Comparative analysis of air and water flows in simplified hydraulic turbine models

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Abstract. This article presents a comparative analysis of flow characteristics behind a hydraulic turbine runner in air and water. Swirling flow with a precessing vortex core (PVC) was investigated using a laser Doppler anemometer and pressure pulsation sensors. The experiments were conducted on aerodynamic and hydrodynamic test rigs over a wide range of hydraulic turbine operating conditions. Part-load modes of hydraulic turbine operation were investigated using the Fourier transform of pressure pulsations obtained from acoustic sensors. The features of the swirling flow were shown for the range of operating conditions from deep part-load to overload.

1. Introduction

Wind, solar, and thermal power plants are very slow in terms of power regulation. Hydropower plants provide stable operation of electricity grid. The regulation of hydro turbine operation based on the amount of energy required leads to operation in non-optimal modes in which the flow coming off the edges of the water turbine runner is highly swirled and highly susceptible to disturbances. The expanding part of the hydraulic turbine draft tube leads to the formation of a powerful precessing vortex core (PVC) of spiral shape in the flow. The vortex rope causes significant periodic pressure pulsations in the flow path, which are extremely dangerous because of the possibility of resonance with the natural frequencies of various oscillations occurring at hydroelectric power plants [1–3]. The vortex rope significantly narrows the allowable operating range and generates strong oscillations of the entire water column in the penstock, directly affecting the runner [4, 5].

A detailed experimental study of swirling flow behind the turbine runner on full-scale experimental rigs is complicated and expensive. Alternative approaches include the use of scaled-down laboratory models of hydro turbines [6, 7] and simplification of experimental modeling by using air instead of water as a working medium [8–11]. Although these approaches have been developed over many years, modern measurement and design techniques, such as the use of additive 3D printing technologies to create various elements of a turbine model, makes it possible to obtain many more flow regimes and test the geometry of various individual elements (runner, draft tube) in a fairly short time to find the most optimal one.

In our previous work, velocity distributions similar to real turbine velocity distributions were obtained using a system of two vane swirlers in a turbine aerodynamic model [9, 12] and a turbine hydrodynamic model [11, 13]. The present study is an attempt to compare these two cases in terms of
available flow characteristics. This is not entirely straightforward since operating parameters such as flow rate and runner rotation speed vary considerably. To solve this problem, we use the integral characteristics of the flow and the dimensionless speed of the runner relative to the optimal operating points of the turbine models.

2. Test rigs

2.1. Turbine aerodynamic model
A detailed description of the aerodynamic setup is presented in [9]. A significant simplification of the experimental modeling is achieved by replacing an aqueous medium with an air medium [14]. The required flow distribution at the diffuser inlet was created using a pair of swirlers: a stationary swirler acting as guide vanes and a rotating swirler—an analogue of a turbine runner. The flow regimes corresponding to the non-optimal (off-design) operating points of the turbine models were obtained by adjusting the rotation speed of the movable swirler in the draft tube cone. These flow regimes involve the formation of a pulsating vortex rope, which is of greatest interest. A pair of swirlers was designed for the optimal operating conditions (the best efficiency point, BEP) corresponding to a volumetric flow rate \( Q_c = 0.049 \text{ m}^3/\text{s} \) and a runner rotation speed \( n_c = 2430 \text{ rpm} \). The Reynolds number \( (Re=QD/\text{A} \cdot \nu) \) calculated for \( Q_c \) is about \( 4 \cdot 10^4 \). The test section consists of a reduced geometric model of a Francis-99 draft tube with inlet diameter \( D=100 \text{ mm} \). The flow velocity at local points of the test section was measured with a LAD-06i two-component laser-Doppler anemometer (LDA) with the addition of tracers (small particles of vegetable oil with a diameter of 1 µm) to the air flow. The tracers were formed in a Laskin atomizer. The relative error in measuring the LDA velocity is 0.2%, as follows from the specifications of the instrument. The experimental setup included an automated control system for air flow rate and rotor speed with an error of 1.5% and 0.5%, respectively.

2.2. Turbine hydrodynamic model
The turbine hydrodynamic model is also based on the idea of using a swirl generator approach [15] instead of a Francis turbine runner to obtain the same velocity profiles at the draft tube inlet of a Francis turbine operated at partial discharge. This experimental test rig is described in detail in [11, 13]. A 3D view of the test section is shown in Figure 1. The diameter of the draft tube throat is also 100 mm as in the aerodynamic case. The maximum flow rate is up to 0.042 m\(^3\)/s (Re = 5 \cdot 10^5), and the

![Figure 1. 3D view of the aerodynamic (left) and hydrodynamic (right) turbine models.](image-url)
3. Results
In comparing the flow in the investigated turbine models, we will use the main dimensionless quantities characterizing the unsteady turbulent flow: the Reynolds number ($Re$), the swirl parameter ($S$), the dimensionless flux of moment of momentum ($M^*$), and the Strouhal number. Despite the fact that the $Re$ number for the air model ($4 \cdot 10^4$) is an order of magnitude lower than that for the water model ($4 \cdot 10^5$), all measurements are in a self-similar region where the dimensionless quantities cease to depend on $Re$.

The measured tangential and axial velocity components at the inlet of the draft tube cone were used to calculate the integral swirl parameter $S$ for each flow regime:

$$S = \frac{\int_0^\infty \rho u w r^2 dr}{\int_0^R \rho u^2 r dr}$$  \hspace{1cm} (1)

This is still one of the main dimensionless quantities characterizing the swirling flow intensity that allows one to roughly compare the operating conditions of swirling devices.

As the runner speed increases, the flow shifts to the walls. At a certain point, the swirling flow is already pressed against the wall; as a consequence, the corresponding swirl parameter stops increasing after reaching a certain saturation level (Figure 2). Thus, the change in the swirl parameter ceases to obey the linear law. The same scenario is observed for the turbine hydrodynamic model. When the dimensionless ratio $N/N_0 = 2.5$ is reached, the integral swirl parameter ceases to increase with increasing dimensionless runner speed and is about of 0.95. For the air model, this value is slightly lower, 0.8, probably due to the difference between the velocity profiles, which, in turn, depend on the shape of the runner blades.

Another parameter by which the flow characteristics can be compared is the Strouhal number ($St = f D A / Q$). Since the dependence of the swirl parameter on the runner speed is nonlinear, the Strouhal number is plotted in Figure 2 as a function of $N/N$, and not as a function of the swirl parameter. Figure 2 shows that for both turbine models, the relationship is well described by a linear regression. Moreover, despite the fact that the slope differs starting from $N/N_0 = 3$, the Strouhal numbers take the same values. It is interesting that although for the air model, the vortex precession frequency was measured starting from $N/N_0 = 2$ and for the water model, it was measured starting from $N/N_0 = 1.6$, if we turn again to the swirl parameter, then, due to the difference in the slope of the curves, we obtain the same value $S \approx 0.5$. 

![Graph showing comparison of swirl parameter and Strouhal number for both models.](image-url)
From the point of view of generalization, one can also consider the approach proposed in [16], which allows comparing the dependences of the dimensionless flux of moment of momentum $M^*$ on the dimensionless flow rate $Q^*$:

$$M^* = \frac{2\pi \omega \int_0^R u w r^2 dr}{(\omega R)^2 \pi R^2}$$  \hspace{1cm} (2)$$

$$Q^* = \frac{Q}{\omega \pi R^3}$$  \hspace{1cm} (3)$$

where $\omega$ is the angular speed of the runner (rad/s), $u$ and $w$ are the axial and tangential velocities, and $R$ is the radius of the runner.

Figure 3. Dimensionless flux of moment of momentum calculated for the aerodynamic (left) and hydrodynamic (right) models.

Figure 3 shows $M^*$ as a function of $Q^*$ for the both cases. One can see that the second-order regression curve fits the experimental data well. The versatility of this approach is that it does not matter under what flow conditions the experiments are carried out. According to [16, 17], the point of intersection of the curve with zero value ($M^*=0$) corresponds to the optimal operating conditions of the turbine model. In this study, corresponding flow rates $Q^*$ are 0.7 in the case of air and 0.52 in the case of water. A decrease in the dimensionless flow rate relative to the optimal one corresponds to part-load operation. Furthermore, a change in the sign of the dimensionless flux of moment of momentum indicates that the flow is swirled in the opposite direction; i.e. these are forced load conditions. Qualitatively, the curves of the dimensionless flux of moment of momentum look similar, the maximum value of $M^*$ in both cases is approximately equal to 0.075.

4. Conclusion
A comparison was made for hydraulic turbine models operating in air and water. The comparison of the integral characteristics of the flow, such as the impulse flux, and the swirl parameter, as well as the frequency characteristic (the Strouhal number) leads to the conclusion that the replacement of water with air is acceptable for investigating vortex phenomena in swirling flows. The advantages of the air environment include the simplicity and low cost of manufacturing the elements of a model turbine, which makes it possible to sort out a large number of their modifications. It also can be concluded that the effects of cavitation or concentration of dissolved air in the low-pressure region of the PVC observed in the turbine hydrodynamic model play a less significant role in terms of the frequency of pressure pulsations associated with the most powerful vortex mode.
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