Flow characterization in an axial micro-hydropower model

D A Suslov¹,²*, S I Shtork¹,², I V Litvinov¹,² and E U Gorelikov¹,²

¹S S Kutateladze Institute of Thermophysics SB RAS, 630090, Russia, Novosibirsk, Lavrentyev Avenue, 1
²Novosibirsk State University, 630090, Russia, Novosibirsk, Pirogova Street, 1

E-mail: d.suslov@g.nsu.ru

Abstract. The flow characteristics behind the runner of an air model of a propeller-type micro-hydropower were studied in detail by varying the operation conditions from part-load to high overload. The Reynolds number was varied from $3 \times 10^4$ to $9 \times 10^4$, and the swirl number from 0.7 to -0.4. An automated laser-Doppler anemometer (LDA) system for non-contact optical diagnostics was used to perform detailed measurements of the flow field distribution, including the profiles of two components of averaged velocities and pulsations and LDA signal spectra. Based on the results, a correlation was found between the identified features of the development of the flow structure under changing operating conditions of the hydroturbine and the nature of the evolution of the integral swirl number, which determines the state of the swirling flow. This can be used to develop recommendations for expanding the range of regulation of hydroturbine operation while maintaining high efficiency.

1. Introduction

Increasing interest in micro hydropower is associated with the general trend toward the increasingly wider use of renewable energy sources to reduce the environmental impact of fossil fuel usage[1–4]. The development of small hydropower is of great social importance because small hydropower plants can be more reliable sources of energy than solar and wind power systems in remote areas where centralized power supply is impractical due to the high cost of the network [3, 5].

Cost is an important parameter of micro-hydroturbines, especially for small pico-hydroturbines with a capacity of less than 5 kW, which are often used to supply power to individual houses [1, 3, 5]. In this regard, propeller hydraulic turbines have attracted particular attention [1, 2, 6–8]. They have a simple design, a low production cost, and a high specific speed. The latter allows the use of cheaper electric generators that can be directly connected to the runner [9]. However, in practice, the range of efficient operation of axial propeller hydroturbines with fixed blade angles is relatively limited [2]. This necessitates a precise adjustment of the hydroturbine operating mode to the conditions of the available water resource. In the context of the above, this paper presents an experimental study of the flow characteristics behind the runner of a micro-hydroturbine model with a rated output of approximately 1 kW under operating conditions ranging from part-load to high overload. Emphasis is placed on the determination of the physical parameters characterizing the flow structure and the degree of deviation of the operating mode from the nominal operating conditions.

2. Experimental technique and determination of flow parameters

In this work, a device with a pair of axial vane swirlers was used as a model of a micro-hydroturbine (figure 1). The first downstream stationary swirler acts as a guide vane. The second swirler, simulating
the runner, is forced to rotate by means of an external servo drive [10, 11]. The runner inlet is connected to a cowl with a height of 39 mm and a base diameter equal to the diameter of the runner hub $d = 40$ mm. Behind the runner, the flow enters a conical outlet that acts as a draft tube. The diameter of the cone at the point of connection to the runner section is $D = 100$ mm, the cone angle is $4^\circ$, and the length is $L = 280$ mm. Geometrically complex elements of the working section, such as vane swirlers, were produced using a 3D prototyping technology. As in previous studies [10–12], atmospheric air was used as a working medium. Air flow rate was controlled by a frequency converter, to which a blower was connected, and was measured by a hot-wire anemometer in the converging nozzle, with additional verification by integrating the axial velocity profile in the control section. During the experiments, the air flow rate $Q$ was varied from 0.03 to 0.09 m$^3$/s, which corresponds to the range of Reynolds number $Re = 4QD/\pi v(D^2 - d^2)$ from $3 \times 10^4$ to $9 \times 10^4$ ($v$ is the kinematic viscosity of air). These conditions correspond to highly turbulent flow, for which the self-similarity condition must be satisfied. Therefore, in the experiments, the angular velocity of the runner was constant $\omega_n = 238$ rad/s. It is assumed that with a change in the angular velocity $\omega$ the flow characteristics remain constant during displacement along isogonal lines (i.e., in the case of a proportional change in the flow rate and the angular velocity of the runner). The value of $\omega_n$ for a flow rate $Q_n = 0.049$ m$^3$/s corresponds to the nominal mode for which the geometry of the swirlers was calculated using the procedure described elsewhere [12]. Thus, the investigated range of operating conditions was varied from part-load (60% of the nominal) to high overload (80% excess over the nominal flow).

![Figure 1](image.png)

Figure 1. Air model of a micro-hydroturbine: 1 – servo drive, 2 – shaft, 3 – stationary swirler (guide vanes), 4 – rotating swirler (runner), 5 – conical draft tube, 6 – cowl, 7 – inlet tube, and 8 – ventilation system to remove spent aerosol.

For the operation of a micro-hydroturbine of this scale on water, the calculated momentum on the runner shaft in the nominal mode is $T_n = 4 \, N \cdot m$ with a head in front of the working section $H_n = 2.6 \times 10^4$ N/m$^2$. The total power of the flow is $P_{tot} = Q_n H_n = 1.3$ kW, and the turbine output $P_{turb} = \omega_n T_n = 0.95$ kW [3, 7]. The specific speed of the micro-hydroturbine is $n_s = 30\omega_n P_{turb}^{1/2}/(H_n/\rho g)^{5/4} = 655 \, (\text{min}^{-1}, \text{kW}, \text{m})$, where $\rho$ is the density of water and $g$ is the acceleration due to gravity [7]. The efficiency of the micro-hydroturbine in the nominal mode is $\eta_n = 100\% P_{turb}/P_{tot} = 73\%$. This value is comparable with the literature data on the efficiency of micro turbines of a comparable scale [7]. If the velocity triangles at the runner inlet and outlet are similar, the hydroturbine efficiency will not change upon displacement from the nominal mode along an isogonal line. When the working point is displaced beyond the isogonal, e.g., along the line of constant angular velocity, the hydroturbine efficiency will decrease. The measured
velocity fields for partial-load and overload modes presented below make it possible to analyze the causes of this effect and to develop recommendations for expanding the range of regulation of hydroturbine operation while maintaining the operating efficiency.

Velocity distributions were measured using a LAD 06-i two-component laser-Doppler anemometer (LDA). The flow seeding required for LDA operation was carried out using an aerosol of paraffin oil with 1–3 µm droplets, which track the flow well [10]. The channel walls were fitted with transparent windows to provide optical access to the flow region. The LDA optical system was mounted on a programmable three-axis stage to carry out measurements in an automated mode, which made it possible to obtain large data arrays. In the studies, profiles of two velocity components with a detailed spatial scanning for more than 25 different modes were obtained. The average number of LDA bursts was $5 \cdot 10^3$ per one velocity component, the maximum measurement time at a point was $\sim 60$ sec, and the average particle density was $\approx 10^5$ bursts per second.

When the operation of the hydroturbine deviates from the nominal conditions, the flow at the runner outlet becomes swirling. In the literature, the degree of flow swirling is characterized by the swirl number $S$ which is defined as the ratio of the axial flux of the angular momentum $M$ to the product of the axial flux of the axial momentum $K$ and the characteristic channel radius $R$ [13]

$$S = \frac{M}{RK}.$$  \hspace{1cm} (1)

The momenta $M$ and $K$ in equation (1) can be determined from the measured velocity distributions ($U$ is the axial component and $W$ is the tangential component)

$$S = \frac{\int_0^R UWR^2dr}{R \int_0^R U^2 rdr}.$$ \hspace{1cm} (2)

Here the integration is performed from the flow axis to the wall of the channel of radius $R$.

The swirl number can also be determined based on the swirler design [14]. For a hydroturbine containing an element rotating at constant frequency, the degree of flow swirling at the hydroturbine outlet can be calculated as a function of the flow rate using the approach proposed in [15]. A diagram of the flow velocity behind the runner is shown in figure 2. Here $V_\alpha$ is the relative flow velocity at the runner outlet in a coordinate system moving with the turbine, $V_s$ is the absolute flow velocity in a stationary coordinate system, $V_b$ is the absolute velocity of motion of the runner tips, $U$ and $W$ are the axial and tangential components of the absolute flow velocity, and $z$ is the longitudinal coordinate axis. Bold type indicates vector quantities. Our studies were performed for a constant angular velocity of the runner $\omega = \omega_0$; i.e., for a given radius $r$, $V_b = \omega_0 r$ = const. In this case, the operating modes of the turbine are varied by changing the flow rate. The slope $\alpha$ of the relative velocity is determined by the geometry of the turbine and remains constant when changing the operating mode (figure 2). 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 (figure 2(a)). Denoting the axial velocity for this mode as $U(Q_0)$, we can write the relation $V_b = \omega_0 r = U(Q_0)\tan(\alpha)$ or

$$\tan(\alpha) = \frac{\rho \omega_0 r}{U(Q_0)}.$$ \hspace{1cm} (3)

When the flow rate decreases below $Q_0$, the absolute flow velocity acquires a nonzero tangential component (figure 2(b)). In this case, we can write $V_b - W = U \tan(\alpha)$ or

$$W = \omega_0 r (1 - U/U(Q_0)).$$ \hspace{1cm} (4)

Relation (4) is also valid when the flow rate increases above $Q_0$ (figure 2(c)). In this case, the tangential velocity changes sign because, for this mode, $U > U(Q_0)$. Assuming as a first approximation that the distribution of the axial velocity over the runner throat is uniform and using relations (2) and (4), we obtain an analytical expression for the swirl number as a function of the flow rate
\[ S = \frac{\pi D \omega_n}{16} (D^2 - d^2) \left( \frac{1}{Q} - \frac{1}{Q_0} \right). \]  

It can be seen that expression (5) gives an inversely proportional dependence of the swirl number on the flow rate and a change in the swirl direction at \( Q > Q_0 \). Since the angle \( \alpha \) remains unchanged, it follows from relation (3) that a change in the angular velocity of the runner leads to a proportional change in \( U(Q_0) \) and hence in the flow rate \( Q_0 \).

3. Results and discussion

Figures 3–6 show examples of the obtained flow characteristics. The data are nondimensionalized using the flow rate \( Q_0 \) and flow-rate-averaged velocity \( U_n = 4Q_0/\pi(D^2-d^2) \) at the runner outlet for the nominal mode. The value \( Q_0 = 0.053 \text{ m}^3/\text{s} \) is determined from the experimental \( Q-S \) dependence. The measurements were performed at a distance of 0.36 \( L \) from the start of the conical outlet section with a step size of 0.01 \( D \) in the radial direction.

It can be seen that the flow swirl increases as the mode is shifted to the underload or overload regions (figure 3(b)). In the underload mode, the direction of the flow swirl coincides with the direction of rotation of the runner. The flow swirled by the stationary swirler passes into the conical tube with high residual swirl, without having fully interacted with the runner. In overload modes, the flow changes the swirling direction, having reflected from the runner blades. The velocity distributions show that the flow pattern in underload modes differs from that in overload modes. At \( Q < Q_0 \), along with an increase in swirl, the flow is displaced to the channel walls, and the axial velocity acquires a pronounced central minimum, up to the occurrence of reverse flow (figure 3(a)). This flow structure is accompanied by the formation of a precessing vortex core, giving rise to a central peak in the distributions of tangential velocity pulsations [10], which can be seen, for example, in the mode with \( Q/Q_0 = 0.64 \) (figure 4(b)). In addition, the vortex core precession effect manifests itself in the presence of a discrete peak in the tangential velocity spectrum measured near the center (figure 5(a)).

For flow rates \( Q > Q_0 \), one can see the formation of an axisymmetric vortex with vorticity concentration near the center, as evidenced by an increase in the absolute maximum of the tangential velocity and its approach to the center (figure 3(b)). In this case, the distribution of the axial velocity remains fairly uniform, without a distinct central minimum. This suggests stabilization of the flow with the central secondary vortex. The central vortex, however, performs a low-amplitude precessional motion, which manifests itself in localized central peaks in the distributions of tangential velocity.
pulsations; this is clearly seen for the mode with $Q/Q_n = 1.67$ (figure 4(b)). The axial velocity component also shows an increased level of pulsations in overload modes (figure 4(a)). As in underload modes, in the case of overload, a discrete peak is observed in the tangential velocity spectrum (figure 5(b)). It is interesting to note that the precession frequency coincides with the runner rotation frequency. That is, the effect of low-amplitude precession of the columnar vortex is modulated by the rotation of the cowl, to which the concentrated vortex is apparently attached.

A comparison of the experimental dependence of the swirl number obtained by integrating the velocity profiles using formula (2) and the dependence calculated using the analytical expression (5) shows that they are in good agreement with each other in the investigated range of operating modes (figure 6). It should also be noted that for the nominal flow rate $Q_n$ for which the blade elements of the hydroturbine were designed, the swirl number is different from zero and is of the order of 0.1. This is due to the practice of designing modern hydroturbines, where a small positive swirl of the flow is initially taken as the nominal (optimal) mode [10]. The practical significance of the result shown in Figure 6 is due to the fact that the adjustment of the optimal mode is carried out relative to the mode with zero swirl (normal output [10]), which can be easily determined using the analytical relation (5) even for a single measurement point.

Experimental data and the analytical dependence show similar trends for the swirl number: rather a sharp increase with decreasing flow rate and a slower increase when passing to the region with negative swirl. This may explain the above-mentioned difference in the evolution of the flow in underload and overload modes. In the first case, the flow swirl number quickly reaches a critical value of 0.5, at which the vortex breaks down, resulting in the formation of a central dip in the axial velocity profile and displacement of the flow to the channel walls [10]. In the second case, the swirl number does not exceed 0.4 in absolute value even for high overload; i.e., it remains below the threshold level. As a result, the maxima of the tangential velocity profile are near the flow axis and the axial velocity profile remains uniform over the cross section of the draft tube. Only for the mode with the highest overload $Q/Q_0 = 1.86$, one can observe the onset of vortex breakdown, which manifests itself in the displacement of the maximum of the tangential velocity away from the center (figure 3(b)) and the occurrence of a central dip in the axial velocity profile (figure 3(a)). This is accompanied by a decrease and broadening of the central peak in the distribution of tangential velocity pulsations (figure 4(b)).

![Figure 3](image.png)

Figure 3. Profiles of the averaged velocity versus flow rate runner at a constant angular velocity of the runner $\omega = \omega_n$: a – axial component, b – tangential component.
Figure 4. Profiles of the pulsating velocity component: a – axial component, b – tangential component. Notation is the same as in figure 3.

Figure 5. Spectra of LDA signals of the tangential velocity component near the center of the vortex ($r/D = 0.03$): a – mode with $Q/Q_n = 0.64$; b – $Q/Q_n = 1.34$. The vertical axis is given in relative units (data are normalized to the maximum value for a given spectrum).

Figure 6. Flow swirl number: comparison of the results of the experiment, formula (2), and the calculation using the analytical dependence (5).
4. Conclusions
Flow characteristics behind the runner of an air model of a propeller micro-hydturbine were studied by varying the operation conditions from part-load to high overload. The measurement data obtained with the automated LDA system include the distributions of time-averaged and pulsation velocity components and pulsation spectra of the LDA signal. The data show that when the operating conditions deviate from the nominal mode, the flow acquires a high residual swirl. In underload modes, the flow swirl quickly reaches a critical value at which the vortex breaks down, resulting in the occurrence of a central dip in the axial velocity profile and flow displacement to the channel walls. In contrast, in overload modes, an axisymmetric vortex is formed with vorticity concentration near the center, as evidenced by an increase in the absolute maximum of the tangential velocity and its approach to the center. The central vortex performs a low-amplitude precessional motion, which manifests itself in a discrete peak in the LDA signal spectrum. The precession frequency coincides with the runner rotation frequency, indicating that the effect of low-amplitude vortex precession is modulated by the rotation of the cowl. In underload regimes, precession also takes place, but its nature is determined by the natural hydrodynamic instability of the flow in the vortex breakdown zone. The flow features identified for various operating modes can be used to adequately predict the parameters of micro-hydturbines and develop recommendations for expanding the range of regulation of hydturbine operation while maintaining high efficiency.

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