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Procedure Selecting Pumps Running as Turbines in Micro Hydro Plants

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Abstract

The authors present a combined method using statistical and numerical models for selecting a pump running as turbine in micro hydro plants. The data of the site (head and capacity) allow calculating two coefficients, C_Q and C_H, which identify the pump to use successfully as turbine in that place. A one dimensional model, starting from data available on the pumps manufacturers catalogues, reconstructs a virtual geometry of the PAT, then calculates the performances curves, head vs. capacity, efficiency vs. capacity. The procedure has been applied with the aim to select a PAT recovering energy from a pipeline whose characteristic curve is known.

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1. Introduction

In the last years, the request of energy from renewable resources has increased more and more tacking into account the depletion of traditional sources like oil, gas, nuclear [1]. Wind, sun, tides, rivers offer huge quantities of clean, green and renewable energy. Among these, hydraulic sources are the most easily used and their exploitation is always desirable. However, small resources, under 100 kW, often are not considered or discarded, because of the

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specific cost of energy, which is higher respect to that of big hydro-electrical power plants. The traditional turbines are very expensive and their cost can reach the 25% of the entire plant one. Moreover, they need specific maintenance and have components hardly available on the market. The pumps working as turbines (PATs) [2] constitute an alternative surely cheaper and manageable. 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. From the economic point of view, realizing a micro hydro plant using PATs having power up to 500 kW, imply payback periods less than two years [3], surely lower with respect to installations using conventional turbines.

This work is framed in this context, with the goal to find the performance curves of a PAT suitable to a hydrological site, whose head H_{site} and capacity Q_{site} are known. To do this, the authors propose a procedure, which involves a statistical method and a numerical model.

The statistic method allows calculating two conversion factors [4]: C_Q as ratio between the capacity of the pump running as turbine and that of the pump one at best efficiency point (BEP) and C_H as ratio between the two heads, defined in a similar way. Both the coefficients are expressed in relation with n_{sp}, the specific speed of the PAT working in pump mode, which can be expressed as function of that of the same machine working as turbine n_{st}. This last parameter can be obtained from the hydrological data of the site H_{site} and Q_{site}, unique input data known when the selection of the PAT has to be done. Once calculated head and capacity at BEP of the pump suitable for the chosen site, it is possible to select, on the composite performance chart of the manufacturer, the pump one.

The authors propose a numerical model, developed during the past years [5-6], which is able to estimate the performance of the chosen pump in the reverse mode, as turbine (PAT).

Firstly, the code reconstructs the geometrical parameters of the PAT, which usually are unknown, by using information provided on the manufacturer catalogue. Once deduced these data, the code calculates the losses and determines the characteristics curves of the PAT i. e. head vs. capacity and efficiency vs. capacity. The knowledge of such curves allows the operating point of the plant to be assessed. In this way, the annual yield of energy can be estimated. The case study of a pipeline whose characteristic curve is known is presented with the aim to better expose the proposed methodology.

2. Statistical Model

The main issue using PATs is represented by the choice of the machine because the performance curves of a pump running as turbine are not immediately available. In fact, on the catalogues of the pumps manufacturer the only performances of the pump operations are reported. Therefore, it is important to connect the pump performances to the turbine performances of the same machine, by using mathematic correlations having universal validity and involving simple data obtainable from the pump catalogue. The easiest way to pass from the pump working mode to the turbine working mode is to put in relation the best efficient point (BEP) of the pump to that of the turbine. In this way two coefficients can be detected: the best head ratio and the best flow rate ratio. Childs [7], Sharma [8], Alatorre [9] and Stepanoff [10] link these coefficients to the global efficiency of the pump. Hancock [11] links them to the global efficiency of the turbine while Schmield [12] relates them to the hydraulic efficiency of the pump.

Some authors, like Grover [13] and Hergt [14], consider statistic correlations involving the specific speed of the pump by considering that the shape of the impeller, and consequently the losses typology, changes when the specific speed increases, as showed in Fig. 1.

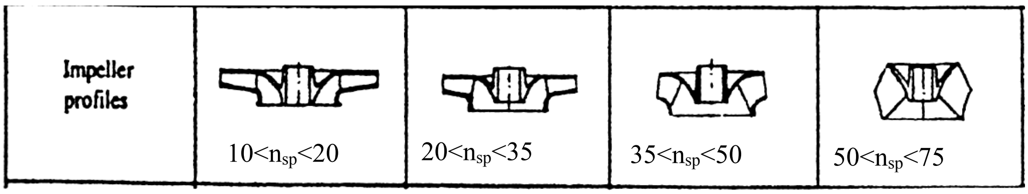

Fig. 1. Shape of the impeller with n_{sp} increasing

Usually, the selection of a suitable PAT for a given site starts from two conversion factors which are:

- C_Q as ratio between the capacity of the turbine and that of the pump one, at best efficiency point (BEP):

$$C_Q = \frac{Q_T}{Q_P} \tag{1}$$

- C_H as ratio between the two heads, defined in a similar way.

$$C_H = \frac{H_T}{H_P} \tag{2}$$

The authors propose a correlation involving the specific speed of the pump n_{sp} defined as:

$$n_{sp} = \frac{n\sqrt{Q_P}}{H_P^{\frac{3}{4}}} \tag{3}$$

The dependence between C_Q, C_H and n_{sp} was fixed on a sample of 26 pumps. A sample of 12 pumps was measured at DIMEG in direct and reverse operation [15]; other 4 pumps measurements were found in [16], while the remaining 11 in [17]
In Tab. 1 the values of flow, head, specific speed, efficiency at BEP point, both in direct (P) and in reverse mode (T), together with the conversion factors C_Q and C_H, of the pumps measured at DIMEG are reported.

Tab. 1. Measurements at BEP point of the pumps sample of University of Calabria

PUMP	Q_P [l/s]	H_P [m]	n_{sp}	η_P	Q_T [l/s]	H_T [m]	n_{st}	η_T	C_Q	C_H
Ksb 40-335	7.39	33.01	9.08	0.44	13.08	93.28	5.54	0.43	1.77	2.83
Ksb 40-315	7.52	31.41	9.43	0.45	14.11	110.80	5.09	0.35	1.88	3.53
Ksb 40-250	6.97	20.00	12.82	0.55	10.65	43.66	8.82	0.51	1.53	2.18
Ksb 40-200	5.28	12.00	16.34	0.55	9.72	25.50	12.60	0.59	1.84	2.13
Av 65-250	16.5	19.3	20.23	0.65	26.02	38	15.28	0.65	1.58	1.97
Av 80-250	26.77	19.6	25.43	0.73	40.28	33.2	21.04	0.73	1.50	1.69
Ksb 50-160	9.72	8.50	28.72	0.67	15.28	13.10	26.03	0.73	1.57	1.54
Ksb 80-220	24.16	14.52	30.31	0.74	36.52	22.40	26.91	0.78	1.51	1.54
Ksb 80-200	23.19	12.06	34.11	0.72	31.22	17.60	29.82	0.76	1.35	1.46
Ksb 100-200	41.67	12.90	43.48	0.76	50.00	18.80	35.91	0.84	1.20	1.46
Ksb 125-200	57.93	9.59	53.01	0.82	84.33	13.30	50.04	0.84	1.46	1.39
Ksb 100-160	34.95	5.32	64.07	0.78	43.63	7.82	53.59	0.70	1.25	1.47

It was therefore possible to obtain the following correlations, which cover a specific speed range from 10 to 70:

$$C_H = -0.00003\, n_{sp}^3 + 0.00440\, n_{sp}^2 - 0.20882\, n_{sp} + 4.64293 \tag{4}$$

$$C_Q = 0.00029\, n_{sp}^2 - 0.02771\, n_{sp} + 2.01648 \tag{5}$$

In Fig. 2 the data collected for the two conversion factors on the pump sample are represented together with the interpolating curves, reporting respectively an error band of 20% (C_H curve) and of 15 % (C_Q curve).
By disposing of the hydrological data of the chosen site i.e. head H_{site} and flow Q_{site}, the required specific speed of the site can be calculated. This parameter will be equal to the available specific speed of the PAT n_{st} defined as:

$$n_{st} = \frac{n\sqrt{Q_T}}{H_T^{\frac{3}{4}}} \tag{6}$$

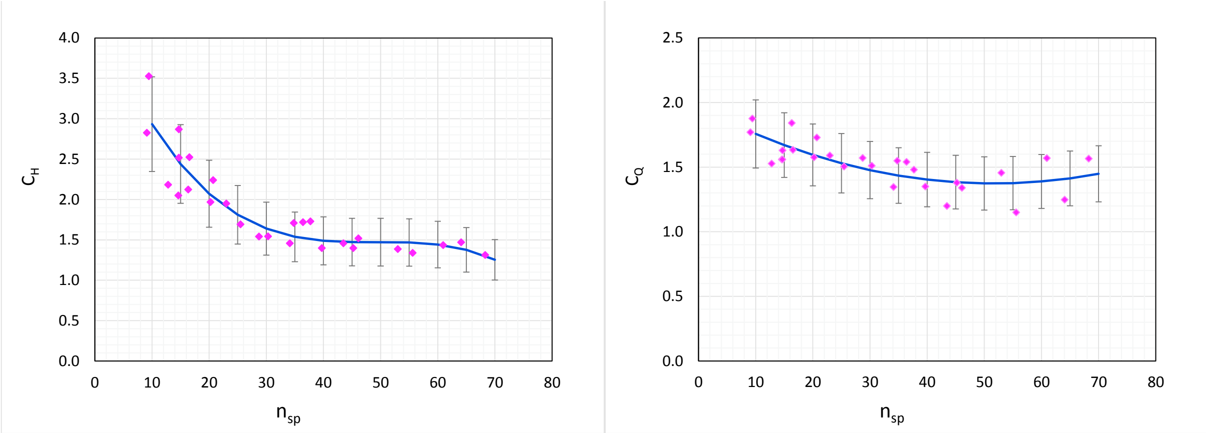

Fig. 2 Interpolating curves for the conversion factors CH and CQ.

This parameter changes almost linearly with the specific speed of the pump and it can be correlated from the pumps sample as it follows:

$$n_{sp} = 0.9867\, n_{st} + 5.2818 \tag{7}$$

Fig. 3 shows a flow chart illustrating the various steps. Obviously, it is necessary to know the hydrological data of the site i.e. the head curve and the yearly frequencies distribution of flow rates. Starting from these data, it is necessary to select the design flow rate Q_{site}, determine the correspondent head H_{site} and the specific speed of the site n_{st} which will be equal to that of the PAT.

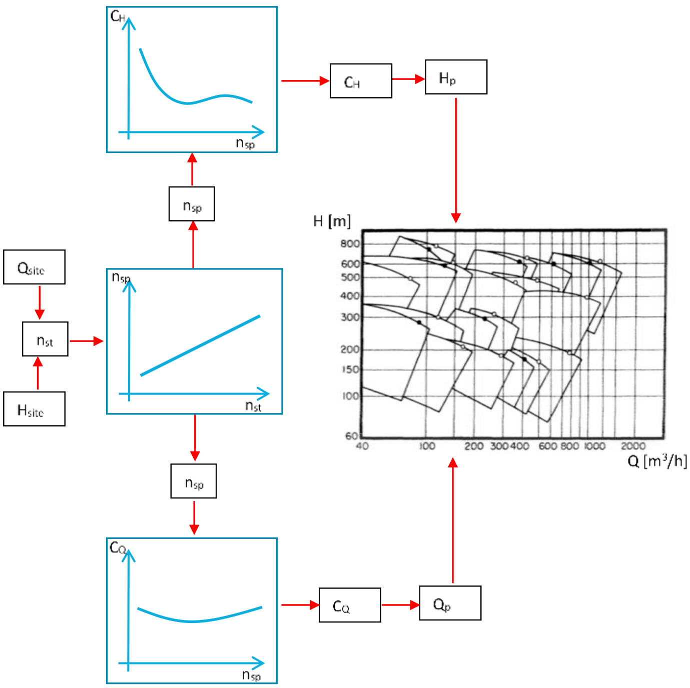

Fig. 3 PAT selection procedure - flow chart

Then it is possible, by means of eq. 7, to calculate the specific speed of the pump n_{sp} and consequently to calculate, via the eqs. 4 and 5, the two conversion factors C_Q and C_H.
At this point, head and capacity at BEP of the pump can be calculated through the following equations:

$$Q_P = \frac{Q_{site}}{C_Q} \tag{8}$$

$$H_P = \frac{H_{site}}{C_H} \tag{9}$$

By using Q_P and H_P as input on the composite performance chart of the manufacturer, it is possible to select the pump to use as turbine. With the aim to calculate the performances curve of the pump in reverse mode, it is possible to apply the UNICAL numerical model [5-6] described briefly in the following section.

3. Numerical Model

The one dimensional model is a code homemade, developed by authors in several years of research. In this work, for brevity, the main features are synthetized, by remanding the reader to accurate description in [5-6] where all the details are found.

The model is capable to operate by starting from data just available on the manufacturer catalogues. It has the aim to supply an extended information. In fact, it can determine the whole *H(Q)* and *η(Q)* curves instead of the BEP only, by providing more accurate responses, without requiring data not immediately accessible. Indeed, the model, by applying standard design criteria, can determine itself many useful geometrical parameters.

In order to perform a standard design it is necessary to take a precise example of centrifugal pump as a reference: a machine prototype, as illustrated in Fig. 4. The shape has been chosen with a straight conical suction, rectangular volute section having height linearly variable from 0 to the throat size, and a final diffuser with square sections as a truncated pyramid.

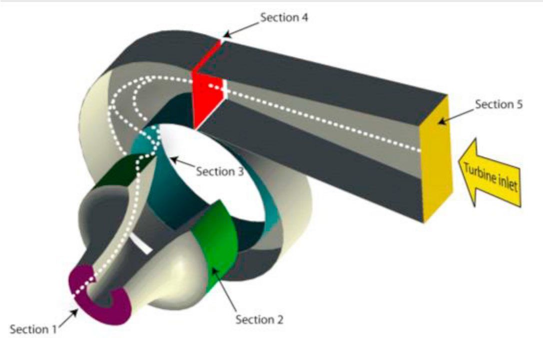

Fig. 4 Reference geometry

The reference pump shape and the related geometric parameter calculations were obtained as Lobanoff [18] suggests. The sizing procedure of the reference prototype needs the knowledge of 6 parameters available in the manufacturers catalogues: head (H_P) and flow rates (Q_P) at BEP of the pump, maximum power (P_{max}), the shut off head (H_{mo}), impeller diameter (D_2) and height of the pump (y_P).

If the geometry of the pump is unknown, the model performs the design step and it operates in design mode, otherwise the design step isn't necessary and it can be by-passed. The model will accept, in this case, the actual geometrical parameters *(geometry known mode)*. In order to have a greater flexibility, the design step can be performed also for a limited number of unknown geometrical parameters and so the model will operate in *mixed mode*.
When the geometry of the machine is defined by applying the sizing procedure or by inputting the known data, the

geometrical parameters become input data for the next calculation section of the model, which allow to calculate the average velocity in each flow section of the PAT and the corresponding losses. These are split in friction losses and dynamic losses, as delineated in Tab. 2, and they are calculated by applying the following formulas [19]:

$$h_f = \lambda \frac{c^2}{2g}\left(\frac{l}{D_h}\right) \tag{10}$$

$$h_d = \zeta \frac{c^2}{2g} \tag{11}$$

The details related to the coefficients λ, ζ and to the geometrical passage areas are better reported in [6].

Tab. 2 Hydraulic losses typology

SECTION	LOSSES TYPOLOGY	
Sect. 5 - diffuser	Friction losses	Dynamic losses (Idel'cick [20])
Sect. 4 - volute	Friction losses calculated by dividing the volute in sectors	
Sect. 3 - vaneless	Friction losses Diffusion losses	Dynamic losses (Idel'cick [20])
Sect. 2 - impeller	Friction losses Drag losses	Shock losses Sudden restriction losses
Sect. 1 - discharge	Discharge losses	

Once the losses have been calculated, the real head (H_m) can be expressed as sum of the Eulerian work (H_{th}) and of the losses themselves:

$$H_T = H_{th} + losses \tag{12}$$

Then the hydraulic efficiency can be calculated as:

$$\eta_H = \frac{H_{th}}{H_T} \tag{13}$$

Taking into account the volumetric efficiency (η_v) and the disc efficiency (η_D), the power output of the PAT is expressed as:

$$P = \eta_H \eta_v \eta_D \rho g\, QH \tag{14}$$

while the total efficiency of the PAT is:

$$\eta_{tot} = \eta_H \eta_v \eta_D \tag{15}$$

4. Case study

A case study of a water pipeline to which install a PAT is illustrated. The pipeline brings water with a difference in level about 23 m, while the design flow rate is 10 l/s (36 m3/h). In the next table (Tab. 3), the parameters of the above-illustrated procedure are reported (see Fig. 3).

Tab. 3 Calculation of the pump parameters

Q_{site} $[m^3/h]$	H_{site} $[m]$	rpm	n_{st}	C_Q	C_H	Q_P $[m^3/h]$	H_P $[m]$
36	23	1450	13.80	1.61	2.18	22.3	10.6

By entering in the composite performance chart of the manufacturer, the suitable pump for the proposed application can be selected.

As illustrated in Fig. 5, the pump selected is KSB 65-40-200. The input parameters necessary for running the numerical model are: H_P = 12.2 m, Q_P = 20.5 m^3/h, P_{max} = 1.25 kW, H_{mo} = 14 m, D_2 = 0.202 m, y_P = 0.225.

Fig. 6 illustrates the two curves of the selected PAT KSB 65-40-200 provided by the numerical model (head and efficiency of the PAT versus flow rate), together with the site characteristic curve (head of the site versus flow rate).

From Fig. 6, flow rate, head and efficiency at operating point of the plant can be deduced, which are respectively: $Q_T = 37\ m^3/h$, $H_T = 29\ m$, $\eta_T = 56\%$

The efficiency is a little lower than BEP, but the loss is very slight. The power supplied by the PAT, by assuming an electrical efficiency (η_{el}) of 0.96, will be equal to:

$$P_e = \eta_{el}\, \eta_T\, \rho\, g\, H_T\, Q_T = 1.6\ kW \tag{16}$$

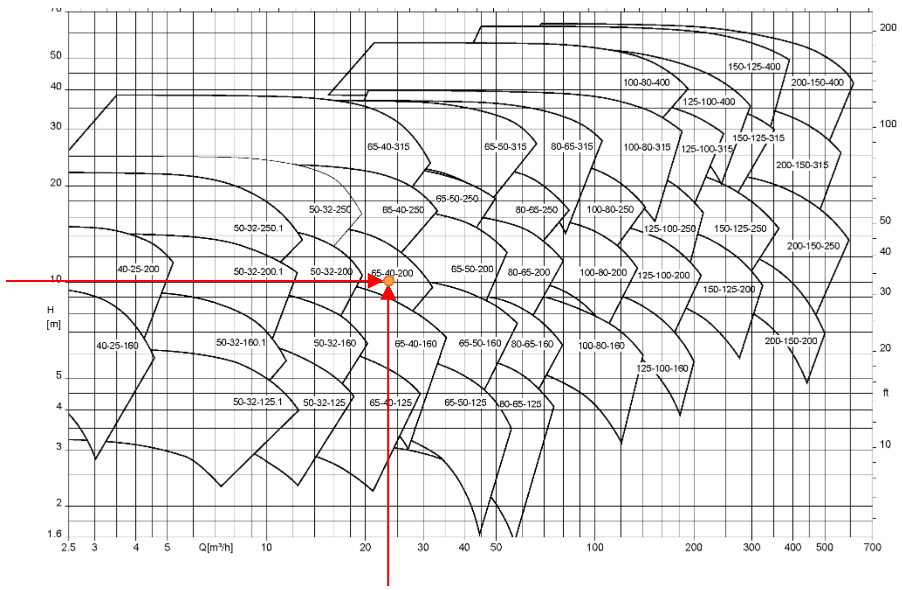

Fig. 5 Selection of the pump on the manufacturer performance composite chart

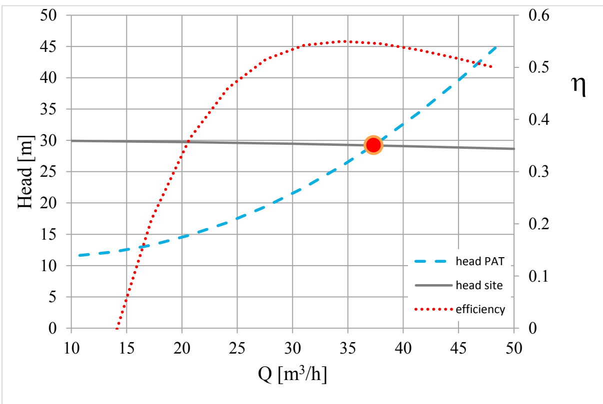

Fig. 6 Characteristic head curves: operating point

5. Conclusions

In the present work, a new methodology for selecting a PAT suitable for installation in a particular site is proposed. The method involves a statistical model and a one-dimensional code: the statistical model allows calculating the conversion factors C_Q and C_H, able to find capacity and flow rate of the suitable pump for the chosen site. By inputting these data on the composite performance chart of the manufacturer, it is possible to select the PAT. The numerical model developed by authors will provide the performances curves head-capacity and efficiency-capacity.

With the aim to give an exemplification of the proposed procedure, the case study of a pipeline whose characteristic curve is known is presented. The pump KSB 65-40-200 has been selected as turbine and calculation results show an output power of about 1.6 kW.

6. Nomenclature

c	velocity [m/s]	l	duct length [m]	**Greek letters**	
C_H	head conversion factor [-]	n	rotational speed [rpm]		
C_Q	flow rate conversion factor [-]	n_{sp}	specific speed of the PAT [$rpm \cdot m^{3/4} \cdot s^{-1/2}$]	ζ	dynamic loss coefficient [-]
D_2	impeller diameter [m]	n_{st}	specific speed of the PAT [$rpm \cdot m^{3/4} \cdot s^{-1/2}$]	η_D	disc efficiency [-]
D_h	hydraulic diameter [m]	N_{site}	specific speed of the site [$rpm \cdot m^{3/4} \cdot s^{-1/2}$]	η_{el}	electrical efficiency of the PAT [-]
g	gravity [m/sec^2]	P	power [kW]	η_H	hydraulic efficiency of the PAT [-]
h_d	dynamic losses [m]	P_{max}	maximum power [kW]	η_T	efficiency of the PAT at BEP [-]
h_f	friction losses [m]	P_{el}	electrical power [kW]	η_{tot}	total efficiency [-]
H	head [m]	Q	flow rate [l/s]	η_v	volumetric efficiency [-]
H_{mo}	shut off head [m]	Q_P	flow rate of the PAT as pump [l/s]	λ	friction loss coefficient [-]
H_P	head of the PAT as pump [m]	Q_{site}	flow rate of the site [l/s]	ρ	density [kg/m^3]
H_{site}	head of the site [m]	Q_T	flow rate of the PAT as turbine [l/s]		
H_{th}	theoretical head [m]	y_p	pump height from axis to diffuser [m]		
H_T	head of the PAT as turbine [m]				

References

[1] Gianluigi Lo Basso, Benedetto Nastasi, Davide Astiaso Garcia, Fabrizio Cumo. How to handle the Hydrogen enriched Natural Gas blends in combustion efficiency measurement procedure of conventional and condensing boilers. Energy, Volume 123, 15 March 2017, Pages 615-636.
[2] Williams, A. A., 1994. The turbine performance of centrifugal pumps: a comparison of prediction methods. Proc. Instn. Mech. Engrs, vol. 208, Pt A, pp. 59-66.
[3] K H Motwania, S V Jainb, R N Patelb. Cost analysis of pump as turbine for pico hydropower plants – a case study. Procedia Engineering 51 (2013) 721 – 726
[4] Jean-Marc Chapallaz, Peter Eichenberger, Gerhard Fischer. Manual on Pumps Used as Turbines. Informatica International, Incorporated, 1992.
[5] Amelio M., Barbarelli S. A one-dimensional numerical model for calculating the efficiency of Pumps As Turbines for implementation in micro-hydro power plants. Proceedings of "7th Biennial ASME Conference Engineering Systems Design and Analysis", Manchester, UK, July 19-22, 2004.
[6] S. Barbarelli, M. Amelio, G. Florio. Predictive model estimating the performances of centrifugal pumps used as turbines. Energy, Volume 107, 15 July 2016, Pages 103-121.
[7] Childs S.M., 1962. Convert pumps to turbines and recover HP. Hydrocarbon Processing an Petroleum Refiner.
[8] Sharma K. R., 1985. Small hydroelectric project-use of centrifugal pumps as Turbines. Kirloskar Electric Co., Bangalore, India.
[9] Alatorre-Frenk, C., and Thomas, T. H., 1990. The pumps as Turbines approach to small hydropower. Word Congress on Renewable energy, Reading.
[10] Stepanoff, A. J., 1957. Centrifugal and Axial flow pumps. John Wiley, New York, pp. 276
[11] Hancock, J. W., 1963. Centrifugal pumps or water turbine. Pipe Line News, June 1963, pp. 25-27
[12] Schmield, E., 1988. Serien-Kreiselpumpen im Turbinenbetrieb. Pumpentagung Karlsruhe, Sec. A6 (in German).
[13] Grover, K.M., 1980. Conversion of pump to turbines. GSA Inter Corp. Katonah, New York.
[14] Lewinsky-Kesslitz, H.P., 1987. Pumpen als Turbinen fur Kleinkraftwerke. Wasserwirtschaft, 77(10), pp. 531-537, (in German).
[15] S. Barbarelli, M. Amelio, G. Florio. Experimental activity at test rig validating correlations to select pumps running as turbines in microhydro plants. Energy Conversion and Management. https://doi.org/10.1016/j.enconman.2017.03.013
[16] Williams A. A. The selection and application of centrifugal pumps as water turbines. Proc. MechE, Vol. 208 Pt A, pp. 59-66, 1994.
[17] Shahram Derakhshan, Ahmad Nourbakhsh. Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds. Experimental Thermal and Fluid Science 32 (2008) 800–807
[18] Lobanoff, V. S., Ross, R. R., 1992. Centrifugal Pumps & Design and Applications. Gulf Publishing Company, Houston, II edition.
[19] Barbarelli, S., Bova, S., Piccione, R. (2009). Zero-dimensional model and pressure data analysis of a variable- displacement lubricating vane pump. SAE Technical Papers
[20] Idel'cick E. Memento des Pertes de Charge. Paris, France: Eyrolles Edition; 1986 [in French].