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Investigation of Oil Vapours in Hydro-Generator

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Abstract

The article discusses engineering approaches to solving the problem of oil vapours in generators. Due to the design characteristics of each power plant, it is difficult to find a typical solution. Nevertheless, the article suggests ways to reduce the amount of oil mist. For their implementation, a calculation methodology based on the substitution scheme of the hydraulic path of the bearing support unit is created which takes into account the design features of the bearing and the parameters of the environment around the oil bath. The methodology employed is presented. The numerical method of modelling the aerodynamic fields of the entire hydro generator is used to estimate the air flows and pressures in the oil bath zone of the thrust bearing. Additionally, the method made it possible to track suspected oil particles that could get from the thrust-bearing bath into the surrounding area of the generator. Measures are proposed to reduce the level of oil mist through the competent design of the oil vapour removal system from the bearing bath.

Keywords

Hydro-Generator, Cooling, Oil Mist, Bearings, Fluid Dynamics

1. Introduction

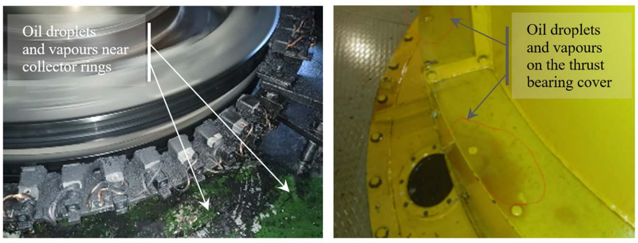
According to the published study by the authors [1], a slight decrease in the growth rate of the hydropower industry in Russia is expected in the period up to 2035, as well as the absence of new large hydropower projects. The majority of investments will be directed at the modernization and reconstruction of existing plants. Russian hydropower equipment manufacturers serve domestic needs and export their products mainly to developing countries in Asia and Latin America. At the same time, the IEA (International Energy Agency) predicts a decline in the rate of commissioning of hydropower plants (HPP) after 2030 in almost all major countries of the world. As a result, competition in the supply of hydroelectric equipment is intensifying.

Competitiveness of products can be ensured by improvement of product quality, service and care for the comfort of equipment operation.

One of the issues of HPP equipment quality improvement is the reduction of oil mist from bearing baths and bearings of the hydraulic unit into the turbine shaft and into the inner zone of the hydro-generator.

The problem of oiling hydraulic unit surfaces has several aspects, which are listed below:

- Oil mist or oil vapours contaminate the constructive and active parts of the hydro-generator, particularly the current-carrying parts. The presence of oil affects the conductor's insulation properties, which directly affects the reliability of the hydro-generator, especially when oil is mixed with dirt, coal dust particles (see **Figure 1 (a)**), etc [2]. In the presence of oil, the thickness of the dust layer retained on the insulation surface increases significantly, resulting in an increase in the temperature of the active elements of the structure generator [3].
- Oiling can lead to loosening of the winding fixing structures, which in turn can lead to mechanical damage to the insulation.
- Oil droplets and vapours accumulate on the surfaces of the equipment and on the bearing housing (see **Figure 1 (b)**), especially on the turbine cover where personnel moves when the unit is shutdown. This poses a risk of injury to operating personnel from falls. In addition, operating personnel working in the area of the turbine shaft may suffer from respiratory issues due to inhaling oil mist.
- It requires constant checking and replenishment of the oil supply if the oil level in the bath drops. The operating instructions for hydro-generators require a visual inspection of the stator winding for oiling and its cleaning if necessary.
- Oil mist escaping from bearing oil baths leads the harmful effects on the environment and the decline in water quality in the downstream parts of rivers due to the infiltration of oil mist and droplets through the turbine top cover into the spiral scroll [3].



(a) (b)

Figure 1. Oil droplets and vapours on equipment surfaces: (a) in the brush contact area, (b) on the bearing housing.

Since the mid-20th century, constructions of hydro-generators' oil baths have been developed to minimize oil mist leakage. Various seal designs were created, patented and introduced into current units. Emphasis was placed on using rubber gaskets, cords, and epoxy varnish to seal the oil baths [4]. Leather or rubber sleeves, along with sets of brass rings (labyrinth seals) were installed in the gaps between rotating surfaces and bath covers [5]. However, during unit operation, excessive vibration can cause an increase in seal gaps [3]. Focus on minimizing the oil mist leakage is also addressed in both Russian and foreign standards [6] [7]. In addition to the standard labyrinth seal system, most hydro generators produced by the "Electrosila" plant feature oil vapour extraction. This is achieved through a pipeline that connects the upper region of the oil bath to a dedicated oil trap tank located outside the equipment. Oil, which passes into the pipeline due to the pressure difference between the internal oil bath and atmospheric air pressure, cools and condenses in the tank.

A properly designed system for extracting oil vapour can lower the leakage of oil mist. However, a literature review shows that enhancing the oil circulation system can only decrease, but not entirely eradicate, the formation of oil mist. For instance, the research by the authors [8] made modifications to the oil bath cover of the generator thrust bearing, decreasing oil mist leakage from 4.9 m^3/h to 0.36 m^3/h. In addition to enhancing seal quality, methods to lower oil vapour concentrations involve fitting oil deflectors to condense and return oil to the oil bath.

Evaporation transforms lubricating oil from a liquid to an oil mist, which is common in equipment that uses oil as a lubricating and cooling fluid. Using a simulation-based evaporation-condensation model by simulating a three-phase oil-air-oil mist environment, the authors of the study [9] showed that increasing the oil temperature by 6 degrees significantly increases the amount of oil mist escaping from the thrust bearing bath. During bearing operation, the operating oil temperature is between 50 and 70 °C, which means that the bath is filled with a mixture of oil, oil mist and air. The authors state that in oil bath simulation modelling, many researchers assume the oil bath is a closed system without considering the clearance between the shaft and the seal, assuming that the oil mist leakage is inevitable and judging the amount of leakage from the pressure distribution in the oil bath.

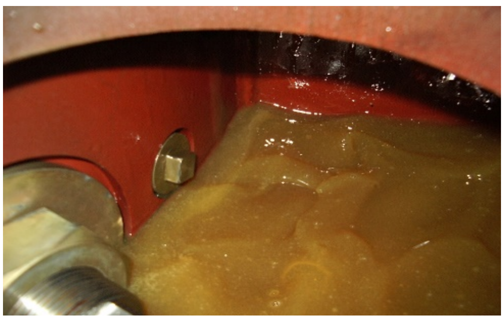


Figure 2. Formation of air-oil emulsion at intensive mixing of oil with air due to rotation in the oil bath of the thrust bearing.

Thus, intensive mixing of oil and air by rotating oil in the oil bath of the thrust bearing (see **Figure 2**) forms a substantial amount of air-oil emulsion. The rotating oil's surface profile becomes parabolic, and foam is formed upon collision with the bath's internal parts. The foamed oil triggers more evaporation due to a considerable increase of evaporation surface. The accumulation of oil mist causes an increase in pressure difference between the inside and outside of the oil bath, ultimately resulting in the leakage of oil mist.

In spite of the fact that the process of studying the processes of oil vapour formation in oil baths of hydraulic unit support units, as well as detailed study of theoretical and practical issues of heat exchange of support units continues to involve a large number of researchers [3] [10] [11] [12] [13], at the same time not enough in modern publications is given to the methods of reducing oil vapours spreading inside the generator.

The main goals of this research are:

- To reduce the amount of oil mist by a calculation methodology based on the substitution scheme of the hydraulic path of the bearing support unit which takes into account the design features of the bearing and the parameters of the environment around the oil bath.
- To estimate the air flows (pressures) in the oil bath zone of the thrust bearing and to track suspected oil particles that could get from the thrust bearing bath into the surrounding area of the generator by numerical method.

In order to achieve these objectives, the article presents the system of air purification from oil mist and the numerical method of modelling the aerodynamic fields of the entire hydro generator.

2. Calculation Methodology for Reduction Oil Vapours in Generators

The system of air purification from oil mist can be used to evacuate oil vapors from the oil baths of hydro-generator bearings. To ensure the efficient operation of various types of hydro-generators, a calculation methodology has been developed in order to select suitable equipment with the appropriate capacity, pipeline parameters, number, and size of compensation holes.

2.1. The Method Utilizes

The method utilizes replacement scheme of the hydraulic path [14]. The system model under study and its corresponding replacement scheme are depicted in **Figure 3**, showcasing the rotor bushing holes which provide hydraulic resistance while concurrently producing pressure in the line; the pipeline; the gap between oil bath wall and rotor bushing; the dedicated oil trap tank; the oil trap, or filter; and the fan. In the model, the fan is connected to the gap and oil trap tank through the atmosphere. This connection is based on the assumption that there is equal atmospheric air pressure both inside and outside the shaft at the location of the filter.

The pressure developed by the holes in the rotor sleeve can be determined by the equation (1):

$$H = \frac{\gamma}{2g}\left(u_2^2 - u_1^2\right) \tag{1}$$

The pressure losses, equitation (2), in each element of the circuit are calculated as follow:

$$\Delta H = \xi_i\left(\mathrm{Re}\right)\frac{\rho}{2}\upsilon_i^2 = z_i Q_i^2 \tag{2}$$

where Re = $f(v)$. The total hydraulic resistance of each circuit element in the general case can be define by the equation (3):

$$z_i = z_f + z_t + z_e = \frac{\rho}{2}\left(\sum_{q=1}^{n1}\frac{\xi_f}{S_q^2} + \sum_{j=1}^{n2}\frac{\xi_{tj}}{S_j^2} + \sum_{q=1}^{n1+1}\frac{\xi_{eq}}{S_q^2}\right) \tag{3}$$

where:

index f means friction,

index t means turn of the flow,

index e means expansion of the output flow,

S_q is duct cross section in m^2,

S_j is duct cross section after the turn of the flow in m^2,

$n1$ is number of sections of the scheme,

$n2$ is number of turns of the flow.

The solution uses the transformation of a second-degree equations system to a first-degree equations system and the method of successive approximations requiring compliance with Kirchhoff's law [14], which is implemented using a program written in VBA.

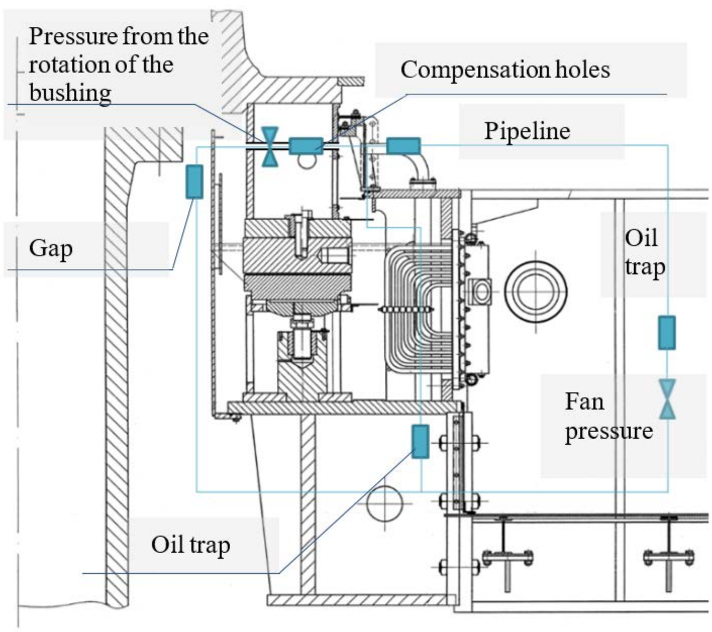


Figure 3. The system model under study and its equivalent scheme interpretation. The rectangles indicate the aerodynamic resistances.

2.2. The Numerical Method for Estimating Airflows and Pressures within the Oil Bath and the Contamination Tracks

A numerical method was utilized to model the aerodynamic fields (computational fluid dynamics—CFD) of an entire hydro-generator, estimating airflows and pressures within the oil bath of the thrust bearing [15] [16]. Additionally, the method tracked suspected oil particles that may have escaped from the thrust bearing bath and into the generator's surrounding area.

The results of the CFD calculation are shown by the example of two types of ventilation systems for hydrogenerators: single-sided ventilation system and double-sided ventilation system. The generators with these systems are operated on Volga HPP (Russia) and HPP Nuojua (Finland).

The modelling involved the stationary Reynolds-averaged Navier-Stokes (RANS) equations (4) with two eddy viscosity turbulence model equations, solved using the finite volume technique and the frozen rotor approach [17] [18].

$$\begin{aligned}
&\frac{\partial U}{\partial t}+\nabla(U\mathbf{U})=-\frac{1}{\rho}\frac{\partial P}{\partial x}+\nu\Delta(U)+\frac{1}{\rho}\left[\frac{\partial\left(-\rho\overline{u'^2}\right)}{\partial x}+\frac{\partial\left(-\rho\overline{u'v'}\right)}{\partial y}+\frac{\partial\left(-\rho\overline{u'w'}\right)}{\partial z}\right],\\
&\frac{\partial V}{\partial t}+\nabla(V\mathbf{U})=-\frac{1}{\rho}\frac{\partial P}{\partial y}+\nu\Delta(V)+\frac{1}{\rho}\left[\frac{\partial\left(-\rho\overline{u'v'}\right)}{\partial x}+\frac{\partial\left(-\rho\overline{v'^2}\right)}{\partial y}+\frac{\partial\left(-\rho\overline{v'w'}\right)}{\partial z}\right],\\
&\frac{\partial W}{\partial t}+\nabla(W\mathbf{U})=-\frac{1}{\rho}\frac{\partial P}{\partial z}+\nu\Delta(W)+\frac{1}{\rho}\left[\frac{\partial\left(-\rho\overline{u'w'}\right)}{\partial x}+\frac{\partial\left(-\rho\overline{v'w'}\right)}{\partial y}+\frac{\partial\left(-\rho\overline{w'^2}\right)}{\partial z}\right],
\end{aligned} \tag{4}$$

where:

$-\rho\overline{u'^2}, -\rho\overline{v'^2}, -\rho\overline{w'^2}$ are the normal stresses in Pa,

$-\rho\overline{u'v'}, -\rho\overline{u'w'}, -\rho\overline{v'w'}$ are the Reynolds stresses (tangential stresses),

t is the time in s; is the gas density in kg/m^3; is the kinematic viscosity in m^2/s;

When the Navier-Stokes equations are averaged over time, a new term known as the turbulent stress tensor (or Reynolds stress tensor) emerges.

The Reynolds stresses are calculated using the realizable k - ε turbulence model, which introduces a novel equation (5) for turbulent viscosity in contrast to the standard k-ε model [19] [20]. Characteristics ofthe realizable k - ε turbulence model are presented in **Table 1**.

Table 1. Characteristics of the realisable k - ε model.

$C_{1\varepsilon}=\max\left[0.43,\left(\frac{\eta}{\eta+5}\right)\right]$ $\eta=S\frac{k}{\varepsilon}$	$C_{2\varepsilon}=1.9$	$U^*=\sqrt{S_{ij}S_{ij}+\tilde{\Omega}_{ij}\tilde{\Omega}_{ij}}$	$\phi=\frac{1}{3}\cos^{-1}\left(\sqrt{6}W\right)$
$S=\sqrt{2S_{ij}S_{ij}}$ $S_{ij}=\frac{1}{2}\left(\frac{\partial u_j}{\partial x_i}+\frac{\partial u_i}{\partial x_j}\right)$	$\nu_t=C_\mu\frac{k^2}{\varepsilon}$	$A_0=4.04$ $A_s=\sqrt{6}\cos\phi$	$\sigma_k=1.00$
$W=\frac{S_{ij}S_{jk}S_{ki}}{\tilde{S}^3}$ $\tilde{S}=\sqrt{S_{ij}S_{ij}}$	$C_\mu=\frac{1}{A_0+A_s\frac{kU^*}{\varepsilon}}$	$\tilde{\Omega}_{ij}=\Omega_{ij}-2\varepsilon_{ijk}\omega_k$ $\Omega_{ij}=\bar{\Omega}_{ij}-\varepsilon_{ijk}\omega_k$	$\sigma_\varepsilon=1.20$

$$\frac{\partial \varepsilon}{\partial t}+U_j\frac{\partial \varepsilon}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\left(\nu+\frac{\nu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}S\varepsilon-C_{2\varepsilon}\frac{\varepsilon^2}{k+\sqrt{\nu\varepsilon}} \quad (5)$$

Assumptions are made in the numerical simulation:

- The end parts of the stator winding are used to describe its geometric shape while preserving the winding end length and angle of inclination. However, winding connections are not included in the model.
- Rectangular section spacers are used between stator steel packages, with one spacer per slot pitch of the stator. The spacer is deepened 2 mm from the air gap.
- To minimize calculations, the calculation model area is limited by the circumference of the pressure element comprising the generator rotor spider.
- Periodic conditions are applied to the air gap, radial ducts in the stator, poles, frame, cold and hot air chambers, and air openings in the upper and lower generator brackets.
- The model was partitioned into finite elements using adaptive techniques that consider the assumed air flow velocity in the actual geometry and achieve an appropriate dimensionless coefficient y+ in the boundary layer.
- An additional area (volume) has been incorporated to facilitate air flow and replicate the actual spacing in the separating shields that were installed between the upper (lower) end chamber and the air space in front of the fan (if presented).

Rotation is simulated using the MRF technique, where the computational model comprises of rotating and stationary regions. The air coolers are defined by a porous volume, as an axisymmetric tensor in a cylindrical coordinate system. The tensor coefficients in the radial direction are determined from the dependence of pressure drop on air velocity at the outlet of the air coolers, chosen from the manufacturer's experimental data. The accuracy of the hydro-generator ventilation system's numerical modelling is verified through a satisfactory correlation between the calculations and experimental findings, pertaining to the entire air flow quantity and local velocity parameters at various points along the ventilation pathway.

3. Research Results and Discussion

The generator of the Boguchanskaya HPP utilizes the system of air purification from oil mist, which is fully manufactured at the "Electrosila" plant. The system comprises a pipeline (see **Figure 4**) that connects the seals of the bearing oil bath and the oil bath of the thrust bearing with an oil trap and a 1.5 kW centrifugal fan. Main design parameters of the system are following: H_{fan} = 3600 Pa, Q = 0.1 m^3/s. A check valve is installed behind the fan to prevent air from flowing back into the system. The fan blows air mixed with oil vapour into the oil trap, where the oil settles and condenses on the reflectors. The oil trap features a window for visual inspection of the accumulated oil and a tap for draining it. The system has been noted for its positive operational experience.

BOGUCHANSKAYA HPP, Russia	
Hydrogenerator	
Rated power	370 MVA
Rated stator current	13563 A
Rated speed of rotation	90.9 rpm
Total air flow rate through the air coolers	116.7 m^3/s

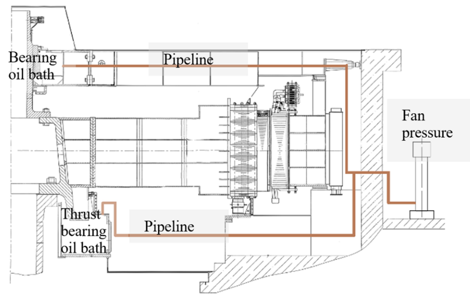
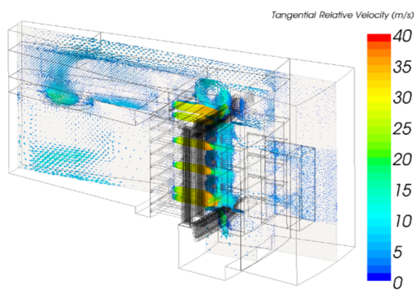
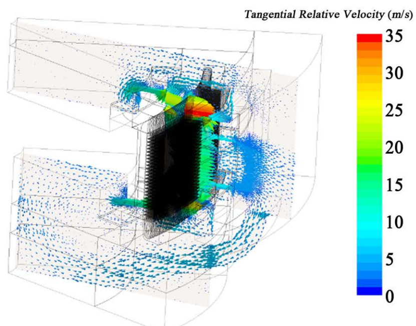


Figure 4. The schematic pipelines of the system under study on the example generator of the Boguchanskaya HPP.



VOLGA HPP, Russia	
Hydrogenerator	
Rated power	139.4 MVA
Rated speed of rotation	68.2 rpm
Total air flow rate through the air coolers	66.5 m^3/s

Figure 5. Velocity distribution of the hydro-generator single-sided ventilation system by CFD.



HPP NUOJUA, Finland	
Hydrogenerator	
Rated power	38 MVA
Rated speed of rotation	136.4 rpm
Total air flow rate through the air coolers	48 m^3/s

Figure 6. Velocity distribution of the hydro-generator double-sided ventilation system by CFD.

Figures 5 and 6 display the numerical results for the velocity vector distributions.

By analyzing the velocity vector distribution, it is evident that the generator's active parts are more prone to contamination in the end region of the generator. This includes the surfaces of the insulation of the rotor poles and the end parts of the stator winding. The contamination occurs when oil particles mixed with dirt penetrate the rotor spoke (Refer to Figure 7 for a visual representation).

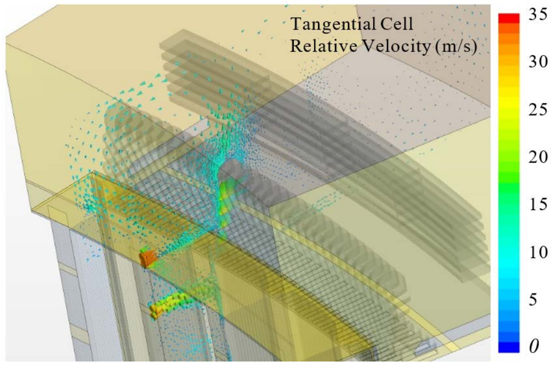
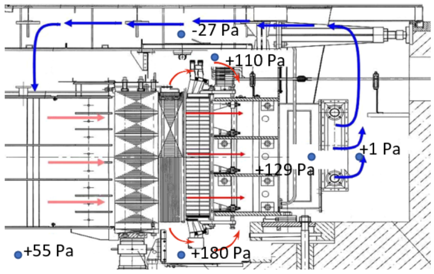


Figure 7. Velocity distribution in the end winding zone of Volga HPP hydro-generator by CFD.

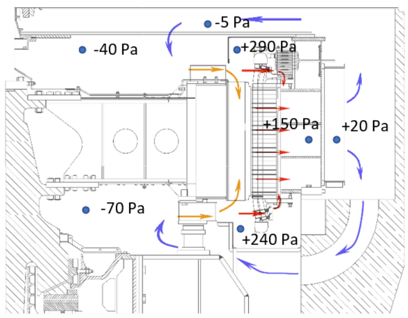


VOLGA HPP, Russia

Hydrogenerator

Rated power	139.4 MVA
Rated speed of rotation	68.2 rpm
Total air flow rate through the air coolers	66.5 m^3/s

Figure 8. Overpressure in different design of the hydro-generator single-sided ventilation system by CFD.



HPP NUOJUA, Finland

Hydrogenerator

Rated power	38 MVA
Rated speed of rotation	136.4 rpm
Total air flow rate through the air coolers	48 m^3/s

Figure 9. Overpressure in different design of the hydro-generator double-sided ventilation system by CFD.

Based on the numerical simulation overpressure data, it is evident that in umbrella-type hydro-generators, the air pressure is positive under the rotor (near the thrust bearing oil bath) with a single-sided ventilation system (**Figure 8**) and negative with a double-sided ventilation system (**Figure 9**). If the pressure is negative, oil vapour may escape from the oil bath through the seal and into the low-pressure zone under the rotor. To prevent this, an even lower pressure must be created at the inlet of the oil vapour extraction system. This allows the mixture of air and oil to be directed via pipework to the oil vapour filter. It's essential to have a compensating opening in the support sleeve allowing fresh air to enter the oil compartment. If the air pressure beneath the rotor is at or above zero, then the required fan power for the extraction of oil vapour reduces.

4. Conclusions

The literature outlines engineering methods allowing to tackle the issue of oil vapour in generators. Solutions vary with power plant designs, resulting in unique approaches. Nevertheless, in general, existing techniques have been created to diminish the presence of oil mist, but not to eradicate it entirely. Efforts should be made to optimize heat exchange between heated oil and coolers to lower oil temperatures, thus reducing the formation of oil vapour and foam, as well as eliminating oil vapour from bearing baths.

When designing an oil vapor extraction system, it is important to choose the appropriate equipment and its capacity, based on the design features of each system, as well as taking into account the cooling scheme of the generator and the air pressure values in the bearing area.

Further development of scientific research is possible in the formulation of a multiphase problem (Lagrangian multiphase, Volume of fluid method) with modeling of the direct processes of oil flow and the particle trajectories.

Collecting statistical data on oil mist leakage from bearings in hydroelectric power plants is crucial. This data should include the frequency of degreasing of shaft and stator rod surfaces, and the concentration of oil vapour in the turbine shaft area. This data is necessary for further investigation of the problem, to identify new patterns, as well as for long-term planning to solve oil mist problems in power plants.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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