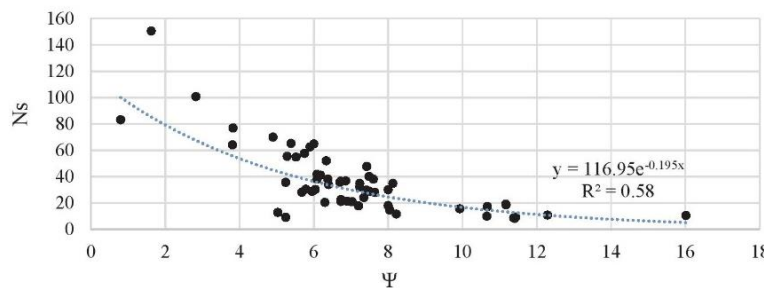


Fig. A1. Correlation between the head parameter (Ψ) and N_s for a subset of the available PAT database.

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3.3. COST MODEL FOR PUMPS AS TURBINES IN RUN-OF-RIVER AND IN-PIPE MICROHYDROPOWER APPLICATIONS

Cost Model for Pumps as Turbines in Run-of-River and In-Pipe Microhydropower Applications

D. Novara¹; A. Carravetta²; A. McNabola³; and H. M. Ramos⁴

Abstract: Pumps as Turbines (PATs) are standard water pumps utilized as hydraulic turbines by reversing the flow direction across them. The off-the-shelf availability of water pumps and their reduced purchase price with respect to conventional hydro turbines makes them an ideal technology for exploiting large portions of uncapped hydro potential that is technically viable but not financially convenient. Such low-cost technology could help to expand hydropower exploitation in water resources worldwide, helping to reduce climate change greenhouse gas emissions. However, in the literature, the few available cost figures relative to PAT purchase price are discordant and often outdated, and such a lack of information is likely a severe barrier to a more widespread PAT implementation. In order to overcome this limitation, data from 343 pumps and 286 generators were compiled to show graphically and analytically how the purchase price of PATs varies over different nominal powers and hydraulic conditions. Besides, a set of equations was developed to allow designers to predict the PAT and generator cost from the nominal flow rate and available hydraulic head. DOI: 10.1061/(ASCE)WR.1943-5452.0001063. © 2019 American Society of Civil Engineers.

Author keywords: Pump as Turbine (PAT); Cost analysis; Energy recovery; Microhydropower; PAT cost.

Introduction

Hydropower applications can have significant positive implications for the management of natural and human-made water resource systems. In its *Special Report on Renewable Energy Sources and Climate Change Mitigation*, the Intergovernmental Panel on Climate Change highlighted the potential of the use of hydropower to contribute toward the reduction of CO₂ emissions in the coming years; this report also highlighted the main challenges facing current hydropower technology and its applications (IPCC 2012). Potential for further development and climate change mitigation is highlighted in the opportunities that exist for multipurpose hydropower (e.g., turbines in water supply pipe networks or in existing dams without a turbine). Challenges facing the technology and its applications include the lack of low-cost alternatives to conventional large hydropower machines. A lack of reliable information on the cost of hydropower in the literature was also highlighted as a barrier to future development (IPCC 2012).

Pumps as Turbines (PATs) are standard water pumps operating in reverse mode as hydraulic turbines, which are therefore able to exploit the pressure held by a stream of water and generate electricity. Water pumps are commonly sold as combined units with asynchronous induction-type electric motors assembled on

the same shaft. When a pump is used in reverse mode as a PAT, the same class of induction motors can act as a generator capable of converting the kinetic energy at the shaft into electricity. PATs are considered an unconventional solution for hydropower generation, adapted to fit in the many scenarios where a conventional turbine would not be economically or technically viable (Ramos and Borga 1999; Fontana et al. 2011; Binama et al. 2017). As such, these are a low-cost hydropower technology that could contribute to the growth of the sector and reductions in climate change in the coming years.

Water pumps are easily available industrial equipment in most countries around the world, and their fabrication is highly standardized. It is generally accepted that the off-the-shelf availability of hydraulic pumps extends up to a nominal power of around 250 kW (Alatorre-Frenk 1994), whereas larger units in the range of megawatts normally require dedicated manufacturing (Garey 1990). Generally speaking, this factor restricts their application range as PATs to the field of minihydropower, microhydropower, and pichydropower (i.e., schemes having nominal power lower than 1 MW).

The use of PATs has shown to have several advantages over other turbine types, namely (1) compact dimensions, (2) mass manufacturing, (3) availability and short delivery time, (4) easy installation, operation, and maintenance, and (5) reduced investment costs when compared with conventional hydro turbines (Chapallaz et al. 1992; Williams 1996; Caxaria et al. 2011; Binama et al. 2017). The physical behaviour of PATs is similar to Francis turbines, but the lack of movable guide vanes prevents the possibility of flow regulation and may result in poor part-load performances. Another limitation to the applicability of PATs is the lack of performance information from pump manufacturers, who are normally not willing to publicize how their units behave when installed as turbines. Therefore, the performance charts of PATs must be predicted by the users during the engineering design phase (Novara and McNabola 2018).

Among the various applications of PATs, these have proved to be very effective if used for microhydropower plants and in-pipe energy recovery with associated power output lower than 100 kW

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(Williams 1996; Lahimer et al. 2012; Carravetta et al. 2012, 2013; McNabola et al. 2014; Lima et al. 2017; Pérez-Sánchez et al. 2017). The cost of such small-scale hydro schemes is heavily influenced by the value of the generating set, consisting of the turbogenerator group. Such equipment commonly represents between 30% and 40% of the total required investment cost (Chapallaz et al. 1992; Ogayar and Vidal 2009). Furthermore, for plants having capacity in the range of a few kilowatts, this figure could be up to 70% (Gallagher et al. 2015). This fact makes the PAT technology a natural choice for powering small-size hydro schemes because cost savings on the purchase of the generating set could make a large impact on the final project viability.

Indeed, in the literature, it is universally acknowledged that the mass manufacturing process allows for the purchase price of pumps to be considerably less expensive than conventional hydro turbines (e.g., Pelton, Francis, Crossflow/Banki, and Kaplan). However, little detailed information is available in the literature about the purchase price of a PAT, and what is there is often out-of-date and contradictory. Some authors in 2009 calculated the cost of a PAT to vary between 200 and 400 €/kW for a nominal power lower than 40 kW (Ramos et al. 2009). Other authors provided an estimate of PAT turbogenerator unit cost as the sum of two components: 230 €/kW to be multiplied by the nominal power accounting for the turbine alone, plus 115 €/kW multiplied by the maximum PAT power accounting for the cost of the generator (Carravetta et al. 2013). Conversely, others proposed a fixed unit cost of 2,000 €/kW for assembled turbine plus generator PAT units (De Marchis et al. 2014).

The cost model proposed by Ramos et al. (2009) shows a realistic decrease of unit cost with increasing PAT size due to economies of scale. However, the validity of this cost model is hampered by the lack of information on its data sources, by the length of time since its publication, and by its applicability, which is limited to a relatively narrow range of PAT sizes. Conversely, the other two commonly cited cost models assume a constant unit cost of equipment. Using these cost models outside their range of applicability would be an oversimplification and subject to significant errors. Additionally, a common limitation of all the presented cost models is their independence from values of PAT nominal pressure head and power. Previous investigations aimed at predicting the purchase price of conventional hydraulic turbines or pumps have found these factors to be significant for the estimation of the equipment cost (Couper et al. 2009; Ogayar and Vidal 2009). Moreover, the aforementioned studies by Ramos et al. (2009), Carravetta et al. (2013), and De Marchis et al. (2014) did not have as their primary objective the development of a PAT cost model with wide applicability. Instead, they examined the technical performance of PATs in specific scenarios. Furthermore, in the literature, there is no agreement on the extent of the gap between the unit cost of PATs and that of conventional hydro turbines. Some authors suggested that PATs are five times less expensive than equivalent conventional hydro turbines (Alatorre-Frenk 1994), whereas others pointed out that for $P > 40$ kW, the price difference between them drops to below 35% (Ramos et al. 2009).

The lack of reliable information on the cost of microhydropower plants has been deemed as one of the obstacles to the diffusion of such plants (Carravetta et al. 2014). Indeed, in order to assess the financial viability of a hydropower project adopting the PAT technology, it is fundamental to have access to realistic estimates of the cost of the generating set. The lack of this information is a clear barrier to the exploitation of this low-cost technology in natural or piped water resource systems. Therefore, the aim of the present paper is to develop a comprehensive model to estimate of PAT plus generator unitary purchase prices and compare these with prices of

conventional hydropower generating sets as well as previously published PAT cost figures.

Cost Determination of Asynchronous Generators

As a first step, the price list [excluding Value Added Tax (VAT)] of 286 asynchronous induction motors belonging to the IE3 efficiency class and rated for nominal power in the range 1–200 kW has been compiled. The analyzed machines belonged to four different manufacturers, and their listings were relative to the years 2009–2018 (Omec Motors 2015; Speroni S.p.a. 2009; Koupas Motors 2016; Vallorani 2016, MA.PI S.a.s. 2018). All generators were selected to have one, two, or three pairs of magnetic poles (pp), which when operated at a 50-Hz electric grid frequency corresponding to a nominal shaft speed of 3,000, 1,500, and 1,000 revolutions per minute (rpm) respectively. Generators rotating at lower speeds (e.g., 8-pole units, rated for 750 rpm at 50 Hz) have not been considered because typical pump performance charts only include curves relative to motors having $pp = 1, 2$, and 3 .

The cost of the generators (C_{gen}) has been found to be approximately proportional to the nominal power rating of the machine, as shown in Fig. 1. Even if for a given power rating, a motor from one manufacturer can be more than two times as expensive as an equivalent one from another manufacturer, as can be noticed in Fig. 1, the authors are confident that the choice of an average cost function is representative of the market. However, because the market for electric motors is a very competitive one, it has been shown that large discounts on the price listings are possible (De Almeida et al. 2008). The set of linear equations obtained from fitting linearly the cost versus power of the induction generators having one, two, or three pairs of magnetic poles ($pp = 1, 2$, and 3) is displayed in Eqs. (1)–(3)

$$\text{For } pp = 1: C_{gen} = 40.03 \times P \text{ (kW)} + 172.24 \quad (1)$$

$$\text{For } pp = 2: C_{gen} = 40.51 \times P \text{ (kW)} + 183.03 \quad (2)$$

$$\text{For } pp = 3: C_{gen} = 60.19 \times P \text{ (kW)} + 163.15 \quad (3)$$

The results show that on increasing the number of magnetic poles, the purchase price of the motor/generators rises as well, even though the difference between $pp = 1$ and $pp = 2$ is not marked.

Cost Determination of PATs

The estimation of cost per kilowatt of centrifugal PATs coupled with induction generators having two, four, or six magnetic poles ($pp = 1, 2$, and 3) has been carried out based on commercially available data (Grundfos 2016; Abacus Resale 2018). In particular, the cost (excluding VAT) of 343 radial end-suction and vertical multistage close-coupled pumps connected to IE3 class generators has been compiled and related to their predicted power output when operated as turbines at their best efficiency point (BEP). The features of such pumps are summarized in Table 1.

For every unit considered, the main parameters obtained from the pump technical specification were the pump mode peak efficiency (η_{BEP}), shaft speed (N), and the coordinates of the pump mode BEP in terms of nominal flow rate and head ($Q_{p,BEP}$ and $H_{p,BEP}$).

Given the difficulty in obtaining information about PAT performance, the characteristic parameters of pumps operated in reverse have to be estimated by the designers. Several methods exist to predict the BEP of a PAT, each with different requirements in

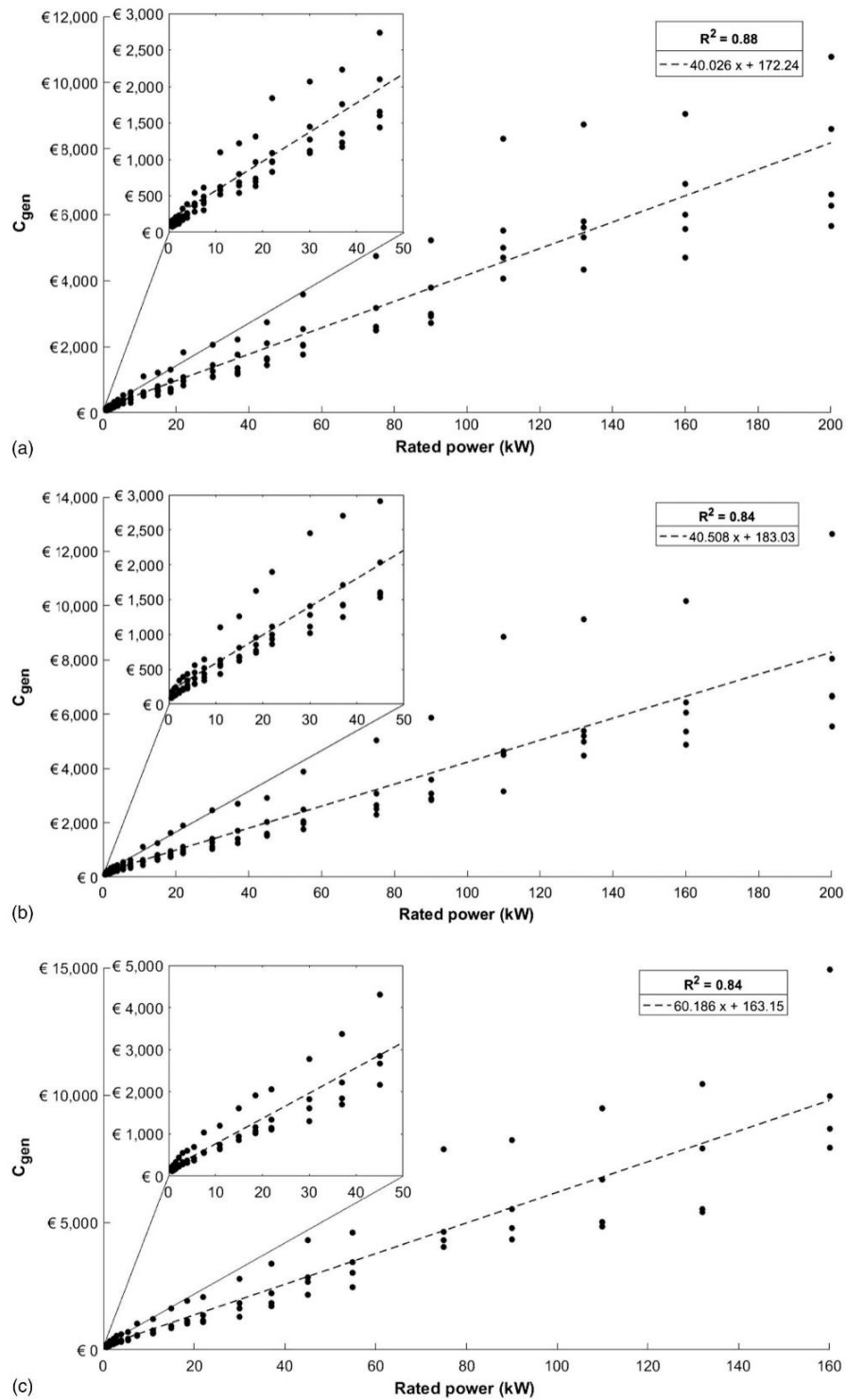


Fig. 1. Cost of 286 asynchronous generators as function of their nominal power for machines having (a) $pp = 1$; (b) $pp = 2$; and (c) $pp = 3$.

Table 1. Features of the selected pumps/PATs

Pump type	Feature	
	Radial, end-suction	Vertical multistage
Sample size	301	42
Volute material	Cast iron	
Shaft material	Stainless steel	
Motor/generator class	Induction type, IE3 class efficiency, having one, two, or three pairs of poles (pp)	
Head range as pumps (m)	2–148	5–200
Capacity range as pumps (L/s)	2–240	1–43
Head range as PATs (m)	3–352	10–311
Capacity range as PATs (L/s)	3–320	1–62

terms of resources needed and complexity: full laboratory tests, computational fluid dynamics (CFD) analysis, and mathematical semiempirical correlations. Because laboratory testing programs are very time- and resource-consuming and CFD data generally require validation, the most straightforward approach is to rely on nondimensional one-dimensional (1D) coefficients relating the head and flow rate values at the BEP ($Q_{t,BEP}$ and $H_{t,BEP}$) with those of the same machine operating as a pump ($Q_{p,BEP}$ and $H_{p,BEP}$), which are known.

Even if a large number of methods can be found in literature to determine such coefficients from one or more machine parameters (most often the pump specific speed and/or η_{BEP}), it has been shown that none of them can provide an accurate estimate of BEP location for any pump (Singh 2005). One of the latest methods available in literature has been shown to provide higher accuracy with respect to previously published approaches against a large database of PATs (Yang et al. 2012) and consists of

$$h = \frac{H_{t,BEP}}{H_{p,BEP}} = \frac{1.2}{(\eta_{p,BEP})^{1.1}} \quad (4)$$

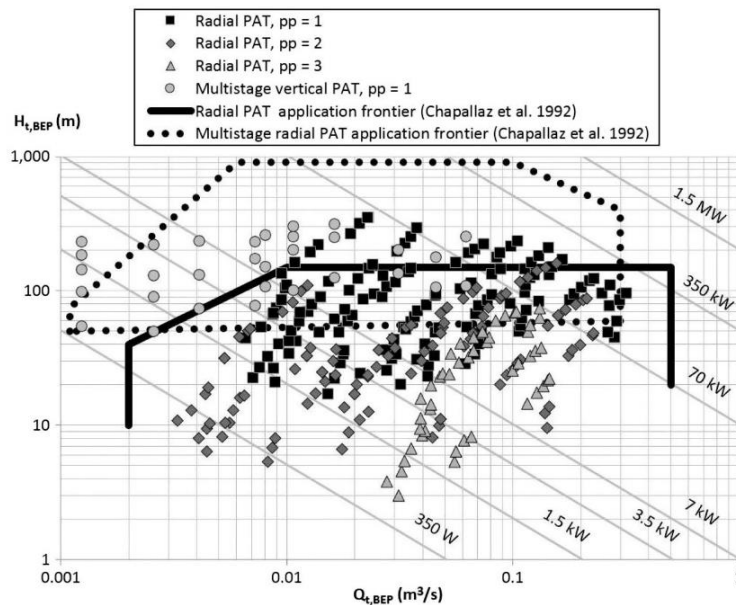
$$q = \frac{Q_{t,BEP}}{Q_{p,BEP}} = \frac{1.2}{(\eta_{p,BEP})^{0.55}} \quad (5)$$

Eqs. (4) and (5) have been used within the present research to estimate the values of $H_{t,BEP}$ and $Q_{t,BEP}$ of the chosen set of pumps when running in turbine mode. Even though such a 1D mathematical method may occasionally result in underestimation or overestimation of the BEP location for a determined machine, it was assumed that their application to a large number of machines, 343 in this case, should dampen such scattered errors.

Finally, the maximum efficiency of pumps operating in turbine mode was considered equal to that in pump mode, which many authors have shown to be a good approximation (Garey 1990; Derakhshan and Nourbakhsh 2008).

The BEP values in terms of predicted nominal flow rate and head ($Q_{t,BEP}$ and $H_{t,BEP}$) of the investigated PATs are displayed in Fig. 2. This shows that the PATs selected for building the cost model may generate from as little as 300 W up to more than 300 kW. Additionally, when compared with the PAT application range chart proposed by Chapallaz et al. (1992), the selected set of PATs fit well within the proposed application boundaries.

As a general rule, a pump is able to process a larger flow rate and hydraulic head when operated in reverse as a turbine compared with its performance as a pump. Therefore, when a pump is bought to be used as PAT, its attached motor unit will need to be replaced by another one rated for a higher nominal power. Besides, when using an asynchronous induction motor as generator, it has been suggested to derate it by 20% of the nominal capacity as a motor in order to avoid overheating and reliability issues (Williams 1996). In order to account for this, Eqs. (1)–(3) have been used to calculate the price of the motor originally sold with pumps ($C_{gen,p}$) and the price of the larger unit able to run as generator at 80% of its nominal capacity ($C_{gen,PAT}$).

**Fig. 2.** Location of the estimated PAT BEP for the selected 343 machines over the H - Q space.

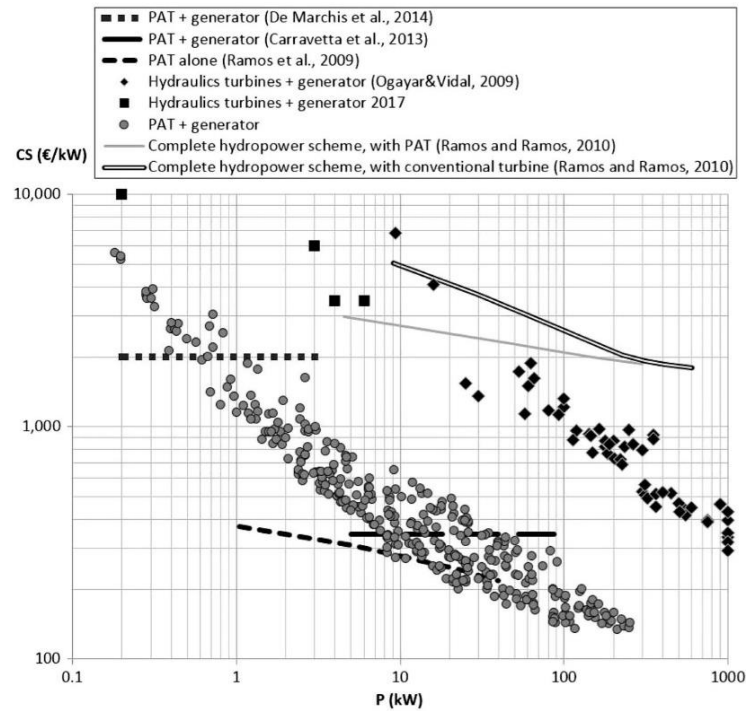


Fig. 3. Unit cost per rated power $CS_{PAT+gen}$ of the 343 evaluated PATs coupled to IE3 asynchronous generators compared with previously available PAT cost figures and unit cost of conventional hydro turbine plus generator units. (Adapted from Novara et al. 2017.)

The resulting cost of a PAT plus generator has been therefore calculated as follows in Eq. (6):

$$C_{PAT+gen}(\text{€}) = C_p(\text{€}) - C_{gen,p}(\text{€}) + C_{gen,PAT}(\text{€}) \quad (6)$$

where C_p = selling price of a pump with its original motor. The nominal power of PATs was calculated as follows in Eq. (7):

$$P(\text{kW}) = \frac{\gamma \times \eta_{BEP} \times Q_{t,BEP} \left(\frac{\text{m}^3}{\text{s}}\right) \times H_{t,BEP}(\text{m})}{1,000} \quad (7)$$

where γ = specific weight of water (N/m^3); and η_{BEP} = maximum efficiency of a PAT and was considered equal to the maximum efficiency of the same machine operated as a pump, in agreement with the available literature (Derakhshan and Nourbakhsh 2008; Garey 1990). The PAT plus generator unit cost ($CS_{PAT+gen}$) was then computed as follows in Eq. (8):

$$CS_{PAT+gen} \left(\frac{\text{€}}{\text{kW}}\right) = \frac{C_{PAT+gen}(\text{€})}{P(\text{kW})} \quad (8)$$

The resulting figures of PAT specific cost ($CS_{PAT+gen}$) have been compared with cost per rated power of conventional hydro turbines (Francis, Pelton, and Kaplan) as gathered by the authors in 2017 and by Ogayar and Vidal (2009). The cost figures proposed by Ogayar and Vidal (2009) have been updated to account for the inflation that has occurred between 2008 (year of original data collection) and the end of 2016. Also, the specific cost of PAT generating sets has been compared with other previously published costs. The resulting chart is displayed in Fig. 3, where it can be

noticed how for a given installed power according to the choice of PAT type (radial or multistage) and shaft speed ($pp = 1, 2$, or 3) the cost of a turbogenerator may even double. For instance, at 3 kW, the unit cost $CS_{PAT+gen}$ varied from 500 to 1,000 €/kW, and at 20 kW, it was in the range 200–500 €/kW.

The PAT and generator assembly unit cost shown in Fig. 3 decreased with the increase of nominal power output, showing realistic economies of scale. At the same time, the previously published cost estimates (Ramos et al. 2009; Carravetta et al. 2013; De Marchis et al. 2014) can be considered accurate only within specific intervals of nominal power. On the other hand, generating sets relying on PAT technology proved to be less expensive per unit of installed power than conventional hydraulics turbines by a factor of 5 for $P > 20$ kW and up to 15 for lower powers.

In the literature, the existence of a linear relationship between the parameter $Q\sqrt{H}$ and the price of centrifugal pumps has been validated (Couper et al. 2009). Plotting graphically such parameters for the analyzed set of PATs, a significant linear correlation was also found, relating the $Q_{t,BEP}$ (m^3/s) and $H_{t,BEP}$ (m) of a machine to its purchase price. For every considered PAT (single-stage radial or vertical multistage, having generators with $pp = 1, 2$, or 3) a best linear fit for the cost function was derived as shown in Fig. 4.

The results of the linear regressions in Fig. 4 proved to be a reasonable estimate in determining the purchase price of radial PATs with two and three pole pairs ($R^2 = 96\%$ and 97% , respectively). The slightly reduced accuracy of the linear fit of radial PATs with $pp = 1$ generators was mostly due to the large number of data points available for this category in comparison with other types of machines. Instead, the value of R^2 relative to vertical multistage

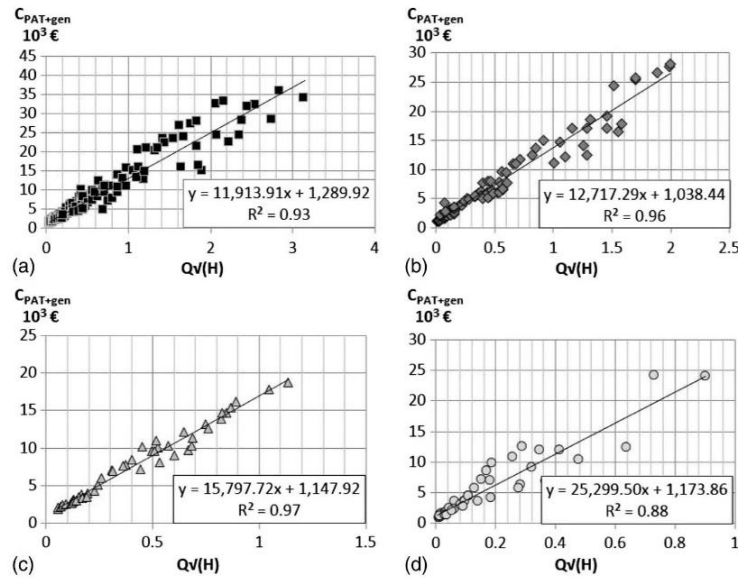


Fig. 4. Linear functions predicting the purchase price of (a) radial PATs, pp = 1; (b) radial PATs, pp = 2; (c) radial PATs, pp = 3; and (d) vertical multistage PATs, pp = 1. (Adapted from Novara et al. 2018.)

PATs has been calculated as 88%. The set of linear equations arising from Fig. 4 are displayed in Eqs. (9)–(12)

Radial PATs

$$pp = 1: C_{PAT+gen}(\epsilon) = 11,913.91 \times Q\sqrt{H} + 1,289.92 \quad (9)$$

Radial PATs

$$pp = 2: C_{PAT+gen}(\epsilon) = 12,717.29 \times Q\sqrt{H} + 1,038.44 \quad (10)$$

Radial PATs

$$pp = 3: C_{PAT+gen}(\epsilon) = 15,797.72 \times Q\sqrt{H} + 1,147.92 \quad (11)$$

Vertical multistage PATs

$$pp = 1: C_{PAT+gen}(\epsilon) = 25,299.50 \times Q\sqrt{H} + 1,173.86 \quad (12)$$

As could be expected, PATs linked to a generator with a higher number of magnetic poles, i.e., rotating at a lower shaft speed, also had a higher purchase price.

Comparing the real cost of the analyzed PAT plus generator units with the cost estimate obtained by applying the set of Eqs. (9)–(12), the distribution of errors displayed in Fig. 5 can be obtained. The mean error corresponded to -5.1% , meaning that the application of Eqs. (9)–(12) may lead to an average overestimation of the PAT plus generator purchase price of about 5%. However, in

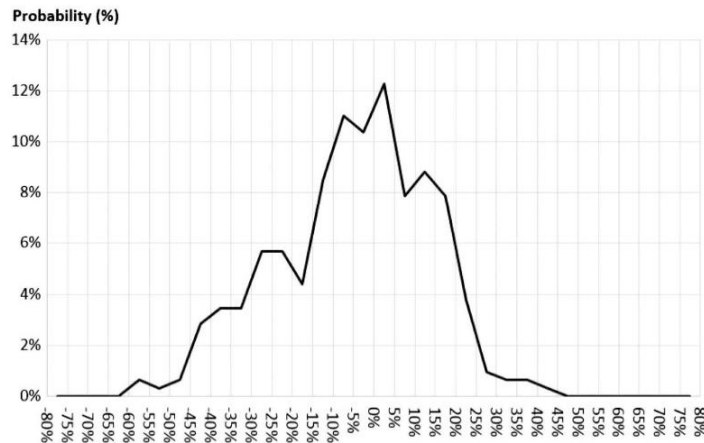


Fig. 5. Distribution of the percentage errors resulting from applying Eqs. (9)–(12) to the 343 analyzed PATs.

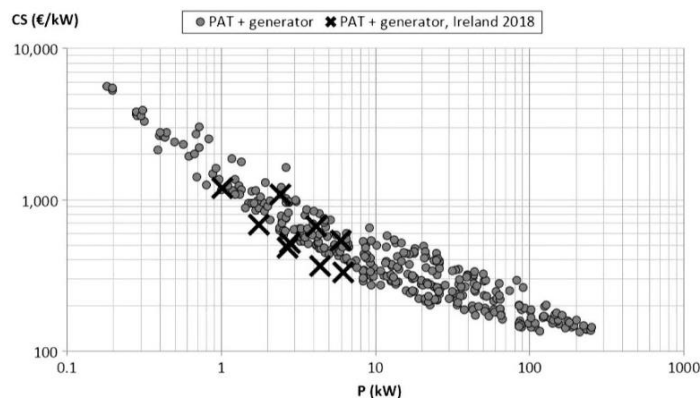


Fig. 6. Comparison of the unit cost of nine PATs from different manufacturers as gathered in Ireland, 2018, with the proposed cost model.

81% of cases, the relative error between actual and estimated prices was found to be within the $\pm 25\%$ interval, which is generally acceptable at a predesign phase.

The developed PAT cost prediction model has been compared with the selling price of nine radial end-suction pumps to be used as turbines as gathered in Ireland in the first trimester of 2018. The investigated units belonged to different manufacturers (KSB, Ebara, Johnson Pumps, and DAB), and as shown in Fig. 6, their cost per unit power agrees well with the proposed PAT purchase price prediction model.

Review of Other Project Costs in a PAT Installation

The expenditure necessary to the purchase of the turbine and generator is only a portion of the total project cost of any hydropower scheme. Some of the additional required cost items, such as civil works, engineering design, site management, and grid connection fees, can reasonably be considered as highly site-specific and independent of the chosen turbine technology. By contrast, the implementation of a control system with different degrees of automation is a reflection of the operator requirements and the choice of turbine type. The cost of such a control system can have a strong impact on the total cost figure, as reviewed by Ogayar and Vidal (2009), who established that on average for hydro schemes having capacity lower than 2 MW and featuring conventional turbine types (Francis, Pelton, and Kaplan), the control system accounted for the 22% of the total investment, representing nearly 2/3 of the turbogenerator cost. Because a turbogenerator unit featuring a PAT would be less expensive than an equivalent one based on a conventional turbine technology as seen in Fig. 3, it is reasonable to assume that the share of the control system cost over the total investment may grow to above 22% if a PAT is used.

Additionally, given its rigid geometry and lack of in-built regulation devices, in some contexts the adoption of a PAT would require the installation of a relatively sophisticated and expensive control system in order to achieve a degree of system flexibility comparable with that of a conventional turbine (Novara and McNabola 2018). An example of this is the application of PATs as energy recovery devices within drinking-water networks designed to exploit local excesses of water pressure otherwise dissipated by means of pressure-reducing valves (PRVs). Due to operational reasons, network operators may require a specific

backpressure level to be maintained downstream of the PAT independently from the instantaneous flow rate and upstream pressure, which the machine alone would not be capable of accomplishing. In order to achieve this, three alternative designs are possible (Carravetta et al. 2012, 2013):

- hydraulic regulation (HR), in which a PRV is placed in series with the PAT and an automated by-pass circuit is installed parallel to the main branch;
- electric regulation (ER), when the turbine shaft speed can be varied by means of a four-quadrant inverter connected to the main grid; and
- hydraulic and electric regulation (HER), when both of the aforementioned options are implemented.

All three mentioned designs require a closed-loop control device able to perform a real-time reading of the connected sensors (typically, flow, and pressure meters) and adjust accordingly the opening of the bypass in case of HR and the turbine shaft speed in case of ER (Carravetta et al. 2012, 2013).

Very little information can be found in the literature regarding existing PAT schemes, especially concerning their economics and construction details. An example of PAT installations within water networks featuring an HR control strategy are the twin power plants of Cartuja and Conejeras near Granada, Spain. The two installed PATs are designed for a flow of 278 L/s and head of 39 m (Cartuja) and 36 m (Conejeras) and are coupled to a 90-kW generator each (Carravetta et al. 2017). In both cases, the reported cost of the control system was 1.2 times the purchase cost of the PAT and generator, which is higher than the ratio suggested by Ogayar and Vidal (2009) to hydro schemes with conventional turbines. It is reasonable to assume that this phenomenon is due partially to the lower acquisition cost of the PATs and partially to the relative sophistication of the control system needed to ensure HR.

However, the in-pipe installation of PATs within water networks is a very specific application field of the technology that requires special precautions in order to ensure the essential reliability of water supply. Instead, in the vast majority of cases where flow and backpressure control is not required (e.g., run-off-river schemes), the control system of a PAT can be significantly less elaborate and expensive and based on a standard on/off operating procedure.

In addition, given the lower peak efficiency of a PAT compared with that of conventional turbines (Novara and McNabola 2018) and the relatively more sophisticated control system that circumstances may require when using a PAT, it is reasonable to assume

that above a certain threshold value of installed power, the total cost of a scheme adopting either a PAT or a conventional turbine could match. Previous research that analyzed the cost per installed kW of several hydro schemes suggested that such a threshold corresponded to 200–300 kW (Ramos and Ramos 2010), as shown in Fig. 3.

Conclusions

The PAT cost model produced here is an important enabling tool to in the exploitation of low-cost hydropower in natural and piped water resource systems. The results obtained prove that previously published PAT plus generator cost figures are accurate but only in a fraction of the whole microhydropower application range, and the unit cost may vary from 130 to 5,400 €/kW according to the machine rated power. In all cases, PAT plus generator unit cost proved to be between 5 and 15 times lower than generating sets based on conventional hydro turbines. Given that cost models are generally highly sensitive, it is recommended for inflation to be accounted for during its future use.

Linear correlations were produced to estimate the price of radial and multistage PATs connected to a motor generator unit having one, two, or three magnetic pole pairs as function of their nominal head and flow rate in turbine mode. The proposed cost model appears to be a valid tool for estimating the PAT price during the design phase, but its accuracy is limited by several factors: the number sample of pumps and pump types on which it is based, the assumption of equal maximum efficiency in pump and turbine modes, and the inaccuracies of the Yang et al (2012) method used for predicting the PAT operating points.

Future work to improve the accuracy of this PAT cost model should focus on the inclusion of more data on axial flow and other pump families not well covered by the current database. In addition, when more reliable PAT performance prediction methods become available, the cost model can be further refined. However, these must be widely applicable considering the often-limited amount of input parameters available.

Furthermore, scholars and practitioners attempting to predict the total project cost of a PAT installation must be aware that the cost of the control system for in-pipe applications where back-pressure control is needed has been reported to account for up to 1.2 times the acquisition cost of the turbogenerator unit. If more empirical economic data, including the cost breakdown of future PAT installations, were made available, it would help in reducing the uncertainties around the cost estimate of the control system under different scheme configurations.

Acknowledgments

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Notation

The following symbols are used in this paper:

- C = cost (€);
- c = cost per unit of nominal power (€/kW);
- H = hydraulic head (m);
- N = rotational speed (rpm);

- P = power (kW);
- pp = number of magnetic pole pairs;
- Q = flow rate (m^3/s); and
- η = efficiency.

Supplemental Data

Table S1, which shows the key parameters of the 343 pumps/PATs from the analyzed database, is available online in the ASCE Library (www.ascelibrary.org).

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3.4. A MODEL FOR SELECTING THE MOST COST-EFFECTIVE PRESSURE CONTROL DEVICE FOR MORE SUSTAINABLE WATER SUPPLY NETWORKS



Article

A Model for Selecting the Most Cost-Effective Pressure Control Device for More Sustainable Water Supply Networks

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Abstract: Pressure Reducing Valves (PRV) have been widely used as a device to control pressure at nodes in water distribution networks and thus reduce leakages. However, an energy dissipation takes place during PRV operation. Thus, micro-hydropower turbines and, more precisely, Pump As Turbines (PAT) could be used as both leakage control and energy generating devices, thus contributing to a more sustainable water supply network. Studies providing clear guidelines for the determination of the most cost-effective device (PRV or PAT) analysing a wide database and considering all the costs involved, the water saving and the eventual power generation, have not been carried out to date. A model to determine the most cost-effective device has been developed, taking into account the Net Present Value (NPV). The model has been applied to two case studies: A database with 156 PRVs sites located in the UK; and a rural water supply network in Ireland with three PRVs. The application of the model showed that although the investment cost associated to the PRV installation is lower in the majority of cases, the NPV over the lifespan of the PAT is higher than the NPV associated with the PRV operation. Furthermore, the ratio between the NPV and the water saved over the lifespan of the PAT/PRV also offered higher values (from 6% to 29%) for the PAT installation, making PATs a more cost-effective and more sustainable means of pressure control in water distribution networks. Finally, the development of less expensive turbines and/or PATs adapted to work under different flow-head conditions will tip the balance toward the installation of these devices even further.

Keywords: pumps as turbines; pressure reducing valves; energy recovery; leakage reduction; water-energy nexus

1. Introduction

World water use has risen considerably in recent decades and this growing tendency is likely to continue due to the expected population growth in the coming years [1]. In addition, large cities are increasingly concentrating a higher number of people, leading to water scarcity conditions in many urban areas [2]. One quarter of the largest cities in the world were recently estimated to be water stressed [3]. In Europe, and more specifically in its southern regions, a 24% decrease in renewable water resources has been detected during the period 1960–2010 [4]. Moreover, world energy consumption is also expected to increase by 28% in 2040 with respect to 2015 [5]. Therefore, the sustainable use of water and energy resources is currently one of the main concerns worldwide.

Water abstraction and distribution are among the activities in which the water-energy nexus plays an important role. In the European Union, as an example, 8% of the total energy consumption in 2014 was related to water supply [6]. In addition, it is estimated that 32 billion cubic meters per year (66% of the treated water) are lost in the water distribution process globally [7]. This is mainly due to the

ageing infrastructure, the non-optimal design of the water supply systems, and the increase in water stress in urban areas [8]. Different strategies have been proposed to reduce energy consumption from fossil fuels in the water industry sector, by using renewable energy sources [9] or by recovering the excess of heat at the wastewater treatment plant [10], to name a few.

Leakage management can also play a major role in the reduction of energy consumption in the water sector. Thus, pressure control in water distribution networks is one of the most effective measures to reduce leakages because of the direct relation between pressure and leakage rate [11,12]. Hence, Pressure Reducing Valves (PRVs) play an important role in reducing water losses. The determination of the best location and the number of PRVs to install in a network has been widely analysed as an effective measure to decrease leakages. Gupta et al. [11] applied an optimization algorithm to determine the optimal PRVs location in a network, resulting in a leakage reduction of 21%. Fontana et al. [13] used real time control to manage the pressure in a PRV placed in a network inlet. This enabled optimal operation conditions by reducing pressure at nodes and therefore, leakages.

PRVs are a convenient device for reducing leakage. However, the energy dissipation that takes place in a PRV is wasteful of energy resources, and this energy could be recovered by substituting the PRV with a hydropower turbine. Thus, in addition to the reduction of water losses, a certain part of the energy in the network could be recovered, reducing greenhouse gas emissions and making the water supply system more sustainable [14]. Due to the potential of hydropower in these systems, several investigations have focused on the evaluation of the installation of these devices in water networks, showing that up to 40% of the gross power potential available in a PRV could be recovered by replacing the PRV with a PAT [15]. Other potential locations for installing hydropower turbines, such as break pressure tanks, water storages or water treatment plants, were also evaluated, estimating significant energy generation potential in some locations [16].

The main disadvantage historically associated with hydropower installations in water distribution networks was their lack of cost-effectiveness when the power output capacity is small [17]. The majority of power output capacities in typical PRVs in water distribution networks has been shown to lie in the range 1 to 15 kW [18]. However, recent developments in new micro-turbines which can operate under low flow/head conditions or new in-line Banki turbines with mobile regulating flap which enable the adaptation of the characteristic curve according to flow or pressure with reasonable high efficiencies may alter the results of these historical assessments which considered traditional turbine types [19–22].

A particular class of micro-turbines consists of pumps working in reverse mode, i.e., Pump As Turbines (PAT). These are devices that can be installed along distribution pipes to reduce pressure at nodes and recover energy [23–25], with significantly reduced investment costs compared to traditional turbines [26]. Optimization techniques have focused on identifying the best PAT location within water distribution networks to reduce leakage and generate energy. Giugni et al. [27] proposed a methodology to determine the optimal PAT location in water distribution networks considering two approaches: The reduction of leakage, and the maximization of energy. The authors demonstrated that the second approach involved higher energy production without decreasing pressure at nodes significantly. Other methodologies have considered the joint optimization of energy generation by means of PATs, leakage reduction and pump operation in order to reduce the energy costs associated with the water network [25,28]. The use of PATs as energy recovery devices within irrigation water distribution networks has also been proved feasible, with calculated annual energy recovery of up to 58 MWh and 271 MWh in selected networks [29,30].

The replacement of PRVs with PATs as a measure to generate energy and reduce leakage represents an efficient strategy to improve the sustainability of water supply networks. However, the sometimes small amounts of energy generated by a PAT can be interpreted as not achieving economic viability in some cases, making PRVs more attractive. For example, Fecarotta et al. [25] proposed an optimization procedure to determine the best location and the number of PATs considering both the maximization of the net present value (NPV) and the reduction of leakage. They included the water cost saving by leakage reduction in the estimation of the NPV and the algorithm provided a solution with 16 PATs

to install on a selected network. However, 10 out of the 16 PATs were considered not economically feasible due to the low power generated. Therefore the authors proposed the installation of PRVs instead of PATs for those sites.

Some investigations have evaluated the replacement of PRVs by PATs as a measure to control pressure and generate energy, also analysing the most convenient element from an economic point of view [24,31,32]. However a comprehensive study giving guidelines for the installation of pressure reducing devices (PRV or PAT) for pressure reduction at specific nodes taking into account all the cost involved in both, and the eventual power generation have not been provided to date.

Therefore in this work a model has been developed to aid in choice between PAT and PRV in water distribution networks. The model focuses on the determination of the Net Present Value (NPV) taking into account the total cost associated with the installation of a PRV and a PAT as well as the cost savings associated with the reduction of leakage volume and the energy savings related to the PAT performance. The methodology has been applied to two case studies: The first one included a database with 156 PRVs sites located in Wales and West Midland regions of the UK; while the second case study included a water distribution network with three PRVs currently operating, located in a rural area in Ireland.

2. Methodology

2.1. Problem Approach

The model proposed in this work took into account all costs involved in the installation of a PRV and a PAT, as well as the sum of the incomes associated with the operation of both devices during their lifespan. Thus, *NPV* (€), which was previously used to determine the economic feasibility of the installation of PATs [25], was selected in this work to compare both devices:

$$NPV = -TC + \sum_{t=1}^L \frac{CS_t}{(1+r)^t} \quad (1)$$

where *TC* (€) is the total installation cost, *t* is an index related to year, *L* is the lifespan of the device considered, PRV or PAT (assumed as 15 years for both elements in this work). *CS* is the total cost saving at year *t* (€), and is determined by adding up the water cost saving, *WCS*, associated with the reduction of leakage volume after installing these devices and the energy cost saving, *ECS*, obtained by the PAT performance. *r* is the discount rate. A value of 0.05 for *r* was assumed in this work [33]. No management costs have been considered since the same cost has been assumed for both installation types.

The total installation cost was determined using Equation (2):

$$TC = C_{PRD} + C_{HE} + IC \quad (2)$$

where *C_{PRD}* is the cost of the pressure reducing device, PRD (PRV or PAT), *C_{HE}* the cost of the hydraulic elements required in the installation of each pressure reducing device, and *IC* is the installation cost. The determination of each term in Equation (2) is described in the following sections.

The estimation of *WCS* was carried out by the calculation of the water saving derived from the installation of a PRV or a PAT. When any of these devices is installed in a water supply network, pressure at nodes decreases and hence the leakage flow is reduced.

When the leakage flow at each pipe is known, this value can be assigned to each node and the discharge coefficient, *c_i*, which relates the leakage flow to pressure can be determined according to Equation (3):

$$ql_i = c_i \cdot P_i^\beta \quad (3)$$

where q_i is the leakage flow at node i (l s^{-1}), β is the emitter exponent which takes into account the pipe material and the shape of the orifice (1.18 has been assumed in this work [34]) and P_i is the pressure (m) at node i . The coefficient c_i was determined by Equation (4):

$$c_i = \alpha \cdot 0.5 \cdot \sum_{j=1}^{K_{ji}} L_{ji} \quad (4)$$

α is a coefficient ($\text{l s}^{-1} \text{ m}^{-1-\beta}$), j is an index related to pipe, K_{ji} is the number of pipes connected to node i , and L_{ji} is the length of the pipe j connected to node i (m). The coefficient α was determined using EPANET [35] which enabled the pressure dependent demand analysis. Once the coefficient α was determined, the effects of the performance of a PRD were evaluated in EPANET and WCS was estimated by:

$$WCS = c_w \cdot (VI_0 - VI_{PRD}) \cdot ndays \quad (5)$$

where c_w is the water cost (0.3 € m^{-3} was assumed in this work [25]) and VI_0 and VI_{PRD} ($\text{m}^3 \text{ day}^{-1}$) are the leakage volume in the current situation and after installing the pressure reducing device, respectively. $ndays$ was the number of days of the year. The leakage volume was obtained by applying Equation (6):

$$VI = \frac{1}{1000} \cdot \sum_{h=1}^{24} Q_{lh} \cdot \Delta h \quad (6)$$

where the term $1/1000$ is used to convert units from l s^{-1} to $\text{m}^3 \text{ s}^{-1}$, Q_{lh} is the total leakage flow at time h and Δh (s) is the time in which Q_{lh} (l s^{-1}) is applied.

However, in most of the cases, the leakage flow of the entire network is the only available data. For these analyses, methodologies to estimate the leakage flow at each node from the total leakage rate have been proposed. In this work, the methodology proposed by Araujo et al. [36] was used. From the current leakage rate, which is usually estimated as a percentage of the minimum total night flow, an iterative process was carried out to determine the value of c_i which accomplished Equations (3) and (4) using the software EPANET [35]. This iterative process was required to set the value of c_i at each node and thus determine the leakage flow according to the pressure received at each node [37]. After that, two types of demand were applied to the nodes, one associated with the human consumption, while the other was pressure-dependent and related to leakage. Therefore, a genetic algorithm was applied to determine the new demand pattern which, multiplied by the base demand and added up the leakage flow, matched with the actual measured hourly demand of the network. A detailed description of the above methodology can be found in [36]. Once the leakage flow after installing a PRV or a PAT was obtained, the water cost saving was determined by Equation (5).

The procedure to determine the water saving was carried out in MATLAB [38].

As for the energy cost saving, Equation (7) was used to estimate its value:

$$ECS = \frac{1}{1000} \cdot ndays \cdot \sum_{h=1}^{24} \rho \cdot g \cdot Q_{PAT,h} \cdot H_{PAT,h} \cdot \eta_h \cdot ep \quad (7)$$

where the term $1/1000$ is used to convert units from W to kW, ρ is the water density (kg m^{-3}), g the gravity acceleration (m s^{-2}), $Q_{PAT,h}$ ($\text{m}^3 \text{ s}^{-1}$) and $H_{PAT,h}$ (m) the flow and head provided by the PAT at time h , respectively. η_h was the efficiency of the PAT and ep the savings from displaced electricity costs (0.17 € kWh^{-1} has been assumed in this work, [39]).

Different methodologies to estimate the friction losses in a PAT have been developed [40,41]. A simplified approach similar to the one proposed by [23] was considered in this work. According to [42], the maximum PAT efficiency, i.e., the efficiency of a PAT at the Best Efficiency Point (BEP) is achieved when the flow at the PAT is 75% of the maximum flow, considering 0% as the minimum flow

rate through a PAT and 100% as the maximum flow at the same speed. Therefore, two considerations were assumed in this work: (1) $Q_{PAT,h}$ was Q_{BEP} when the input flow was higher than Q_{BEP} and η_h matched with the maximum PAT flow-to-wire efficiency, η_{max} , assumed here as 0.65 [18,25]. In this case, the rest of the flow was bypassed; (2) $Q_{PAT,h}$ matched with the input flow when it was lower than Q_{BEP} and the η_h was determined according to the methodology proposed by [42], in which they evaluated the performance of 113 PATs by characterising the relationship between head and specific speed. The 113 PATs performance database was included in EPANET, to estimate the hydropower potential according to the instant flow and head at a certain site.

A flow chart with a summary of the methodology above described is shown in Figure 1.

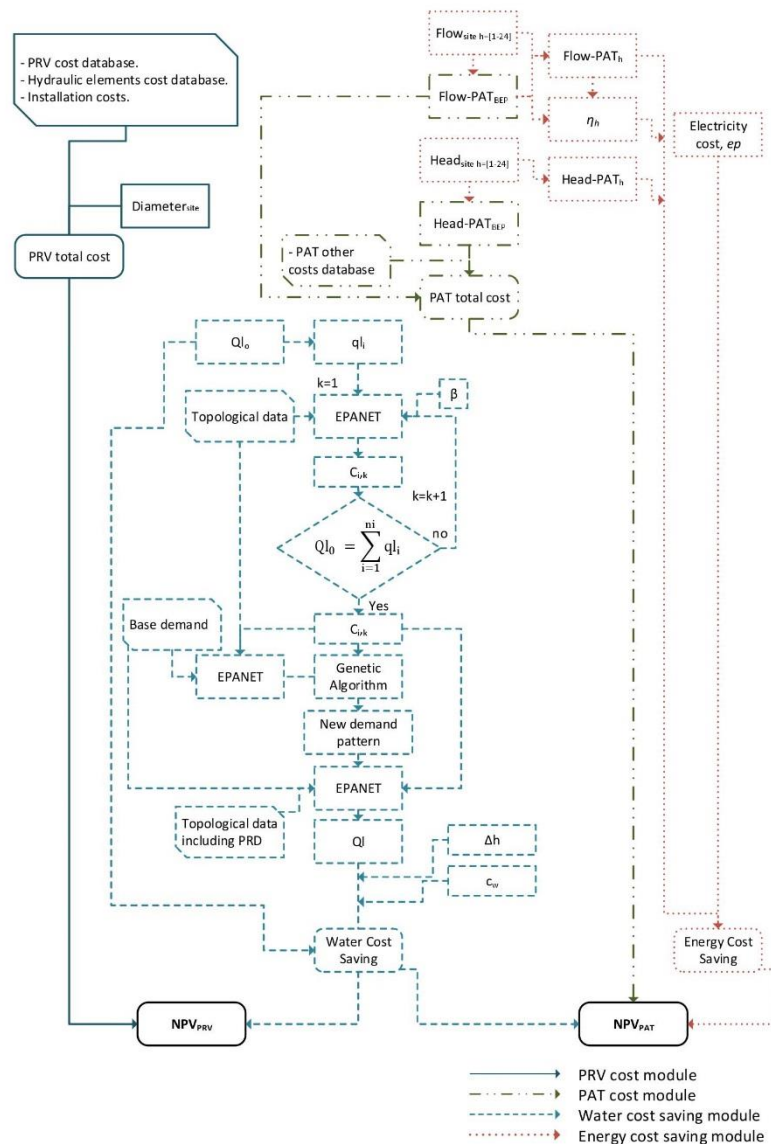


Figure 1. Flow chart of the model to determine the most cost-effective device.

2.2. PRV Total Installation Costs

The installation of a PRV in addition to the valve itself usually requires an upstream isolation valve, a strainer, a downstream isolation valve and an air valve (Figure 2a). In order to select the PRV to be installed in a certain pipe, the flow through it, its diameter and the pressure setting range need to be considered.

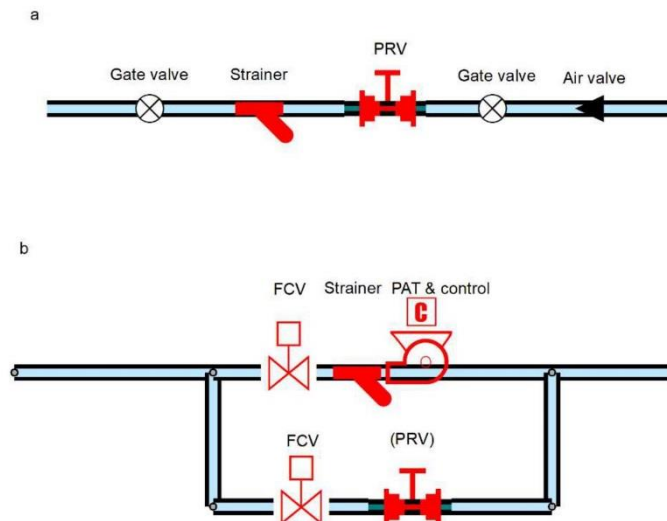


Figure 2. Installation scheme of a Pressure Reducing Valve (PRV) (a) and of a Pump As Turbine (PAT) (b).

Data related to the cost of PRVs from several manufacturers for a wide range of diameters (50, 65, 80, 100, 125, 150, 200, 250, 300, 400, 500 and 600 mm) and for different pressure settings, were collected [43–51]. Likewise, the costs of the hydraulic elements required in the installation of a PRV for different diameters were gathered. Civil works were also considered, and assumed here as the 10% of the total costs [52].

2.3. PAT Total Installation Cost

The cost in the European context of a centrifugal PAT with a connected four-pole asynchronous motor used as a generator can be estimated as function of its nominal flow and head working conditions at the Best Efficiency Point (BEP) through a set of linear correlations [53]:

$$C_{PAT+gen}(\text{€}) = 12,717.29 \cdot Q_{PATBEP} \sqrt{H_{PATBEP}} + 1038.44 \quad (8)$$

The above equation was determined from a database of 343 commercially available pumps and 286 generators [53]. Only asynchronous induction motors which can efficiently work as generators and are commonly sold as standard prime movers of hydraulic pumps were selected. More specifically, four-pole asynchronous motors have been considered as their moderate nominal speed of about 1500 RPM prevents PATs from reaching an excessive speed under runaway conditions, compared to two-poles units.

Apart from the purchase of the turbine and the generator, the final cost of a typical Micro-Hydropower (MHP) scheme includes additional contributions (see Figure 2b) which can be grouped as follows:

- Civil works and hydraulic equipment, including:

- a bypass pipe;
- a set of actuated or manually operated control and sectioning valves;
- a PRV (in certain installations). The necessity of installing this device depends on whether the PAT will operate far from its BEP at a given site, and the capability of the actuated valves to control the pressure received at downstream nodes. If the pressure is too high to be controlled by the actuated control valves, a PRV installed in the bypass is required.
- a Y-strainer;
- a powerhouse hosting the equipment.
- Electric cabinet and control system
- Grid connection fee
- Commissioning
- Other project costs (including consultancy)

The cost of the above elements in PAT energy recovery installations is unclear, since no comprehensive study has provided a cost breakdown of PAT-based MHP schemes embedded in water infrastructures to date. Some authors have suggested that the contribution given by the purchase of a conventional turbine and generator may account typically for the 35% [54] up to a maximum of 70% of the final cost figure of a MHP scheme [18]. However, such figures are not directly applicable to the context of this research since their authors did not consider the use of PATs, which allows for a considerable reduction of the turbine purchase price (5 to 20 times less expensive). In addition, they did not refer specifically to hydropower stations within water distribution networks.

In order to evaluate more accurately the cost of such plants, data from 9 energy recovery schemes in water networks from different countries have been compiled [24,55–57]. All of the selected plants adopted a PAT as a generating device, and had nominal powers ranging from 9 to 120 kW. The location and power rating of all 9 schemes is displayed in Figure 3. According to the available information, it was possible to sub-divide the total cost of the installation into single components:

- turbogenerator alone;
- turbogenerator and control system;
- commissioning;
- civil works and hydraulic equipment;
- grid connection;
- other project costs.

From the purchase price of a PAT and a generator by means of Equation (8), the resulting cost breakdown allowed the realistic quantification of the total expected cost of a PAT-based MHP plant in water infrastructures from its nominal values of flow rate and hydraulic head.

2.4. Case Studies

The first case study consisted of a database with 156 PRVs placed in water infrastructure in Wales and the West Midland regions of the UK (see Figure 4). Information related to average flow, pressure and pipe diameter were collected at those sites [18]. The average flow ranged from 0.2 l s^{-1} to 296.5 l s^{-1} with associated head values of 32 m and 15 m, respectively. As for diameters, the values were 150 mm, 200 mm, 250 mm, 300 mm and 400 mm.

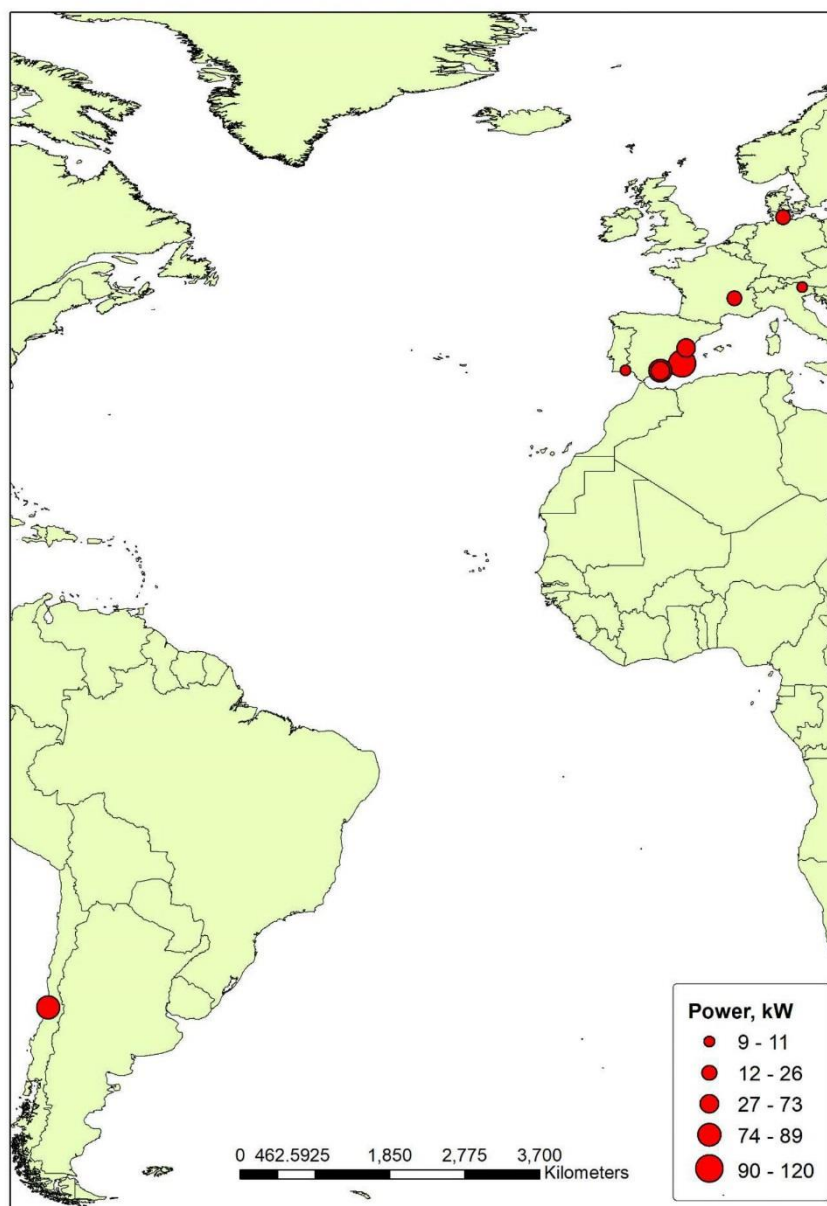


Figure 3. Location of the PAT plants used to estimate the total PAT cost.

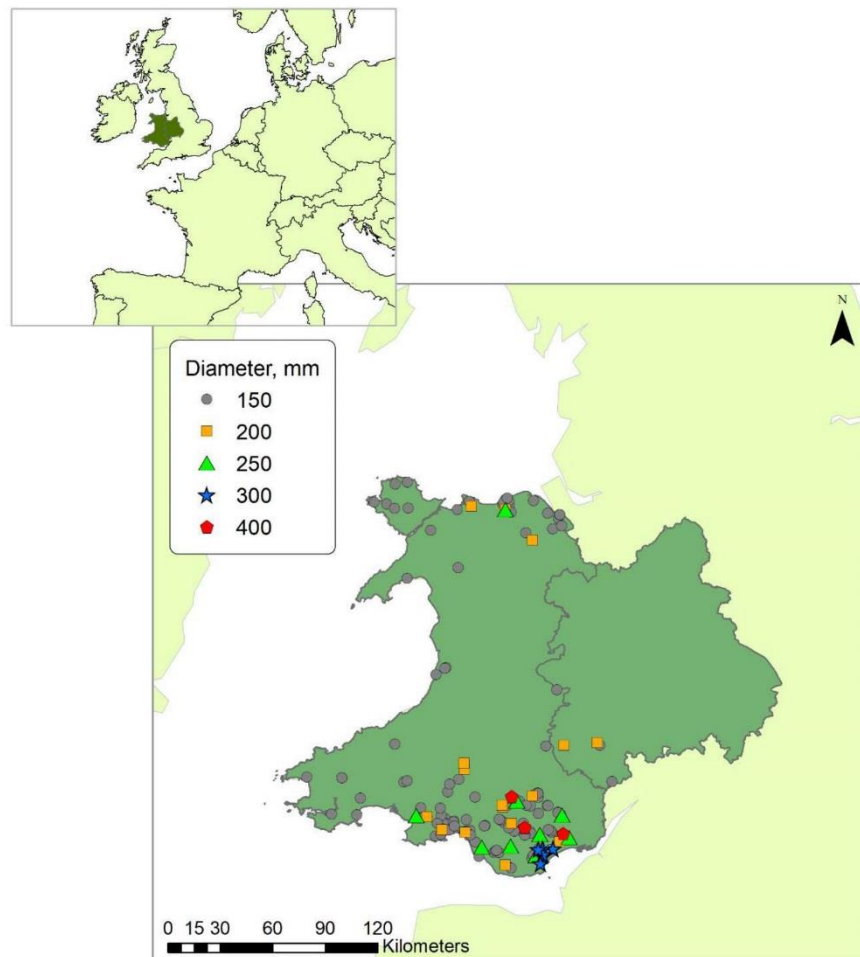


Figure 4. Location of the selected PRV sites.

The second case study focused on Ballinabranna Group Water Scheme (BGWS), a water supply network located in a rural area in Ireland. BGWS is a gravity network which supplies an average demand of 6.16 l s^{-1} to 44 consumption nodes. This network covers an area of almost 54 km^2 , with diameters ranging from 50 mm to 150 mm. Three PRVs are currently operating in the network. However, in an analysis to determine the optimum PAT locations in this network, a new potential site was detected [37]. This site showed higher MHP potential than that of the current PRV locations. Hence, the analysis carried out in this work took into account the determination of the NPV in two of the current PRVs and at the proposed additional site (site 3 in Figure 5).

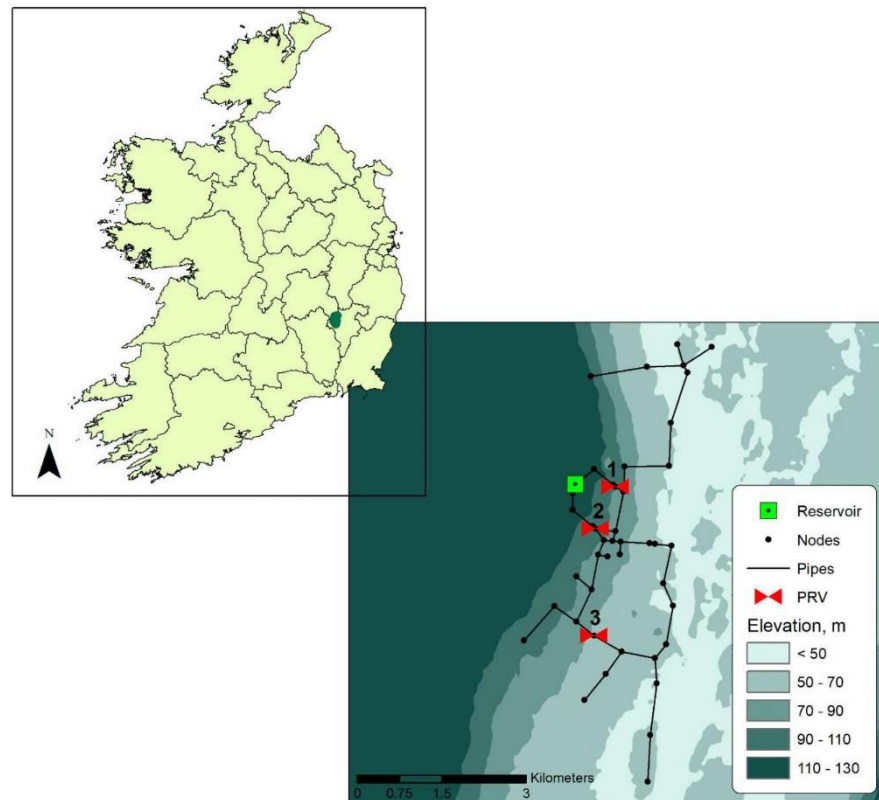


Figure 5. Ballinabranna water supply network layout.

3. Results

3.1. PRV Cost Model

The cost of 110 PRVs from different manufacturers were collected. The diameter of the selected PRVs ranged from 50 mm to 600 mm while the pressure setting range was 6–60 m, 15–60 m, 10–100 m, 10–160 m, 14–72 m, 40–120 m, 50–120 m, 10–70 m, 17–86 m and 7–70 m. The average PRV cost was 3516 € with a minimum value of 162 € for a 50 mm diameter and a maximum price of 31,765 € for a 500 mm diameter (see Figure 6). As could be expected, the greater the diameter the higher the PRV cost. However, a clear difference between the range of the pressure setting and the PRV cost was not detected. The differences in the PRV cost were mainly related to the different manufacturers who provided the data.

To estimate the total cost, the cost of the two required gate valves, the strainer and the air valve were also considered (Table 1). The cost of the two gate valves ranged from 146 € to 6370 € for diameters of 50 mm and 600 mm respectively, whereas the strainer cost varied from 148 € to 18,063 € for the same diameters. As for the air valve, the cost ranged from 543 € for the 50 mm diameter to 19,286 € for the 600 mm diameter.

These additional costs and the amount of civil works were added to the PRV cost to estimate the total installation cost. Considering an average PRV cost for each diameter according to the values provided by the different manufacturers, the average total cost was 18,762 €, ranging from 1738 € for a 50 mm diameter pipeline to 72,621 € for a 600 mm diameter (see Figure 7). When the minimum PRV

cost for each diameter was taken into account the average total cost was 16,931 €, varying from 1099 € to 72,621 € for diameters of 50 mm and 600 mm. As for the total cost when the maximum PRV cost for each diameter was selected, the average value was 21,258 € ranging from 2520 € to 72,621 € for diameters of 50 mm and 600 mm.

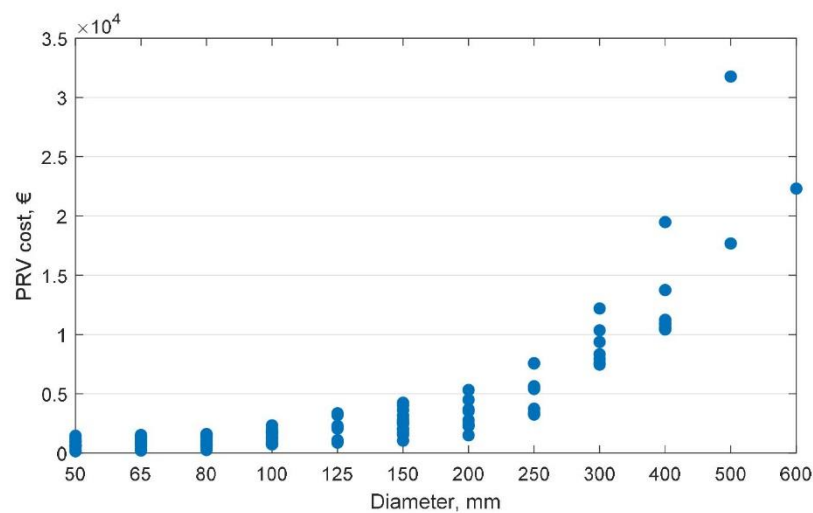


Figure 6. PRV cost for diameters ranging from 50 mm to 600 mm.

Table 1. Additional costs related to the PRV installation.

	Diameter, mm											
	50	65	80	100	125	150	200	250	300	400	500	600
Gate valve (2)	146	190	218	274	374	514	800	1216	1676	2899	4462	6370
Strainer	148	159	208	227	410	391	717	1377	2104	5331	11,669	18,063
Air valve	543	543	615	854	1215	1290	2520	4430	4947	8953	13,624	19,286
TOTAL	837	892	1041	1355	1999	2195	4037	7023	8727	17,183	29,756	43,719

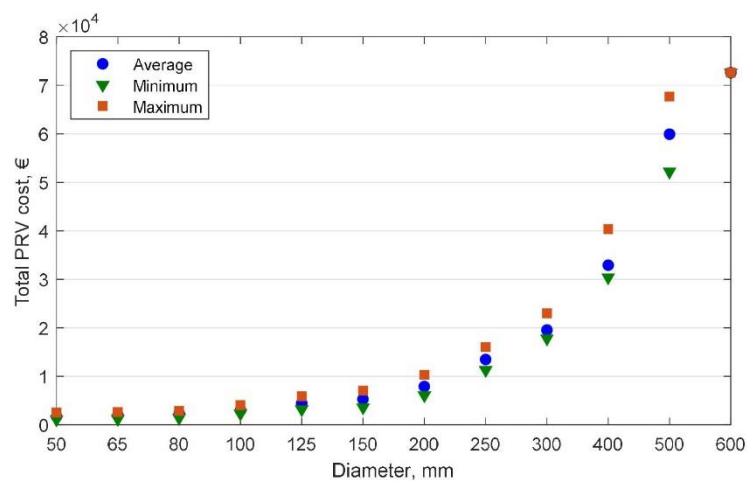


Figure 7. Total PRV cost for diameters ranging from 50 mm to 600 mm.

3.2. PAT Cost Model

The model developed by [53] was used to determine the PAT cost. A flow range between $0.01 \cdot 10^{-3} \text{ m}^3 \text{ s}^{-1}$ and $0.4 \text{ m}^3 \text{ s}^{-1}$ and pressure values varying from 5 m to 100 m were considered. The average PAT cost was 18,666 €, ranging from 1039 € for the minimum values of flow and pressure to 51,908 € associated with a flow of $0.4 \text{ m}^3 \text{ s}^{-1}$ and a pressure of 100 m (see Figure 8).

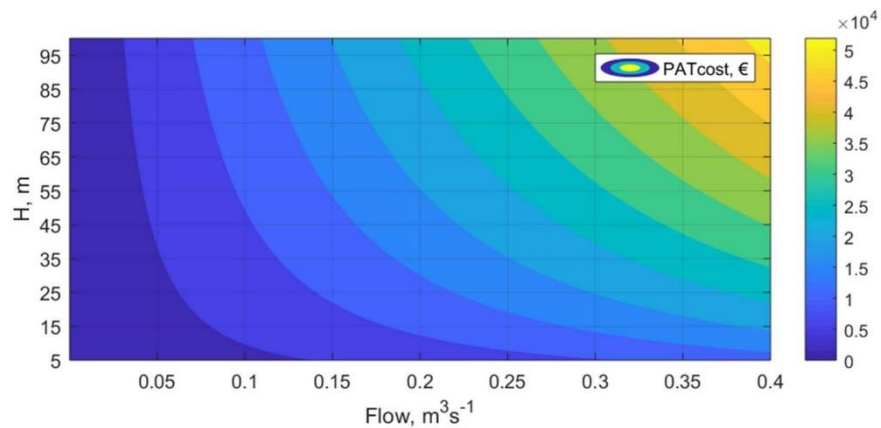


Figure 8. PAT cost according to flow and head.

The analysis of the cost components from the 9 evaluated PAT schemes according to the categories listed in Section 2.3, did not lead to any identifiable trends according to the nominal power of the scheme. Instead, it is reasonable to assume that a number of site-specific conditions (e.g., amount of civil works required, distance from the electric grid, and effectiveness of the design) resulted in varied percentages of each analysed cost category. The overall results of the analysis are shown in Figure 9, where the vertical error bars indicate the minimum and maximum percentages for each cost category.

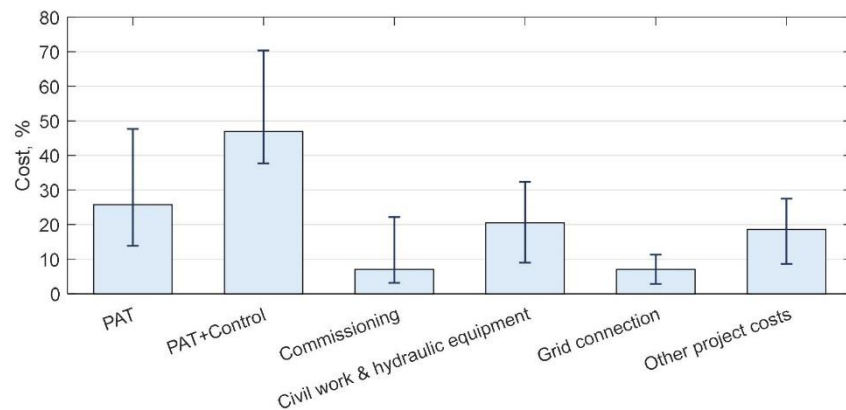


Figure 9. Costs involved in the PAT installation.

The results from Figure 9 showed that across the 9 analysed schemes the purchase of the PAT and generator accounted for 26% of the total cost of the installation on average, ranging from 14% to 48% (see Figure 9). Hence, the results from Equation (8) were divided by a factor of 0.26 in order to achieve

a realistic estimation of the total expected cost for a PAT-based energy recovery scheme according to its mean operating conditions (Figure 10).

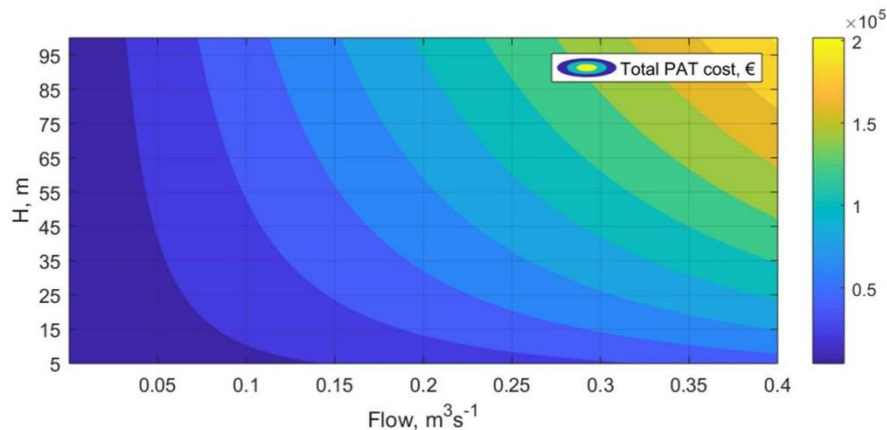


Figure 10. Total PAT cost according to flow and head.

Apart from the turbine and generator, the PAT control devices (with an average cost share of 24%) the civil work and hydraulic equipment (involving an average of 21% of the total cost) and the other project costs (accounting for 19%) were found to be the most expensive elements in a typical PAT installation.

Considering the PAT cost as 26% of the total cost, the total PAT installation cost was estimated for the flow range $0.01 \cdot 10^{-3} \text{ m}^3 \text{ s}^{-1}$ – $0.4 \text{ m}^3 \text{ s}^{-1}$ and pressure values from 5 m to 100 m. The average total PAT cost was 72,518 €, varying from 4035 € to 201,662 €.

When the PAT cost was assumed as 48% of the total cost, the average PAT installation cost was 39,214 €, with a minimum cost of 2182 € for the lowest values of flow and pressure, and a maximum cost of 109,050 €. Taking the PAT cost as the 14% of the total cost, instead, the average total cost was 134,288 €, varying from 7473 € to 373,436 €.

3.3. Determination of the Most Feasible Pressure Reduction Device, PRV or PAT, in the Case Studies

For case study 1, only average flow and head data were available. Hence, the generated power was determined considering that the average values matched with the flow and head of the PAT at the BEP. 140 sites out of 156 showed an average power higher than the minimum threshold value (400 W), with an average value of 4338 W, ranging from 460 W to 31,455 W (see Table 2). As for the total installation cost, the PRV cost was related to the pipe diameter while the PAT cost varied according to the generated power (see Figure 11).

Table 2. PRV/ PAT analysis in case study 1.

	Power, kW	Total PRV Cost, €	Total PAT Cost, €			PAT NPV, €			PAT Payback Period, years		
			26% ¹	48% ²	14% ³	26% ¹	48% ²	14% ³	26% ¹	48% ²	14% ³
Ave.	4338	7236	10,397	5567	20,831	57,960	61,489	59,163	4	3	6
Max.	31,455	32,909	60,770	32,862	112,534	451,467	467,421	421,875	10	7	10
Min.	460	5284	4761	2575	9097	2695	4197	5079	2	2	3

¹ Considering the PAT cost as 26% of the total cost. ² PAT cost as 48% of the total cost. ³ PAT cost as 14% of the total cost.

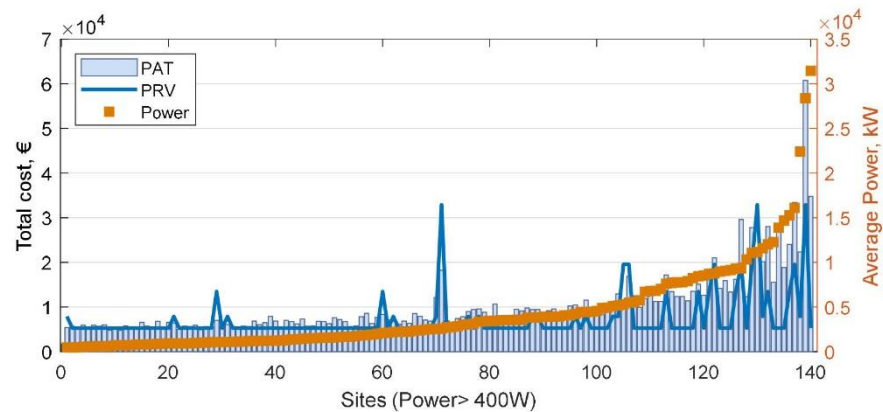


Figure 11. PAT vs. PRV total cost according to the average power at the evaluated sites.

Considering in this case the average total cost of both PRV and PAT, the PRV was the less expensive option in the majority of the sites, with an average total cost of 7236 € (see Table 2). In only 15 sites (10%) the PAT showed lower capital costs. Analysing these sites, the PAT total cost was slightly cheaper in 6 sites with a 150 mm diameter and power lower than 1 kW. When the generated power was higher the PRV was the best option considering this diameter. As for the 200 mm diameters, the PAT was the best option in 3 sites with an average power lower than 2300 W. When the generated power at the sites with a 200 mm diameter was higher, the PRV was cheaper. A similar tendency was found for pipes of 250, 300 and 400 mm. The lower the generated power the higher the probability that the PAT would be less expensive (see Figure 11).

As for the economic analysis in case study 1, only the energy cost saving was estimated since information about the effects of the PRV performance on nodes pressure was not available. The NPV was determined considering the energy that could be generated in each evaluated site during the PAT lifespan (15 years). Considering that the PAT cost was 26% of the total cost, the NPV showed positive values in all the 140 sites. The estimated average NPV was 57,960 €, ranging from 2695 € to 451,467 € (see Figure 12). The average payback period was 4 years (see Table 2). Under this assumption, three sites showed payback periods higher than the threshold established (see Figure 13).

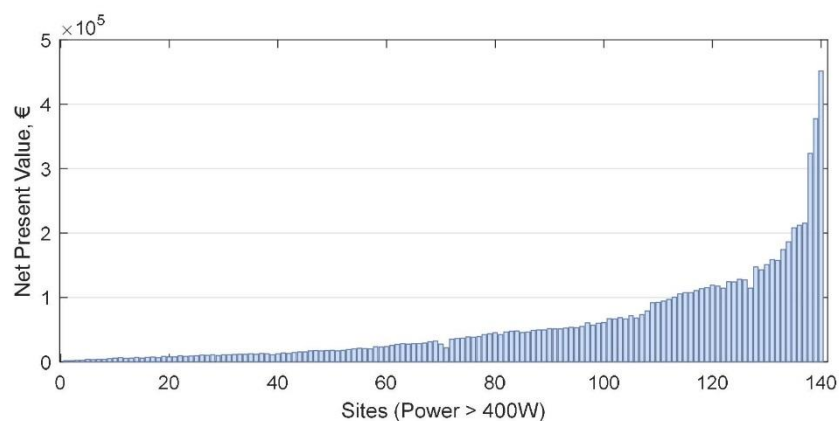


Figure 12. Net present value in the evaluated sites considering that the PAT represents 26% of the total installation cost.

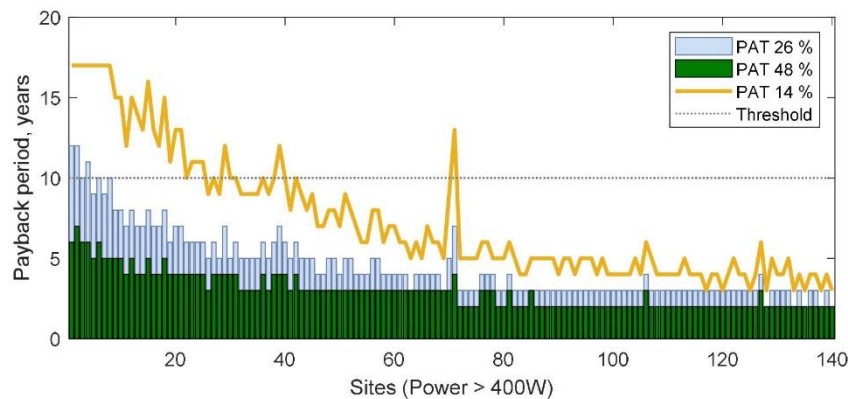


Figure 13. Payback period in the evaluated sites.

When the PAT cost was considered as 14% of the total cost, 113 out of the 140 sites with power higher than 400 W showed a payback period lower than 10 years, with an average payback period of 6 years (see Figure 13). As for the NPV, under this assumption the average value was 59,163 €, varying from 5079 € to 421,875 € (see Table 2). Finally, when the PAT cost was assumed as 48% of the total cost, the payback period was lower than 10 years in all 140 sites with average power higher than 400 W (see Figure 13), showing an average payback period of 3 year. In this case, the average NPV was 61,489 € with a minimum and a maximum value of 4197 € and 467,421 € (see Table 2). Considering the evaluated sites in case study 1, the PAT installation costs could be even lower since a current PRV is already operating, making the PAT installation as a retro-fit, more feasible.

For case study 2, the replacement of the three PRVs with PATs would involve powers at the BEP of 1223 W, 1533 W and 457 W (see Table 3). As for the total installation cost, the PRV showed the lowest values in the majority of the cases. The PAT installation cost was only lower at site 1, with a diameter of 150 mm, when the PAT cost was assumed as the 48% of the PAT total cost. The average installation cost ranged from 3009 € to 5284 € when the PRV was considered and from 5323 € to 6268 € for the PAT installation.

Table 3. PRV/ PAT analysis in Ballinabranna Group Water Scheme (BGWS).

Site Number	PRV			PAT		
	1	2	3	1	2	3
Diameter, mm	150	100	100	150	100	100
Power BEP, W				1223	1533	457
Energy, kWh year ⁻¹				7346	9203	2744
Ave cost, €	5284	3009	3009	5919	6268	5323
Min cost, €	3574	2303	2303	3201	3390	2879
Max cost, €	7074	4067	4067	10,984	11,633	9879
Water saving, m ³ year ⁻¹	15,559	15,559	15,559	15,559	15,559	15,559
Water cost saving, € year ⁻¹	4668	4668	4668	4668	4668	4668
Energy cost saving, € year ⁻¹				1249	1565	466
Payback period, year	3.0	2.0	2.0	3.0	3.0	3.0
NPV, €	43,164	45,440	45,440	55,493	58,420	47,967
NPV/Water saving, € m ⁻³	0.18	0.19	0.19	0.24	0.25	0.21

The economic analysis in case study 2 was carried out considering both the energy cost saving and the water saving as a result of the PRV/PAT operation. As initial leakage rate, 80% of the minimum night-time flow was assumed [36,58]. The same pressure drop was considered for the PRVs and the

PATs. Hence, the water saving and the water cost saving showed the same values for both devices: $15,559 \text{ m}^3 \text{ year}^{-1}$ and 4668 € year^{-1} per pressure reducing device (see Table 3). The operation of the three PATs would involve an annual energy cost saving of 1249 €, 1565 € and 466 €. Thus, the NPV considering the PAT lifespan and the average installation cost showed values of 55,493 €, 58,420 € and 47,967 € at sites 1, 2 and 3, respectively. This was 29%, 29% and 6% higher than the estimated NPV considering the average PRV installation cost. The ratio between the NPV and the water leakage that could be saved during the PAT/ PRV lifespan offered values for the PATs of 0.24 € m^{-3} , 0.25 € m^{-3} and 0.21 € m^{-3} , again higher than the estimated values for the PRVs (see Table 3).

Therefore, this work shows that although the required investment for a PRV installation is lower in most cases, the energy and hence, the cost savings that could be achieved by considering a PAT justify its installation. While both devices are equally efficient in terms of leakage reduction, the additional energy cost savings achieved by the PAT performance make its installation as a device to reduce leakage and generate energy more feasible.

4. Discussion

PATs are considered to be a cheaper technology compared to traditional turbines for small hydropower energy recovery [26,59]. However, information related to the total PAT cost, also including installation cost, is not easily accessible. Overall, methodologies focused on the use of PATs took into account a PAT cost according to the generated power. Thus, [60] considered an average PAT investment cost of 545 € kW^{-1} for generated power lower than 10 kW, including the PAT cost and the civil, electrical and electronic equipment. [25] determined the total installation cost considering an average PAT cost related to the generated power of 220 € kW^{-1} plus two additional costs of 450 € and 2500 € (associated to piping and grid connection). Other methodologies established the installation cost either as a percentage of the total installation cost or as a cost associated to the generated power. [30] considered the installation cost as 65% of the total cost, whereas [23] proposed an installation cost of 350 € kW^{-1} .

The methodology presented in this work analysed the installation cost and categories of 9 MHP plants in different countries, thus determining an average installation cost and the range. The installation cost determined in the sites of case study 1, using the methodology presented in this work compared with previous methodologies, showed higher total PAT costs using the values obtained from the 9 MHP plants. Thus, the average total PAT cost at the sites in case study 1 was 3904 €, 2364 €, 4159 € and 4251 € applying the methodologies proposed by [23,25,30,60], respectively. These were all considerably lower than the average total value presented in this work, 10,397 €.

Above all, the model presented here shows that when the total costs and benefits of the installation of PATs or PRVs as pressure reduction devices are compared, the PAT is the more feasible option with greater long term benefits. These greater long term benefits were also consistently found when a sensitivity analysis of the assumed variables was conducted (i.e., using min, average and maximum values for percentage contribution of differing cost components).

5. Conclusions

A model to determine the most feasible device, PRV or PAT, considering all the costs involved in a PRV and a PAT installation, as well as the water cost savings associated to leakage reduction, and the energy cost saving related to the PAT operation has been proposed. This work highlights the potential of MHP installations in general and PATs in particular as a tool to improve the sustainability of water distribution networks and reduce electricity costs. The analysis carried out showed that a wider database with the real PAT installation cost would be required to better estimate the feasibility of the installation of these devices. Limitations of the current methodology include: The limited amount of information available on PAT costs at very low powers; the lack of significant amounts of case studies on which to base estimates of civil works and other project costs; and the lack of information on the cost and performance of PAT types other than centrifugal.

The main finding of the work is that the use of a PRV over a PAT as a pressure reduction tool is not the most economically viable or sustainable option. While installing a PRV reduces leakage and thus energy consumption, it wastes the opportunity to recover electricity in the network. The development of less expensive turbines and/or pump as turbines adapted to work under low flow-head condition will tip the balance toward the installation of these devices even more.

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3.5. THE DEVELOPMENT OF A DECISION SUPPORT SOFTWARE FOR THE DESIGN OF MICRO-HYDROPOWER SCHEMES UTILIZING A PUMP AS TURBINE



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The Development of a Decision Support Software for the Design of Micro-Hydropower Schemes Utilizing a Pump as Turbine [†]

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Abstract: Pumps As Turbines (PATs) are a class of unconventional hydraulic turbines consisting of standard water pumps working in reverse mode as the prime mover. Such devices can be well suited for either in-pipe energy recovery or small-scale hydropower, but their practical application is hampered by the lack of comprehensive guidelines able to assist the designer in the determination of the optimal plant layout and the choice of equipment. In fact, the performances of a PAT will depend on factors such as its construction type, its size and the flow conditions under which the machine is expected to operate. Ultimately, the design of a PAT-based hydro scheme is a matter of trade-offs which are in most cases not trivial. An innovative software was developed in order to assist hydro designers and provide a visual aid when choosing between different layouts of the analyzed hydro scheme (e.g., more than one PAT in series/parallel, different shaft speeds), and has been applied to a real case study of energy recovery in a water network.

Keywords: decision support system; energy recovery; micro hydropower; Pumps as Turbines (PAT)

1. Introduction

Pumps as Turbines (PAT) are a class of hydraulic energy converters consisting of standard water pumps working in reverse mode as turbines. Being mass manufactured, they distinguish themselves from classic hydraulic turbines for having a lower purchase price, being available down to small power ratings, and being inexpensive to maintain [1–3].

These are ideally suitable for in-pipe energy recovery in water networks, irrigation or process industry, and belonging to the class of reaction turbines they allow the fluid to hold a residual pressure at outlet which is fundamental to maintain the integrity of the network. Besides, water pumps are vastly known and adopted pieces of equipment that can be found in any water network, therefore network operators are generally familiar with the technology and its maintenance aspects. Finally, due to their robustness and availability, PATs have been considered an appropriate technology for Micro-Hydropower schemes in developing countries [1,4,5].

One of the major limitations of the use of PATs is their generally low part-load efficiency, which makes the choice of the equipment and plant layout for a determined site critical. This is particularly true for drinking water or irrigation networks, which commonly experience marked fluctuations of water demand daily and seasonally [3].

Another limitation is the uncertainty about the PAT performances, since pump manufacturers tend not to divulge the characteristic charts of their units when operated as turbines. Such uncertainties in the location of the Best Efficiency Point (BEP) of PATs are intrinsic to the act of

utilizing an off-the-shelf piece of equipment for a use not originally envisaged by its maker. The effect of such uncertainties can be multiple, ranging from a lower energy yield than the expected power output up to failing to maintain the desired backpressure at the turbine's outlet.

Another important aspect to be considered is that choosing the most appropriate PAT for a determined site is a matter of non-trivial tradeoffs between ideal targets which are often contrasting. The typical design targets can be of technical or economic nature:

- technical aspects
 - maximization of the energy yield over a set period of time;
 - maximization of the power output;
 - maintenance of an adequate backpressure at the turbine's outlet;
 - maximization of CO₂ savings over a set period of time;
 - mitigating the risk of pressure surges under runaway conditions, which occur if the connection to the grid is temporarily lost during turbine operations;
 - space constraints in presence of physical obstacles to the installation.
- economic aspects
 - maximization of the Net Present Value (NPV) of the project;
 - minimization of the project payback time.

A practical example of contrasting design parameters is when a pump is selected to operate as PAT in order to have as short payback time as possible, which would most likely lead to the selection of an undersized PAT with respect to other models which may have resulted in a higher energy yield and CO₂ savings. Ultimately, the selection of a PAT is a problem of multi-objective nonlinear optimization. Since the priorities and the relative weight of the above mentioned parameters may be relative to specific cases, the aim of the software is to enable the designer to select any parameter relevant to their needs and for each of them identify on a chart the most appropriate families of commercially available pumps. Besides, the choice of giving a visual output also enables the software user to assess the convenience of selecting a different equipment layout (e.g., more than one PAT in parallel or in series).

2. Architecture of the Software

The key concept around which the software revolves is that of Best Efficiency Point (BEP), which corresponds to the set of hydraulic working conditions in terms of flow rate and head drop across a machine that allow the same unit to work under the highest possible hydraulic conversion efficiency. The software generates a 2D contour chart in which each point represents a hypothetical turbine whose BEP corresponds to the Q (m³/s) and H (m) coordinates shown on the horizontal and vertical axis respectively. For each hypothetical turbine the hydraulic behavior and purchase cost are computed, and its performances are evaluated against the provided flow rate and available head profiles. The logic of the software is schematically represented in Figure 1.

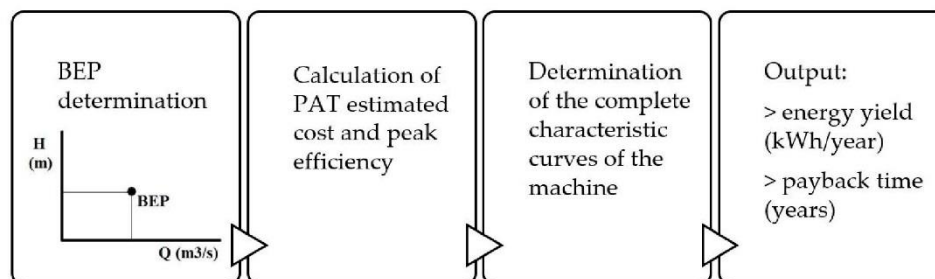


Figure 1. Basic layout of the software logic.

2.1. PAT Cost Model

A recent research proposed a set of equations able to predict the purchase price of PATs connected to induction generators having 1, 2 or 3 pairs of magnetic poles (pp) [6]. Three equations have been proposed, based on the values of Q (m³/s) and H (m) at the BEP of a PAT:

For radial PATs, pp = 1

$$C_{PAT+gen} (\text{€}) = 11,589.32 Q_{BEP} \sqrt[3]{(H_{BEP})} + 1380.79 \quad (1)$$

For radial PATs, pp = 2

$$C_{PAT+gen} (\text{€}) = 12,864.77 Q_{BEP} \sqrt[3]{(H_{BEP})} + 949.93 \quad (2)$$

For radial PATs, pp = 3

$$C_{PAT+gen} (\text{€}) = 15,484.97 Q_{BEP} \sqrt[3]{(H_{BEP})} + 1172.72 \quad (3)$$

The above equations have been integrated in the PAT design software, since they allow an easy calculation of the PAT + generator purchase price for a range of nominal shaft speed N of 3020, 1510 and 1005 RPM which correspond respectively to the speed of asynchronous generators having 1, 2 and 3 pairs of poles.

2.2. Efficiency Prediction

The magnitude of peak hydraulic efficiency at BEP of a centrifugal PAT has been found to vary according to its nominal flow and head characteristics in term of Q (m³/s), H (m) and N (RPM) at the BEP [7]. An equation has been proposed to predict the efficiency at the BEP of any machine based from experimental data of 280 radial PATs:

$$\eta_{MAX} = 0.89 - (0.024/Q_{BEP}^{0.41}) - 0.076 (0.22 + \ln (Ns/52.933))^2 \quad (4)$$

The term Ns in Equation (4) refers to the specific speed of a PAT defined as:

$$Ns = N ((\sqrt[3]{Q_{BEP}})/(H_{BEP}^{0.75})) \quad (5)$$

where N is in RPM, Q_{BEP} in m³/s and H_{BEP} in m. The nominal power output of a PAT P (kW) can therefore be calculated as:

$$9.81 \eta_{MAX} Q_{BEP} H_{BEP} \quad (6)$$

2.3. Prediction of the BEP of a PAT

Several methods exist to predict the BEP location of a PAT with respect to its known BEP as a pump. The most reliable is the physical lab test of pumps in specialized test rigs, which in contrast is very expensive and resource-demanding. Another option is Computational Fluid Dynamics (CFD) simulations, which can give accurate results but require a powerful computational capacity and need an extensive knowledge of the internal geometry of the machine [8].

Instead, several 1-D algebraic methods exist which are capable of predicting the location of the BEP in turbine mode from the BEP in pump mode [9]. They require minimal data, typically only the Q , H coordinates of the BEP in pump mode, the machine rated efficiency and the nominal shaft speed. The 1-D method developed by Yang et al. (2012) has the advantages of being developed from a large database of PATs [10]. It consists of the set of equations described in Equations (7) and (8).

$$h = H_{t,BEP}/H_{p,BEP} = 1.2/(\eta_{p,BEP})^{1.1} \quad (7)$$

$$q = Q_{t,BEP}/Q_{p,BEP} = 1.2/(\eta_{p,BEP})^{0.55} \quad (8)$$

In this analysis the peak efficiency of a pump has been considered equal to that of the very same unit in turbine mode, as suggested in the literature [11,12].

A database of BEP of pumps from several manufacturers has been gathered, which included standardized centrifugal pumps from 5 manufacturers. Only units with full size impellers were selected. The units are displayed in the charts, according to their predicted BEP location in turbine mode obtained through Equations (7) and (8).

2.4. PAT Curve Extrapolation from BEP

In the literature some second and third-degree polynomials are available which allow to draw the approximate Q-H and P-H characteristic curve of a PAT only knowing its Q, H and P (or η , based on Equation (6)) at the BEP. An example of such curves is shown in Figure 2, where the point of coordinates (1,1) represents the BEP.

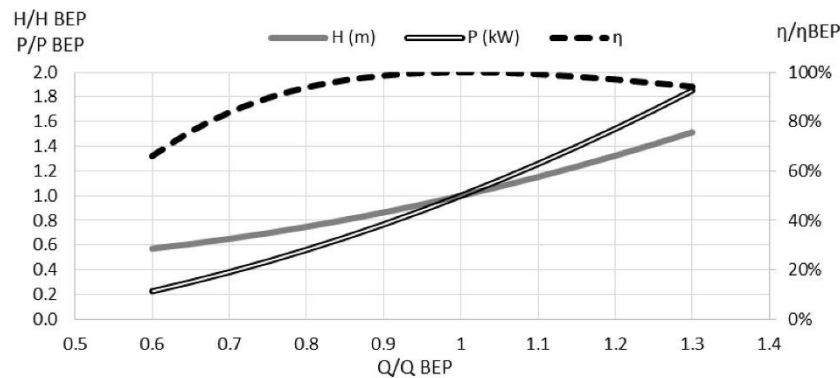


Figure 2. Typical Q-H and Q-P characteristic curves of a PAT relative to its BEP.

Several authors have proposed empirical equations to perform this task [12–14]. The most recent set of equations has been published based on the characteristic curves of 12 radial PATs experimentally tested [15]. For each machine the head and power curves relative to the BEP have been interpolated to second and third degree polynomials respectively. Subsequently, the average values of each coefficient of the relative head and power polynomial has been obtained as shown in Equations (9) and (10).

$$H_t/H_{t,BEP} = 0.922 (Q_t/Q_{t,BEP})^2 - 0.406 (Q_t/Q_{t,BEP}) + 0.483 \quad (9)$$

$$P_t/P_{t,BEP} = 0.040 (Q_t/Q_{t,BEP})^3 + 1.185 (Q_t/Q_{t,BEP})^2 - 0.043 (Q_t/Q_{t,BEP}) - 0.183 \quad (10)$$

3. Limits of the Analysis

In order to predict realistically the performance and cost of a continuous series of hypothetical PATs the model relies on a series of semi-empirical mathematical correlations whose applicability constraints and uncertainties consequently limit the results provided by the software.

Besides, although several pump—and therefore PAT—design families exist (e.g., multistage, submersible, double suction split-case, lobe pumps) by far the most common pump type corresponds to the centrifugal or radial units [16]. Such category of pumps is not only the most easily available and sold worldwide, but also the one whose behavior as turbine has been studied more in depth. Therefore, the present version of the software focuses solely on centrifugal PATs. At the same time, it has been chosen for the first version of the software to consider uniquely the use of asynchronous induction generators which are the most chosen option for turbines in the Micro-Hydropower range. The asynchronous motor/generator units are classified into four efficiency classes; IE1 the lowest and IE4 the highest [17]. Since standard hydraulic pumps being sold in Europe come with IE3-class motor/generators, such type of machines have been considered in the analysis.

The permissible operating range of each PAT was considered as a precautionary measure to extend from 0.7 to 1.2 of Q/Q_{BEP} in order to ensure an acceptable reliability [18]. Besides, the value of N_s for each PAT was limited to the range 6–70 because of all of the units from which Equations (9) and (10) were derived fall within that interval [15].

4. Case Study and Results

A research from the University of Trento (Italy) has evaluated the opportunity for energy recovery downstream of an elevated tank from which water is distributed to the city of Merano [19].

4.1. Available Flow Rate and Hydraulic Head

The flow duration curve follows a daily profile as displayed in Figure 3, showing a peak flow rate more than double than the minimum flow occurring during night time. In the absence of more detailed information, the available hydraulic head has been considered as constant and equal to 60 m [19]. In reality, this value will change slightly according to the head losses in the piping.

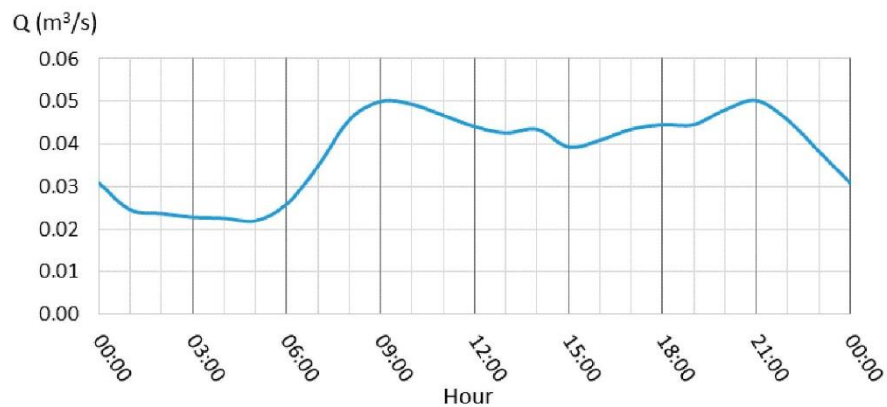


Figure 3. Recorded flow rate daily profile at the outlet of a water storage tank, adapted from [19].

4.2. Additional Input Data

The presence of an automated by-pass duct has been introduced, in order to guarantee the network reliability under all operational regimes. Therefore, a portion of water can be diverted through the bypass instead of the turbine at any given time. Additional inputs were:

- the rest of project costs were assumed to be equal to the 300% of the cost of PAT and generator calculated through Equations (1)–(3) [20];
- the revenues from energy sales to the national grid were evaluated as 210 €/MWh which is the feed-in tariff from in-pipe micro hydro in Italy at the time of this study [21];
- the yearly maintenance costs have been considered equal to the 5% of the PAT + generator cost [22].

4.3. Results Chart and Interpretation

In order to compare the results with those reported by the original study, a similar shaft speed was selected (3020 RPM). The results are shown in Figure 4.

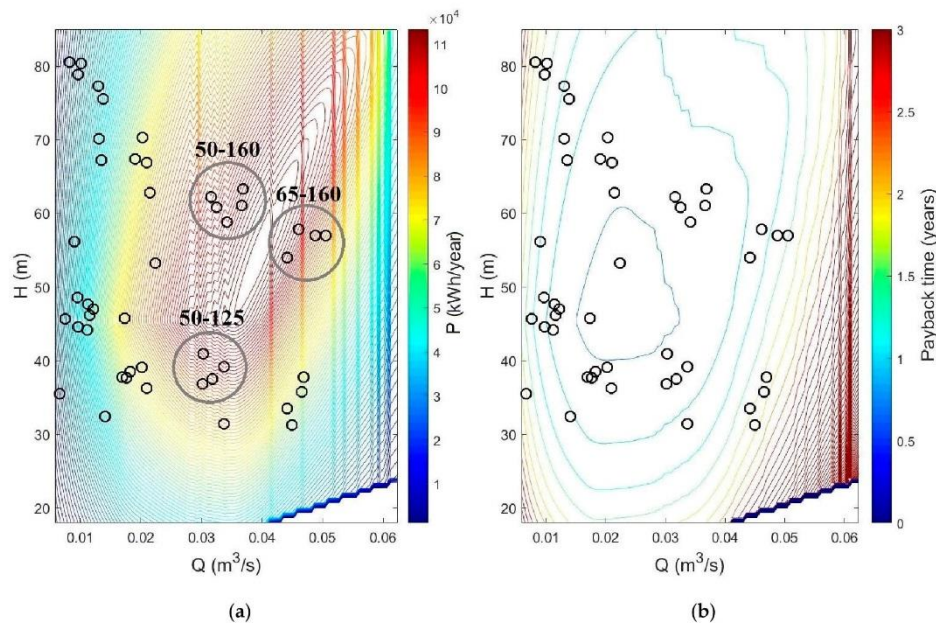


Figure 4. Algorithm results for the PAT downstream of Merano aqueduct: (a) yearly energy yield; (b) simple Payback time.

The chart of Figure 4a indicates that the maximum yearly energy yield that can be obtained by a PAT rotating at 3020 RPM in the selected location is equal to approximately 100 MWh/year which is close to the 123.4 MWh/year forecasted by the authors of the article [19]. The families of standardized centrifugal pumps most likely to achieve the highest energy output are the 50–160, 50–125 and 65–150. The first two digits correspond to the nominal size in mm of the pump discharge nozzle (PAT inlet nozzle) while the last three refer to the nominal diameter of the impeller in mm. Instead, Figure 4b shows that the payback of the investment would likely occur in less than three years with any PAT included in the range of the chart axis. However, the shortest possible payback time corresponds to a hypothetical turbine having nearly half of the nominal flow rate with respect to the ideal PAT that would maximize the energy yield as seen in Figure 4a.

5. Conclusions

The main scope of the proposed algorithm is the selection of PAT family and system configuration for any given potential site in order to meet the existing constraints and satisfy the stakeholder's expectations. The generated results can be useful for a designer in order to shortlist one or more families of commercially available PATs for determinate locations. The software has been tested against a previously investigated site for in-pipe energy recovery within a water network whose data was found in the literature. The results compared favourably with the expectations of the authors of the original study and provided useful information for a successful choice of equipment.

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Conflicts of Interest: The authors declare no conflict of interest.

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ANNEX 4: COMPILED PAT DATABASES

4.1. COMPARISON OF PAT PERFORMANCE PREDICTION METHODS

Pump mode				PAT mode					
Q (l/s)	H (m)	η_{BEP}	N (RPM)	η_{BEP}	D (m)	N (RPM)	H _{BEP} (m)	Q _{BEP} (l/s)	P _{BEP} (kW)
31.29	35.1	74%	2950	78%	0.174	1200	9.0	17.3	1.2
90	32.5	84%	1450	84%	0.35	1500	44.8	124.5	46.1
1	1.0	60%	402	56%	-	401.6	2.4	2.1	0.03
1539	9.8	88%	700	86%	0.686	730	13.2	1744.0	194.1
1766	9.0	85%	714	85%	0.686	725	13.2	2073.0	228.4
33.8	12.3	76%	2000	73%	0.177	2000	19.4	47.3	6.6
30.62	111.3	69%	3000	69%	0.325	3000	221.2	49.5	73.9
6.37	16.4	65%	2000	71%	0.18	2000	30.0	10.5	2.2
80.5	45.9	83%	2900	83%	0.215	2900	67.0	114.0	62.4
43.06	21.7	80%	1475	82%	0.262	1475	28.6	51.5	11.8
43.06	13.5	83%	1475	82%	0.219	1050	10.3	40.5	3.3
41.67	7.8	80%	1475	74%	0.185	1250	11.2	55.0	4.5
5.51	66.5	47%	3000	44%	0.2	3000	177.1	10.1	7.6
103.6	10.5	84%	1450	76%	0.224	1450	17.6	138.0	18.1
10.71	41.8	67%	3000	69%	0.16	3000	64.7	14.1	6.2
19.62	21.4	74%	3000	79%	0.125	3000	29.8	29.1	6.7
84.4	62.9	78%	1800	77%	0.356	1800	78.2	109.1	64.3
114.4	11.6	80%	1750	79%	-	1750	17.7	178.3	24.5
10.67	24.0	72%	3000	61%	-	3000	44.5	17.1	4.5
42.47	4.5	78%	1450	64%	-	1450	9.0	53.4	3.0
208.9	72.2	79%	4500	89%	-	4500	127.5	306.2	341.5
112	50.3	84%	1160	81%	0.505	837	30.3	112.0	26.9
52.53	6.5	84%	700	85%	0.312	578.2	6.3	50.0	2.6
17.35	44.7	79%	3100	67%	-	3100	69.4	24.9	11.3
183.2	102.1	89%	2500	91%	0.34	2500	128.7	267.3	305.1
210.7	87.4	92%	2133	87%	-	1933	90.9	224.6	173.7
195.5	98.1	90%	2500	90%	0.34	2500	116.2	268.3	273.8
1	1.0	81%	664	85%	-	664.4	1.5	1.4	0.0
1	1.0	70%	735	71%	-	734.6	1.7	1.3	0.0
1	1.0	89%	952	86%	-	813.8	0.9	1.1	0.0
60.06	33.7	81%	1500	74%	0.315	1510	46.5	83.2	27.9
297.5	127.6	91%	2940	89%	0.35	2940	182.0	369.4	583.1
349.4	20.0	87%	1640	79%	0.409	1607	23.2	424.4	76.6
327.6	8.8	87%	1640	78%	0.409	1507	10.7	413.8	34.0
36.1	33.8	82%	1750	73%	0.284	1750	48.5	50.5	17.5
52.1	9.2	76%	1199	74%	0.273	857	9.2	51.9	3.4
19.42	23.1	72%	2000	63%	0.2	2000	50.6	33.0	10.4

8.7	19.9	65%	1450	65%	0.25	1000	21.7	11.9	1.7
1.382	8.1	52%	2500	54%	0.1014	2500	29.1	2.8	0.4
20	15.4	73%	1425	57%	0.246	1450	24.2	25.4	3.5
1408	7.0	83%	665	69%	-	482	6.5	1419.0	62.3
12.53	32.3	62%	1450	52%	0.32	920	28.0	12.2	1.7
51.39	34.7	81%	1450	81%	0.334	1500	49.0	69.4	26.8
42.92	13.4	86%	1450	78%	0.219	1500	22.1	56.2	9.5
1	1.0	84%	867	86%	-	866.8	1.3	1.3	0.0
90.4	5.6	83%	1760	79%	0.21	1173	3.5	90.4	2.4
51.73	0.8	75%	970	76%	0.203	609.2	0.5	51.7	0.2
8.54	4.3	68%	1060	80%	-	1060	8.0	14.2	0.9
2377	25.9	84%	592	73%	1.372	592	40.8	2940.0	852.4
38.48	30.2	84%	1480	78%	0.318	1510	40.8	48.2	15.0
40.94	11.6	87%	1485	75%	0.216	1525	24.6	64.9	11.7
86.1	20.1	78%	1450	78%	0.28	1550	34.0	120.5	31.3
2.226	21.2	43%	2900	47%	0.134	3100	55.9	3.8	1.0
9.86	25.9	57%	2850	52%	0.184	3000	40.3	18.7	3.8
25	12.8	69%	2850	67%	0.14	3000	27.1	44.8	8.0
1	1.0	84%	2070	77%	-	2070	1.3	1.3	0.0
6.94	35.1	64%	3500	65%	0.127	3550	61.4	11.5	4.5
6.1	29.0	54%	2900	46%	0.176	1520	20.0	4.1	0.4
176.96	18.5	87%	725	88%	0.518	760	23.0	242.5	48.1
16.25	20.3	70%	1450	70%	0.258	1510	30.5	19.4	4.0
16.47	11.5	74%	1480	76%	0.203	1520	19.8	22.2	3.3
18.06	19.5	78%	2900	76%	0.14	3020	36.6	25.1	6.8
45.36	21.0	79%	1450	84%	0.269	1450	24.4	45.7	9.2
45.47	13.2	82%	1450	85%	0.219	1450	17.4	50.2	7.3
41.55	7.7	80%	1450	71%	0.185	1450	12.9	59.6	5.4
72.51	5.3	84%	1050	75%	0.224	1050	8.4	94.9	5.9
17.00	45.4	74%	2900	74%	0.2	2900	67.0	21.0	10.2
41.11	39.0	79%	2900	66%	0.19	2900	61.0	50.0	19.8
15.00	3.9	54%	1450	49%	0.12	1450	4.6	22.0	0.5
13.89	10.0	76%	1450	76%	0.19	2900	60.9	42.22	19.1
41.11	39.0	79%	2900	83%	0.189	2900	72.3	60.3	35.5
24.58	22.0	77%	2900	86%	0.146	2900	28.6	30.1	7.3
7.39	33.0	44%	1450	43%		1450	93.3	13.1	5.1
7.52	31.4	45%	1450	35%		1450	110.8	14.1	5.4
6.97	20.0	55%	1450	51%		1450	43.7	10.7	2.3
5.28	12.0	55%	1450	59%		1450	25.5	9.7	1.4
16.50	19.3	65%	1450	65%		1450	38.0	26.0	6.3
26.77	19.6	73%	1450	73%		1450	33.2	40.3	9.6
9.72	8.5	67%	1450	73%		1450	13.1	15.3	1.4
24.16	14.5	74%	1450	78%		1450	22.4	36.5	6.3
23.19	12.1	72%	1450	76%		1450	17.6	31.2	4.1
41.67	12.9	76%	1450	84%		1450	18.8	50.0	7.7

57.93	9.6	82%	1200	84%		1200	13.3	84.3	9.2
34.95	5.3	78%	1200	70%		1200	7.8	43.6	2.3
10.80	14.5	77%	1500	73%	0.225	1500	0.0	0.0	0.0
26.50	21.5	78%	1500	77%	0.258	1500	0.0	0.0	0.0
25.40	12.8	79%	1500	81%	0.206	1500	0.0	0.0	0.0
15.27	8.4	74%	1450	72%	0.174	1450	0.0	0.0	0.0
65.90	19.8	85%	1450	84%	0.264	1450	0.0	0.0	0.0
33.00	10.5	80%	1450	80%	0.2	1450	0.0	0.0	0.0
13.50	5.6	76%	1450	76%	0.139	1450	0.0	0.0	0.0
28.90	6.4	72%	1450	74%	0.165	1450	0.0	0.0	0.0
103.00	10.6	84%	1450	76%	0.224	1450	0.0	0.0	0.0
251.40	21.5	84%	1450	80%	0.338	1450	0.0	0.0	0.0
101.40	8.3	83%	1450	83%	0.236	1450	0.0	0.0	0.0
4.48	8.1	64%	1450	60%	0.176	1520	19.6	5.2	0.6
4.00	13.7	56%	1450	51%	0.209	1520	34.0	6.0	1.0
0.50	4.3	33%	2800	33%	0.01	1100	2.8	0.7	0.0
8.00	54.5	58%	2900	46%	0.209	3020	137.0	13.5	8.3
4.30	21.0	48%	1450	32%	0.26	1520	70.0	8.0	1.8
6.02	13.7	65%	1450	56%	0.209	1520	32.0	10.0	1.8
7.10	20.8	57%	1450	51%	0.26	1520	50.0	11.2	2.8
24.10	19.4	82%	2900	78%	0.142	3020	32.0	30.8	7.5
12.00	4.8	79%	1450	78%	0.142	1520	8.0	15.5	0.9
22.00	62.0	77%	2900	69%	0.219	3010	94.8	33.4	21.4
11.70	20.3	68%	1450	64%	0.26	1520	40.0	16.8	4.2
15.80	5.4	82%	1450	78%	0.141	1520	6.8	22.1	1.1
17.1	14.8	80%	1450	80%	0.219	1525	23.0	23.0	4.1
17.2	21.6	75%	1450	78%	0.26	1520	32.0	24.0	5.9
23.3	32.5	73%	1450	69%	0.32	1520	52.0	30.0	10.5
28	14.5	85%	1450	80%	0.219	1520	23.9	38.0	7.1
32	22.5	81%	1450	80%	0.269	1520	35.0	38.0	10.4
38	35.5	80%	1450	70%	0.334	1520	55.2	52.0	19.6
26.8	21.6	74%	960	66%	0.404	1020	46.0	40.0	11.9
43	13.6	85%	1450	78%	0.219	1510	21.0	55.2	8.9
55	33.5	82%	1450	73%	0.334	1515	57.0	75.0	30.6
55	50.0	76%	1450	76%	0.404	1520	80.0	75.0	44.7
105	11.0	82%	1450	75%	0.224	1520	14.8	134.0	14.6
121	18.0	90%	1450	80%	0.269	1520	25.0	166.5	32.6

4.2. DEVELOPMENT OF A MODEL FOR PAT EFFICIENCY PREDICTION

n.	H _{BEP} (m)	Q _{BEP} (l/s)	N (RPM)	η _{BEP}	Ns
1	22.5	7.1	3020	0.564	24.6
2	26.7	7.6	3020	0.592	22.4
3	32.2	8.3	3020	0.642	20.3
4	40.2	9.2	3020	0.696	18.2

5	21.0	8.9	3020	0.602	29.0
6	26.6	8.8	3020	0.658	24.1
7	34.0	10.3	3020	0.696	21.7
8	42.6	11.1	3020	0.701	19.1
9	44.3	6.6	3020	0.547	14.3
10	53.5	7.7	3020	0.591	13.4
11	66.6	8.7	3020	0.606	12.1
12	74.7	9.0	3020	0.623	11.3
13	42.6	10.2	3020	0.593	18.3
14	50.4	10.4	3020	0.624	16.3
15	59.2	11.4	3020	0.655	15.1
16	70.4	12.5	3020	0.666	13.9
17	67.6	8.8	3020	0.521	12.0
18	85.1	9.6	3020	0.526	10.5
19	105.3	9.9	3020	0.545	9.1
20	109.7	10.0	3020	0.546	8.9
21	69.5	11.5	3020	0.583	13.4
22	79.2	13.0	3020	0.634	12.9
23	90.5	14.1	3020	0.663	12.2
24	117.0	15.3	3020	0.643	10.5
25	134.3	9.6	3020	0.428	7.5
26	162.7	10.8	3020	0.433	6.9
27	194.0	12.7	3020	0.449	6.5
28	221.3	13.8	3020	0.468	6.2
29	16.9	15.2	3020	0.7	44.7
30	22.3	16.4	3020	0.722	37.6
31	28.8	17.3	3020	0.75	31.9
32	33.8	17.8	3020	0.798	28.7
33	36.1	17.9	3020	0.811	27.4
34	48.6	13.9	3020	0.668	19.4
35	49.2	17.0	3020	0.764	21.2
36	59.3	18.6	3020	0.778	19.3
37	65.6	19.4	3020	0.778	18.3
38	61.5	16.4	3020	0.652	17.6
39	72.0	18.1	3020	0.66	16.4
40	86.3	21.8	3020	0.679	15.8
41	95.2	24.4	3020	0.693	15.5
42	100.5	19.0	3020	0.671	13.1
43	123.7	20.7	3020	0.676	11.7
44	129.3	26.7	3020	0.667	12.9
45	147.2	23.2	3020	0.701	10.9
46	157.6	23.9	3020	0.702	10.5
47	266.0	19.2	3020	0.396	6.3
48	314.4	21.0	3020	0.409	5.9
49	351.5	23.1	3020	0.424	5.7

50	351.1	22.9	3020	0.485	5.6
51	24.2	21.2	3020	0.67	40.3
52	25.6	27.8	3020	0.717	44.2
53	31.3	29.2	3020	0.767	38.9
54	33.8	30.6	3020	0.815	37.7
55	36.4	27.4	3020	0.716	33.7
56	43.3	29.8	3020	0.742	30.9
57	53.2	32.9	3020	0.786	27.8
58	58.7	35.5	3020	0.818	26.8
59	90.0	26.4	3020	0.692	16.8
60	106.8	27.8	3020	0.699	15.2
61	115.3	30.3	3020	0.701	15.0
62	138.2	34.8	3020	0.71	14.0
63	147.8	35.4	3020	0.714	13.4
64	194.8	31.0	3020	0.573	10.2
65	224.1	33.3	3020	0.577	9.5
66	252.5	35.6	3020	0.582	9.0
67	292.6	37.8	3020	0.592	8.3
68	20.2	32.0	3020	0.736	56.7
69	23.0	42.1	3020	0.799	58.9
70	28.0	42.6	3020	0.829	51.2
71	33.0	44.9	3020	0.817	46.5
72	30.9	37.7	3020	0.813	44.7
73	40.2	41.1	3020	0.835	38.3
74	49.5	45.6	3020	0.862	34.6
75	52.6	47.0	3020	0.868	33.5
76	51.7	33.7	3020	0.742	28.8
77	64.4	36.7	3020	0.765	25.4
78	70.7	38.0	3020	0.799	24.2
79	78.0	41.7	3020	0.788	23.5
80	92.4	45.5	3020	0.821	21.6
81	94.9	46.0	3020	0.825	21.3
82	163.1	56.0	3020	0.722	15.7
83	191.1	63.4	3020	0.732	14.8
84	204.2	68.0	3020	0.739	14.6
85	223.9	74.8	3020	0.745	14.3
86	28.6	61.7	3020	0.754	60.6
87	33.8	68.1	3020	0.765	56.2
88	40.0	70.7	3020	0.797	50.4
89	43.8	73.4	3020	0.798	48.0
90	49.6	72.9	3020	0.848	43.6
91	54.0	62.6	3020	0.733	37.9
92	67.0	68.6	3020	0.755	33.8
93	77.5	70.9	3020	0.792	30.8
94	84.9	77.6	3020	0.795	30.1

95	92.6	81.0	3020	0.82	28.8
96	102.8	58.8	3020	0.789	22.7
97	114.4	65.4	3020	0.794	22.1
98	133.9	81.1	3020	0.806	21.8
99	145.0	88.4	3020	0.819	21.5
100	176.6	84.8	3020	0.742	18.2
101	191.8	88.0	3020	0.759	17.4
102	212.9	97.8	3020	0.759	16.9
103	236.1	105.1	3020	0.772	16.3
104	50.2	106.6	3020	0.717	52.3
105	54.9	108.6	3020	0.763	49.3
106	61.9	110.3	3020	0.812	45.5
107	71.7	115.2	3020	0.819	41.6
108	83.6	130.0	3020	0.833	39.4
109	80.7	97.4	3020	0.764	35.0
110	97.6	112.7	3020	0.816	32.7
111	109.9	114.0	3020	0.834	30.0
112	124.9	118.8	3020	0.841	27.9
113	132.5	124.9	3020	0.847	27.3
114	145.8	113.0	3020	0.761	24.2
115	162.5	121.7	3020	0.773	23.1
116	183.0	130.1	3020	0.786	21.9
117	207.7	143.5	3020	0.803	20.9
118	85.0	178.7	3020	0.781	45.6
119	95.6	187.7	3020	0.796	42.8
120	105.8	202.0	3020	0.811	41.2
121	118.2	219.7	3020	0.828	39.5
122	122.9	229.2	3020	0.835	39.2
123	132.6	145.4	3020	0.771	29.5
124	150.5	149.3	3020	0.799	27.2
125	170.7	165.6	3020	0.811	26.0
126	45.0	282.6	3020	0.791	92.3
127	49.6	264.7	3020	0.837	83.1
128	59.5	289.1	3020	0.841	75.8
129	77.6	267.5	3020	0.748	59.7
130	85.2	297.2	3020	0.78	58.7
131	96.4	320.1	3020	0.825	55.5
132	96.4	249.6	3020	0.848	49.1
133	109.9	270.5	3020	0.864	46.3
134	8.0	4.1	1520	0.614	20.3
135	9.5	4.4	1520	0.685	18.6
136	10.3	4.6	1520	0.69	17.8
137	6.4	4.4	1520	0.642	24.9
138	8.3	5.2	1520	0.679	22.3
139	10.5	5.6	1520	0.688	19.4

140	10.8	3.3	1520	0.563	14.5
141	13.0	3.8	1520	0.586	13.6
142	16.9	4.4	1520	0.583	11.9
143	19.1	4.5	1520	0.589	11.1
144	10.5	5.4	1520	0.546	18.9
145	12.9	5.8	1520	0.588	16.9
146	16.1	6.2	1520	0.626	14.8
147	16.7	6.5	1520	0.626	14.8
148	31.7	5.3	1520	0.432	8.2
149	45.1	6.1	1520	0.45	6.8
150	50.6	6.7	1520	0.468	6.5
151	52.3	6.9	1520	0.47	6.5
152	5.4	8.2	1520	0.718	38.9
153	6.8	8.6	1520	0.761	33.3
154	8.0	8.9	1520	0.811	29.9
155	26.3	10.9	1520	0.66	13.5
156	32.5	12.3	1520	0.665	12.3
157	37.7	12.8	1520	0.678	11.2
158	70.4	9.6	1520	0.393	6.1
159	82.9	10.7	1520	0.404	5.7
160	99.7	11.6	1520	0.42	5.2
161	109.1	12.4	1520	0.42	5.0
162	14.4	12.4	1520	0.747	22.8
163	17.3	13.4	1520	0.757	20.6
164	21.1	14.9	1520	0.807	18.7
165	23.5	16.4	1520	0.786	18.2
166	24.9	14.3	1520	0.698	16.2
167	30.1	16.0	1520	0.703	14.9
168	36.7	16.5	1520	0.719	13.0
169	6.7	17.5	1520	0.81	48.2
170	8.9	18.7	1520	0.822	40.0
171	11.0	21.1	1520	0.845	36.3
172	12.6	23.1	1520	0.867	34.2
173	13.6	18.0	1520	0.752	28.5
174	17.2	20.0	1520	0.778	25.2
175	20.0	20.4	1520	0.819	22.8
176	22.9	22.7	1520	0.822	21.7
177	23.7	23.0	1520	0.721	21.3
178	27.2	25.0	1520	0.729	20.0
179	32.5	28.3	1520	0.742	18.7
180	37.7	30.1	1520	0.754	17.2
181	36.5	25.6	1520	0.699	16.3
182	44.2	29.1	1520	0.703	15.0
183	53.4	33.5	1520	0.731	14.0
184	55.8	35.6	1520	0.732	14.0

185	24.2	35.7	1520	0.782	26.1
186	30.0	39.4	1520	0.797	23.4
187	35.6	40.5	1520	0.831	20.9
188	39.3	44.2	1520	0.754	20.2
189	48.3	48.1	1520	0.766	18.1
190	53.4	49.1	1520	0.773	16.9
191	58.2	49.9	1520	0.792	16.0
192	68.9	48.0	1520	0.654	13.8
193	75.8	51.3	1520	0.66	13.3
194	87.3	60.5	1520	0.672	13.0
195	97.9	66.4	1520	0.687	12.5
196	105.9	69.1	1520	0.704	12.0
197	8.1	44.1	1520	0.777	66.2
198	9.9	47.0	1520	0.816	58.6
199	11.0	47.9	1520	0.83	54.5
200	38.7	61.2	1520	0.764	24.1
201	44.0	65.1	1520	0.77	22.6
202	48.7	67.3	1520	0.783	21.2
203	54.6	72.9	1520	0.808	20.3
204	48.6	66.7	1520	0.738	21.2
205	65.1	74.1	1520	0.758	17.9
206	75.8	77.9	1520	0.764	16.4
207	88.5	87.5	1520	0.766	15.5
208	91.6	88.9	1520	0.767	15.2
209	20.1	87.8	1520	0.807	47.2
210	23.5	93.9	1520	0.817	43.3
211	26.6	101.2	1520	0.827	41.0
212	29.0	105.5	1520	0.842	39.3
213	31.1	108.8	1520	0.852	37.9
214	96.6	93.2	1520	0.684	15.0
215	116.9	107.3	1520	0.692	13.9
216	126.5	117.0	1520	0.698	13.7
217	140.8	127.9	1520	0.706	13.2
218	148.8	139.8	1520	0.717	13.3
219	160.4	156.6	1520	0.733	13.3
220	9.5	142.0	1520	0.763	104.9
221	12.2	139.6	1520	0.83	86.3
222	13.7	144.9	1520	0.844	80.6
223	32.6	175.5	1520	0.785	46.4
224	36.2	183.0	1520	0.812	43.8
225	39.7	199.2	1520	0.832	42.6
226	46.9	226.4	1520	0.845	40.1
227	48.5	227.6	1520	0.855	39.2
228	55.1	143.0	1520	0.822	28.2
229	65.7	158.8	1520	0.837	26.1

230	72.0	170.9	1520	0.847	25.3
231	79.4	190.5	1520	0.859	24.8
232	85.1	204.4	1520	0.872	24.4
233	88.5	212.6	1520	0.879	24.1
234	3.0	31.3	1020	0.747	78.4
235	3.8	27.7	1020	0.808	61.5
236	4.6	32.4	1020	0.808	58.0
237	5.4	33.1	1020	0.778	51.7
238	6.7	35.3	1020	0.826	45.2
239	8.5	39.7	1020	0.836	40.3
240	9.1	40.9	1020	0.843	38.7
241	9.4	38.8	1020	0.77	36.8
242	11.2	39.3	1020	0.828	32.5
243	13.2	43.1	1020	0.834	30.1
244	14.2	43.5	1020	0.842	28.7
245	15.9	39.0	1020	0.762	25.0
246	19.7	43.5	1020	0.775	22.4
247	23.1	47.5	1020	0.798	20.8
248	24.3	49.0	1020	0.808	20.3
249	24.2	52.3	1020	0.742	21.1
250	34.2	53.1	1020	0.757	16.4
251	40.6	58.6	1020	0.76	15.1
252	5.4	55.4	1020	0.842	67.1
253	6.4	56.2	1020	0.856	59.2
254	7.7	60.9	1020	0.87	53.7
255	8.3	65.6	1020	0.882	52.8
256	29.4	57.6	1020	0.717	19.1
257	31.9	63.9	1020	0.737	18.9
258	36.1	67.4	1020	0.752	17.7
259	42.0	76.5	1020	0.764	16.9
260	44.6	78.7	1020	0.761	16.3
261	44.7	67.4	1020	0.696	15.1
262	49.8	73.1	1020	0.699	14.5
263	59.9	83.6	1020	0.705	13.5
264	65.9	92.3	1020	0.716	13.2
265	69.3	101.9	1020	0.732	13.3
266	70.1	103.2	1020	0.733	13.3
267	14.5	116.3	1020	0.805	46.2
268	17.5	127.7	1020	0.814	42.0
269	19.5	136.7	1020	0.818	40.1
270	21.8	143.2	1020	0.82	37.7
271	22.4	144.5	1020	0.82	37.1
272	25.8	99.7	1020	0.83	27.7
273	29.1	105.7	1020	0.839	26.1
274	32.7	120.1	1020	0.858	25.5

275	36.2	126.6	1020	0.874	24.2
276	37.6	133.9	1020	0.88	24.2
277	49.9	116.7	1020	0.766	18.3
278	55.8	118.9	1020	0.773	17.0
279	62.8	131.8	1020	0.783	16.4
280	74.8	131.2	1020	0.774	14.3

4.3. PREDICTION OF PAT PERFORMANCE AWAY FROM BEP

n	Type	Ns	η_{BEP}	D (m)	N (RPM)	H _{BEP} (m)	Q _{BEP} (m ³ /s)	P _{BEP} (kW)
1	Radial, end-suction	18.5	73%	0.225	1520	26.5	0.019	3.6
2	Radial, end-suction	18.6	77%	0.258	1520	48.6	0.051	18.6
3	Radial, end-suction	28.1	81%	0.206	1520	21.2	0.033	5.6
4	Radial, end-suction	30.1	72%	0.174	1520	15.8	0.025	2.7
5	Radial, end-suction	35.7	84%	0.264	1520	30.5	0.093	23.3
6	Radial, end-suction	41.1	80%	0.2	1520	16.2	0.048	6.0
7	Radial, end-suction	38.1	76%	0.139	1520	9.6	0.019	1.3
8	Radial, end-suction	57.6	74%	0.165	1520	10.2	0.047	3.5
9	Radial, end-suction	70.0	76%	0.224	1520	16.1	0.137	16.3
10	Mixed flow	54.9	80%	0.338	1520	41.3	0.345	111.7
11	Mixed flow	76.9	83%	0.236	1520	13.9	0.133	15.1
12	Radial, end-suction	9.9	46%	0.176	1520	21.6	0.004	0.4
13	Radial (multistage, n=3)	11.4	61%	0.142	1520	18.3	0.004	0.5
14	Radial, end-suction	20.9	61%	0.139	1020	4.0	0.003	0.1
15	Split case, double entry	35.6	88%	0.518	760	23.0	0.243	48.1
16	Radial, end-suction	14.7	70%	0.258	1520	35.0	0.019	4.6
17	Radial, end-suction	24.1	77%	0.203	1520	19.8	0.022	3.3
18	Radial, end-suction	32.2	76%	0.14	3020	36.6	0.025	6.8
19	Radial, end-suction	28.2	84%	0.269	1450	24.4	0.046	9.2
20	Radial, end-suction	38.1	85%	0.219	1450	17.4	0.050	7.2
21	Radial, end-suction	52.1	71%	0.185	1450	12.9	0.060	5.4
22	Radial, end-suction	65.2	75%	0.224	1050	8.4	0.095	5.9
23	Radial, end-suction	9.1	42%	0.33	2525	103.0	0.014	5.8
24	Radial, end-suction	12.8	51%	0.26	2186	46.0	0.011	2.5
25	Radial, end-suction	28.7	73%	0.174	1540	12.1	0.015	1.3
26	Radial, end-suction	30.3	79%	0.219	1562	20.0	0.034	5.2
27	Radial, end-suction	29.7	76%	0.198	1520	19.0	0.032	4.5
28	Radial, end-suction	64.1	70%	0.185	1518	8.5	0.044	2.6
29	Radial, end-suction	17.7	75%	0.2	3020	74.4	0.022	12.1
30	Radial, end-suction	28.8	66%	0.19	3020	69.9	0.053	24.3
31	Radial, end-suction	64.8	49%	0.12	1520	5.7	0.024	0.7
32	Radial, end-suction	47.1	74%	0.177	2000	19.4	0.047	6.6

33	Radial, end-suction	11.6	69%	0.325	3000	221.2	0.050	73.9
34	Radial (multistage, n=6)	16.0	71%	0.18	2000	30.0	0.011	2.2
35	Radial, end-suction	41.8	83%	0.215	2900	67.0	0.114	62.4
36	Radial, end-suction	24.7	81%	0.262	1475	28.6	0.043	9.8
37	Radial, end-suction	36.8	82%	0.219	1050	10.3	0.041	3.3
38	Radial, end-suction	47.8	74%	0.185	1250	11.2	0.055	4.5
39	Radial, end-suction	62.6	76%	0.224	1450	17.6	0.138	18.1
40	Radial, end-suction	6.2	44%	0.2	3000	177.1	0.010	7.6
41	Radial, end-suction	15.6	69%	0.16	3000	64.7	0.014	6.2
42	Radial, end-suction	40.1	79%	0.125	3000	29.8	0.029	6.7
43	Radial, end-suction	22.6	77%	0.356	1800	78.2	0.109	64.3
44	Radial (multistage, n=3)	24.5	74%	0.315	1510	46.5	0.083	27.9
45	Radial (multistage, n=2)	21.4	73%	0.284	1750	48.5	0.050	17.5
46	Radial, end-suction	19.1	63%	0.2	2000	50.6	0.033	10.4
47	Radial, end-suction	10.8	65%	0.25	1000	21.7	0.012	1.7
48	Radial, end-suction	10.5	57%	0.1014	2500	29.1	0.003	0.5
49	Split case, double entry	21.1	57%	0.246	1450	24.2	0.025	3.5
50	Radial, end-suction	8.3	52%	0.32	920	28.0	0.012	1.8
51	Radial, end-suction	21.3	81%	0.334	1500	49.0	0.069	26.8
52	Radial, end-suction	34.9	78%	0.219	1500	22.1	0.056	9.5
53	Split case, double entry	20.4	78%	0.318	1505	40.8	0.048	15.0
54	entry	34.9	75%	0.216	1515	24.6	0.065	11.7
55	Radial, end-suction	38.2	78%	0.28	1550	34.0	0.121	31.3
56	Radial, end-suction	9.3	47%	0.134	3100	55.9	0.004	1.0
57	Submersible, mixed flow	53.4	67%	0.14	3000	27.1	0.045	8.0
58	Radial, end-suction	17.4	65%	0.127	3550	61.4	0.012	4.5
59	Mixed flow, bowl	62.5	78%	\	2070	1.2	0.001	0.0
60	End suction, radial	19.3	85%	\	664	1.3	0.001	0.0
61	End suction, radial	19.0	70%	\	735	1.6	0.001	0.0
62	End suction, radial	25.0	61%	\	3000	36.7	0.015	3.4
63	End suction, mixed flow	61.5	67%	\	1450	9.9	0.056	3.6
64	Mixed flow, bowl	93.0	80%	\	1750	14.8	0.160	18.5
65	End suction, mixed flow	9.9	57%	\	402	2.2	0.002	0.0
66	Tubular propeller, axial	83.2	63%	0.2	750	0.5	0.004	0.0
67	Multistage!	4.0	52%	\	3066	194.3	0.005	4.6
68	Multistage	6.8	40%	\	3015	116.6	0.006	2.9
69	?	10.0	65%	\	1550	30.1	0.007	1.3

70	?	10.3	69%	\	3050	109.3	0.013	9.6
71	?	8.2	61%	\	1550	52.7	0.011	3.4
72	?	10.2	62%	\	3050	120.1	0.015	10.7
73	Multistage	5.2	69%	\	1550	73.7	0.007	3.5
74	?	8.3	55%	\	1550	74.8	0.019	7.5
75	?	12.2	61%	\	3050	122.1	0.022	15.9
76	?	7.2	71%	\	1550	69.3	0.012	5.9
77	?	24.8	41%	\	3050	60.3	0.031	7.6
78	?	22.0	68%	\	1550	56.1	0.085	31.9
79	?	17.7	51%	\	3050	82.6	0.025	10.5
80	?	21.4	52%	\	3050	61.8	0.024	7.6
81	?	14.1	60%	\	3050	99.1	0.021	12.2
82	?	22.4	65%	\	1550	45.6	0.064	18.8
83	?	27.3	65%	\	1550	44.5	0.092	25.9
84	?	50.3	61%	\	780	7.7	0.089	4.1
85	?	51.9	65%	\	1050	13.1	0.116	9.6
86	?	63.5	68%	\	1550	19.5	0.145	18.7
87	?	61.8	68%	\	1550	20.4	0.146	19.9
88	?	57.8	66%	\	1050	10.6	0.104	7.1
89	?	13.9	67%	\	3031	94.7	0.019	12.0
90	?	14.3	67%	\	3033	83.7	0.017	9.4
91	?	41.3	69%	\	780	10.0	0.089	6.0
92	?	42.6	73%	\	1050	16.9	0.115	13.8
93	?	43.9	73%	\	1260	22.8	0.132	21.4
94	?	51.7	70%	\	1550	23.4	0.126	20.2
95	?	54.4	37%	\	1860	25.1	0.108	9.8
96	?	44.2	72%	\	1050	15.5	0.108	11.8
97	?	51.2	72%	\	1550	24.3	0.131	22.3
98	?	53.0	66%	\	1050	10.0	0.080	5.2
99	?	51.2	68%	\	1550	23.7	0.126	19.9
100	?	42.9	68%	\	780	9.0	0.082	5.0
101	?	43.1	73%	\	1260	23.6	0.134	22.5
102	?	40.7	60%	\	780	9.3	0.078	4.3
103	?	51.6	67%	\	1260	15.3	0.100	10.1
104	?	50.4	54%	\	1550	23.9	0.123	15.7
105	?	63.8	84%	\	750	8.2	0.170	11.4
106	?	60.5	85%	\	1000	16.5	0.246	33.8
107	?	75.5	66%	\	1550	23.6	0.272	41.3
108	Radial	8.6	60%	0.176	1520	19.0	0.005	1.0
109	Radial	34.1	79%	0.174	3020	50.0	0.045	17.4
110	Radial	36.5	78%	0.219	1510	20.8	0.056	8.8
111	Radial	55.5	81%	0.269	1520	25.0	0.167	33.2
112	Radial	8.6	51%	0.209	1520	32.5	0.006	1.0
113	Propeller	100.7	61%	0.3	1000	7.2	0.196	8.4

4.4. PREDICTION OF PAT COST

4.4.1. GENERATOR COST DATASET

pp = 1						
	Brand A	Brand B	Brand C	Brand D	Brand E	Brand F
kW	€	€	€	€	€	€
0.75	546	\	114	76	145	90
1.1	625	\	122	85	161	97
1.5	722	\	154	109	211	124
2.2	929	\	176	120	228	140
3	1050	\	222	171	325	198
4	1240	\	264	206	391	247
5.5	1560	\	394	286	546	371
7.5	1980	1223	424	308	618	397
11	2740	1558	626	579	1102	520
15	3580	2005	684	642	1222	545
18.5	4270	2401	740	693	1318	630
22	5170	2709	974	966	1837	835
30	6710	3612	1271	1085	2064	1114
37	8430	4393	1353	1172	2230	1230
45	9800	5255	1602	1437	2735	1655
55	11700	6363	2044	1770	3580	2068
75	16000	7966	2611	2490	4738	2509
90	18900	9491	2924	2726	5225	2983
110	22600	11732	5003	4061	8290	5513
132	27100	14030	5314	4327	8740	5788
160	33700	17345	5556	4703	9056	6009
200	42100	21469	6278	5663	10774	6615
pp = 2						
0.55	541	\	\	\	\	88
0.75	591	\	120	97	184	98
1.1	700	\	120	118	223	126
1.5	830	\	140	131	248	139
2.2	956	\	160	180	341	192
3	1100	\	216	204	388	225
4	1340	335	243	227	432	278
5.5	1690	456	294	285	560	374
7.5	2140	1297	382	339	644	439
11	2800	1588	439	579	1103	550
15	3640	2022	687	663	1261	620
18.5	4390	2401	739	855	1625	772
22	5100	2771	935	997	1897	860
30	6620	3510	1022	1285	2446	1114

37	7850	4269	1252	1422	2705	1412
45	9240	5070	1600	1530	2912	1577
55	11200	6199	1772	2044	3889	1963
75	15100	7702	2290	2651	5044	2510
90	17700	8964	2889	3081	5861	2845
110	22100	11262	3156	4648	8843	4642
132	26200	13465	4474	4992	9498	5193
160	31700	16121	4883	5348	10175	6064
200	39400	20151	5558	6652	12656	6670
pp = 3						
0.37	525	\	\	\	\	\
0.55	607	\	\	\	\	\
0.75	686	\	162	116	220	\
1.1	850	\	186	138	263	\
1.5	945	\	238	171	328	\
2.2	1200	\	251	219	435	\
3	1500	\	335	281	548	\
4	1850	\	374	317	602	\
5.5	2320	\	427	364	693	\
7.5	2950	1417	564	549	1033	\
11	4070	1888	679	639	1204	\
15	5450	2377	937	855	1609	\
18.5	6660	2884	1074	1021	1923	\
22	7710	3371	1147	1098	2066	\
30	10100	4579	1608	1308	2770	\
37	12300	5534	1837	1705	3367	\
45	15000	7118	2662	2162	4310	\
55	17700	8569	3012	2443	4603	\
75	24100	11601	4301	4027	7860	\
90	27100	13804	4771	4329	8240	\
110	32700	16686	5016	4827	9473	\
132	38900	19774	5510	5394	10443	\
160	47200	24859	8664	7925	14925	\

4.4.2. PUMP COST DATASET

4.4.2.1. Radial, end suction

Model	H _{BEP} (m)	Q _{BEP} (l/s)	P (kW)	η _{BEP}	N (rpm)	Cost (Excl. VAT) €
32-125.1/100	10	4.3	0.75	56%	2900	1290
32-125.1/110	12.5	4.75	1.1	59%	2900	1338
32-125.1/121	16.5	5.4	1.5	64%	2900	1403
32-125.1/140	22.5	6.3	2.2	70%	2900	1450
32-125	19	7	2.2	70%	2900	1497

32-125/106	10	5.6	1.1	60%	2900	1348
32-125/115	14	5.8	1.5	66%	2900	1417
32-125/130	19	7	2.2	70%	2900	1455
32-125/142	24	7.6	3	70%	2900	1686
32-160.1/139	19	3.95	1.5	55%	2900	1475
32-160.1/155	25	4.8	2.2	59%	2900	1530
32-160.1/169	32	5.5	3	61%	2900	1776
32-160.1/177	37	5.8	4	62%	2900	2024
32-160.1	37	5.8	4	62%	2900	2038
32-160/139	20	6.4	2.2	59%	2900	1550
32-160/151	25	6.7	3	62%	2900	1791
32-160/163	31	7.5	4	66%	2900	2001
32-160/177	37.5	8.3	5.5	67%	2900	2380
32-200.1/172	27.5	5.1	3	52%	2900	1899
32-200.1/188	35	5.6	4	53%	2900	2122
32-200.1/205	45	5.9	5.5	55%	2900	2460
32-200.1/207	47	6	7.5	55%	2900	2686
32-200	48	9.4	7.5	66%	2900	2281
32-200/176	32	7.1	4	58%	2900	2137
32-200/190	40	8.4	5.5	63%	2900	2475
32-200/206	48	9.4	7.5	66%	2900	2676
32-200/219	60	10	11	64%	2900	3424
32-250/199	44	5	5.5	43%	2900	2682
32-250/219	54	5.7	7.5	43%	2900	2888
32-250/244	67	6.8	11	45%	2900	3551
32-250/262	80	7.6	15	47%	2900	3929
40-125/105	9.5	10.4	1.5	70%	2900	1528
40-125/116	13	11.4	2.2	72%	2900	1570
40-125/127	17.5	12.3	3	75%	2900	1816
40-125/139	22	13.1	4	80%	2900	2036
40-125/142	23.9	13.3	5.5	81%	2900	2338
40-160/144	26	9.3	4	67%	2900	2053
40-160/158	30.5	12.2	5.5	76%	2900	2450
40-160/172	37.5	13.5	7.5	78%	2900	2680
40-160/177	41.5	14.1	11	78%	2900	3412
40-200/172	32	10.8	5.5	65%	2900	2535
40-200/188	38	12	7.5	66%	2900	2765
40-200/206	47	14.7	11	68%	2900	3444
40-200/219	53	16.6	15	69%	2900	3782
40-250/211	54	12.7	11	67%	2900	3524
40-250/230	67	13.9	15	68%	2900	3878
40-250/245	69	17.8	18.5	67%	2900	4149
40-250/255	83	15.9	22	70%	2900	4805
40-250/260	89	16.4	30	70%	2900	6127
40-315/273	80	9.6	22	40%	2900	5374

40-315/298	98	10.7	30	41%	2900	6727
40-315/318	114	12	37	42%	2900	7750
40-315/336	132	12.8	45	49%	2900	10026
50-125/111	13	14.2	3	67%	2900	1917
50-125/121	14.8	19.3	4	72%	2900	2102
50-125/135	19.5	21	5.5	77%	2900	2480
50-125/144	22.5	22.8	7.5	82%	2900	2696
50-160/136	21	19	5.5	72%	2900	2535
50-160/150	26	21.1	7.5	74%	2900	2771
50-160/167	34	24	11	79%	2900	3489
50-160/177	39.2	26.5	15	82%	2900	3782
50-250/205	50	18	15	69%	2900	5034
50-250/222	60	19	18.5	70%	2900	5265
50-250/233	65	20.8	22	70%	2900	6110
50-250/254	79	24	30	71%	2900	7476
50-250/263	85	24.5	37	71%	2900	8154
50-315/267	88	19	30	57%	2900	6838
50-315/285	102	20.5	37	58%	2900	7808
50-315/300	116	22	45	58%	2900	9036
50-315/321	137	23.6	55	59%	2900	11171
65-125/120-110	12	22.5	4	74%	2900	2177
65-125/127	15	31	5.5	80%	2900	2550
65-125/137	19	32	7.5	83%	2900	2832
65-125/144	22	33.5	11	82%	2900	3553
65-160/143	20.5	28	7.5	81%	2900	2934
65-160/157	27.5	31	11	84%	2900	3655
65-160/173	35	35	15	86%	2900	3893
65-160/177	37.5	36.2	18.5	87%	2900	4197
65-200/162	31	23.8	11	74%	2900	3835
65-200/177	40	26.4	15	77%	2900	4121
65-200/190	46	28	18.5	80%	2900	4373
65-200/198	50	30.5	22	79%	2900	5039
65-200/217	62	34	30	82%	2900	6295
65-200/219	64	34.5	37	83%	2900	7852
65-315/272	95	39	55	72%	2900	10213
65-315/295	113	44.5	75	73%	2900	12481
65-315/308	122	48	90	74%	2900	14262
65-315/320	135	53	110	75%	2900	18729
80-160/147-127	17.5	44	11	75%	2900	3810
80-160/151	21	49	15	77%	2900	4079
80-160/161	26	52	18.5	80%	2900	4338
80-160/167	28.5	54	22	80%	2900	5154
80-160/177	34.5	55.5	30	85%	2900	6377
80-200/171	32	44	22	73%	2900	4982
80-200/188	41	49	30	76%	2900	6587

80-200/200	50	52	37	79%	2900	7376
80-200/211	55	57	45	80%	2900	8586
80-200/222	62	60.5	55	82%	2900	10072
80-250/220	66	43	45	79%	2900	8670
80-250/234	74	48	55	79%	2900	10157
80-250/257	88	60	75	81%	2900	11718
80-250/270	97	66	90	82%	2900	13612
80-315/278	106	60	90	74%	2900	13895
80-315/295	118	63	110	76%	2900	19153
80-315/310	131	70	132	76%	2900	21009
80-315/328	148	76	160	77%	2900	23851
100-200/170	29	74	30	72%	2900	6349
100-200/181	34	78	37	76%	2900	6952
100-200/192	41	82	45	81%	2900	8303
100-200/203	48	86	55	82%	2900	9649
100-200/219	57	98	75	83%	2900	11351
100-250/205	50	70	55	76%	2900	9875
100-250/229	65	84	75	82%	2900	11774
100-250/242	75	86	90	83%	2900	13245
100-250/258	86	90	110	84%	2900	18588
100-250/269	92	95	132	85%	2900	20726
100-315/269	90	81	110	76%	2900	19153
100-315/284	102	88	132	77%	2900	21291
100-315/301	117	95	160	79%	2900	24416
100-315/322	136	106	200	80%	2900	28884
125-250/222	54	130	90	78%	2900	13527
125-250/236	62	138	110	80%	2900	18871
125-250/249	70	150	132	81%	2900	21009
125-250/263	80	165	160	83%	2900	24133
125-250/269	84	173	200	84%	2900	28602
125-315/262	83	105	132	77%	2900	21856
125-315/277	98	110	160	80%	2900	25546
125-315/297	113	123	200	81%	2900	30014
150-200/216-176	29	207	75	79%	2900	13212
150-200/218-202	34	200	90	84%	2900	14716
150-200/224	41	219	110	84%	2900	19909
150-250/226-220	47	190	132	75%	2900	21928
150-250/235	54	216	160	78%	2900	25103
150-250/250	65	240	200	83%	2900	29610
150-315.1/261	67	190	160	85%	2900	28239
150-315.1/277	78	208	200	86%	2900	31434
32-250	67	6.8	11	45%	2900	2840
50-125	22.5	22.8	7.5	82%	2900	2398
50-160	26	21.1	7.5	74%	2900	2395
50-315	137	23.6	55	59%	2900	11180

65-160	20.5	28	7.5	81%	2900	3399
65-315	95	39	45	72%	2900	9679
80-160	26	52	18.5	80%	2900	4747
80-250	74	48	55	79%	2900	3998
32-125.1/121	3.9	2.6	0.25	61%	1490	1035
32-125.1/139	5.2	3	0.25	69%	1490	1035
32-125.1/140	5.7	3.1	0.37	69%	1490	1045
32-125/115	3.3	2.9	0.25	64%	1490	1017
32-125/130	4.5	3.5	0.25	68%	1490	1017
32-125/142	5.8	3.8	0.37	69%	1490	1045
32-160.1/137	4.8	2	0.25	56%	1490	1074
32-160.1/155	6	2.35	0.25	59%	1490	1074
32-160.1/172	7.8	2.7	0.37	58%	1490	1105
32-160.1/177	8.9	2.8	0.55	59%	1490	1186
32-160/138	4.5	3.2	0.25	55%	1490	1074
32-160/154	6	3.6	0.37	59%	1490	1102
32-160/172	8	4	0.55	63%	1490	1180
32-160/173	8.3	4.2	0.75	63%	1490	1332
32-250/206	10.5	2.8	0.75	43%	1490	1571
32-250/236	15.6	3.3	1.1	45%	1490	1646
32-250/260	18.3	3.7	1.5	47%	1490	1765
32-250/262	19	3.8	2.2	47%	1490	1916
40-125/116	3.1	5.7	0.25	72%	1490	1206
40-125/130	4.2	6.2	0.37	76%	1490	1220
40-125/142	5.3	6.6	0.55	81%	1490	1301
40-250/219	13.9	7.2	1.5	66%	1490	1735
40-250/245	17.3	8.2	2.2	67%	1490	1924
40-250/260	20.5	8.6	3	68%	1490	2013
40-315/283	21	4.8	3	39%	1490	2505
40-315/305	25.5	5.4	4	40%	1490	2742
40-315/334	32	6	5.5	42%	1490	3129
40-315/344	35	6.4	7.5	42%	1490	3256
50-200/171	8.7	8.8	1.1	75%	1490	1602
50-200/188	10.6	9.6	1.5	76%	1490	1663
50-200/210	13.9	11	2.2	81%	1490	1747
50-200/219	15	12	3	79%	1490	1856
50-250/221	14	9.8	2.2	70%	1490	1867
50-250/241	17	11	3	70%	1490	1969
50-250/263	21.3	11.5	4	72%	1490	2186
65-160/135	4.4	13	0.75	81%	1490	1473
65-160/149	6	14	1.1	82%	1490	1540
65-160/165	7.6	16	1.5	85%	1490	1572
65-160/177	9	17.8	2.2	87%	1490	1712
65-200/170	8.3	12.8	1.5	75%	1490	1861
65-200/189	10.9	14.5	2.2	78%	1490	1952

65-200/205	13.4	15.2	3	82%	1490	2067
65-200/219	15.4	17	4	82%	1490	2337
65-250/215	13.8	16	3	72%	1490	2421
65-250/232	16	17.5	4	73%	1490	2680
65-250/254	19.5	20	5.5	74%	1490	3150
65-250/270	23	21.5	7.5	75%	1490	3548
65-315/261	20.5	17.5	5.5	70%	1490	3339
65-315/282	25	20	7.5	70%	1490	3666
65-315/314	31.5	23.5	11	73%	1490	4441
65-315/320	33	25	15	73%	1490	4844
80-250/225	15.4	26	5.5	78%	1490	3283
80-250/247	19.5	29	7.5	80%	1490	3661
80-250/270	24.2	30.5	11	83%	1490	4441
80-315/280	24	31.5	11	75%	1490	4605
80-315/305	30	34.6	15	77%	1490	4945
80-315/320	33.5	35.5	18.5	77%	1490	5579
80-315/334	37.5	36.6	22	79%	1490	5969
80-400/347	36	31.7	18.5	65%	1490	7269
80-400/365	40	34	22	66%	1490	7680
80-400/397	47	40.5	30	67%	1490	9061
80-400/419	54	45	37	69%	1490	10376
80-400/438	60	47.5	45	70%	1490	10994
100-160/160-140	5.1	32	2.2	78%	1490	2168
100-160/169	6.6	35	3	82%	1490	2271
100-160/176	7.5	36	4	83%	1490	2503
100-315/279	24	44	15	76%	1490	5175
100-315/295	27.5	47	18.5	77%	1490	5755
100-315/312	31	49	22	78%	1490	6165
100-315/334	36	54	30	81%	1490	7458
100-400/315	29	47	22	74%	1490	7652
100-400/360	40	53	30	76%	1490	8948
100-400/375	47	56	37	76%	1490	10263
100-400/395	55	63	45	77%	1490	11305
100-400/415	57	64	55	77%	1490	12766
125-250/220	13.2	65	11	81%	1490	4774
125-250/236	15.7	70	15	82%	1490	5090
125-250/249	18	76	18.5	83%	1490	5710
125-250/262	20	80	22	84%	1490	6099
125-250/269	21.7	83	30	85%	1490	7503
125-500/406	53	63	55	68%	1490	14178
125-500/447	65	73	75	69%	1490	15881
125-500/473	71	80	90	70%	1490	17075
125-500/500	80	88	110	71%	1490	22570
125-500/526	86	97	132	72%	1490	23370
125-500/548	95	110	160	73%	1490	25081

150-200/210-158	5.9	102	7.5	76%	1490	4845
150-200/218-208	8.3	105	11	83%	1490	5562
150-200/224	9.5	110	15	84%	1490	5890
150-315/275	20.8	128	37	79%	1490	10405
150-315/291	24	136	45	81%	1490	11305
150-315/310	27	150	55	83%	1490	13048
150-315/336	32.5	172	75	85%	1490	15033
150-315/338	34	174	90	86%	1490	16793
150-400/343	37	107	55	82%	1490	13737
150-400/375	45	120	75	84%	1490	15805
150-400/394	50	130	90	85%	1490	17634
150-400/412	56	146	110	86%	1490	23674
150-400/431	61	158	132	87%	1490	24501
150-400/438	64	165	160	88%	1490	26282
032-200.1	10.5	2.8	0.75	43%	1490	2172
040-250	17.3	8.2	2.2	67%	1490	4199
050-250	14	9.8	2.2	70%	1490	2609
065-200	10.9	14.5	2.2	78%	1490	2649
065-315	20.5	17.5	5.5	70%	1490	3154
080-315	30	34.6	15	77%	1490	5200
100-160	7.5	36	5.5	83%	1490	3493
125-250	20	80	22	84%	1490	6292
150-400	45	120	75	84%	1490	11122
150-400	50	130	90	85%	1490	15506
100-160/160-140	1.8	22.2	0.55	75%	995	1840
100-160/169	2.5	20.5	0.75	81%	995	2085
100-160/176	3	24	1.1	81%	995	2166
100-200/182	3.4	24	1.1	78%	995	2374
100-200/193	4.55	26.5	1.5	83%	995	2454
100-200/214	5.8	30	2.2	84%	995	2561
100-200/219	6.3	31	3	84%	995	2945
100-250/216	5.9	28	2.2	77%	995	2622
100-250/236	7.6	29.5	3	83%	995	3002
100-250/260	9	32.5	4	83%	995	3191
100-250/269	9.8	33	5.5	84%	995	3722
100-315/272	9.8	28	4	76%	995	3379
100-315/301	12.4	31.5	5.5	78%	995	3750
100-315/326	15	35	7.5	80%	995	4107
100-315/334	16	36.3	11	81%	995	5019
100-400/340	14.5	37	7.5	74%	995	5757
100-400/385	21	38	11	76%	995	6658
100-400/415	25	42	15	76%	995	7390
125-200/176-150	3.7	42	1.5	84%	995	2856
125-200/196-182	4.5	43	2.2	86%	995	2927
125-200/215	5.5	47	3	87%	995	3254

125-200/224	6	51	4	88%	995	3349
125-400/351	17	40	11	72%	995	6658
125-400/384	19	45	15	74%	995	7390
125-400/410	22	48	18.5	75%	995	8289
125-400/434	26	55	22	76%	995	9053
125-400/438	27.5	56.4	30	76%	995	9887
125-500/421	25	46	18.5	70%	995	9701
125-500/445	28	50	22	70%	995	10466
125-500/493	34	57.5	30	71%	995	11300
125-500/524	38	64	37	72%	995	12191
125-500/546	41	71.5	45	73%	995	13507
125-500/548	41.5	72.5	55	73%	995	14678
150-315/280	9.5	86	11	81%	995	6912
150-315/305	11.6	95	15	81%	995	7645
150-315/322	13	102	18.5	82%	995	8543
150-315/337	14.6	107	22	82%	995	9166
150-315/338	15	108	30	82%	995	10057
150-400/357	17.5	75	18.5	83%	995	9066
150-400/375	20	80	22	84%	995	9703
150-400/408	23	92	30	86%	995	10624
150-400/430	26	98	37	87%	995	11839
150-400/438	27.2	104	45	88%	995	13351
150-500/457	31	84	37	77%	995	13618
150-500/483	35	86	45	77%	995	15129
150-500/513	40	96	55	78%	995	16284
150-500/548	47	95	75	77%	995	17634

4.4.2.2. Vertical Multistage

Stages n°	H _{BEP} (m)	Q _{BEP} (l/s)	η _{BEP}	P _{BEP} (kW)	N (RPM)	Cost (Excl. VAT) €
5	25	0.77	58%	0.3	2900	636
9	45	0.77	58%	0.6	2900	776
13	65	0.77	58%	0.8	2900	944
17	85	0.77	58%	1.1	2900	1182
21	105	0.77	58%	1.4	2900	1293
5	25	1.67	63%	0.6	2900	783
9	45	1.67	63%	1.2	2900	1026
13	65	1.67	63%	1.7	2900	1237
22	110	1.67	63%	2.9	2900	1620
1	8	2.76	68%	0.3	2900	929
5	40	2.76	68%	1.6	2900	1531
9	72	2.76	68%	2.9	2900	1960
16	128	2.76	68%	5.1	2900	3046
1	11	5	71%	0.8	2900	1097

4	44	5	71%	3.0	2900	1905
9	99	5	71%	6.8	2900	2988
12	132	5	71%	9.1	2900	3714
1	12.5	5.6	72%	1.0	2900	1090
5	62.5	5.6	72%	4.8	2900	2492
7	87.5	5.6	72%	6.7	2900	2995
12	150	5.6	72%	11.4	2900	4832
1	16	7.8	78%	1.6	2900	1764
4	64	7.8	78%	6.3	2900	3798
8	128	7.8	78%	12.5	2900	6206
10	160	7.8	78%	15.7	2900	7297
12	192	7.8	78%	18.8	2900	8453
1	20	12	79%	3.0	2900	2356
4	80	12	79%	11.9	2900	5994
8	160	12	79%	23.8	2900	8970
10	200	12	79%	29.8	2900	10272
1	22	18	80%	4.9	2900	3048
4	88	18	80%	19.4	2900	4911
6	132	18	80%	29.1	2900	9830
1	23	23	80%	6.5	2900	3566
3	69	23	80%	19.4	2900	7610
5	115	23	80%	32.4	2900	9698
1	21	32	72%	9.1	2900	4985
3	63	32	72%	27.4	2900	8469
7	147	32	72%	64.0	2900	19800
1	20	43	71%	11.9	2900	5877
3	60	43	71%	35.6	2900	9874
6	120	43	71%	71.2	2900	19203