Features of Using Pumps in Turbine Mode

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Abstract: One of the possible directions of using renewable energy sources to save fuel and energy resources in Uzbekistan is the use of the hydropower potential of small rivers. The cost of electricity generated at micro and small hydroelectric power plant is already lower than the cost of electricity generated at traditional types of power plants, including gas turbine, wind, nuclear power plant and thermal power plant. In addition, due to constantly rising energy prices, the cost of electricity at traditional power plants is constantly increasing. The use of the hydropower potential of small rivers will contribute to the decentralization of the integrated energy system and improve energy supply to remote and hard-to-reach rural areas. Analysis of literature sources shows that the use of serial pumps as hydraulic turbines is a successful alternative to the use of specially designed hydraulic turbines for mini and small hydroelectric power plant. To reliably use pumps as hydraulic turbines, manufacturers need to obtain experimental performance of pumps in turbine mode, which can help expand their markets and better utilize available hydraulic potential.

1 INTRODUCTION

One of the possible directions of using renewable energy sources (RES) to save fuel and energy resources in Uzbekistan is the use of the hydropower potential of small rivers. The cost of electricity generated at micro and small hydroelectric power plant (HPP) is already lower than the cost of electricity generated at traditional types of power plants, including gas turbine, wind, nuclear power plant (NPP) and thermal power plant (TPP). In addition, due to constantly rising energy prices, the cost of electricity at traditional power plants is constantly increasing. The use of the hydropower potential of small rivers will contribute to the decentralization of the integrated energy system and improve energy supply to remote and hard-to-reach rural areas. This is the way the EU countries have gone. For example, in Switzerland, the percentage of electricity production at small HPP has already reached 8.3%, in Spain – 2.8%, in Sweden – almost 3%, and in Austria – all 10%. China, which has become a leader in the creation of small hydropower in recent decades (about 18-20% of all electricity in the country is supplied by more than 80 thousand small hydropower plants) and the former Soviet republics (Lithuania, Kyrgyzstan, Armenia, etc.) are moving in the same way. An important component of this process is the use of mini and microHPP for decentralized energy supply and thermal energy generation for consumers remote from industrial networks, for example, in mountainous areas, farms, etc. The increase in the cost of energy resources has stimulated interest in the development of new potential energy sources, which can serve as existing differences in irrigation canals, aqueducts, water supply and technological networks, and even transport oil pipelines.

As a first step in Uzbekistan to stimulate the use of renewable energy sources at the legislative level, one can note the adoption of so-called “green tariffs”, which oblige the purchase of electricity produced from them at specially established prices, similar to the practice that exists in European countries.

One of the reasons hindering the widespread introduction of mini and microHPP in Uzbekistan is the high cost per kilowatt of installed equipment capacity, which is associated with the virtual absence of specialized companies performing the entire range of work - from equipment development, its manufacture and installation with subsequent commissioning and service. Abroad, there are more than 300 manufacturers of hydraulic turbine equipment of various types of mini and micro hydroelectric power stations for various parameters, but the cost of the equipment is quite high and amounts to about 1,500-2,000 $/kW, which leads to significant payback periods for the equipment.

An alternative is the proposal to use commercially produced pumps as hydraulic turbines for microHPP, which in the power range of 5÷1000 kW can successfully compete with most used types of hydraulic turbines with significantly shorter payback periods.

Research into the possibility of using pumps as turbines to generate energy was carried out in the 90s in Germany [1], England [2], Iran [3], India [4, 5] and other countries. The recent increase in the number of publications on research on this topic in England, the USA, Russia, as well as on specific applications in developing countries [5, 6] confirms the need and prospects for using pumps in small-scale power generation.

This proposal is based on the proposition known from the theory and practice of hydraulic engineering that the hydraulic machine is reversible, and the pumps have fairly high efficiency in turbine mode and have some advantages over existing hydraulic turbines for mini and microHPP in the specified power range:

- there is no need to develop designs and establish small-scale production of equipment, since the pumps are already being produced, and the range is quite wide;
- serial pumps are usually supplied with an asynchronous motor, which can be used as a generator;
- high energy conversion efficiency, often corresponding to the level of specially designed hydraulic turbines;
- relatively low cost of equipment, which ensures quick payback for mini and microHPP, ease of maintenance and repair;
- the simplicity and compactness of the design makes it possible to consider various layouts of mini and microHPP units, which leads to a reduction in the cost of designing the construction part of mini and microHPP and construction time.

The disadvantages of using pumps as hydraulic turbines include:

- a steeply falling shape of the efficiency curve, close to the shape of a propeller hydraulic turbine, which requires either a narrow range of actuated pressures or the installation of a frequency converter that allows the generator to operate at a variable speed with subsequent conversion to an industrial frequency, which increases the cost of the installation;
- lack of experimental characteristics of pumps in turbine operating mode, which are necessary for the optimal selection of equipment for given parameters;
- the absence, as a rule, of regulatory bodies in the pump (except for a valve on the pressure pipeline, the use of which leads to a deterioration in operating conditions at partial loads).

In Figure 1 shows the areas of use for various types of hydraulic turbines used as mini and microHPP power equipment, and the area of pressures and powers in which serial pumps can be used [1]. From the above diagram it follows that serial pumps can replace mini and microHPP equipment for Banky type turbines, radial-axial Francis type and Pelton type active turbines in the pressure range H = 10-100 m.

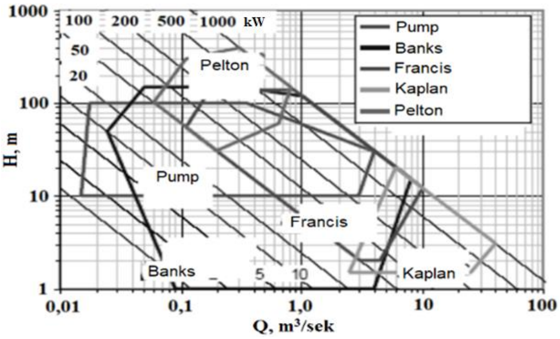

Figure 1: Areas of use of standard pumps as turbines according to [2].

To make a reasonable choice of the type of pump that provides the required parameters of the turbine mode in certain operating modes, it is necessary to have a characteristic in this mode. The lack of experimental characteristics of pumps in turbine mode leads to the need to establish, on the basis of a known pump characteristic, the relationship between the values of pressure $H_T\eta_{max}/H_P\eta_{max}$ and flow rate $Q_T\eta_{max}/Q_P\eta_{max}$ for the point with maximum efficiency in turbine and pump modes approximately using theoretical analysis [1-7], including using computational fluid dynamics (CFD) methods [8, 9].

Let us consider the flow features in pump and turbine modes for the pump impeller. These relationships can be approximately established based on the analysis of velocity triangles and the Euler equation

$$H_{Theory} = \frac{1}{g}\left(V_{Uguide} \cdot U_{Uguide} - V_{U\,BC} \cdot U_{U\,BC}\right). \quad (1)$$

Assuming that maximum efficiency values occur under the condition of flow supply to the impeller in pump mode without preliminary swirling ($V_{UBC} = 0$) and axial flow exit from the blade system in turbine mode ($V_{UBC} = 0$), (1) taking into account hydraulic the efficiency of the flow path η_g will be for the pump mode

$$\frac{H_{pump}}{\eta_{g\ pump}} = \frac{1}{g}\left(V_{U\ pump} \cdot U_{U\ pump}\right) \qquad (2)$$

and for turbine mode

$$H_{Turbine}\eta_{g\ turbine} = \frac{1}{g}\left(V_{U\ pump} \cdot U_{Turbine}\right). \qquad (3)$$

If the circulations created or activated by the blade system are equal, it follows from (2) and (3) that for points with maximum efficiency, the ratio of the pressures of the turbine and pump modes

$$H_{Turbine}\,\eta_{max}/H_{Pump}\,\eta_{max} = 1/\eta_{gP} \cdot \eta_{gT}. \qquad (4)$$

To determine the relationship between the flow rates $Q_{Turbine,\ max}/Q_{Pump,\ max}$ for the point with maximum efficiency in the turbine and pump modes, we consider the combined speed triangles in the turbine and pump modes on the pressure side of the impeller, shown in Figure 2.

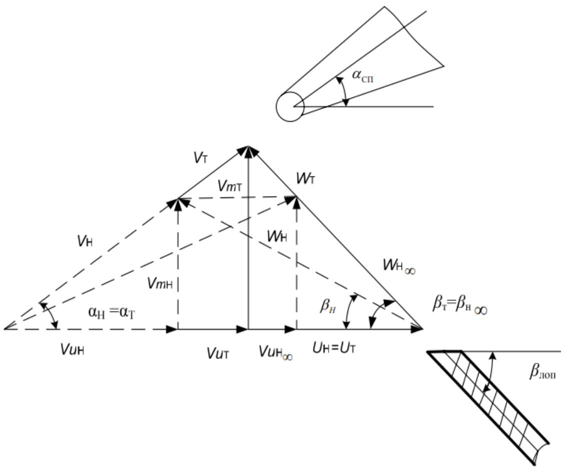

Figure 2: Combined speed triangles in turbine and pump modes on the pressure side of the impeller.

When calculating the geometry of the pump outlet elements, in order to achieve maximum efficiency of the flow path, the flow exit angle in absolute motion α_n must correspond to the outlet angle αsp. Consequently, in the turbine mode, the flow flows into the valve at an angle $\alpha_{sp} = \alpha_n$ in absolute motion, and the optimal mode is characterized by a minimum of shock losses at the inlet, i.e., the equality of the angles $\beta_t = \beta_{lop}$ in relative motion.

To achieve this, the flow rate V_{mT} must be higher in turbine mode than in pump mode. For the same reason, a pump in turbine mode has maximum efficiency at high flow rates compared to discharge mode, which leads to approximately the same maximum efficiency value or, especially for low speeds, even higher in turbine mode than in pump mode [3]. The relative magnitude of volumetric and disk losses at the point of maximum efficiency is lower due to the greater power developed by the river. in turbine mode.

2 METHODS

For a scheme with an infinite number of blades, the flow angle in relative motion β_n at the outlet in pump mode coincides with the blade installation angle β_{lop}, which also occurs in turbine mode. Due to the flow deviation due to the influence of a finite number of blades in pumping mode, the flow angle β_p decreases while maintaining the value of the flow component of the velocity V_{mP} compared to the scheme of an infinite number of blades. Thus, the ratio of flow rates at the optimum characteristics $Q_{T,max}/Q_{P,max}$ mainly depends on the magnitude of the flow deviation, which in turn is determined by the geometric parameters of the blade system.

To determine the magnitude of the flow deviation within the framework of the jet theory, there are several types of corrections for a finite number of blades, specified by empirical dependencies proposed by a number of authors. However, this method is based on very rough assumptions, since the shape of the blade profiles, the geometry of the inlet and outlet, and spatial effects in the flow part of the pump are not taken into account.

In most of the reviewed works [1-7], this issue was studied on the basis of experimental data and theoretical analysis. In practice, in the absence of information about the parameters of the turbine mode, approximate calculation formulas proposed by a number of authors are used.

For example, the relationships between the values of pressure $H_T\eta_{max}/H_P\eta_{max}$ and flow rate $Q_T\eta_{max}/Q_P\eta_{max}$ for the point with maximum efficiency in turbine and pump modes are, according to Stepanov's monograph [10]:

$$h = \frac{H_T\eta_{max}}{H_P\eta_{max}} = \frac{1}{\eta_{gP} \cdot \eta_{gT}}, \qquad q = \frac{Q_T\eta_{max}}{Q_P\eta_{max}} = \frac{1}{\sqrt{\eta_{gP} \cdot \eta_{gT}}}, \qquad (5)$$

which coincides with (4), and according to Sharma's data [11]:

$$h = \frac{H_{T\eta_{max}}}{H_{P\eta_{max}}} = \frac{1}{\eta_{maxP}^{0,8}},$$
$$q = \frac{H_{T\eta_{max}}}{H_{P\eta_{max}}} = \frac{1}{\eta_{maxP}^{1,2}}. \quad (6)$$

The results obtained from these relationships, depending on the speed studied, the type and design of the pump, had a very large scatter of results - from ±2 to ±20% deviations from the experimental data [9].

Theoretical experimental studies conducted model studies of two versions of impellers I-1 and I-3 for turbine and pump operating modes in the flow part of a reversible hydraulic machine with a speed of 200. The flow part of the hydraulic machine model consists of a spiral chamber having a speed coefficient in the inlet section K = 0,87; stator with toroidal rings and number of columns Z_1 = 8 of constant height b_1 = 0,149 D_1; guide vane (GV) with 16 blades (Z_0 = 16) height b_0 = 0,149 D_1 and symmetrical profile, diameter of the rotation axes D_0 = 1,16 D_1; RK ORO 70/5215 (I-1) with diameter D_1 = 0,35 m and number of blades Z_1 = 6, diameter on the suction side D_{ss} = 0,6 D1; curved suction pipe with a turbine-type elbow.

In addition to the described flow part of the reversible hydraulic machine, theoretical experimental studies were carried out on two versions of the I-1 and I-3 in the flow part with the dismantled GV. In the absence of GV, the flow part is identical to the pump flow part, which makes it possible to use the results of theoretical experimental studies to obtain experimental characteristics and refine approximate formulas for calculating hydraulic parameters in turbine mode.

The experimental characteristics of the studied version of the flow path with the I-1 I for turbine (TM) and pump modes (PM) are shown in Figures 3–5. In pump construction, it is customary to use dimensionless coefficients of pressure ψ, supply - flow F and power π, which are calculated using $\psi = \frac{H}{n^2 \cdot D^2}$, $\psi = \frac{Q}{n \cdot D^3}$, $\pi = \frac{P}{n^3 \cdot D^5}$ and are related to the given values $n_1^!$ and $Q_1^!$, usually used in turbine construction, by the following relationships:

$$\psi = \left(\frac{60}{n_1^!}\right)^2, F = Q_1^! \cdot \sqrt{\psi}.$$

The research results confirm the main provisions obtained from literary sources and the analysis performed. The maximum efficiency value in turbine mode occurs at higher pressures and flow rates than in pump mode, which leads to an increase in turbine power.

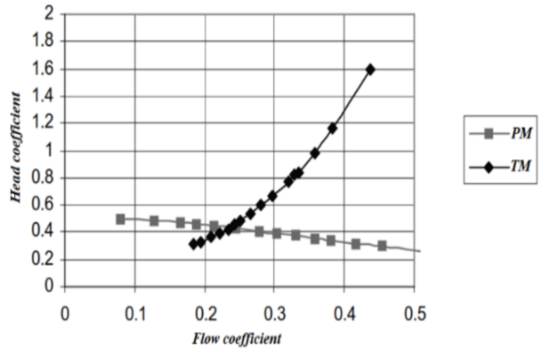

Figure 3: Dependence of the pressure coefficient in TM and PM on the flow coefficient.

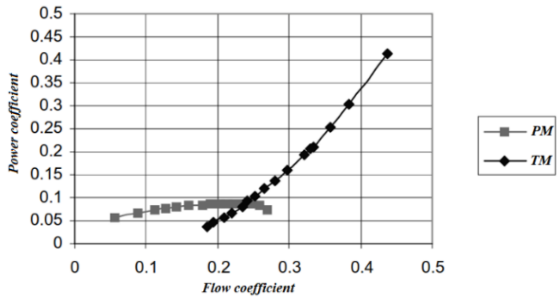

Figure 4: Dependence of the power factor in TM and PM on the flow coefficient.

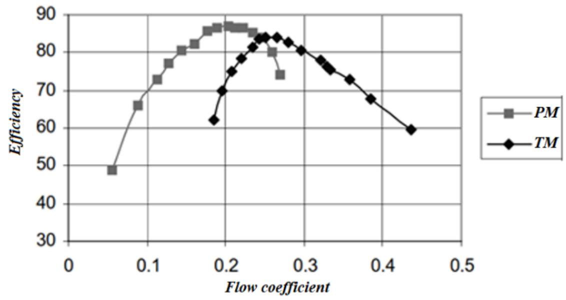

Figure 5: Dependence of efficiency in TM and PM on the flow coefficient.

3 RESULTS AND DISCUSSION

Experimental parameters in the optimum characteristics of turbine and pump modes for the studied flow parts in the absence of n. a. are given in Table 1.

Impeller type	Speed n	Efficiency		Experiment		Calculation by formula (5)		Calculation by formula (6)		Computational fluid dynamics calculation	
		η_{Pump}	$\eta_{Turbine}$	h	q	h	q	h	q	h	q
Pump [12]	297	0,91	0,84	1,25	1,379	1,263	1,238	1,2	1,119	-	-
Impeller - 1	205	0,91	0,882	1,238	1,208	1,245	1,116	1,1198	1,0783	-	-
Impeller - 3	195	0,934	9,894	1,09	1,137	1,195	1,0926	1,091	1,0597	-	-
Pump [5]	144	0,85	-	1,4	1,35	1,38	1,18	1,22	1,14	-	-
Pump [10]	56	0,89	-	1,518	1,25	1,52	1,23	1,15	1,097	1,55	1,219

Figure 6: Comparison of experimental and calculated values of *h* and *q*.

Table 1: Experimental parameters of impeller I-1 and I-3 at performance optimums.

Code of the flow part	Experimental parameters at performance optimums					
	Turbine mode			Pump mode		
	Q'_1, m³/s	n'_1, min⁻¹	η, %	Q'_1, m³/s	n'_1, min⁻¹	η, %
I-1	0,358	89,0	84,2	0,33	98,0	87,0
I-3	0,365	88,8	85,4	0,335	92,0	89,6

Comparison of experimentally obtained relationships between the values of pressure $h=H_T\eta_{max}/H_P\eta_{max}$ and flow rate $q=Q_T\eta_{max}/Q_P\eta_{max}$ at points with maximum efficiency in turbine and pumping modes for I-1 and I-3 with the results obtained from dependencies (5), (6), are given in Figure 6. For I-1, the values of h and q determined experimentally are closest to those determined by (4), which coincides with Stepanov’s recommendations, and for I-3, they are closer to the values determined according to Sharma’s recommendations. The indicated impeller have the same number of blades, values of b_0 and D_{vc} and blade installation angle β_{blade} on the pressure side, but differ in the geometry of the meridional section and the inter-blade channel. Optimization of the blade geometry leads to a significant increase in efficiency in turbine and especially in pump mode for I-3 and, accordingly, to a decrease in the values of *h* and *q* compared to the experimental ones. In addition, in Figure 6 shows a comparison of calculated and experimental data obtained from sources [3, 9] for pumps in the speed range *n* = 56-297. It also confirms the spread of the results of deviations of calculated and experimental data [9], reaching up to 5 in pressure and up to 12% in flow rate, and for the considered range of speeds Stepanov’s recommendations turn out to be closer to the experimental ones. Therefore, statistical methods for processing data from experimental studies, which were used in [3, 6, 9, 19], cannot provide acceptable accuracy in predicting the hydraulic parameters of turbine mode. A promising way to determine the ratio of pressures and flow rates at the optimum characteristics is, along with experimental ones, the use of computational methods of computational fluid dynamics in a three-dimensional formulation, which is confirmed by the results obtained in [8-18].

4 CONCLUSIONS

Analysis of literature sources shows that the use of serial pumps as hydraulic turbines is a successful alternative to the use of specially designed hydraulic turbines for mini and small HPP.

To reliably use pumps as hydraulic turbines, manufacturers need to obtain experimental performance of pumps in turbine mode, which can help expand their markets and better utilize available hydraulic potential.

Subject to the availability of geometry and the accumulation of certain experience, it is possible to use methods for calculating three-dimensional viscous flow in the flow part to obtain the necessary hydraulic parameters of serial pumps in turbine mode with satisfactory accuracy.

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