Experimental modeling of swirl flows in power plants

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Abstract. The article presents an overview of the methods and approaches to experimental modeling of various thermal and hydropower devices - furnaces of pulverized coal boilers and flow-through elements of hydro turbines. The presented modeling approaches based on a combination of experimentation and rapid prototyping of working parts may be useful in optimizing energy equipment to improve safety and efficiency of industrial energy systems.

1. Introduction
Since experiments in full-scale power plants are extremely expensive and time-consuming, experimental modeling using reduced laboratory models is widely used to develop energy unit designs [1,2]. Strict fulfillment of the similarity condition by all criteria is practically impossible. Therefore, the most commonly used is the approximate modeling only on certain parameters [3]. For example, in the case of combustion chambers, isothermal modeling on geometrically similar reduced models based only on one similarity criterion, the Reynolds number, enables the relatively low-cost study of the combustor aerodynamics, which, in turn, cardinally determines the combustion process [4]. Although the results of isothermal modeling are somewhat limited when it comes to full-scale firing conditions, they still allow determining the direction of the design optimization and, thus, reducing the equipment development costs, by diminishing the number of design options, studied on large-scale fire stands and real size facilities [5].

Owing to rapid progress in the computer technique, the numerical modeling is now increasingly becoming the main tool for the power plant development and design [5]. The role of physical modeling is still very important, while the main objective of the experiment is to provide empirical information for the verification of numerical codes [6]. With respect to experimental modeling, the reliability and detailed character of the data obtained on the model become crucial. To this end, it is important to precisely specify the boundary conditions, which implies using a simplified, generalized geometry of the working area. To obtain reliable data, it is also important to use non-contact, non-perturbing measurement techniques [7]. Another relevant objective of the experiment, in addition to verifying the computer simulation, is to reveal the physical mechanisms of the processes and to obtain fundamentally new physical effects that may be further used in practice. To accomplish this task the laboratory model should allow for variation within a wide range of geometry and operating modes.

In this context, the article presents some overview of methods and approaches of experimental modeling of various thermal and hydropower devices - combustion chambers of pulverized coal boilers and flow elements of hydro turbines. The focus is on the study of large-scale concentrated vortices, determining the global structure of the flow and critically influencing the technological processes in industrial equipment.
2. Experimental techniques

2.1. Flow structure diagnostics

A specific feature of the used modeling technique is the replacement of working media: gas by liquid, as it was done in the modeling of furnaces for pulverized coal boilers [4-6], and water by air in flow modeling of hydro turbine equipment [7,8]. In the first case, the use of liquid as a working medium simplified the visual studies, which provide important qualitative information on the flow structure. Small air bubbles were used as liquid flow markers (figure 1), which were illuminated by a light source (a powerful LED spotlight for analyzing the general flow pattern or a narrow light "sheet" formed by sweeping on a cylindrical lens of the laser beam from a solid-state laser to visualize the flow structure in a specific flow cross-section).

To obtain quantitative information about the flow field a two-component laser-Doppler anemometer LAD-06i was used [9,10]. To measure pressure pulsations in the experiments with air the acoustic sensors based on precision measuring microphones were used [7]. In hydrodynamic models, sensitive piezoelectric pressure sensors were employed [6]. To measure the instantaneous velocity fields, the PIV system was applied [6,11].

Figure 1. Scheme of hydraulic model of vortex combustor (a) and flow visualization for test section with central (b) and shifted outlet orifice (d). Trajectories of fluid particles at generation of stationary vortex structures: c – rectilinear vortex, e – spiral vortex.
2.2. Test rigs

2.2.1. Hydraulic model of the vortex combustor. The model test section (figure 1a) is a square-section channel with dimensions of 188×188×625mm³. This geometry corresponds to the design of the tangential furnace E-500, used to burn a pulverized-coal fuel [6]. The square shape of the cross section in the case of large tangential furnaces is conditioned by the fact that the flat side walls are easier to manufacture than the cylindrical ones. It has been shown experimentally that the use of a square shape does not result in vortex destruction or distortion [11]. It is also important that the flat side walls of the model do not give optical aberrations facilitating the use of non-contact optical methods for investigating the flow. Therefore, this unit is widely used not only to simulate the flows in tangential furnaces but also to study the general physical effects, governing the dynamics of vortex flows. To control the structure of the vortex flow the boundary conditions at the ends of the chamber were changed. In particular, a diaphragm with an axisymmetric or offset hole was installed at the chamber outlet [6].

![Image](image1.png)

![Image](image2.png)

**Figure 2.** View of the hydraulic stand with model TURBINE-99 (a) and visualization of precessing vortex rope (b). Pressure pulsations recorded with piezoelectric sensor: (c) - time series; (d) FFT spectrum (vertical axis is in arbitrary units).

2.2.2. Hydraulic models of hydroturbine draft tubes. Figure 2a shows the hydraulic stand with the model of a draft tube with the geometry TURBINE-99, widely used for testing computer codes [12]. The internal geometry of the model is formed by relatively thin Plexiglas plates. The whole construction is inserted into a liquid-filled rectangular casing, made of thicker Plexiglas, taking up fluid pressure inside the working area [13]. Such a design with plane external walls, similar to the model of vortex furnace, provides minimal optical distortion. To simulate the regimes with a large
residual swirl at the input to the working section, a stationary blade swirler with the design swirl parameter \( S=1.1 \) is installed [14]. Such regimes characterized by the development of strong flow nonstationarity behind the impeller (in the conical part of the draft tube) are realized in the operation under non-optimal conditions of underload or overload of the hydroturbine [7].

Figure 3a presents the scheme of the working section, simulating only the inlet conical part of the draft tube. This enables the detailed study of the processes of vortex formation behind the impeller, not complicated by the influence of the draft tube elbow. To ensure optical access the working area is made of Plexiglas and has plane external walls. A stationary blade swirler is used for swirl generation similar to the previous case. For better matching the real conditions of velocity and pressure distributions at the draft tube inlet, a rotating element is added to the swirler design. At the first stage of research, a freely rotating runner was used. This, in principle, can provide distributions of axial and tangential velocity components in the model to be fairly close to the flow distribution under the working wheels of full-scale hydroturbines [15]. However, similar to the previous case, this combination of swirlers ensures simulation of conditions only for a single operating regime of the hydroturbine. In the presented study, we use a stationary swirler with inclination angle of the blades, corresponding to the design swirl parameter \( S=1.1 \).

Figure 3. Sketch of axisymmetric model of the draft tube (a) and visualization of single spiral (b) and double spiral (c) vortex ropes.

2.2.3. Air models of hydroturbine draft tubes. The replacement of liquid media by air has been widely used for the simulation of hydraulic machines from the previous century up to present [7,8,17]. The advantages of an experiment with air are due to much less difficulties in manufacturing experimental sections (avoiding problems of joint sealing and test section strength; providing easier optical access). In our case, the use of air enables 3-d printing based on available plastic materials for rapid and accurate reproduction of complex geometry of the channel elements.

The scheme of the working section is shown in figure 4a. In this unit, similar to the hydraulic model, the velocity distribution close to the distribution behind a real hydroturbine is reached by means of a system of two blade swirlers [16,17]. One of the swirlers is fixed (stator) and functions as a spiral chamber and a guiding vane of the hydroturbine (figure 4b). The air flow with a volume flow rate \( Q \) (m\(^3\)/h) is uniformly flowing onto it. The second swirler (runner) being an analog of the hydroturbine impeller rotates. Unlike the above design of the hydraulic model, the runner in the air model is rotated at a frequency \( n \) (rpm) with the aid of an external servo drive. Thus, the unit with the air model allows simulating various operating regimes of the hydroturbine based on specifying the parameters \( Q \), \( n \) and the geometry of the swirler arrays.
3. Sample results

3.1. Stationary spiral vortices in the combustor model.

The initial geometry of the tangential chamber with no diaphragm is characterized by the development of large-scale flow instability. It is expressed in the vortex core shift relative to the channel center and its high-amplitude precession motion around the channel center [6]. Diaphragming the chamber allows stabilizing the flow by fixing the vortex core near the channel axis [11] and obtaining a generally axisymmetric pattern of the flow (figure 1b). The flow below the diaphragm is, in general, insensitive to the conditions behind the diaphragm, in particular to the asymmetrical side exit from the working channel. However it drastically changes with the introduction of asymmetry at the ends of the vortex [6]. For example, when the outlet hole of the diaphragm is displaced the axis of the vortex takes the form of a stationary (immobile) single helix (figure 1d).

The results obtained on the tangential furnace model for the conditions of forming the vortex structures with spiral axes may be interesting from a practical point of view to demonstrate possibilities of control over the combustion chamber aerodynamics. In the case of the formation of spiral vortex structures the fluid particles move along spiral trajectories around the spiraled rotation axis. Therefore, it is a case of elongation of the trajectory (in comparison with the rectilinear vortex) and accordingly of an increasing residence time. This is important for a more complete burn-out of fuel and improved combustion efficiency. Modes with a single and especially a double spiral vortex are characterized by a greater involvement of the internal chamber volume in the intense vortex motion that may assumingly intensify the mixing processes and improve the heat exchange in the combustion chamber. These assumptions were verified by calculating the length of trajectories and the residence time using numerical LES modeling [18]. The comparison of the calculation results shows (figure 1c, e) that the trajectory lengths for regimes with spiral vortices are noticeably larger than for a

Figure 4. Scheme of aerodynamic test section for modelling the flow in draft tubes at different regimes of hydroturbine operation (a) and example of designed pair of stationary (guide vanes) and rotary (runner) elements (b). Pulsation characteristics of the flow in air model at a constant rotation frequency of the runner and varied flowrate: (c) – levels of pulsation RMS; (d) – amplitude spectra (arbitrary units) of difference signals for three flowrates (0.4Qn, 0.5Qn, Qn). n – rotation frequency of the runner.
rectilinear vortex. For example, in the case of a single-helical vortex, the average residence time increases almost 2 times.

3.2. Precessing spiral vortices in hydraulic models of draft tubes.
Under conditions of strong swirling an instability develops in the form of a precessing vortex rope (figure 2b), which generates powerful pressure pulsations (figure 2c). This strong unsteadiness is expressed in the pulsation spectrum by the presence of a dominant peak (figure 2d). The frequency of precession is linearly dependent on the volumetric flow rate and the amplitude of the pulsations at the precession frequency varies quadratically with the flow rate. In a dimensionless form, the dependence of the precession frequency demonstrates a self-similarity with respect to the Reynolds number [15].

Studies show that under conditions of strong swirling the dominant mode is a single-helical vortex rope [14]. However, in some regimes there is an abrupt transition from single-spiral (figure 3b) to a double-helical mode (figure 3c). This kind of instability has a special practical and scientific interest in view of the random changes in the characteristics of the vortex rope: vorticity field, period and frequency of precession [15]. This behavior of the precessing rope in draft tubes is poorly predicted and can cause additional unexpected vibrations of hydraulic equipment.

3.3. Unsteady regimes in air model of a draft tube.
The experiments with the air model included the determination of regimes with an increased level of flow pulsations, caused by the precessing vortex rope. Such regimes arise, in particular, at moving away from the nominal flow rate $Q_0$ into the region of lower flowrates, corresponding to partial loads of the hydroturbine. To determine the modes with maximum pressure pulsations, the flow rate was varied from $0.4Q_0$ to $Q_n$ with a step of $0.05Q_0$ at a constant $n$. Based on the pressure pulsation signals recorded with the precision B&K microphone on the cone wall, the level of RMS fluctuations of the flow was determined as a function of the ratio $Q/Q_n$ (figure 4c). The data shows the formation of a local extremum (labeled as “max” on the graph). This extremum arises due to generation of a precessing vortex rope in the draft tube cone. To prove this two acoustic sensors (Behringer microphones) were mounted flush with the wall opposite each other in the mean horizontal section of the cone [7,16,17]. Figure 4d shows the spectra of the difference signals of the pressure pulsations, obtained from two sensors for the modes $0.4Q_n$, $0.5Q_n$ and $Q_n$. On the difference spectra, there is the dominating frequency $f^*(Q)$, which differs from the frequency $n$. Starting at a flow rate $Q=0.4Q_n$ a peak with a frequency $f^*$ is formed, then at a flow rate of $0.5Q_n$ the peak amplitude $f^*$ reaches its maximum. When approaching the nominal mode ($Q_n$) the rope frequency $f^*$ completely disappears from the spectrum.

Thus, the studies have resulted in determining the regimes with maximum pressure pulsations, which appeared in the region of low (relative to the optimal mode) flowrates. Based on the analysis of pressure pulsation signals, the development of the precessing vortex rope in the cone of the draft tube is demonstrated. The modeling approaches presented in this paper and based on a combination of experiment and rapid prototyping of working parts may be useful for optimizing the geometry of the flowing parts of a hydroturbine with the view of increasing the dynamic reliability and efficiency of a hydropower plant operation.

Conclusions
The presented sample results demonstrate the ability of the developed approaches in experimental modelling of power equipment to reveal new physical phenomena, which may drastically influence the technological processes. Among them, the formation of large-scale stationary spiral vortices noticeably increasing the residence time has been studied. This is important for a more complete burn-out of fuel and improved combustion efficiency in thermal power plants. Regarding the hydropower engineering, the new features of emergence of spiral precessing vortices at off-design operation regimes of hydroturbines, such as sudden switching between single and double helical modes, have been described. Most dangerous regimes associated with maximum pressure pulsations have been
determined using a novel laboratory model, allowing for simulation of different conditions of a hydroturbine work.

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