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PAT efficiency variation with design parameters

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Abstract

The use of pump operating as turbine (PAT) instead of traditional turbines is an effective source of reducing the equipment cost in small hydropower plants. However, the lack of information on the PATs performances represents a limit for their wider diffusion. Furthermore, while for the design of traditional hydropower plants the knowledge of the best efficiency point of the turbomachine is sufficient, additional difficulties arise under variable operating conditions for the exploitation of hydraulic power within water distribution networks: a procedure for the design of the optimal machine operating under variable hydraulic conditions, named variable operating strategy (VOS), has been recently developed, and for its implementation a full set of characteristic curves of several different PATs rotating at different speeds is needed. In order to face the hydraulic variability, variable operating strategy considers either a hydraulic or an electric regulation. The hydraulic regulation system consists in a series-parallel circuit with a PAT and two regulating valves, while for the electric regulation the PAT generator is connected to an inverter which modifies the rotational speed of the machine. If the performances curves of a single PAT are known, all the information needed for the application of VOS can be obtained by the application of the affinity law of turbomachines, wich allows to extent the given information to a complete set of similiar machines. This paper focuses on the study of the affinity law for the evaluation of the behaviour of a single machine under variable speed. A large database of experimental curves of several PATs operating at different speeds is available and the experimental data are compared with the results of the application of the affinity law herein.

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Keywords: Water distribution networks; PATs; Energy recovery; Turbomachinery affinity law.

1. Introduction

In the management of water supply networks, savings of water and energy are among the major concerns(Gonçalves et al., 2011). Water leakages of corrupted networks are strictly related to the pressure in the pipelines (Nazif et al., 2010, Araujo et al., 2006, Vairavamoorthy and Lumbers, 1998), and good results in the reduction of leakages can be obtained by a pressure control strategy (Almandoz et al., 2005, Tucciarelli et al., 1999, Walsky et al., 2006, Prescott and Ulanicki, 2008), avoiding expensive rehabilitation of pipes. Furthermore, if hydraulic turbines are used instead of pressure reducing valves, an additional energy production can be pursued together with water savings (Carravetta and

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Giugni, 2009). Nevertheless, the small amount of available power, together with the large variability of the hydraulic characteristics (Karadirek et al., 2012), limits the diffusion of hydropower plants within water distribution networks, and the installation of power plants is presently limited to the transmission pipelines, where the available hydraulic power is considerable and fairly constant.

Therefore, the chance to obtain a sustainable energy production within water distribution networks constitutes a topical issue and a fashionable challenge for hydraulic and mechanical engineers. Among the different alternatives of special turbines that have been suggested to face with small and variable power (Sammartano et al., 2013, Carravetta and Giugni, 2009, Paish, 2002) pump as turbines (PATs) can be considered a good alternative (Chapallaz et al., 1992), which combines low installation costs with an acceptable energy production, with best efficiencies ranged between 0.6 and 0.7 (Carravetta et al.). Unfortunately, pump producers usually do not provide the performances curves of PATs and the designer should face a lack of data that constitutes an obstacle to the choice of the right machine.

A specific procedure, named Variable Operating Strategy (VOS), aimed to the design of a PAT operating in variable conditions of both flow discharge and pressure drop, was recently developed by Carravetta et al. (2012, 2013b). The impeller diameter of the PAT and its rotational speed are the main results of VOS application that allows the designer to select the appropriate machine from the pump market. Two regulation modes are considered by the procedure: the PAT could be either inserted in a series-parallel circuit or connected to an inverter. In the former case (hydraulic regulation - HR) a valve placed in series produces an additional dissipation, while the parallel valve can bypass a part of the flow. In the latter case (electric regulation - ER) the inverter modifies the rotational speed of the asinchronous generator to adapt the PAT operating point to the network needs. The two regulation modes are shown in Fig. 1, where the installation schemes are drawn together with the effect of the regulations on the flow-rate (Q) and the head drop (H). The scarcity of data about the PAT behaviour and the performaces curves is overcome by the application of the affinity law: once both the characteristic and the efficiency (η) curves of a prototype PAT are known, the results may be extended to obtain the performances curves of other similar devices with different runner diameter and rotational speed, by using the affinity law and the Suter (Suter, 1966, Wylie et al., 1993) parameters.

It is clear that the more dependable are the $\eta(Q)$ and $H(Q)$ curves resulting from the application of affinity law, the more reliable will be the VOS solution. In this paper the results of the affinity law modeling are compared with a large dataset of experimental characteristic curves, obtained for several machines operating at different rotating speed.

2. PAT characteristic curves and affinity law

PATs are generally used in combination with asynchronous electric generator, having constant rotation speed related to the electrical grid frequency. In such conditions the performances of the machines are described by the characteristic and the efficiency curves, which relates the flow rate with head drop end efficiency respectively. The device efficiency curve, depending on the discharge, presents a maximum: the corresponding values of discharge and head drop (Q_B^T, H_B^T) is called Best Efficiency Point (BEP). Both curves can be obtained in three ways: experimentally (Gantar, 1988, Fernandez et al., 2004, Derakhshan and Nourbakhsh, 2008), by computational fluid dynamics (CFD) (Rodrigues et al., 2003, Natanasabapathi and Kshirsagar, 2004, Carravetta et al., 2011) and by any one-dimensional method (Stepanoff, 1957, Childs, 1962, Hanckock, 1963, Grover, 1980, Sharma, 1985, Schmiedl, 1988, Alatorre-Frenk and Thomas, 1990). The first choice is the most reliable, but a large number of experiments is necessary for the whole range of flow conditions and generator rotation speed and these results are generally not available for PATs. CFD could be considered a valid alternative to experiments since it can be considered a reliable technique (Kerschberger and Gehrer, 2010, Nautiyal et al., 2010, Carravetta et al., 2011). For pumped storage power plants, a large number of one-dimensional methods for the determination of PAT performances (Williams, 1994) based on the pump behaviour have been proposed, but they can be considered the least reliable methods (Fecarotta et al., 2011). Once the prototype performances curves are available, the results may be extended to obtain the characteristic and efficiency curves of other similar devices, by using the Suter (Suter, 1966, Wylie et al., 1993) parameters. Two machines of the same type (e.g. centrifugal, semiaxial) can be considered similar if they have either the same turbine specific speed (N_S^T) or the same pump specific speed (N_S^P):

$$N_S^T = N_B^T \frac{{P_B^T}^{\frac{1}{2}}}{{H_B^T}^{\frac{4}{3}}} \quad N_S^P = N_B^P \frac{{Q_B^P}^{\frac{1}{2}}}{{H_B^P}^{\frac{3}{4}}} \tag{1}$$

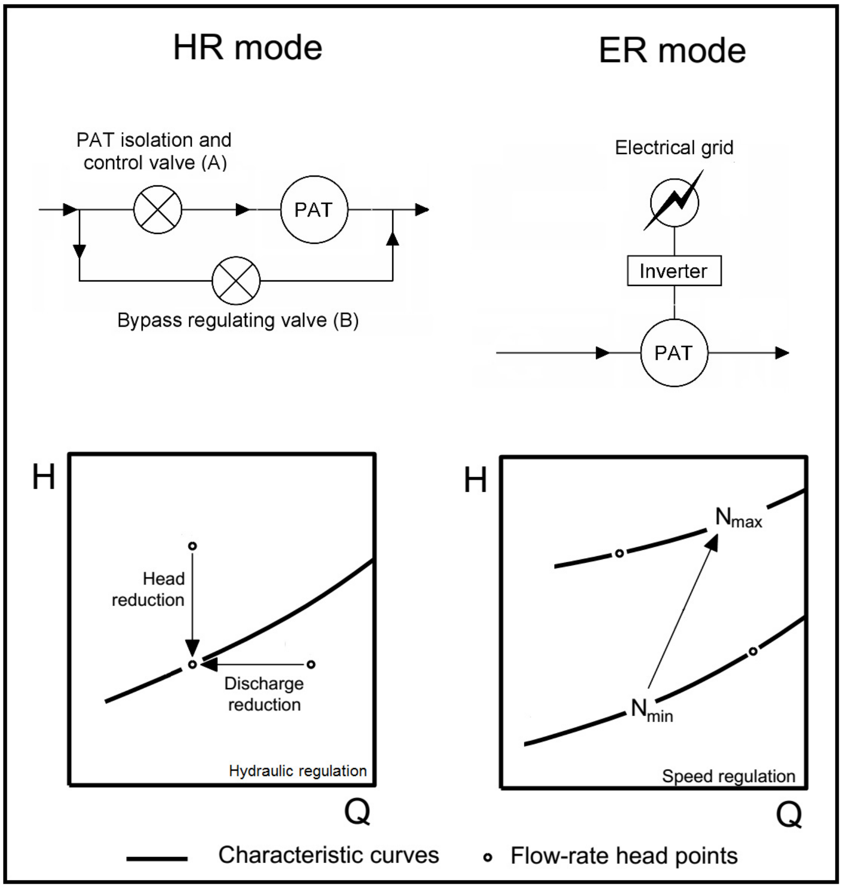

Fig. 1. Hydropower plants regulation modes

where N_B, Q_B, P_B e H_B are rotational speed, flow rate, power and head drop at the best efficiency point (BEP), respectively, and the superscript T and P refers to turbine and pump mode, respectively. Obviously, the same machine at two different rotating speed can be considered as two similiar machines. The turbomachine affinity law allow to calculate the BEP of a similiar machine based on the BEP of the prototype:

$$\frac{N_B^I}{N_B^{II}} = \frac{D^{II}}{D^I}\left(\frac{H_B^I}{H_B^{II}}\right)^{\frac{1}{2}} = \left(\frac{Q_B^{II}}{Q_B^I}\right)^{\frac{1}{2}}\left(\frac{H_B^I}{H_B^{II}}\right)^{\frac{3}{4}} = \left(\frac{P_B^{II}}{P_B^I}\right)^{\frac{1}{2}}\left(\frac{H_B^I}{H_B^{II}}\right)^{\frac{5}{4}} \tag{2}$$

BEPs (Q_B^{II}, H_B^{II}) and produced power at the BEP (P_B^{II}) can be determined for any PAT, having D^{II} diameter and N^{II} rotational speed, similiar (superscript II) to the prototype one. Furthermore, the whole characteristic and efficiency curves can be calculated by the application of Suter model: in 1966 Suter introduced two parameters (see also Wylie et al. (1993)):

$$WH = \frac{h}{q^2(\theta^2+1)}; \quad WT = \frac{t}{q^2(\theta^2+1)}; \tag{3}$$

where

$$\theta = \frac{\omega}{q} \tag{4}$$

represents the operating condition of the machine and

$$h = \frac{H^T}{H_B^T}, \; \omega = \frac{N^T}{N_B^T}, \; q = \frac{Q^T}{Q_B^T}, \; t = \frac{T^T}{T_B^T}, \tag{5}$$

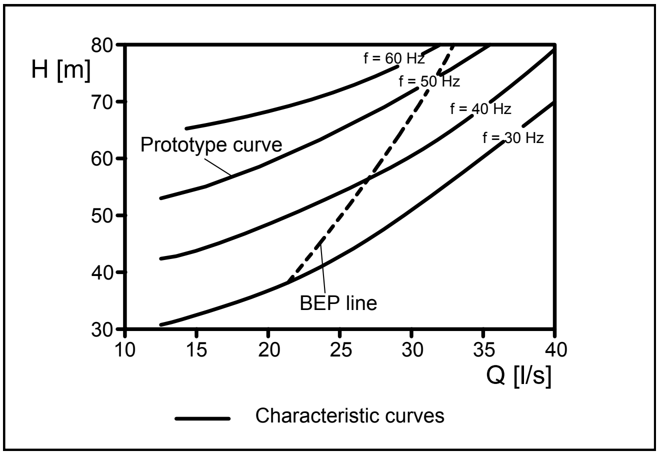

Fig. 2. Characteristic curves of a PAT under variable speed, obtained by affinity law and Suter parameters from the prototype curve.

being T the torque applied to the runner.

The two functions $WH(\theta)$ and $WT(\theta)$ are unique for similar machines. Thus, once they are available for a prototype machine, they can be used to calculate the head drop H^T and the efficiency η^T of similar PATs in any operating condition θ

$$H^T = \left[q^2(\theta^2 + 1)WH\right] H_B^T, \quad \eta^T = \theta \frac{WT}{WH} \eta_B^T \tag{6}$$

where η_B^T is the efficiency of the machine at the BEP.

In Fig. 2 the characteristic curves of a single-stage centrifugal pump, obtained by such procedure from the prototype curve, are plotted together with a dashed line, connecting the BEPs of the different curves at different rotational speed.

3. Comparison between experimental data and affinity law results

A large data set relative to the behaviour of 17 different pumps operating as turbines is available to the authors. For each PAT both the characteristic and the efficiency curves have been measured for different rotational velocities. The characteristics of the pumps are reported in Table 3

Table 1. Characteristics of the machines dataset. (HCM = Horizontal Centrifugal Multi-stage, HCS = Horizontal Centrifugal Single-stage, SSS = Submersible Semiaxial Single-stage)

PAT	Pump type	No. of stages	speed range [rpm]
a	HCM	2	1550 - 3050
b	HCM	2	1550 - 3050
c	HCS	1	1550 - 3050
d	HCS	1	1550 - 3050
e	HCS	1	1550 - 3050
f	SSS	1	780 - 1550
g	SSS	1	1050 - 1550
h	SSS	1	780 - 1550
i	SSS	1	780 - 1860
j	SSS	1	780 - 1860
k	SSS	1	780 - 1550
l	SSS	1	780 - 1550
m	SSS	1	750 - 1550
n	HCM	3	1550 - 3050
o	HCM	4	1550 - 3050

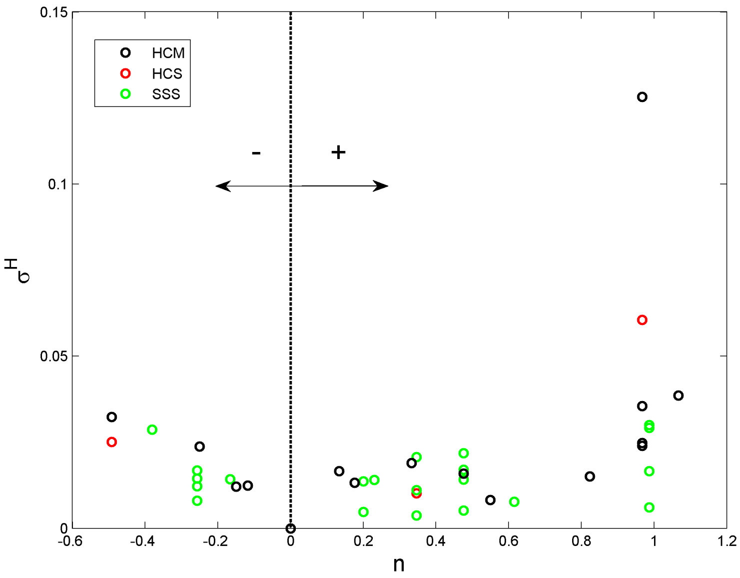

Fig. 3. Values of the global error σ^H for each couple of curves versus the velocity ratio n.

In order to find the agreement between real (experimental) and calculated (affinity law and Suter parameters) $H(Q)$ and $\eta(Q)$ curves, the experimental curve of each PAT has been used as prototype curve. Then the performances curves has been extended to the other velocity values of the dataset and $H(Q)$ and $\eta(Q)$ values have been compared. For each couple of characteristic curves, the error σ^H, wich accounts the discrepancies between the calculations and experiments, has been expressed by:

$$\sigma^H = \frac{1}{m}\sqrt{\sum_{i=1}^{n}\left(\frac{H_i^{exp} - H_i^{calc}}{H_i^{exp}}\right)^2} \tag{7}$$

while, for the efficiency curves, the error σ^η:

$$\sigma^\eta = \frac{1}{m}\sqrt{\sum_{i=1}^{m}\left(\eta_i^{exp} - \eta_i^{calc}\right)^2} \tag{8}$$

where m is the number of experimental measurements for each curve, the superscripts ^{exp} and ^{calc} refers to the experimental and calculated values of the related variable respectively.

The resulting values of σ^H and σ^η are plotted in Fig. 3 and 3 respectively, versus n, where

$$n = \frac{N^{exp} - N^{calc}}{N^{exp}} \tag{9}$$

where N^{exp} is the rotational velocity of the prototype curves and N^{calc} the calculated rotational velocity of the calculated curves. From Eq. 9 it is clear that n could assume both positive (+) and negative (−) values, depending on the respective velocities of prototype and simulated machine.

4. Discussion

Observing the plots of Fig. 3 and 3, several interesting remarks can be figured out:

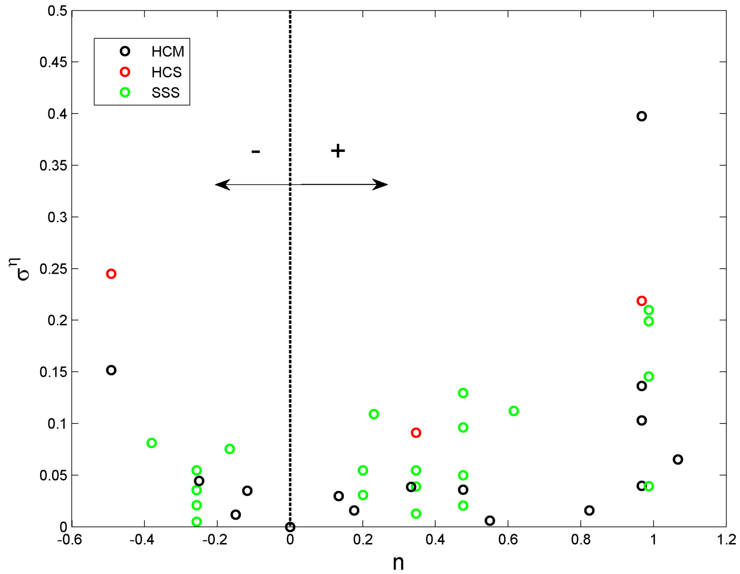

Fig. 4. Values of the global error σ^η for each couple of curves versus the velocity ratio n.

- the agreement between the experimental and the calculated curves worsen when the difference between the velocities of the prototype and the simulated machine increases;
- the calculation error does not appear dependent on the machine type;
- if the calculation by means of affinity law and Suter parameters is limited to a value of difference in velocity of 20%, the error in the evaluation of head drop is lower than 3%; this value can be accounted in the evaluation of margin of error of VOS procedure;
- the error in the evaluation of η is less than 15 percentage points if the difference of velocities is ranged between −40% and +50%;
- the resulting uncertainty in the evaluation of the efficiency of the plant can be token into account as a parameter influencing the plant effectiveness (Carravetta et al., 2013a).

This study shows that the use of performances curves calculated by means of affinity law and Suter parameters produces a limited error in the evaluation of the head drop, granting the satisfaction of the correct hydraulic constraint (pressure level within the network). Meanwhile, the error in terms of mechanical efficiency is greater but still acceptable in a limited range of velocity difference between prototype and simulated machine.

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