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Numerical and Economic Study of Centrifugal Pump as Turbine Performance

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Abstract

In this paper, performance of centrifugal pump as turbine (PAT) is investigated numerically. Three different specific speeds are considered and three pumps are designed using diagrams from catalogues and CFturbo V.9 software. Next, models are analyzed by Ansys CFX 16 software and results are compared with those of CFturbo software. Also, a mesh study analysis for one case is performed in order to show the effect of grid size on the solution. In addition, three different flow rates of 75%, 100%, and 125% of best efficiency point (BEP) are considered for extracting head-flow rate diagrams and comparing results of CFX and CFturbo software. In next step, using relations between pump and turbine modes (PAT formulations) and by changing boundary conditions in CFX, turbine mode is investigated and efficiency is compared with pump mode. Finally, by an economic analysis a comparison between PATs and turbines with same nominal output powers are performed to distinguish which case is more profitable. Results showed that PATs have lower payback time in comparison with turbines with equal output power (in low capacities), although they have lower efficiencies.

Keywords:

Pump as turbine, numerical study, specific speed, economic analysis.

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1. Introduction

Utilizing renewable energies due to lack of fusel fuels and environmental problems is a basic issue of energy policy. On the other hand, turbomachines play the most role among renewable energies. Electrical energy production is a vital problem in areas which are far from national power grid. Hence, using off-grid electrical production has attracted many attentions during past decades. One of the known solutions for areas with water sources, is utilizing pumps operating as turbines (PATs). This solution has some advantages such as low construction, operating, and maintenance costs, and simplicity of design in comparison with water turbines.

In background of PATs many researches have been done. Most recent efforts to predict performance of PAT, have been made using CFD [1– 3]. For instance, in 1961 Kittredge [4] analyzed the capability of centrifugal pump to operate as a turbine where the required power is rather small and low initial cost is more important than high efficiency. In this research, he showed how to select a pump which will meet specified requirements. in 1988 Gantar [5] tested models of propeller pumps in reverse mode and proposed that, like radial type, the working point in turbine mode operation of the pumps is roughly estimated based on pump characteristics. In 1994 Williams [6] used eight methods for predicting the turbine performance based on the data for pump performance at best efficiency point and compared them using an analysis of the effects of turbine prediction on the operation of a pump as turbine at a typical micro-hydro site. The tests were performed on 35 different pumps and the comparison used their results. In 1996 Williams [7] discussed a method with three different types of PATs for micro-hydropower plants. Also, he explained that utilizing induction generator and controller (IGC) design enables the PAT units to be used for isolated micro-hydropower projects. In 1999 Ramos and Borga [8] considering local needs and working conditions of pumps in turbine mode, introduced PATs as a novel way for energy production.

In 2007 Rawal and Kshirsagar [9] worked on numerical simulation on a pump operating in turbine mode and then compared the results with practical tests. They used numerical analysis to investigate various parameters that cannot be measured experimentally, e.g., internal hydraulic losses and flow pattern. In 2007 Isbasoiu et al [10] studied different aspects of standard pump working as turbine. The research indicates that PAT is an economical way for power production in small hydropower sites. The range of heads and flow rates which PAT can operate over them was discussed in the study. Performance curves in pump and turbine modes and different system control methods with PAT were also studied. The procedure for the selection of a pump as turbine for a particular site has been explained. In 2008 Derakhshan and nourbakhsh [11] made an experimental research on characteristic curves of centrifugal pumps working as turbines in different specific speeds (below 60) and by utilizing experimental information offered two equations for investigating characteristic curves of pumps in turbine mode and finally presented a process for determination working point in turbine mode. In 2009 Derakhshan et al [12] by focusing on a pump impeller redesigned the shape of blades to reach a higher efficiency in turbine mode using a gradient based optimization algorithm coupled by a 3D Navier–Stokes flow solver. Also, by rounding the blades' leading edges and hub-shroud interface in turbine mode they made another modification. They tested their model by manufacturing a modified one and showed that the efficiency of the pump in reverse operation can be improved by impeller modification. In 2011 Morros et al [13] used numerical methodology to observe a commercial pump working as a turbine. They focused on determination flow patterns with special interest in the unsteady behavior in order to explain the shape of the performance curves. Complementarily, an experimental study was conducted to validate the numerical model and characterize the pump-turbine performance curves at constant head. The study demonstrated that numerical methodology has reliable results. In addition, the efficiency for the inverse mode is shown to be as high as achieved for the pumping operational mode and the commercial design of the pump allows a reasonable use of the impeller as a turbine runner. In 2011 Fecarotta et al [14] studied CFD and comparisons for a PAT, mesh reliability and performance concerns. They concluded solving reliability is depended on flow regime and number of elements considered for flow meshing comparing the achieved data. They finally attained critical number of mesh elements. In 2012 yang et al [15] studied theoretical, numerical and experimental prediction of pump as turbine performance. They analyzed relations between pump mode and turbine mode performance for a same pump utilizing CFD. Moreover, in order to check the accuracy of the theoretical and numerical results they prototyped the pump and tested on it. In 2015 Dribssa et al [16] started numerical analysis using ANSYS-CFX and presented some corrections to refine the PAT operating in order to find the best operating conditions for a pump in turbine mode. In 2017 Frosina et al [17] presented a method to prognosis the characteristic of industrial centrifugal pumps in reverse mode. The results are based on results of simulations performed by commercial 3D CFD software. Model results have been first validated in pumping mode using data supplied by pump manufacturers. Then, they have been compared to experimental data for a pump running in turbine operation.

In this paper, three centrifugal pumps with different specific speeds are designed using CFturbo 9 software. Next, designed models are meshed in Ansys turboGrid 16 software and then are analyzed in Ansys CFX 16. On the other hand, utilizing relations between design points of pumps and turbines in PAT mode, pervious models are analyzed in turbine mode by Ansys CFX 16. Then, hydrodynamic characteristics of pump and turbine modes are compared. Finally, economic aspects of PATs are discussed and a financial comparison between PATs and turbines is done.

2. Methodology

In this paper first of all three different centrifugal pumps with different specific speeds are designed using CFturbo 9 software. Considered pumps in this paper have design points as shown in Table 1. The effort was selecting the models among high, medium, and low amounts of specific speeds of centrifugal pumps, since the pumps which are utilized to operate as turbines must be centrifugal. Also, for calculating specific speed of each pump, following equation is utilized:

$$N_s = \frac{n\sqrt{Q}}{H^{0.75}} \quad (1)$$

where:

N_s: Specific speed

n: Revolutions (rpm)

Q: Flow rate of design point (m^3/h)

H: Head of design point (m)

The set of solved equations contain continuity and unsteady Navier-Stokes equations in their conservation form. The instantaneous equations of mass and momentum conservation can be written as following in a stationary frame.

The continuity equation is as the following:

Table 1. Design points of considered pumps

Type	Flow rate (m^3/h)	Head (m)	Revolution (rpm)	specific speed
Low specific speed	13.25	13	1450	**12.85**
Medium specific speed	100	32	1450	**17.96**
High specific speed	87	21	1450	**22.98**

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho U) = 0 \quad (2)$$

where ρ and U are density and velocity of the working fluid respectively.

Also, the momentum equation can be written as the following:

$$\frac{\partial (\rho U)}{\partial t} + \nabla \cdot (\rho U \times U) = -\nabla P + \nabla \cdot \tau + S_M \quad (3)$$

where p, τ, and S_M are pressure, stress tensor, and term due to external momentum sources respectively. The stress tensor, τ, is related to the strain rate by the following equation:

$$\tau = \mu \left(\nabla U + (\nabla U)^T - \frac{2}{3} \delta \nabla \cdot U \right) \quad (4)$$

where δ is strain rate. The term S_M is considered to be zero, because there are no external momentum sources.

Total head of pump can be obtained utilizing below equation assuming that difference of heights between inlet and outlet is negligible:

$$H_{pump} = \frac{P_{out} - P_{in}}{\rho g} \quad (5)$$

where P_{out} and P_{in} are pressures at inlet and outlet respectively.

Hydraulic efficiency can be calculated by the following equation:

$$\eta = \frac{\rho g Q H}{(T\omega)_{in}} \quad (6)$$

where T_{in} and ω_{in} are torque and number of revolutions of pump at inlet respectively.

In turbine mode analysis working point must be found. For this purpose, in past researches some equations have been derived. In the following a process will be shown to find the working point in turbine mode:

First step: obtaining α_p, which is turbine dimensionless specific speed, from Eq. 7 and placing in Eq. 8 [11]:

$$\alpha_P = \frac{N_{SP}}{g^{0.75}} \quad (7)$$

$$\gamma = 0.0233\alpha_P + 0.6464 \quad (8)$$

where γ is a dimensionless parameter that will be utilized for obtaining rotational speed of turbine.

Second step: obtaining h (head number) from the following equation [15]:

$$h = \frac{1.2}{\eta_P^{1.1}} \tag{9}$$

In above equation η_p is hydraulic efficiency which is calculated by ANSYS CFX.

Third step: Putting h and η_p in following equation to obtain optimized turbine rotational speed [11]:

$$N_{turbine} = N_{pump} \cdot \gamma \cdot h^{0.5} \tag{10}$$

where $N_{turbine}$ and N_{pump} are rotational speeds of turbine and pump modes respectively.

Fourth step: Obtaining optimized head and flow rate:

For this purpose head number and flow rate number are defined as [11]:

$$h = \frac{H_t}{H_P} = \frac{1.2}{\eta_P^{1.1}} \tag{11}$$

$$q = \frac{Q_t}{Q_P} = \frac{1.2}{\eta_P^{0.55}} \tag{12}$$

where H and Q are head and volumetric flow rate respectively and t and p indexes are shown turbine and pump modes respectively.

Utilizing below equation, specific speed for turbine mode can be extracted in term of specific speed of pump mode [11]:

$$N_{sp} = 1.125N_{st} + 1.73 \tag{13}$$

Finally, power output of a turbine can be calculated by the following equation:

$$(P_{out})_t = (T\omega)_{out} \tag{14}$$

where T_{out} and ω_{out} are output torque and revolutions of turbine respectively.

3. Numerical modeling

The meshing process is performed on the generated model in CFturbo by Ansys Turbogrid. The meshed model with medium speed is shown in Fig. 1 as a sample. Also, Information of grids for three pumps is mentioned in Table 2.

Also, in order to investigate mesh independency, three different grids have been performed for medium speed pump. Results of mesh independency will be presented at results section. Boundary conditions for pump and turbine modes are illustrated in Figs. 2 and 3 respectively.

Also, types and descriptions of boundary conditions for pump and turbine modes are presented in Tables 3 and 4 respectively:

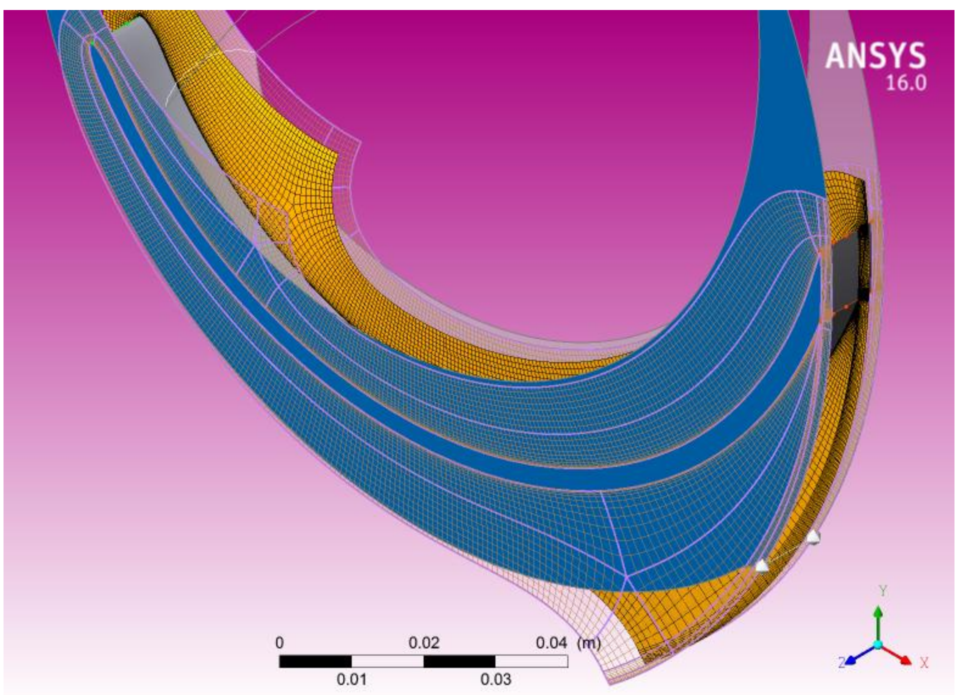
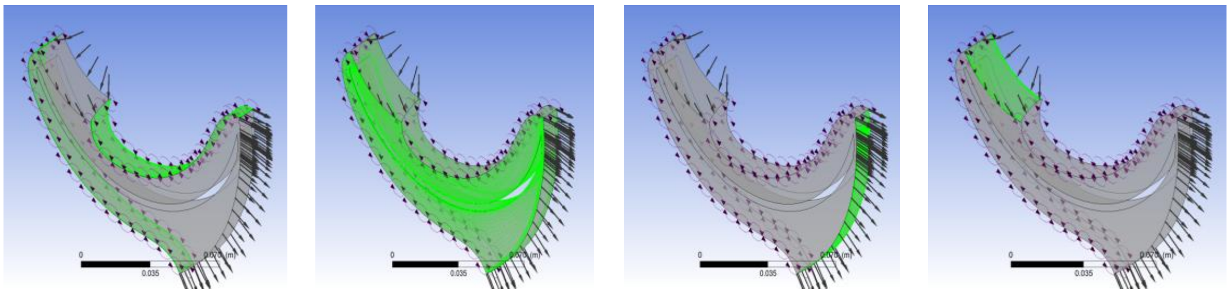


Figure 1. Gridded model of medium speed pump

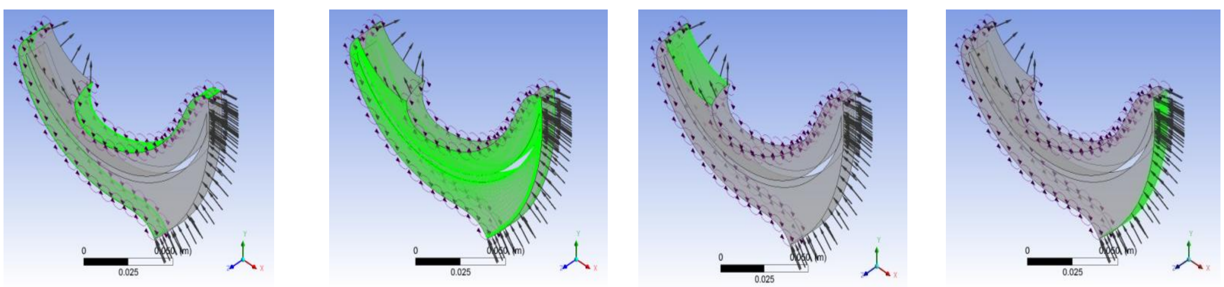
Table 2. Information of grids of pumps

N_s	Global size factor	Total nodes	Total elements
12.85	1.56	591,720	549,003
17.96	1.65	565,360	597,904
22.98	1.52	566,640	531,010



(a) (b) (c) (d)

Figure 2. a) Inlet b) outlet c) rotation d) periodic boundary conditions of pump mode



(a) (b) (c) (d)

Figure 3. a) Inlet b) outlet c) rotation d) periodic boundary conditions of turbine mode

Table 3. Define and description of boundary conditions in pump mode

Boundary	Option	Unit	Description		
			N_s=12.85	**N_s=17.96**	**N_s=22.98**
Inlet (1/6 of total flow rate)	Mass flow	(m^3/h)	2.2	16.66	14.5
Outlet	Stat. frame tot. press	atm	1	1	1
Rotation	Wall type		No slip	No slip	No slip
Periodic interfaces	Interface model		Rotational periodicity	Rotational periodicity	Rotational periodicity

Table 4. Define and description of boundary conditions in turbine mode

Boundary	Option	Unit	Description		
			N_s=12.85	**N_s=17.96**	**N_s=22.98**
Inlet (1/6 of total flow rate)	Mass flow	(m^3/h)	2.91	20.73	18.07
Outlet	Stat. frame tot. press	atm	1	1	1
Rotation	Wall type		No slip	No slip	No slip
Periodic interfaces	Interface model		Rotational periodicity	Rotational periodicity	Rotational periodicity

4. Results and discussion

Results extracted from CFturbo software can be seen in Table 5.

In Table 6, there is a comparison among results from Ansys CFX, CFturbo, and catalogues. As shown in this table, there are acceptable errors between Ansys CFX and CFturbo results. It should be noted that errors are calculated between Ansys CFX and CFturbo results, since CFX model is based on designed model by CFturbo.

Also, as mentioned before, a mesh independency investigation has been organized for medium speed pump with four different grids. One of the grids is base model and three other ones have much and lower nodes than the base one. The results are shown in Fig. 4. As shown in this figure, efficiency does not change significantly with grid (lower than one percent) and it can be said that the solution has acceptable independency of grid.

For more comparison among results of CFX, CFturbo, and diagram, three different volumetric flow rates are considered in order to achieve to the head-flow rate curves of each pump: best efficiency point (BEP) flow rate, 75%, and 125% of BEP flow rate. In fact, this is allowable region of working flow rate in order to prevent cavitation phenomenon and motor failure. So, head-flow rate diagrams of each point resulted from CFX, CFturbo, and diagram are demonstrated in Fig. 5. As shown in this figure, difference among CFX, CFturbo, and diagram heads decreases by increasing volumetric flow rate. This might be for some losses that CFX does not consider such as disk and leakage losses which their effects decrease with increasing volumetric flow rate and cause lower difference between heads resulted from CFturbo and CFX. It is worthy to say that the BEP for diagram and CFturbo are coinciding with each other, because the design of the pumps in CFturbo are based on the values of the diagrams.

Table 5. Design points of considered pumps resulted from CFturbo V.9

Parameter	Unit	Symbol	N_s=12.85	N_s=17.96	N_s=22.98
		Design Point			
Flow rate	m^3/h	Q	13.25	100	87
Revolutions	1/min	n	1450	1450	1450
Casing efficiency	%	η_c	90	90	90
Head	m	H	13	32	21
		Main Dimensions			
Hub diameter	mm	d_H	13.3	32.1	26.3
Suction diameter	mm	d_S	70.6	134	124
Impeller diameter	mm	d_2	210	326	269
Outlet width	mm	b_2	10.05	20.6	19.6
		Blade properties			
Number of blades	-	Z	6	6	6
Thickness leading edge	mm	s_{LE}	3.1	3.9	3.8
Thickness trailing edge	mm	s_{TE}	3.7	4.4	4.3
Angle leading edge	°	β_2	12.5	14.4	16.2
Angle trailing edge	°	β_1	16.7	22.4	23.5
		Efficiency			
Hydraulic efficiency	%	η_h	72.6	86.1	88

Table 6. Comparison between head and efficiency of pumps between different methods

Method	Diagram		CFturbo		Ansys CFX		Error (%)	
N_s	**H (m)**	**η (%)**	**H (m)**	**η (%)**	**H (m)**	**η (%)**	**H**	**η**
12.85	13	56	13	70.2	12.24	72.6	6.21	2.64
17.96	32	74	32	86.1	29.79	87.7	7.38	1.93
22.98	21	76	21	87.2	19.69	88	6.65	0.87

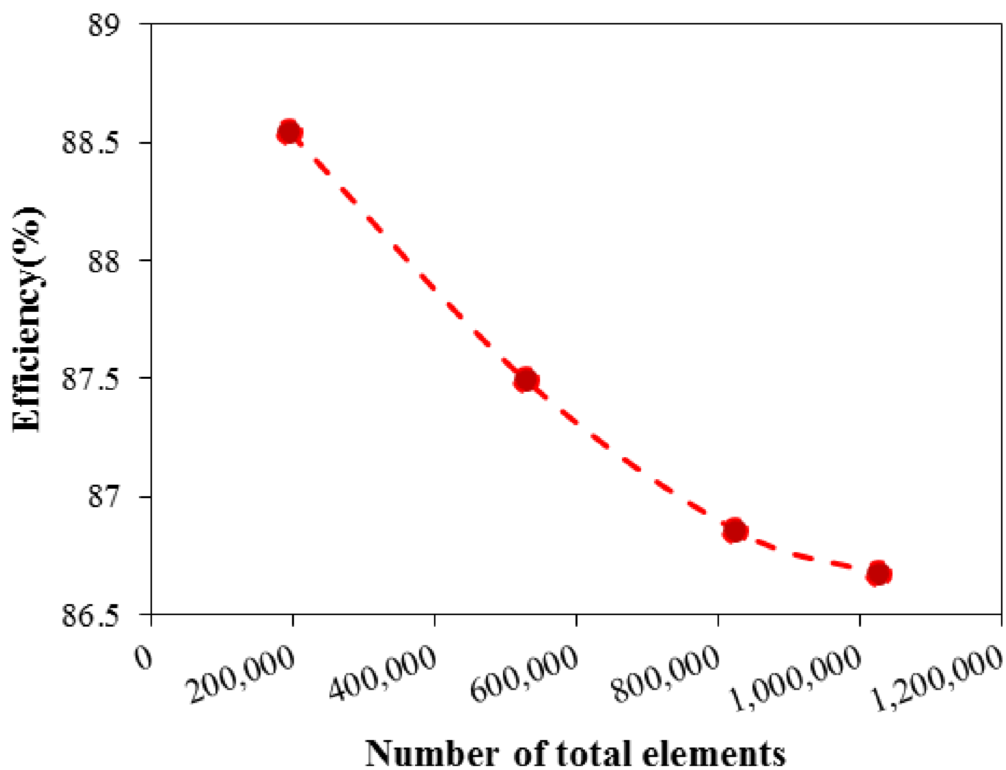


Figure 4. Mesh independency for medium speed pump

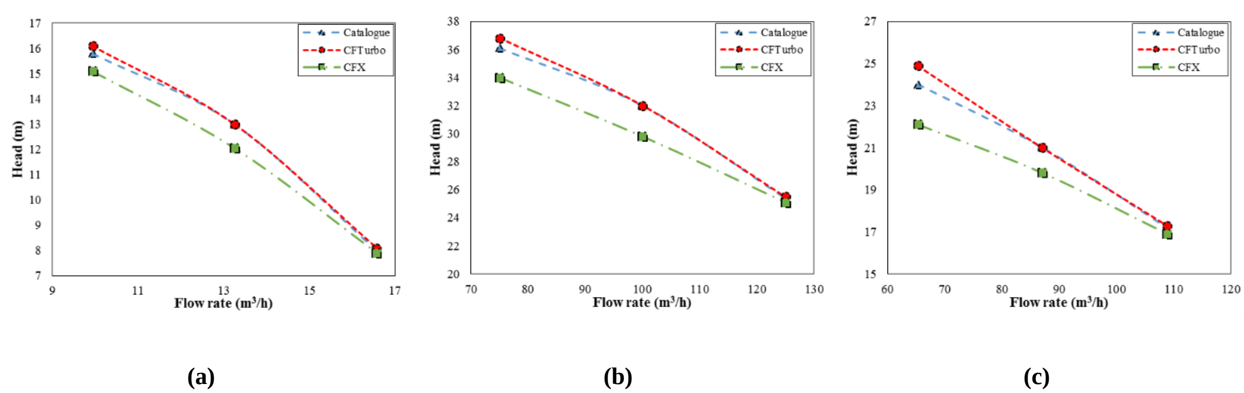


Figure 5. Head-flow rate diagram comparison between CFX, CFturbo, and diagram for a) low specific speed b) medium specific speed and c) high specific speed pumps

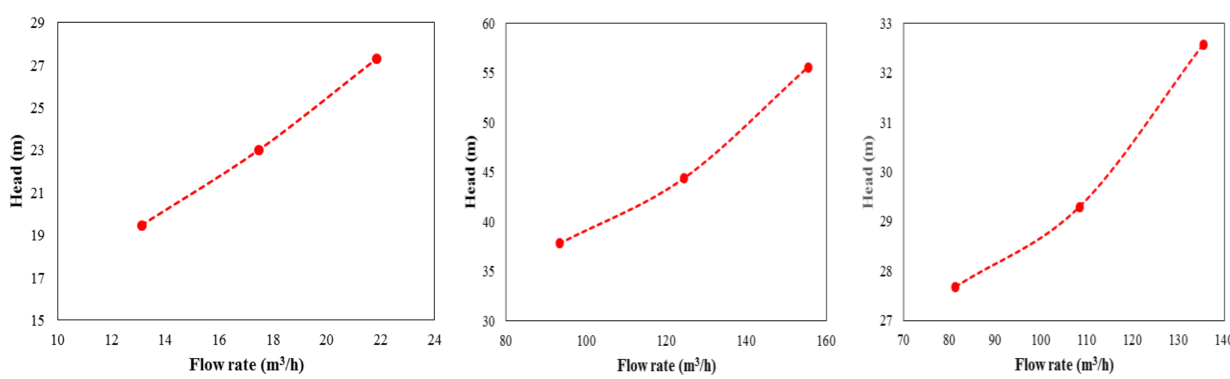
By mentioned relations in methodology section and numerical modeling in CFX software, results in turbine mode are extracted and tabulated in Table 7. Also, comparison with their pump modes is presented in this table. It was predictable that efficiency in turbine mode is less than pump mode, since design of blades and other components are based on pump mode, so for turbine mode they are considered as off design parameters and efficiency reduces as working point goes away from design point.

Table 7. Comparison between pump and turbine modes results

N_s	Mode	n (rpm)	N_s	Q (m^3/h)	η (%)
12.85	Pump	1450	12.85	13.25	70.2
	Turbine	1223.36	9.88	17.48	63.0
17.96	Pump	1450	17.96	100	87.7
	Turbine	1179.63	14.43	124.37	70.4
22.98	Pump	1450	22.98	87	87.2
	Turbine	1218.88	18.88	108.39	69.4

Also, the head-flow rate diagrams of the selected turbines are illustrated in Fig. 6. Finally, an economic analysis should be performed to identify whether PAT or turbine is more profitable. For this purpose, costs of each pump and its equivalent turbine (which has approximately same nominal power as the PAT of current work) are extracted from IRENA reports [18]. Power generation during a year based on their efficiency and by assuming that load factor is equal to 1 are calculated too which are shown in Table 8. Finally, by considering that cost of electricity in Iran is equal to 0.015 $ per kW, power generation investment is also calculated.

As seen in Table 8, utilizing PAT has much lower payback time and is desirable as economic aspect in low capacities. As power increases using both PATs and turbines will be more rational. It should be noted that investigated powers in this work are so low and generally they are not economical. It is necessary to note that usually the application of PATs is generating power for small usages in the regions with proper water accessibilities. For gathering more output power some PATs can be utilized simultaneously and in that case certainly the project has more economic justification. For higher power potentials (usually more than 50 kW) PAT does not have appropriate efficiency and system does not have any economic interest. So, in that case hydropower systems containing water turbines are used.



(a) **(b)** **(c)**

Figure 6. Head-flow rate diagram of the selected turbines for a) low specific speed b) medium specific speed and c) high specific speed

Table 8. Economic comparison between PAT and turbine

N_s	Type	Nominal output power (kW)	Efficiency (%)	Cost (US$)	Power generation during a year (kWh)	Power generation investment during a year (US$)	Normal payback time (year)
12.85	PAT	0.10	79.4	250	695	10.43	23.97
	Turbine	0.10	90.0	450	788	11.83	38.04
17.96	PAT	1.09	83.9	1000	9070	136.05	7.35
	Turbine	1.10	95.0	4950	9154	137.31	36.05
22.98	PAT	0.67	83.3	650	5874	88.11	7.38
	Turbine	0.75	95.0	3375	6242	93.63	36.05

5. Conclusion

Small hydropower as an alternative of off-grid electricity production has been attracted a lot of attentions. PATs are a new kind of applications of pumps that have many attractive aspects such as economic aspect, low cost design and maintenance, etc. In this paper, utilizing pumps as turbines was investigated numerically and also an economic analysis was produced in order to compare between PATs and water turbines. Briefly, results of current work are described below:

- The hydraulic pump efficiency calculated by CFX is more than one calculated by CFturbo, because in CFX some terms of losses such as disk and leakage dissipations are not regarded.
- Based on head-flow rate diagrams extracted from CFX and CFturbo, there are some differences between exported diagrams from this two software in low flow rates. This is due to ignoring some dissipations in CFX analysis. As the flow rate increases, the roll of these losses decreases and consequently the gaps between diagrams reduce.
- Utilizing presented correlations in previous researches, working points for turbine mode of designed pumps were found. So, this is the most effective source of reduction in efficiencies in turbine mode in comparison with pump mode. This is because of not working in designed conditions and boundary layer separation, which is resulted in creating new vortexes in inlet and outlet of the blades.
- According to economic analysis, PATs have lower payback time in comparison with turbines with equal output power (in low capacities) although they have lower efficiencies. Using two or more PATs in parallel for gaining more power is more economical. So, utilizing PATs for rural areas with water sources where do not have access to national power grid is affordable.

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