The swirl number as a method for determining the optimal operating mode of the micro hydro turbine

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Abstract. The paper presents the data of a detailed study of the flow characteristics behind the runner of an air model of a propeller-type micro hydro turbine with varying operating modes from partial load to severe overload. Detailed measurements of the flow field distributions were carried out using an automated system for contactless optical diagnostics (LDA). The obtained data made it possible to link the identified features of the development of the flow structure when changing the operating mode of the installation with the nature of the evolution of the integral swirl number that determines the state of the swirled flow. Eventually, the work results can be used in the elaboration of recommendations for extending the regulation range of the operating regimes of hydraulic microturbines and providing their high efficiency.

1. Introduction
Nowadays, hydropower is the oldest field of renewable energy [1]. Hydroelectric power plants have high efficiency and flexibility in managing energy production. The possibilities of hydropower are largely determined by the territory under consideration. In individual territories, the cheapest energy is generated at small hydroelectric power plants, they are unpretentious and reliable in operation since the failure of one element does not entail disconnecting significant territories from electricity.

Natural height changes on various spillways, irrigation systems, pipelines, and industrial spillways can be used as energy sources for small hydroelectric power plants. However, the issue of automating the management of small hydroelectric power plants is acute, since they are located in hard-to-reach areas and must be serviced with minimal involvement of specialists. A partial solution to the problem may be introducing single, standard equipment. The choice of the type of turbine used depends on the level of the height difference at the inlet and outlet, on the water flow rate, the required turbine power, the degree of uniformity of the water flow, etc. [2].

A significant criterion for micro hydro turbines is their price for consumers, which is particularly important for units with a power of few kilowatts, which are generally employed for energy generation in individual households [3,4]. Due to this, axial hydraulic turbines of the propeller type draw much attention [2,5,6]. They are distinguished by the simplicity of the design and low manufacturing costs, as well as specific speed (specific speed of the impeller rotation). At the same time, the practical use of axial hydraulic units is hampered by the narrow range of effective operation of a propeller hydro turbine with fixed blade mounting angles [7]. This feature requires precise adjustment of the operation mode of the hydro turbine to the conditions of the available water resource. It should be noted that in
relation to large hydraulic turbines, as a rule, underloading modes are studied [8], since they are more common in practice. At the same time, in the literature, much less attention is paid to modes with overload [9].

This work is devoted to the experimental study of the flow characteristics behind the impeller of a model micro hydro turbine with a power of about 1 kW when changing the operating modes from partial load to severe overload, as well as considering the possibility of using the integral swirl parameter to find an effective mode of operation of a micro hydro turbine.

2. Experimental setup
A model of a propeller-type turbine was chosen as a prototype of a micro hydro turbine device. The model of a micro hydro turbine (Fig. 1) consists of a series of fixed and rotating blade swirlers. The geometric shape of the blade swirlers and the order of their location allows simulating the speed distribution at the output of a real hydraulic turbine [10]. The rotating swirler is driven through a shaft connected to a servo drive. The propeller turbine has a high speed among other types of turbines. This makes it possible to obtain a high rotation speed at low flow rates. High turbine speeds lead to using faster, and therefore lighter and cheaper electric generators. The high specific speed makes it possible to connect the turbine directly to the generator [11].

For the used blade swirlers, it was obtained by numerical modeling [10] that the optimal mode providing the highest efficiency corresponded to the flow rate \( Q_c = 176.4 \text{ m}^3/\text{h} \) at the rotation speed of the swirler \( n_c = 37.9 \text{ Hz} \).

All parts of the model section were manufactured using rapid prototyping technology. The diameter of the input part of the cone was \( D = 100 \text{ mm} \), the output part was \( 1.2D \). The length of the cone was \( 2.8D \), the angle of the cone was \( 4^\circ \). A part of the cone wall was replaced with a transparent film to allow laser beams to pass inside the cone. The rotation frequency of the swirler was set with an error of 0.5%. The air flow rate was set via a frequency converter that controlled the blower. For a micro hydro turbine with the specified geometric dimensions, operating on water, the efficiency in the optimal mode is equal to about 75% [2].

When the flow rate and impeller rotation speed are changed proportionally, the flow characteristics are kept to be the same [8]. This allows confining the studied range of operating parameters to a constant impeller speed. So that in these experiments the flow rate was varied from partial load (60% of the nominal flow rate) to high overload (80% excess of the nominal flow rate) while the speed of the impeller was fixed at value \( n_c \).

In the studies, profiles of two velocity components with a detailed spatial resolution for different flow rates were obtained. Velocity distributions were measured using a two-component laser-Doppler anemometer (LDA). The flow seeding required for the LDA operation was carried out using an aerosol of paraffin oil consisting of tiny droplets, which reliably track the flow [8]. The nonintrusive LDA equipment was set on a computerized three-axis coordinate system to perform automated acquiring the velocity distributions. During the experiments around \( 5 \cdot 10^3 \) bursts per each velocity component were collected at an average particle’s rate of around 1 kHz.

3. Results and discussion
When shifting the operation conditions of the hydro turbine apart from the nominal regime, the flow passing the impeller acquires a swirl. The swirling level of the flow is commonly represented by the swirl parameter \( S \) determined in turn as the ratio of the axial flux of the angular momentum \( M \) to the product of the axial flux of the axial momentum \( G \) and the characteristic radius \( R \) [12]:

\[
S = \frac{M}{RG}
\]
The momenta $M$ and $G$ in equation (1) can be calculated based on the measured velocity distributions ($V$ is the axial component and $U$ is the tangential component; $D_0$ is the channel diameter in the measured cross-section):\

$$S = \frac{2 \int_0^{D_0/2} UV r^2 dr}{D_0 \int_0^{D_0/2} V^2 r dr}$$ (2)\

It is common also to define the flow swirling degree through the swirler geometry [13]. In the case of a hydro turbine, the degree of flow swirling at the hydro turbine outlet depends not only on the geometry of the turbine but in addition on its rotation speed and flow rate. To evaluate the resulting swirl number as a function of the flow rate and rotation speed, an approach proposed in [14] can be used. For a given angular velocity of the runner, it is possible to select a flow rate $Q=Q_0$ at which the absolute velocity has only an axial component. For our experimental setup $Q_0 = 1.08Q_c$. As a first approximation, the axial velocity distribution over the impeller area can be set uniform. Then an analytical expression for the swirl number as a function of the flow rate $Q$ at constant rotation speed $n_c$ can be written:

$$S = \frac{\pi^2 Dn_c}{8Q_0} (D^2 - d^2) \left(\frac{Q_0}{Q} - 1\right),$$ (3)\

where $D$ is the diameter of the impeller, $d$ is the diameter of the cowl, $d = 0.4D$. It can be seen that expression (3) gives an inversely proportional dependence of the swirl number on the flow rate and a change in the swirl direction at $Q>Q_0$.

Figure 2 shows a comparison of the swirl number values, obtained by integrating the measured velocity profiles using formula (2), and the values, calculated based on the analytical expression (3). A good agreement of both curves for the studied range of operating parameters can be noted. The measurements were performed at a distance $D$ from the start of the conical outlet section with a step of 0.01$D$ in the radial direction. It should also be noted that for the nominal flow rate $Q_c$ for which the blade elements of the hydro turbine were designed, the swirl number is somewhat different from zero and has a positive value around 0.1. This is due to the practice of designing modern hydro turbines, where a small positive swirl of the flow is initially taken as the nominal (optimal) mode [8]. The
meaning of the presented results is related to the fact that in practice the best operation mode in terms of efficiency is searched close to the regime with normal output [8] or zero swirl. This regime can be readily identified using the analytical dependence (3) based on a single test point. Assuming a proportional changing of zero swirl flow rate \( Q_0 \) at changing the speed of impeller rotation, the characteristic curves for swirl number \( S \) can be calculated for different rotation speeds of the impeller.

Experimental and the analytical dependences presented in the swirl number diagram (Figure 2) show similar ranges for the swirl number variation at overload and part-load regimes. At part-load regimes, with \( Q_0/Q > 1 \) the direction of flow swirling corresponds to the impeller’s swirl sign. At overload mode, the flow is swirled in opposite direction. It can be seen that the swirl number \( S \) just slightly exceeds the absolute value of 0.3 even for the largest flow rates while it reaches levels almost equal to 0.7 in part-load regimes. This difference in the evolution of swirl number \( S \) would imply different flow states at the transition to part-load and overload regimes. In the first case, the flow swirl number exceeds the critical value of 0.5 at which the vortex breakdown normally takes place [8]. This means the development of a central recirculation zone, shifting the flow towards the cone wall, and the emergence of unsteady large-scale vortical structures in the form of precessing vortex core (PVC) [12]. In the second case, the swirl number remains below the threshold level. As a result, the flow structure is stable, the maxima of the tangential velocity profiles are near the flow axis, and the axial velocity profile is uniform over the cross-section of the draft tube without forming an area of the flow deceleration along the axis.

![Figure 2. Flow swirl number: comparison of the results of the experiment, formula (2), and the calculation using the analytical dependence (3)](image)

4. Conclusions
In this paper, detailed studies of the flow characteristics behind the impeller of an air model of a propeller-type micro hydro turbine with varying operating modes from partial load to strong overload are carried out. The measurement data, obtained using the automated LDA system includes the distributions of the time-averaged velocities. The velocity data are used to determine the swirl number variation as a function of flow rate and impeller rotation speed. The results show that in underloaded
modes, the level of flow swirl quickly reaches a critical value. Strong flow swirling under suboptimal conditions leads to energy losses and a decrease in the efficiency of the hydraulic turbine. It is shown that using the integral swirl parameter, it is possible to quickly identify the most effective mode of operation of a micro hydro turbine. This stage will significantly save time when designing equipment for specific field conditions of a water resource.

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