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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

H	hydraulic head (m)
N	rotational speed (RPM)
n	rotational speed (rad/s)
Ns	specific speed (–)
P	power (kW)
Q	flow rate (m^3/s)
R^2	coefficient of determination

Greek symbols

η	efficiency (–)
Ψ	head coefficient (–)

Subscripts

rel	relative

Abbreviations

BEP	Best Efficiency Point
PAT	Pump as Turbine
3D	three-dimensional
CFD	computational fluid dynamics

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:

$$N_s = N((\sqrt{Q_{BEP}})/(H_{BEP}^{0.75})) \tag{1}$$

where N represents the shaft speed (RPM) and Q (m^3/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 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).

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:

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

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

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:

$$\Phi = Q/(nD^3) \tag{4}$$

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 Eq. (3) is [18]:

$$P_t/P_{t,BEP} = 4.000\cdot10^{-3}(Q_t/Q_{t,BEP})^3 + 1.386(Q_t/Q_{t,BEP})^2 - 0.390((Q_t/Q_{t,BEP})) \tag{5}$$

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

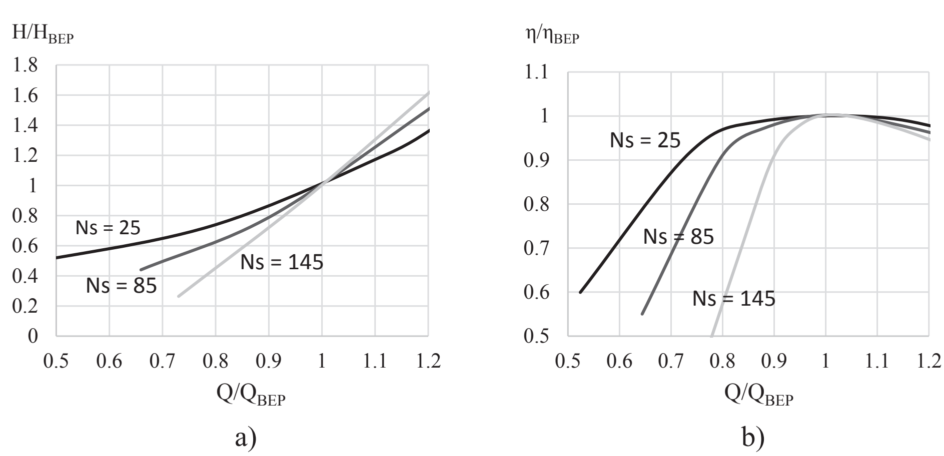

Fig. 1. Comparison of three Pats having different Ns: (a) relative head versus relative flow rate, (b) relative efficiency versus relative flow rate. Adapted from: [5]