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HYDRODYNAMIC LOADS ON THE WALLS OF A TURBINE BLOCK WITH A COUNTER VORTEX DAMPER

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Abstract: The article discusses the use of a counter vortex water flow energy dissipater when using a medium-pressure hydroelectric power station's conduit as a spillway. A design of a counter vortex damper in the chamber of a hydraulic turbine impeller with damping of excess flow energy in the cone of the draft tube is proposed. Research was carried out on pressure pulsations on the walls of a hydraulic model of a turbine block, as a result of which spectra and standards of pressure pulsations on the walls of a draft tube cone were obtained during operation of the model with and without a damper. It has been shown that passing water through a turbine unit of a medium-pressure HPP with the working impeller removed is unacceptable and leads to the destruction of the structure. It was found that the installation of a counter vortex damper in the impeller chamber reduces the dynamic loads on the walls of the draft tube cone by 6–7 times compared to the mode without a damper. It is concluded that the flow of water through a turbine unit of a medium-pressure HPP with a counter vortex damper installed in the impeller chamber satisfies the safety standards of hydraulic structures. The need to supply atmospheric air to the counter vortex damper via a special air duct has been established.

Keyword: counter vortex spillway, flow energy dissipation, pressure pulsation spectra and standards, safety of hydraulic engineering structures

ГИДРОДИНАМИЧЕСКИЕ НАГРУЗКИ НА СТЕНКИ ТУРБИННОГО БЛОКА С КОНТРВИХРЕВЫМ ГАСИТЕЛЕМ

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Аннотация: В статье рассматривается применение контрвихревого гасителя энергии водного потока при использовании энергетического водовода гидроэлектростанции среднего напора в качестве водосброса. Предложена компоновка контрвихревого гасителя в камере рабочего колеса гидротурбины с гашением избыточной энергии потока в конусе отсасывающей трубы. Выполнены исследования пульсаций давления на стенках гидравлической модели турбинного блока, в результате которых получены спектры и стандарты пульсаций давления на стенках конуса отсасывающей трубы при работе модели без гасителя и с гасителем. Показано, что пропуск воды через турбинный блок ГЭС среднего напора с удаленным рабочим колосом является неприемлемым и ведет к разрушению сооружения. Получено, что установка контрвихревого гасителя в камере рабочего колеса снижает динамические нагрузки на стенки конуса отсасывающей трубы в 6 – 7 раз по сравнению с режимом без гасителя. Делается вывод, что пропуск расходов через турбинный блок ГЭС среднего напора с установленным в камере рабочего колеса контрвихревым гасителем удовлетворяет нормам безопасности гидротехнических сооружений. Установлена необходимость подвода атмосферного воздуха к контрвихревому гасителю по специальному воздуховоду.

Ключевые слова: контрвихревой водосброс, гашение энергии потока, спектры и стандарты пульсаций давления, безопасность гидротехнических сооружений

1. INTRODUCTION

The study focuses on improving the reliability and service life of hydroelectric power plants and multi-purpose hydroelectric complexes, many of which were built in the middle of the last century and require reconstruction of their hydraulic structures in line with modern requirements. It seems to be economically reasonable to use decommissioned energy waterways

as spillways when renovating these hydroelectric power plants.

The issue of bypassing idle flows through the energy flow path has been repeatedly raised in specialized literature regarding low-pressure hydroelectric power plants [1–8]. The most comprehensive review of this issue is provided in [9]. According to these works, the conclusion about the possibility of passing idle flows through the flow path of a turbine of an inactive hydroelectric unit and the resulting loads on the structure is based on an analysis of flow characteristics at relatively low speeds. It has been established that passing idle flows under these conditions is only possible with a remote impeller and a reliable supply of sufficient air under the hydro turbine cover. It should be noted that due to the high dynamic loads on the structure, turbine manufacturers do not recommend this procedure and are not responsible for the consequences. There is no experience in the world of bypassing idle flows through HPP turbine units at heads of more than 40 m.

Discharging idle water through the energy water pipes of non-operating hydroelectric power plants of medium-pressure HPPs without special measures is unacceptable due to the risk of destruction, both of the water pipes themselves, operating in uncalculated modes, and of the hydroelectric complex as a whole due to erosion in the lower reaches. Such a measure is the dissipation of flow energy within the pressure flow path in a special dissipation chamber, which allows the flow to be released into the suction pipe and the lower reach at low speeds close to the design turbine modes.

The idea of installing an energy dissipation chamber on a pressure water pipeline is not new. As an example, we can mention the operational deep spillway with a pressure chamber in the form of a sudden expansion of the flow at the Mica hydroelectric complex in Canada [10]. This spillway with a chamber diameter of 13.7 m has a flow rate of 850 m^3/s at a head of 137 m. At the same time, pressure pulsations on the chamber walls due to high flow turbulence in the zone of intense energy dissipation reach 30% of the difference between the velocity heads of the flow at the chamber inlet and inside it. Such a pressure energy dissipator for the discharge flow is effective, but cannot be considered reliable due to the extremely high dynamic loads on the energy dissipation unit. It should be noted that such high hydrodynamic loads on the structure prevent the use of pressure dampers in spillways of medium- and high-pressure hydraulic structures. Thus, the problem of reducing the level of hydrodynamic loads on the walls of the structure becomes crucial.

In this regard, let us consider a pressure counter-vortex energy dissipator based on the interaction of coaxial layers of flows swirling in opposite directions [11–13]. In a counter-vortex energy dissipator, the zone of intense dissipation of the mechanical energy of the flow is located between the interacting concentric counter-rotating layers at a considerable distance from the walls of the dissipation chamber. This prevents the appearance of high hydrodynamic loads on the chamber walls, especially when air from the atmosphere is supplied to its central near-axis zone.

Objective of the study is to determine the hydrodynamic loads on the walls of the turbine block of a medium-pressure hydroelectric power plant with a counter-vortex energy dissipator for idle water flow.

2. DAMPER DESIGN

The design was developed for a hydroelectric power plant undergoing reconstruction with an operating head of H_0 = 50 m. One of the hydroelectric power plant's hydroelectric units, which has been decommissioned, is equipped with a spillway after the turbine has been dismantled, ensuring that water is discharged into the lower reach when the level in the upper reach rises when the hydroelectric power plant is shut down. The working range of levels in the upper reach of the spillway is from the operating level of 90.5 m to the forced level of 93.5 m, with levels in the lower reach of the power plant's

discharge channel ranging from a minimum of 39.5 m to a maximum of 42.0 m. The upper tail-race level of 93.0 m is taken as the initial level for shutting down the HPP units and activating the spillway. The design discharge capacity of the spillway is Q = 63.3 m^3/s. The design brief specifies that the damper must be simple to manufacture, require minimal changes to the flow path design, and be removable so that it can be dismantled for the installation of the hydroelectric unit's working wheel.

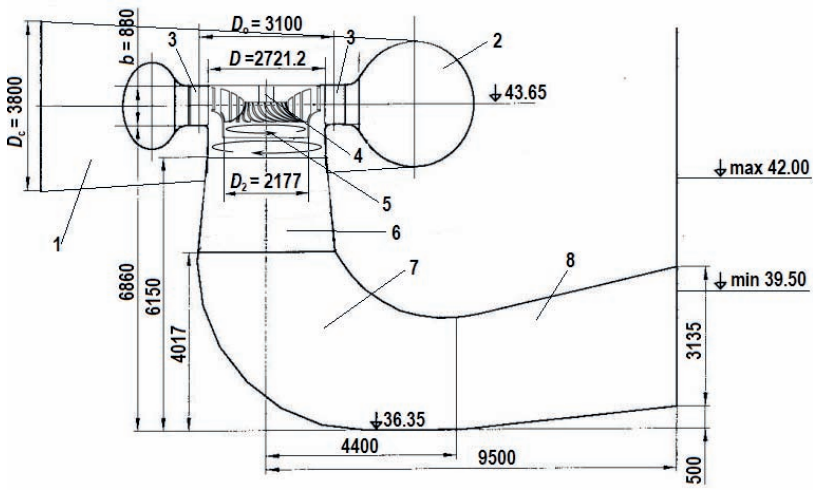


Figure 1. Flowing part of the turbine unit: 1 – water supply pipe, 2 – spiral chamber, 3 – guide vanes of the hydro turbine, 4 – air duct, 5 – working wheel chamber with counter-vortex damper, 6 – suction pipe cone, 7 – suction pipe elbow, 8 – diffuser

According to the task, the layout of the pressure counter-vortex damper is adopted in the chamber of the remote working wheel of the hydro turbine with energy damping in the cone of the suction pipe (Figure 1). The main dimensions of the flow part of the turbine chamber are as follows: Dc = 3800 mm is the diameter of the pressure water pipe at the inlet to the spiral chamber; D0 = 3100 mm is the diameter of the installation of the axes of rotation of the guide vane blades; b = 880 mm is the height of the guide vane blades of the hydro turbine; D = 2721.2 mm is the diameter of the inlet section of the suction pipe cone.

The anti-vortex damper includes the guide apparatus of the remote hydro turbine and a fairing with blades rigidly fixed to it. The guide apparatus of the hydro turbine with rotating blades performs the functions of a regulating valve and swirls part of the idle flow. The blades rigidly attached to the fairing swirl the other part of the idle flow in the opposite direction to the swirl of the guide apparatus blades. The fairing directs one part of the flow coaxially into the other, oppositely swirled part and leads them into a common damping chamber in the cone of the suction pipe.

In the cone of the suction pipe, coaxial concentric oppositely twisted parts of the flow mutually cancel each other's twist, thereby achieving the effect of damping the mechanical energy of the flow as a whole.

Figure 2 shows the design of the counter-vortex damper, carried out in accordance with proven methods for calculating pressure spillway hydraulic structures [9], regulatory documents [15-20], and recommendations contained in specialized literature [11-13].

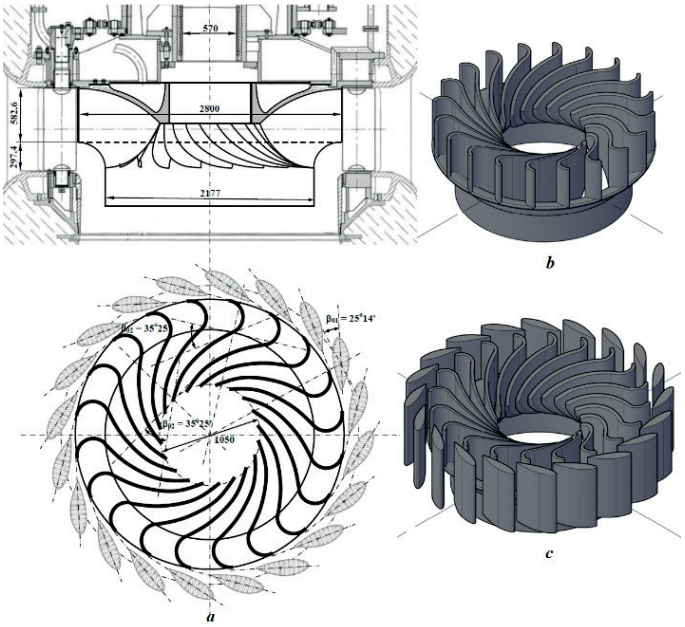


Figure 2. Counter-vortex damper: a – preliminary design of the damper layout in the impeller chamber of a remote hydro turbine, b – fairing with blades for swirling the internal flow in isometric view, c – assembled unit with the hydro turbine guide apparatus

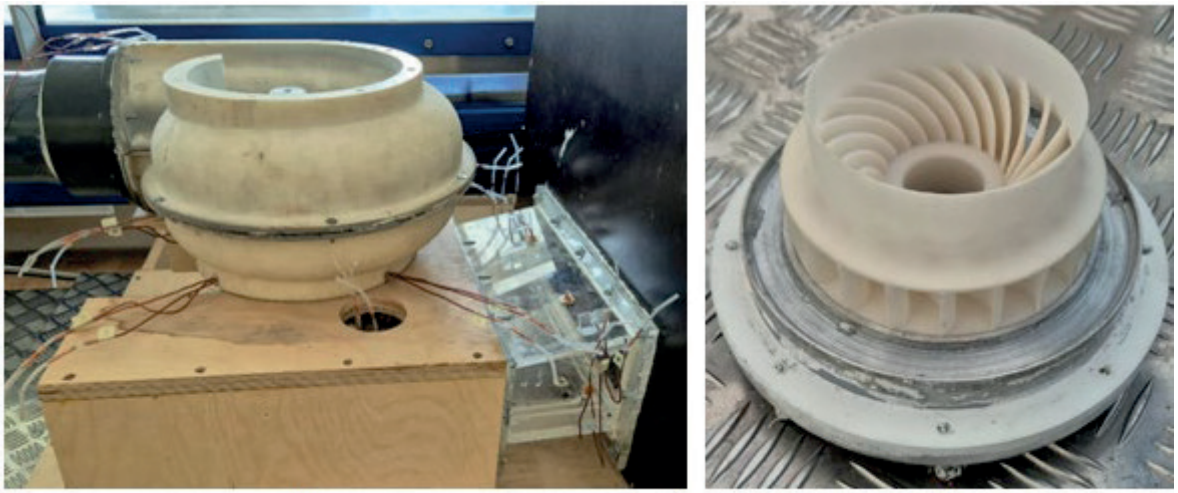
A distinctive feature of the project was that the internal flow swirler blades, fixed on the fairing, have two sections. An inlet section reverses the flow in the opposite direction to the swirl of the peripheral flow formed by the blades
of the guide apparatus of the hydro turbine. And a main section, where the internal flow is given the required swirl parameters. In the main section, the blades have the shape of a logarithmic spiral [14].

HYDRODYNAMIC INVESTIGATIONS

Hydrodynamic loads on the walls of the damping chamber located within the cone of the suction pipe presented in Figure 1 are of crucial importance when deciding on the use of a counter-vortex damper for discharging water into the lower reach through the power water conduit of the reconstructed hydroelectric power plant. To evaluate these loads, physical studies of a model of a turbine unit with a counter-vortex damper on a scale of 1:27.2 shown in Figure 3 were carried out in the hydraulic laboratory of the Moscow State University of Civil Engineering.

Using a laboratory model, pressure pulsations on the walls of the suction pipe cone were measured with silicon piezoresistive sensors XGZP6857A manufactured by CFSensor, China. The range of measurable pressures was from –40 to +40 kPa, the measurement error was ±8.0 Pa, and the range of measurable pressure pulsation frequencies was up to 1000 Hz.

Measurements were taken at two points at distances of 26 and 52 mm from the beginning of the suction pipe cone. Six pressure sampling points were evenly distributed around the perimeter of the cone at each location. At each of the 12 points, the pressure change $P(t) = Pi$ was recorded during an exposure time of approximately $T = 180$ seconds with a discretization step of $\Delta t = 0.001$ seconds.



a *b*

Figure 3. Turbine block model: a – general view, b – turbine cover with counter-vortex damper

The experimental data sets were further subjected to statistical processing and spectral analysis. During statistical processing, the following were calculated:

– average pressure value at the point of sampling during the exposure period (mathematical mean value)

$$\overline{P} = \frac{1}{T}\sum_{i=1}^{N} P_i \Delta t \,,$$

where $N = T/\Delta t$ is the number of measurements;

– pressure pulsation standard (square root of dispersion)

$$\sigma_p = \sqrt{\sum_{i=1}^{N} \frac{\left(P_i - \overline{P}\right)^2}{N}} \,,$$

which was reduced to the pressure H_0 acting on the model

$$\sigma = \frac{\sigma_p}{\rho g H_0} \,. \qquad (1)$$

Spectral analysis was performed based on discrete Fourier transform. Assuming that the pressure change function over time P(t) is non-periodic (finite), i.e., a function completely defined on the interval [0, T]. The spectral analysis consisted of finding the coefficients a_j, b_j of the Fourier series.

$$P(t) = \overline{P} + \sum_{j=1}^{\infty}\left[a_j \cos\left(2\pi\omega_j t\right) + b_j \sin\left(2\pi\omega_j t\right)\right] \,, \qquad (2)$$

where ω_j is the frequency of the j^{th} mode.

In addition to series (2), the following series was used in the spectral analysis

$$P(t) = \overline{P} + \sum_{j=1}^{\infty} C_j \cos\left(2\pi\omega_j t + \varphi_j\right) \,,$$

where C_j is the amplitude of the j^{th} mode

$$C_j = \sqrt{a_j^2 + b_j^2} \,, \qquad (3)$$

φ_j is the phase of the j^{th} mode

$$\varphi_j = -\mathrm{arctg}\left(b_j / a_j\right) .$$

It is known that Fourier coefficients are determined by integrals

$$a_j = \frac{2}{T}\int_0^T P(t)\cos(2\pi\omega_j t)dt \quad (4)$$

,

$$b_j = \frac{2}{T}\int_0^T P(t)\sin(2\pi\omega_j t)dt \quad (5)$$

.

For finite functions, the components of the spectral density are also calculated.

$$S_{Cj} = \int_0^T P(t)\cos(2\pi\omega_j t)dt = \frac{a_j T}{2} \quad (6)$$

,

$$S_{Sj} = \int_0^T P(t)\sin(2\pi\omega_j t)dt = \frac{b_j T}{2} \quad (7)$$

,

spectral density module of the j^{th} mode

$$S(\omega_j) = \sqrt{S_{Cj}^2 + S_{Sj}^2} = \sqrt{\left(\frac{a_j T}{2}\right)^2 + \left(\frac{b_j T}{2}\right)^2} = \frac{C_j T}{2} \quad (8)$$

,

and phase of spectral density of the j^{th} mode

$$\varphi(\omega_j) = -\text{arctg}\left(S_{Sj}/S_{Cj}\right) = -\text{arctg}\left(b_j/a_j\right) = \varphi_j$$

.

Numerical spectral analysis, realized in the course of processing measurements of pressure pulsations on the walls of the cone of the suction pipe of the model, consisted in finding the coefficients a_j, b_j, C_j, S_{Cj}, S_{Sj}, $S(\omega_j)$ for finite functions defined on the interval [0, T] by discrete values P(t) = Pi with a time sampling step Δt = 0.001 s. It was reduced to calculating integrals (4)-(7) using numerical integration formulas by the rectangle method.

$$a_j = \frac{2}{N}\sum_{i=0}^{N-1} P_i \cos(2\pi\omega_j i\Lambda t)$$

,

$$b_j = \frac{2}{N}\sum_{i=0}^{N-1} P_i \sin(2\pi\omega_j i\Delta t)$$

,

$$S_{Cj} = \Delta t\sum_{i=0}^{N-1} P_i \cos(2\pi\omega_j i\Lambda t) = \frac{a_j T}{2}$$

,

$$S_{Sj} = \Delta t\sum_{i=0}^{N-1} P_i \sin(2\pi\omega_j i\Delta t) = \frac{b_j T}{2}$$

.

Next, using formulas (3) and (8), the amplitude C_j and the spectral density modulus $S(\omega_j)$ at the frequency of the j^{th} mode ω_j were calculated. As a result, the doubled spectral density was normalized (reduced to a dimensionless form) according to the pressure H_0 and frequency ω_j.

$$\frac{2S(\omega_j)\omega_j}{\rho g H_0} = \frac{C_j}{\rho g H_0}\cdot\frac{T}{T_j}$$

,

where $T_j = 1/\omega_j$ is the period of the j^{th} mode.

The diagrams of the normalized spectral density of pressure pulsations on the stacks of the suction pipe cone of the model are shown in Figure 4. The caption below the figure indicates the operating mode of the model and the normalized pressure pulsation standards calculated using formula (1).

The spectral density curve obtained for the direct discharge of idle flow through the turbine flow path with the remote impeller of the hydraulic unit is shown in red in Figure 4. In this case, the guide vane of the hydro turbine was set to a blade rotation angle of β_{01} = 25°14' (see Figure 2), no counter-vortex damper was installed in the impeller chamber, and no air was supplied under the chamber cover.

In this mode, the spectral density diagram of pressure pulsations on the walls of the suction pipe cone has a clear sharp peak with a leading frequency of 19.5 Hz. There is no doubt that this is the so-called cable frequency. It is obvious that a highly swirled flow descends from the blade system of the guide apparatus into the impeller chamber, in which, under the action of centrifugal forces in the near-axis zone, a vortex cord with a pressure significantly lower than atmospheric pressure is formed in the center of the swirled flow, even in the conditions of a low-pressure laboratory model.

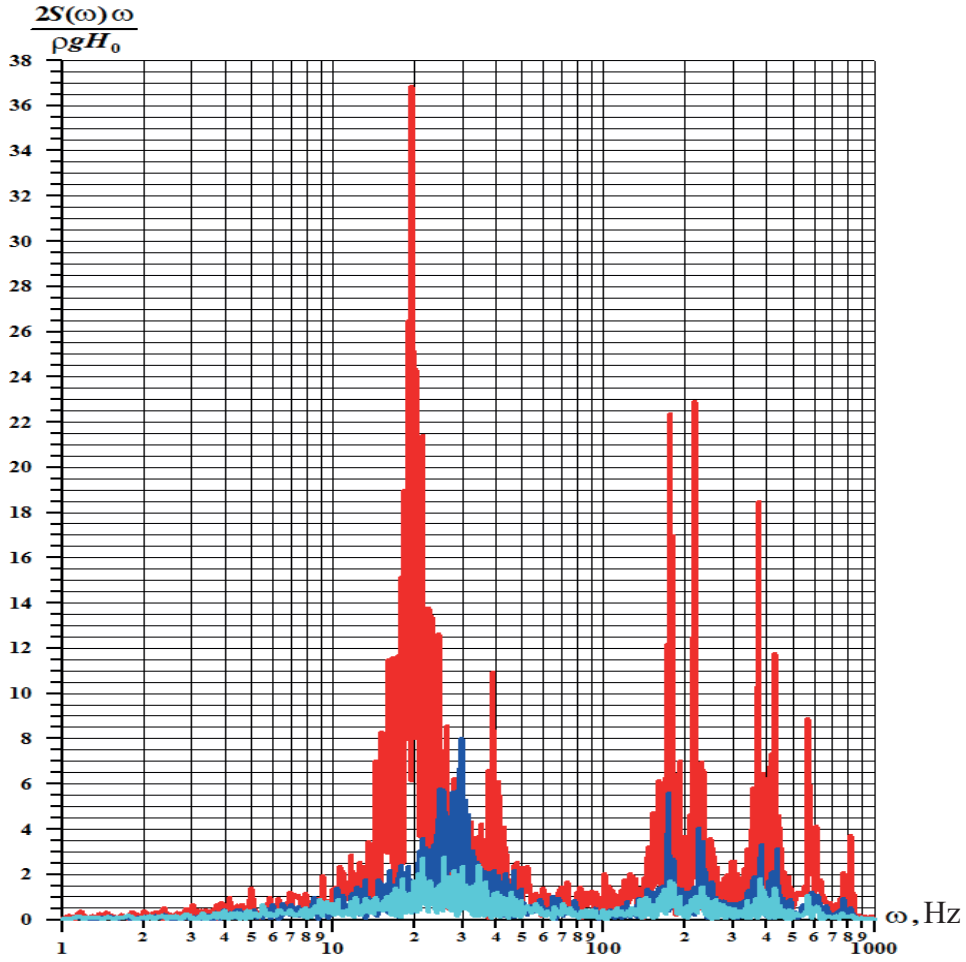


Figure 2. Normalized spectral density of pressure pulsations on the walls of the suction pipe cone: mode 1 (σ = 0.3834) is shown in red; mode 2 (σ = 0.0915) is shown in blue; mode 3 (σ = 0.0574) is shown in light blue

Under natural conditions, the vortex bundle is a cylindrical cavity with a pressure equal to the pressure of saturated water vapor or atmospheric pressure (when the bundle is closed to the atmosphere). The vortex bundle is a source of high dynamics.

A large number of studies have been devoted to the investigation of vortex dynamics and the associated phenomenon of vortex decay. Examples include [3, 21], an excellent review covering 45 years [22], and related articles [23-27]. It is known that cable dynamics is characterized by low frequency and manifests itself in the precession of the core (cable) of a swirling flow, when the entire mass of the flow is drawn into the dynamic process, causing the flow to lose its stability as a whole. Due to the involvement of large water masses, cable dynamics pose an exceptional danger. For cable dynamics to occur, the pressure in the near-axis zone (core, cable) of the swirling flow must be lower than in the downstream sections. Then, under the influence of a negative pressure gradient, significant masses of water from the downstream sections are drawn into the core of the counter-rotating flow, interact with it, rotate, and are thrown back. The process of periodic splashes changes the structure of the swirling flow. The straight axis of the swirling flow becomes curved and begins to rotate around the geometric axis of the water pipe at a certain angular frequency, i.e., the flow loses its axial symmetry and stability. This phenomenon of loss of stability of the swirling flow is called vortex breakdown. These

conditions appear in the cone of the suction pipe in the absence of a working wheel and the formation of a swirling flow by the guide apparatus of the hydraulic turbine. The swirling flow formed in this way, which has a vacuum cavity in the axial zone, encounters a pressure drop determined by the level of the lower tailwater. As a result, the very negative pressure gradient arises, which activates the splash mechanism with subsequent cable dynamics.

If it is established that the dynamics of the cable are related to the loss of stability of the swirling flow as a whole, then it has the rotation frequency of the flow itself, or that part of the flow that is directly adjacent to the cable. Based on this position, let us consider what dynamics should be expected in the cone of the suction pipe under these conditions at the actual facility.

The geometric characteristic of the local swirler, calculated using Abramovich's formula [28, 29] and called the Abramovich number, for the guide apparatus of a hydroturbine is determined by the formula

$$A = \frac{R}{2b\left(1-\frac{nc}{\pi R_0}\right)\mathrm{tg}\beta_{01}} = \frac{1.3606}{2\cdot 0.88\left(1-\frac{20\cdot 0.0569}{3.14\cdot 1.55}\right)\mathrm{tg}(25^{\circ}14')} = 2.141,$$

where R is the radius of the suction pipe cone, $R = D/2 = 2.7212/2 = 1.3606$ m; R_0 is the radius of installation of the guide vane rotation axes, $R_0 = D_0/2 = 3.1/2 = 1.55$ m; b is the height of the guide vanes of the hydro turbine, b = 0.88 m; β_{01} is the angle of installation of the guide vanes, $\beta_{01} = 25^{\circ}14'$; n is the number of guide vanes, n = 20; c is the profile half-thickness of the guide vane on its axis of rotation, c = 56.9 mm.

Following the method of hydraulic calculation of local swirlers [11–13], the relative area of the critical annular cross-section of the swirling flow at A = 2.141 will be $\omega_0 = 0.4969$, and the flow rate passed through will be Q = 48.3 m^3/s.

On the other hand, the geometric characteristic of the swirling flow device is equal to the Higer-Barr swirl number [30–32], defined as the ratio of the azimuthal angular momentum (M) of the swirling flow to twice the product of its axial angular momentum (I) and the hydraulic radius (R_h) [12, 33]

$$A = \frac{M}{2R_h I},$$

where $M = \rho Q u_R R$; $I = \rho Q V$; $R_h = \omega_0 D/4$; V is the average axial velocity across the flow cross-section at the inlet of the suction pipe cone, $V = Q/\omega_0 \pi R^2$; u_R is the tangential flow velocity at the walls at the inlet of the suction pipe cone.

From the Higer-Bera swirling number, the following equality follows [11]

$$u_R = \frac{QA}{\pi R^2}.$$

Then the period of one flow cycle around the perimeter of the cone of the suction pipe is

$$T_R = \frac{2\pi R}{u_R} = \frac{2\pi^2 R^3}{QA},$$

and its rotation frequency will be equal to

$$\omega_R = \frac{1}{T_R} = \frac{QA}{2\pi^2 R^3}.$$

This period and frequency correspond to one revolution of the flow at the walls of the cone of the suction pipe. On the surface of the vortex bundle, the period and frequency will be different. Assuming a radial distribution of tangential velocities according to the law of potential rotation, where $u_{RR} = \mathrm{const}$, it can be shown that on the free surface of the vortex bundle they will be equal to

$$T_d = \frac{1-\omega_0}{\omega_R},$$

and

$$\omega_d = \frac{\omega_R}{1-\omega_0}.$$

Thus, the cable dynamics associated with the loss of flow stability as a whole will occupy the frequency band $\omega_R < \omega < \omega_d$, i.e.

$$\frac{QA}{2\pi^2 R^3} < \omega < \frac{1}{1-\omega_0} \cdot \frac{QA}{2\pi^2 R^3}.$$

By substituting numerical values, we obtain that the cable frequencies on the full-scale object will occupy the band $2.08 < \omega < 4.13$ Hz. Considering the scale of the model M 1:27.2, the range of cable frequencies in hydraulic modeling according to Euler's determining criterion should be between 10.8 and 21.5 Hz. In fact, this is the frequency range we observe in Figure 4.

The recorded normalized pressure standard of pulsations on the walls of the cone of the suction pipe on the model at this mode was $\sigma = 0.3834$. Assuming the maximum pressure pulsation amplitude equal to the Gaussian triple standard, at the actual operating pressure $H_0 = 50$ m, we obtain

$$P' = \pm 3\sigma\rho g H_0 = \\ = \pm 3 \cdot 0.3834 \cdot 1000 \cdot 9.81 \cdot 50 = \pm 564 \text{ кПа}$$

or $\Delta H = \pm 57.5$ m of water column.

It is obvious that under these conditions, half of the time spent discharging idle waste through the turbine flow path, the pressure at the faces of the suction pipe cone will drop to the vaporization pressure, meaning that they will be subject to intense cavitation.

In this regard, appropriate requirements must be imposed on the lining of the suction pipe. For example, the calculated alternating excess pressures on streamlined surfaces from –100 to +200 kPa require reliable interaction between the lining and the concrete structure. A disruption in their interaction, even in a small area, can lead to rapid destruction of the entire lining. As far as we know, there is no satisfactory technical solution to this problem for such a complex structure as a suction pipe.

In general, we consider the passage of idle flow through a hydro turbine unit with a remote working wheel to be unacceptable due to the high dynamic loads on the structure, leading to its destruction and a serious accident at the HPP.

Figure 4 shows in blue the diagram of the normalized spectral density of pressure pulsations on the walls of the cone of the suction pipe in the mode of passing cold flow through the turbine flow path when a counter-vortex damper is installed in the remote impeller chamber. Air was not supplied under the chamber cover.

It can be seen that the installation of a counter-vortex damper in the impeller chamber sharply, judging by the normalized pulsation standard equal to $\sigma = 0.0915$, reduced the dynamic loads on the structure by a factor of four compared to the previous mode without the damper installed.

The installation of the counter-vortex damper also changed the frequency distribution of the spectral density of pressure pulsations, shifting the band of cable dynamics toward higher frequencies with a maximum in the 30 Hz range and increasing the relative spectral density in the turbulent range at frequencies of 150–500 Hz. However, in the turbulent range, there is no shift in spectral density by frequency between the modes without and with the damper installed.

The calculation of the cable frequency band, performed according to the above method, with regard to the design and hydraulic characteristics of the developed counter-vortex damper, showed that under natural conditions this band will lie in the range of 2.17–5.35 Hz. On a 1:27.2 scale hydraulic model, it will occupy the range of 11.3–27.9 Hz, thus confirming the shift of the cable dynamics towards higher frequencies.

The diagram shown in Figure 4 in blue depicts the normalized spectral density of pressure pulsations on the walls of the suction pipe cone. It refers to the idle flow mode when a counter-vortex damper is installed and air is supplied from the atmosphere through the air duct under the impeller cover (see Figures 1 and 2).

It has been established that supplying air from the atmosphere through the impeller cover into the vacuum vortex core of the swirling flow formed by the blade system of the counter-vortex damper sharply reduces the hydrodynamic loads on the structure. Thus, the recorded normalized standard of pressure pulsations on the walls of the suction pipe cone, equal to s = 0.0574 in this mode, is 1.6 times lower than the standard of pressure pulsations in a similar mode, but without air supply. This indicates the need to supply atmospheric air to the counter-vortex damper at idle water outlets of medium-pressure hydraulic structures. It should be noted that the overall reduction in dynamic loads on the structure compared to the mode without the counter-vortex damper was 6.7 times.

As a result, with the installation of an anti-vortex damper and the supply of air from the atmosphere, the maximum amplitude of pressure pulsations on the walls of the suction pipe cone under field conditions will be

$$P' = \pm 3\sigma\rho g H_0 = \\ = \pm 3 \cdot 0.0574 \cdot 1000 \cdot 9.81 \cdot 50 = \pm 84.5 \text{ kPa}$$

or $\Delta H = \pm 8.61$ m of water column.

It can be stated that the considered option of passing the idle flow through the turbine unit with a counter-vortex damper installed in the working wheel chamber of the hydraulic turbine, while ensuring air supply, is quite satisfactory. A positive environmental effect is the significant aeration of the flow discharged into the lower reach.

The diagram of the normalized spectral density of pressure pulsations on the walls of the suction pipe cone shows that when air is supplied to the counter-vortex damper, the spectrum in the central part becomes flatter and wider, covering the range from 10 to 50 Hz, where there are no pronounced leading frequencies. This indicates suppression of the bundle dynamics. Instead, a spectrum corresponding to large vortex structures formed by the intense interaction of concentric counter-rotating flows is observed. In the turbulent frequency range from 150 to 500 Hz, suppression of pulsations is also observed, manifested in the flattening of the spectral density graph with a decrease in its modulus.

4. CONCLUSION

1. The article has examined the possibility of using a pressure counter-vortex energy dissipator for water flow in the power water pipeline of a medium-pressure hydroelectric power plant during its reconstruction into an operational spillway.
2. A configuration of a pressure counter-vortex damper in the chamber of a dismantled hydro turbine impeller with damping of idle flow energy in the suction pipe cone has been proposed
3. The study established the expediency of the design of the counter-vortex damper with its attachment to the cover of the remote hydraulic turbine.
4. A counter-vortex damper design has been developed. It includes a hydro turbine guide apparatus and a fairing with non-rotating blades rigidly fixed to it. In this design, the guide apparatus of the hydraulic turbine with rotating blades simultaneously performs the functions of a regulating valve and a swirl generator for part of the idle flow. The blades rigidly fixed to the fairing swirl the other part of the idle flow in the opposite direction to the swirl of the guide apparatus blades. The fairing directs one part of the flow coaxially into the other, oppositely swirled part and leads them into a common damping chamber in the cone of the suction pipe. In the cone of the suction pipe, the coaxial concentric oppositely twisted parts of the flow mutually dampen each other's twist, thereby achieving the effect of damping the energy of the flow as a whole.
5. Experimental studies of pressure pulsations on the walls of a hydraulic model of a turbine block with a counter-vortex damper were carried out. As a result of the studies, normalized spectra and standards of pressure pulsations on

the walls of the suction pipe cone were obtained for various operating modes of the model.

6. It has been established that in the direct discharge mode of idle flow through the turbine unit with the remote impeller of the hydraulic unit, the normalized pressure standard of pulsations on the walls of the suction pipe cone reaches $\sigma = 0.3834$. Thus, the walls of the suction pipe cone are subjected to high dynamic loads. The diagram of the spectral density of pressure pulsations has a sharp peak with a leading frequency of 19.5 Hz on the model (3.74 Hz in real conditions). This corresponds to the cable frequencies at the loss of flow stability in the process of the so-called vortex breakdown. It is shown that in this mode, in addition to dynamic loads, the walls of the suction pipe cone will be subjected to intense cavitation. It is concluded that passing idle flows through the hydro turbine unit of a medium-pressure HPP with a remote working shaft is unacceptable and leads to the destruction of the structure.

7. The method for calculating the range of cable frequencies associated with vortex decay has been experimentally confirmed

8. The installation of an anti-vortex damper in the remote impeller chamber significantly reduces the dynamic loads on the walls of the suction pipe cone, bringing the pulsation standard to a level of $\sigma = 0.0574$. This is 6.7 times lower than the standard in the idle flow discharge mode through the turbine unit without the damper installed.

9. Bypassing cold losses through the hydro turbine unit of a medium-pressure HPP with a counter-vortex damper installed in the remote impeller chamber meets the standards and requirements for the safety and reliability of hydraulic structures and hydroelectric power plants.

10. It has been established that atmospheric air must be supplied to the counter-vortex damper through a special air duct. This study was con-

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