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Predictive model estimating the performances of centrifugal pumps used as turbines

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ABSTRACT

A one-dimensional numerical code estimating the performances of centrifugal PATs (pumps used as turbines) is presented. Firstly, the code calculates the geometrical components of the PAT using information provided in manufacturer catalogues. Then, once these parameters are deduced, it calculates the losses and determines the characteristics curves of the PAT. The method was validated by comparing the theoretical curves with some experimental measurements acquired at the Department of Mechanical, Energy and Management Engineering (DIMEG) of the University of Calabria, at the Mechanical Propulsion National Centre (CNPM) in Milan and at the University of Trento (Italy) on PATs working in a range of specific speed from 9 to 65 rpm $m^{3/4}$ $s^{-1/2}$. The estimation error was comprised in a range of 5 ÷ 20%, generally acceptable for this kind of application.

1. Introduction

In an energy scenario today characterized by an increasing impoverishment, the exploitation of small power resources is attractive. Installations exploiting micro-hydraulic resources [1,2] (less than 100 kW), although interesting, are penalized by the excessive cost of small turbines.

Using PATs (pumps as hydraulic turbines) [3,4] is a smart and important alternative. Centrifugal pumps are mass-produced for a wide range of heads and flow rates so their prime cost is lower than that of the turbine and their maintenance is easier, because of the availability of spare parts, even in developing countries. The efficiency of these machines will be lower but, as they exploit otherwise wasted energy sources, this is not a critical issue. K H Motwania et al. [5] produced an interesting economic analysis on a case study of a 3 kW capacity pico hydropower plant. They conclude that application of PATs is recommended in the pico/micro-hydro range for power generation in rural, remote and hilly areas.

From the economic point of view, it is often stated that pumps working as turbines in the range of 1–500 kW involve capital payback periods of two years or less [5], which is considerably less than that of a conventional turbine.

Unfortunately, although a wide number of centrifugal pumps are commercially available for micro-hydro engineering plant, manufacturers do not provide information regarding the performance of centrifugal pumps in turbine mode.

Therefore, establishing a correlation enabling the passage from the "pump" characteristics to the "turbine" characteristics is the main challenge in using a PAT (pump as a turbine). Many theoretical/empirical relations for predicting the PAT characteristics in the BEP (best efficiency point) are available in the literature. Childs [6], Sharma [7], Alatorre [8], and Stepanoff [9] link the best head ratio and the best flow rate ratio to the global efficiency of the pump. Hancock [10] links these ratios to the global efficiency of the turbine. Schmield [11] relates these ratios to the hydraulic efficiency of the pump while Grover [12] and Hergt [13] link these ratios to the specific speed of the turbine. The results predicted by these methods are compared with test results for a wide range of pumps by Williams [14] who concludes that only the method of Sharma is able to predict the BEP of most of the analysed sample in a more accurate way.

Unlike a specifically-designed turbine, a pump cannot accommodate changes in the flow through the machine so that for a given resource in a specific site, the choice of a suitable PAT, among several available pumps, becomes critical. Sanjay V. Jain et al. [15]

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Nomenclature

Symbols

A	generic area
A_f	fore leakages passage area
A_b	back leakages passage area
$A_{\theta j-1}$	inlet area of the jth volute sector
$A_{1,\ 2}$	passage area at different points of the impeller
$A_{1r,\ 2r}$	real passage area at different points of impeller
A_3	diffusion region passage area
A_4	volute final section area
A_5	final diffuser inlet passage area
$b_{1,\ 2}$	width at different points of impeller
b_3	vaneless diffuser width
b_4	final section volute width
b_5	final section diffuser width
b_v	volute throat section width
$c_{1,\ 2,\ 3,\ 4}$	absolute fluid velocities at different points of PAT
$c_{m1,\ m2,\ m3}$	meridional velocities at different points of PAT
$c_{u1,\ u2,\ u3,\ u4}$	peripheral velocities at different points of PAT
C	prediction coefficient
C_f	disc loss coefficient
C_D	drag coefficient
D	generic diameter
D_o	impeller eye diameter
D_1, D_2, D_3	diameter at different points of PAT
D_{1H}	hub diameter
D_h	hydraulic diameter
D_{hj}	hydraulic diameter of a volute segment
$D_{h1,\ h2,\ h3}$	hydraulic diameter at different points of PAT
D_{hm}	average hydraulic diameter
D_{heq}	equivalent hydraulic diameter
D_{shf}	shaft diameter
D_{sf}	annular gap diameter of fore seal
D_{sb}	annular gap diameter of back seal
$e\%\_H$	relative error on head
$e\%\_Q$	relative error on capacity
$e\%\_\eta$	relative error on efficiency
Eff_D	disc efficiency
Eff_H	hydraulic efficiency
Eff_Meas	measured efficiency
Eff_Tot	total efficiency
Eff_V	volumetric efficiency
g	gravity
$h_{4,\ 5}$	heights at different points of the final diffuser
h_v	volute throat section height
h_d	dynamic loss
h_{dd}	dynamic diffuser loss
h_{drag}	profile drag loss
h_{dsc}	discharge loss
h_f	friction loss
h_{fc}	friction vaneless diffuser loss
h_{fd}	friction diffuser loss
h_{fi}	friction impeller loss
h_{fv}	friction volute loss
$h_{lowflow}$	low flow loss
h_{inlet}	diffusion region inlet loss
h_{shock}	shock loss
h_{sr}	sudden restriction loss
H	head
H_e	head at BEP of the pump
H_m	real head
H_{m_calc}	calculated real head
H_{m_meas}	measured real head
H_{mo}	head at the shut off
H_{th}	theoretical head (Euler's head)
H_{BEP}	head at BEP of the PAT
H_{BEP_calc}	calculated head at BEP of the PAT
H_{BEP_meas}	measured head at BEP of the PAT
j	generic sector or segment of volute
k	low flow loss constant
k_1	annular gap diameter coefficient
k_2	diffusion region diameter coefficient
k_3	hub diameter coefficient
K_v	volute velocity coefficient
l_v	vane length
l_d	diffuser length
n	rotational speed
n_s	characteristic speed
N	number of sectors of the volute
P	power
P_e	maximum pump power
Q	capacity
Q_e	capacity at BEP of the pump
Q_j	flow rate in a volute sector
Q_s	leakage flow
Q_{BEP}	capacity at BEP of the PAT
Q_{BEP_calc}	calculated capacity at BEP of the PAT
Q_{BEP_meas}	measured capacity at BEP of the PAT
r	generic radius
R_B	vanes camber
Re	Reynolds number
Re_j	Reynolds number of the jth volute sector
$R_{\theta j}$	radius of a volute segment
R_4	final section volute radius
$t_{1,\ 2}$	vane thickness
$u_{1,\ 2}$	peripheral velocity at different points of impeller
$w_{u1,\ u2}$	peripheral components of relative velocity
$w_{m1,\ m2}$	meridional components of relative velocity
w_∞	average relative velocity
y_P	pump height
z	number of blades

Greek letters

α_2	absolute flow angle in the vaneless diffuser
α_d	final diffuser opening angle
β	inclination of relative flow to peripheral direction
$\beta_{1f,\ 2f}$	relative flow direction
$\beta_{1p,\ 2p}$	blades angles at different points of impeller
β_∞	average relative flow direction
ΔS_{es}	external segmental surface area
ΔS_{inn}	increment of inner wall surface
ΔS_{pv}	increment of peripheral volute surface
ΔP_D	disc loss
$\Delta\delta$	increment of volute angle
ζ	dynamic loss coefficient
η	efficiency
η_{calc}	calculated efficiency
η_H	hydraulic efficiency
η_D	disc efficiency
η_{meas}	measured efficiency
η_v	volumetric efficiency
η_{tot}	total efficiency
θ	inclination of blade to radial direction
λ	friction coefficient