Processes of Outflow of the Boiling Steam-Water Mixture in the Widening Part of Hydro-Steam Turbine Nozzles

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Summary
Renewable energy sources based on solar radiation, wind energy, geothermal energy, and biomass energy have now reached the level of industrial application. A new way to generate electricity using low-potential heat is to install a hydro-steam turbine. In hydro-steam turbines, hot water is supplied directly to turbine rotor nozzles without prior separation into steam and water in separators, which significantly increases the efficiency of hot water energy use. Such turbines are suggested to be used as autonomous energy sources in geothermal heating systems, heating water boilers and cooling systems of chemical reactors, metallurgical furnaces, etc. The authors conclude that the installation of hydro-steam turbines in heating plants and process boiler plants can also be effective if the used exhaust steam-water mixture at the turbine outlet is used to heat the network water or as return water.

Keywords:
hydro-steam turbine, geothermal well, thermal scheme.

1. Introduction

Reaction hydro-steam turbines (HST) are designed based on the Segner turbine. As their working medium, they use hot water, which enters the nozzles under the high pressure created by the centrifugal force of the rotating impeller and starts boiling in the widening part of the nozzles [1]. The source of hot water can be a geothermal well. The schematic diagram of the HST operation as a part of a geothermal power plant (GeoPP) is shown in Fig. 1.

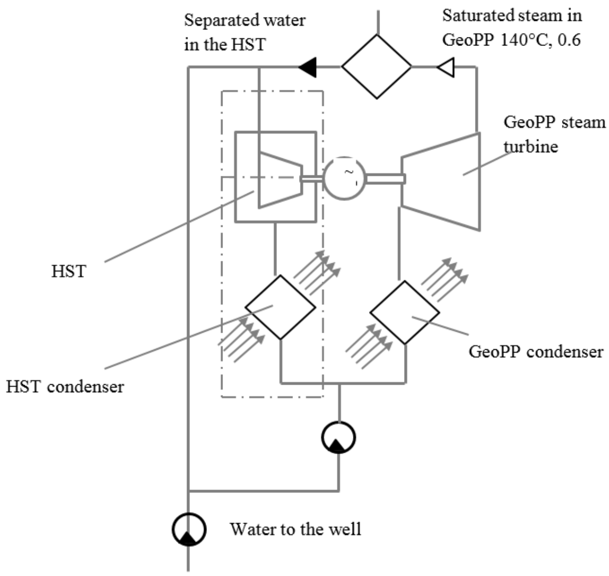

Fig. 1. HST as a part of a GeoPP

The main advantages of a reaction HST include:
• simple design, low capital investment;
• lack of a vane apparatus subject to erosion effects of steam and water flow.
On the other hand, the primary disadvantage of HSTs is their low efficiency.

2. Methods

The work was carried out at the Tsiolkovsky Kaluga State Pedagogical University. The current state of the considered problem of the processes of outflow of boiling steam-water mixture in the expanding part of hydro-steam turbine nozzles.

Physical model of the process of steam-water mixture outflow from nozzles

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The processes of water outflow from divergent nozzles (Fig. 2) occur due to the pressure difference between the nozzle inlet and outlet.

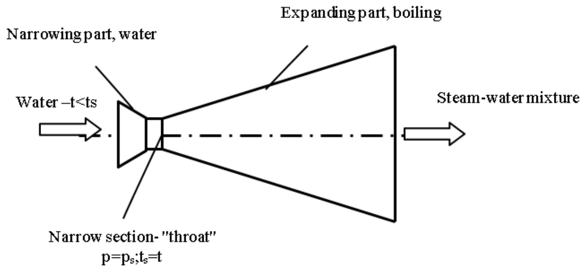


Fig. 2. Nozzle

In the narrowing part of the nozzle, there occurs an increase in velocity and a decrease in pressure. If the pressure at the nozzle outlet is less than the vapor saturation pressure, then the pressure in the narrow nozzle section becomes equal to the saturation pressure, which creates conditions for the onset of boiling. However, the real boiling in the liquid volume begins when the time of stable existence of the metastable liquid is exhausted. The latter depends primarily on the fluid superheat value (the difference between the current temperature and the saturation temperature), yet the onset of boiling is also contingent on such factors as the concentration of solid inclusions, gas bubbles in the liquid flow, cleanliness of the channel surface, and so forth.

The water flow in the nozzle increases as the pressure drop between the inlet and the narrow nozzle section rises until boiling occurs. As back pressure is further reduced, the nozzle is in lock mode, i.e. the flow rate remains unchanged [2].

As the liquid stream moves along the length of the expanding part of the channel, pressure continues to decrease. Experiments by some researchers suggest that boiling begins at the outer boundary of the water stream, the concentration of the vapor phase is minimal at the center of the stream and maximal near the nozzle walls. Evaporation happens due to the heat of the outer layer of the water stream, the temperature of which decreases. The temperature of the center of the stream can decline only due to heat conduction to the relatively cold outer layer [3]. As the fluid moves from the inlet to the outlet, the superheat increases, which shortens the time of stable existence of its metastable state. When the metastable state becomes unstable, intense volumetric boiling occurs, causing the liquid fraction to disintegrate into individual fragments and droplets.

As the vapor phase moves with decreasing pressure, its kinetic energy increases along with its velocity. The velocity of the liquid phase, which exists in separate droplets and fragments and moves by inertia, increases only due to the momentum transferred to it by the vapor. The emerging disbalance of velocities (slip) of the phases leads to their mechanical interaction: the momentum from the vapor phase begins to be transferred to the liquid phase, causing the phases to partially equalize their velocities. The residual slip of the phases strongly depends on the size spectrum of the liquid particles, with larger medium-sized particles having a higher residual slip than smaller ones.

In addition to the phase slip, the structure of the flow coming out of the nozzle and, respectively, the reactive force generated by it, can be influenced by the non-equilibrium of outflow processes. The non-equilibrium is primarily due to the fact that phase transformations require a certain time, the liquid phase can also exist in a metastable state for some time. The time of motion of the working body particle in the expanding part of the nozzle is of the order of 0.001 s, so the non-equilibrium effect can be substantial. The incompleteness of the evaporation process leads to a decrease in the share of the vapor phase in the outlet section of the nozzle. Since it is the vapor phase, which has a higher than average velocity of outflow, that creates additional reactive force increment, the reduction of its share due to outflow non-equilibrium entails a decrease in this force.

Experiments on the organization of steam-water mixture outflow from different nozzles conducted by A.S. Goldin [4] show that the coefficient of velocity determined by the measured reactive force falls in the range of 0.55÷0.7, while a similar parameter for single-phase gaseous media in nozzles with the confusional flow is usually no lower than 0.9. The sharp increase in energy loss at the onset of boiling may be associated with two factors: the interaction of phases and outflow non-equilibrium.

3. Results and Discussion

3.1 Effect of Difference in Average Velocities of Liquid and Vapor Phases (Phase Slip) on the Reactive Thrust and Output of the HST

The above suggests a conclusion that the real distribution of the velocities of droplets and vapor in the outlet section will be consistent with some intermediate option between the two extremes:

1) no impulse exchange between the phases with the maximum difference between the velocities of the phases;

2) so much momentum exchange between the phases that the velocities of both phases in the outlet section are the same and there is no slip.

It can also be argued that the smaller the size of the droplets generated by the fragmentation of the liquid flow, the smaller the phase slip.

Let us now compare two convergent-divergent nozzles with identical initial p_0 and t_0 and the same back pressure p_k with the first nozzle having an uneven velocity field (phase slip) and the second having an even field. Both nozzles have the same diameter of the narrow section ("throat") but different cross-sectional areas at the outlet. Both nozzles are in the calculated mode when the final pressure dictated by the conditions of process development in the nozzle coincides with the pressure provided by the condenser.

In the first nozzle, the liquid phase has a velocity of w' and density of ρ' same for all particles and the vapor phase has a velocity of w'' and density of ρ''.

The specific thrust impulse of the nozzle is:

$$\frac{P}{G} = w_c' \cdot (1 - x) + w'' \cdot x$$

An equivalent of the second nozzle No. 2a is created by adding to the first nozzle an attachment nozzle that equalizes phase velocities through their mechanical interaction without changing the phase composition (Fig. 3).

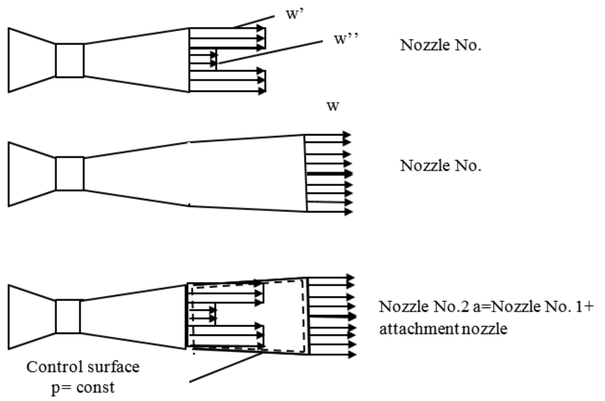


Fig. 3. Schematic design of nozzles

The attachment nozzle has equal pressures at the inlet and outlet, which is achieved through proper selection of the ratio of cross-sectional areas at the inlet and outlet (an increase in the outlet area elevates pressure, a decrease leads to lower pressure). Moreover, by properly profiling the curves of this nozzle, it is also possible to obtain pressure constancy on its lateral surface. Let us identify a control surface limited to the inlet and outlet cross sections and the side surface (dashed line in Fig. 3.1). Throughout the control surface, the pressure is the same.

In the absence of friction, which we disregard, the sum of all external forces over the control surface is 0. According to the law of conservation of momentum in this case the integral momentum $\overline{w}$ of the flow from the inlet section to the outlet, and along with it the specific thrust, must also remain the same:

$$w_c' \cdot (1 - x) + w_c'' \cdot x = w \tag{1}$$

Therefore, the fact of the exchange of momentum between particles with different velocities alone has no effect on the output momentum and specific thrust.

Now we shall consider this as applied to HSTs. Fig. 4 presents an enthalpy–entropy (h-s) chart schematically depicting the workflow in the HST. Herein w is the velocity in relative motion, c – in absolute motion, u – circumferential velocity, and the index "s" denotes the parameters of the isoentropic process. Characteristic sections: "0" – entrance to the HST, "2" – entrance to the narrowing part of the nozzle, "c" – exit from the nozzle.

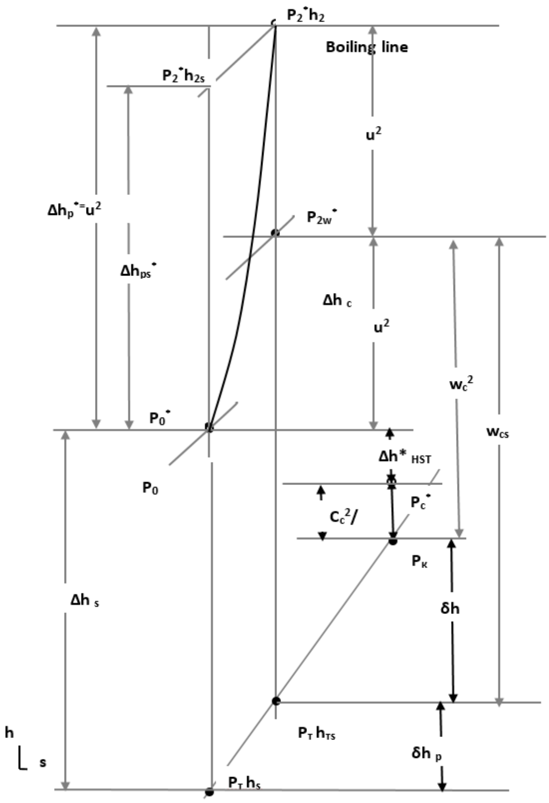


Fig. 4. HST process in an h-s chart

For simplicity, we consider that:

- water velocity at the nozzle inlet is small relative to the circumferential velocity w2<<u2;
- the velocity vector of the steam-water mixture exit from the nozzle coincides with the tangential direction.

Actual (useful) heat drop in HST is the work on the circumference of the wheel:

$$\Delta h_{HST}^{*} = \Delta h_{s\,HST} - \delta h_{pump} - \delta h_c - \frac{\overline{c}_e^2}{2} \tag{2}$$

here δh_{pump} – energy losses in the pumping part $\delta h_{pump} = u^2 \cdot (1 - \eta_{pump})$;

δh_c – energy losses in the nozzle $\delta h_c = \frac{w_{cs}^2}{2}\,(1 - \varphi_c)^2$;

$\frac{\overline{c}_e^2}{2}$ – losses with output velocity.

Hereinafter $\overline{c}_{ce} = \sqrt{\frac{\int_G c_c^{\,2} \cdot dG}{G}} = \sqrt{\int_1 c_c^{\,2} \cdot d\overline{G}}$ is velocity averaged by kinetic energy as opposed to velocity averaged by momentum $\overline{c}_c = \frac{\int_G c_c \cdot dG}{G} = \int_1 c_c \cdot d\overline{G}$, and the relative elementary flow rate $d\overline{G} = \frac{dG}{G}$.

For a stream with non-uniformly distributed velocity in the outlet section, the specific momentum of the reactive thrust:

$$\frac{P}{G} = \overline{c}_c = (\overline{w}_c - u);$$

where $\overline{c}_c$ is the integral specific momentum of the stream in the outlet section:

$$\overline{c}_c = \int c_c \cdot d\overline{G} = u + \int w_c \cdot d\overline{G}$$

The velocity averaged by the kinetic energy of the stream in the outlet section is:

$$\overline{c}_e = \sqrt{c_c^2 \cdot d\overline{G}} = \sqrt{\int (w_c - u)^2 \cdot d\overline{G}}$$

The output velocity values averaged by energy and by momentum coincide only in the case of a uniform velocity field. In all other cases, $\overline{c}_e > \overline{c}_c$. Moreover, the greater the non-uniformity, the higher the losses with the output velocity, and from this, at first glance, the less the work of the HST.

Yet if momentum does not depend on the non-uniformity of the output velocity, as was shown above (1), then neither thrust nor work depends on it. It would seem that there is a clear contradiction.

Now we shall return to the comparison of the performance of nozzles with a uniform (index "cu") and non-uniform velocity field (the current value is denoted by the index "i") at the outlet.

Let us represent the local velocity as the sum of the momentum-averaged velocity and the deviation from it. For an elementary stream in a nozzle with non-uniform and uniform velocity distribution, the following relations are valid:

$$\begin{aligned} w_i &= \overline{w}_c + \Delta w_i; \\ c_i &= \overline{c}_c + \Delta c_i; \\ w_i &= c_i + u; \\ \overline{w}_c &= \overline{c}_c + u; \\ \Delta c_i &= \Delta w_i \end{aligned} \qquad (3)$$

For a nozzle with non-uniform velocity distribution in absolute motion

– integral kinetic energy (index "e") of the flow:

$$\frac{c_e^2}{2} = \int_1 \frac{(\overline{c}_c + \Delta c_i)^2}{2} d\overline{G} = \frac{\overline{c}_c^2}{2} + \int_1 \left(\overline{c}_c \cdot \Delta c_i + \frac{\Delta c_i^2}{2} \right) d\overline{G}$$

– increase in kinetic energy associated with the unevenness of the velocity field:

$$\Delta\left(\frac{c_e^2}{2}\right) = \frac{c_e^2}{2} - \frac{\overline{c}_c^2}{2} = \int_1 \left(\overline{c}_c \cdot \Delta c_i + \frac{\Delta c_i^2}{2} \right) d\overline{G} = \overline{c}_c \cdot \int_1 \Delta c_i \cdot d\overline{G} + \int_1 \frac{\Delta c_i^2}{2} d\overline{G} \qquad (4)$$

Similarly, increase in the kinetic energy of the flow due to non-uniformity of the velocity field in relative motion:

$$\Delta\left(\frac{w_e^2}{2}\right) = \overline{w}_c \cdot \int_G \Delta w_i \cdot d\overline{G} + \int_G \frac{\Delta w_i^2}{2} d\overline{G} \qquad (5)$$

Comparing the change in kinetic energy for uniform and non-uniform velocity distributions in the absolute (3.1) and relative (3.2) coordinate systems, considering (3), we obtain:

$$\Delta w_i = w_i - w_{cu} = c_i + u - (c_{cu} + u) = c_i - c_{cu} = \Delta c_i$$

and note that:

$$\int_G \Delta w_i \cdot d\overline{G} = 0$$

The latter expression is valid since it was previously shown that the momentum of the reactive force for the flow with uniform and non-uniform velocity in the exit section of the nozzle is the same.

Thus, as follows from (4) and (5), the change of kinetic energy in the outlet section for a flow with non-uniform velocity, both in the relative and in the absolute coordinate system, is:

$$\delta K = \int_1 \frac{\Delta w_i^2}{2} d\overline{G} = \int_1 \frac{\Delta c_i^2}{2} d\overline{G}$$

The latter implies that increased energy losses with the exit velocity, caused by irregular flow velocities in the exit section, corresponds to an increase in the kinetic energy of the flow in relative motion by exactly the same value.
The energy-averaged velocity in relative motion will also be higher than the momentum-averaged velocity, and its corresponding velocity coefficient will be greater:

$$\varphi_{ce} = \frac{\sqrt{\int_1 w_i^2 \cdot d\overline{G}}}{w_s} > \varphi_c = \frac{\sqrt{\int_1 w_i \cdot d\overline{G}}}{w_s}$$

Fig. 5 presents an enthalpy-entropy (h-s) chart schematically depicting the workflow in the HST for variants with uniform speed (solid line) and non-uniform speed (dashed line).

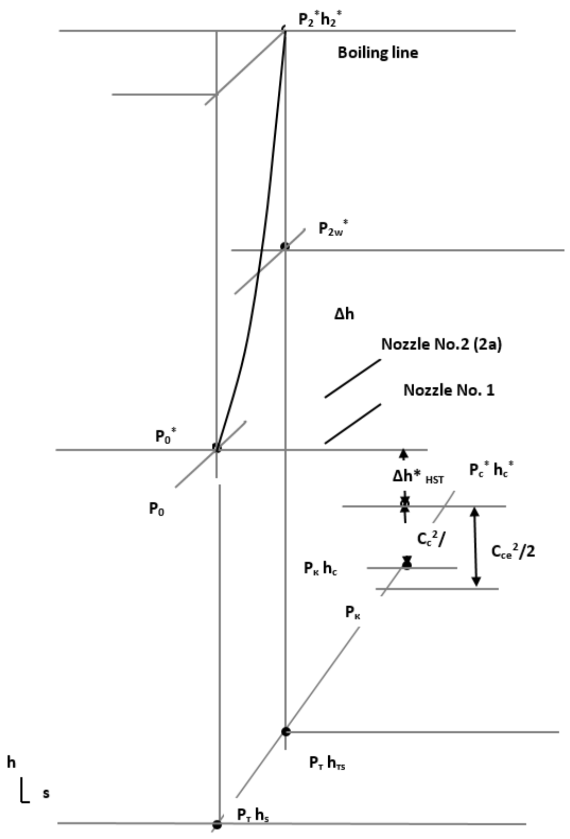


Fig. 5. Schematic diagram of the process in HST with a uniform velocity at the nozzle exit (solid line) and in the presence of phase slip (dashed line)

In this case, the work of the HST calculated from both the momentum-averaged velocity and the energy-averaged velocity remains identical.

3.2 Effect of Non-Equilibrium of Flow Process on the Energy Efficiency of Boiling Water Flow

To estimate the effect of the manifestation of non-equilibrium, we use the experimental data obtained in [4] (Fig. 5).

According to these data, the pressure along the movement in the expanding part of the nozzle decreases until it matches the pressure at the outlet. In the remaining part of the nozzle, pressure remains constant. It can be assumed that flow separation occurs in this specific place where the aforementioned pressures equalize. The part of the nozzle located downstream of the separation point is characterized by constant pressure on the limiting surfaces and according to the law of conservation of momentum does not create a change in the reactive force, if the frictional force is not considered. Thus, it can be argued that the measured reactive force *R* is created by the part of the nozzle located upstream of the separation point.

The attempt to identify the experimental data with the calculation of parameters in the section corresponding to the separation point by the homogeneous equilibrium flow model was unsuccessful: the steam-water mixture flow rate calculated from these data was only 45% of the flow rate calculated from the pressure drop in the "throat" section and close to that measured in the experiment [5]. This residual can be eliminated only if it is assumed that the vaporization process is delayed due to non-equilibrium and that the actual degree of dryness at the separation point is actually less. The calculated estimate of the degree of dryness with respect to the cross-section corresponding to the occurrence of separation is based on the following equations:
Impulse equation:

$$R = G \cdot w_c$$
$$w_c = x \cdot w'' + (1 - x) \cdot w' \qquad (6)$$

Continuity equation (assuming that there are no flowless zones at the separation point yet):

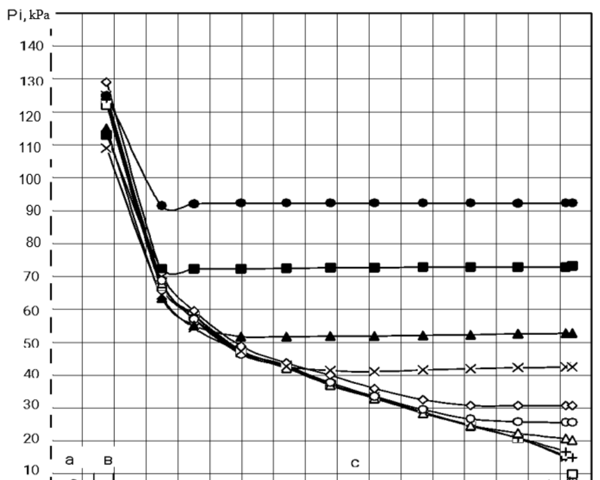
$$G = [(1 - x) \cdot \rho' \cdot w' \cdot f' + x \cdot \rho'' \cdot w'' \cdot f''] \qquad (7)$$

while the sum $f = f' + f''$ is known from the nozzle geometry. Energy conservation law, which in this case can be written as an equality of separation enthalpy, is calculated from the enthalpies of separate phases at the separation point and the enthalpy of separated water flow at the nozzle inlet:

$$h_i^* = h'^* \cdot (1-x) + xh''^* \cdot x = \left(h' + \frac{w'^2}{2}\right)(1-) + (h'' + \frac{w''^2}{2}) \quad (8)$$

The temperature and enthalpies of the liquid h' and vapor h'' phases can be determined from (8) if their temperatures are assumed to be the same. This temperature must be higher than the saturation temperature corresponding to outlet pressure since a part of the liquid phase is in a metastable state with a temperature close to the initial. The temperature of the vapor phase must also be between the initial and saturation temperature at outlet pressure, since a part of this phase formed under a pressure higher than the outlet, which means a higher temperature. Calculations show that a satisfactory residual of data from equations (6)÷(8) is obtained with the outlet temperature of the steam-water mixture close to the half-sum of the initial water temperature and the saturation temperature corresponding to the outlet pressure.

Approximate calculations are made for two back pressures: 30.7 kPa and 42.4 kPa (Fig. 6) by the method of selection of varying parameters, which in this case were the degree of dryness x and the slip ratio $K_w = \frac{w''}{w'}$. Here we considered that the velocity of the liquid phase should be higher than that of the water stream at the point of boiling and that the velocity of the vapor phase is higher than that of the liquid phase.



a – narrowing part, b – cylindrical throat, c – expanding part.
⋆ – P_2 = 6.8 kPa, □ – P_2 = 9.9 kPa, + – P_2 =14.9 kPa, Δ – P_2 = 20.2 kPa, ○ – P_2 = 25.6 kPa, ◊ – P_2 = 30.7 kPa, · – P_2 = 42.4 kPa, ▲ – P_2 = 52.7 kPa, ■ – P_2 = 73.7 kPa, ● – P_2 = 92.3 kPa.

Fig. 6. Experimental data [4] on pressure distribution along the nozzle length at different back pressures at the outlet – Effect of backpressure on pressure distribution along the nozzle No. 1 (P_0=3.1MPa, T_0=100°C)

The results of the calculations are presented in Table 1.

Table 1. Results of identification of calculation and experimental data

Parameter name, dimensionality	Mode No. 1	Mode No. 2
Temperature at the nozzle inlet, °C	100	
Water pressure at the nozzle inlet, MPa	3.1	
Water enthalpy at the nozzle inlet, kJ/kg	421.35	
Saturation pressure, kPa	100.67	
Pressure at the separation point, kPa	42.4	30.7
Water flow rate at the narrowing part of the nozzle (to the "throat") relative to velocity in the "throat"	0.96	0.96
Water velocity in the nozzle throat, m/s	75.92	
Nozzle diameter, mm	5	
Water flow rate by pressure drop and in the "throat", kg/s	1.43	
Parameters of the equilibrium process of homogeneous mixture at the separation point		
Isoentropic expansion enthalpy, kJ/kg	415.24	412.88
Nozzle velocity coefficient prior to separation point, determined by reactive thrust	0.7	0.7
Enthalpy of the homogeneous mixture, kJ/kg	418.36	417.20
Flow rate at the separation point in an equilibrium process, m/s	77.37	91.12
Temperature of steam-water mixture at the separation point in an equilibrium process, °C	77.26	69.62
Degree of dryness at the separation point in an equilibrium process	0.041	0.054
Steam-water mixture flow rate by equilibrium process parameters and cross-section area at the separation point	0.63	0.82
Residual at the nozzle and separation point, %	-56	-42
Parameters of the non-equilibrium process at the separation point		
Degree of dryness	0.020	0.033
Ratio of vapor phase velocity to liquid phase velocity – slip ratio	1.17	1.1
Liquid phase velocity, m/s	77.07	90.82
Liquid phase temperature, °C	88.79	81.56
Liquid phase enthalpy, kJ/kg	371.89	341.51
Liquid phase flow rate	1.40	1.38
Relative area of the cross-section of liquid phase passage, % of the sum	1.48	0.63
Absolute area of the cross-section of liquid phase passage	$1.88 \cdot 10^{-5}$	$1.57 \cdot 10^{-5}$
Enthalpy of deceleration of the liquid phase	374.85	345.64
Vapor phase velocity, m/s	90.5	100
Vapor phase enthalpy, kJ/kg	2,661.33	2,648.96
Vapor phase temperature, °C	88.63	81.56
Vapor phase flow rate	0.029	0.047
Enthalpy of deceleration of the vapor phase, kJ/kg	2,665.25	2,666.56
Relative area of the cross-section of vapor phase passage, %	98.52	99.37
Absolute area of the cross-section of vapor phase passage, m^2	0.00125	0.00143
Sum parameters of the two-phase flow		
Velocity averaged over the impulse, m/s	77.34	91.12

Enthalpy of deceleration, kJ/kg	421.37	421.31
Total cross-section area for the passage of the vapor and liquid phases	0.001269	0.00248
Cross-section area at the separation point according to the nozzle geometry	0.0127	0.00250
Steam-water mixture flow rate	1.43	1.43
Residual %:		
– by section area	-0.1	-0.3
– by impulse	-0.04	0
– by energy	0	0
Decrease in the degree of dryness in a non-equilibrium process, $dx=x_{eq}-x_{non\text{-}eq}$	0.021	0.021

Analysis of data from Table 1 demonstrates that the vapor phase flow rate at the cross-section corresponding to the point of separation is lower than the calculated flow rate based on the parameters of an equilibrium process. This indicates an additional loss of reactive force momentum due to the occurrence of disequilibrium. Measures to reduce the non-equilibrium of the boiling process are one possible way to improve the characteristics of nozzles.

The area of the cross-section at the separation point is at least 98.5% occupied by the vapor phase, despite its small mass fraction. This fact predetermines a preliminary conclusion that the factual dryness of vapor in this section is much lower than the calculated one determined under the assumption of the equilibrium nature of the outflow process. For the studied nozzle, this difference was 0.021 at both modes.

The obtained results need to be confirmed and clarified through additional research, yet already at this stage, they can be used to determine the optimal size of the nozzle outlet cross-section when designing HSTs.

4. Conclusion

1. The process of outflow of boiling water from HST nozzles is significantly different from an equilibrium process, which manifests in reduced vapor dryness and additional loss of momentum. Measures to decrease the degree of non-equilibrium, such as stimulating the fragmentation of the stream into small droplets, may be useful in intensifying the vapor fraction separation and, thus, capable of improving the characteristics of the nozzles.

2. Striving to reduce liquid and vapor phase slip should not be an end in itself, since this alone does not yield a gain in the reactive thrust generated by the boiling liquid.

3. The area of the nozzle outlet cross-section required to achieve maximum efficiency at the design mode may greatly depend on the difference between outlet velocities of the liquid and vapor phases, which needs to be considered in nozzle profiling.

The article uses the results achieved in the implementation of research and development work with the use of state support measures for the development of cooperation between Russian higher education institutions, state scientific institutions, and organizations of the real sector of the economy that implement complex projects to create high-tech production, as stipulated by the decree of the Government of the Russian Federation of April 9, 2010, No. 218 Agreement with the Ministry of Education and Science of the Russian Federation No. 075-11-2022-031 of October 7, 2022 on the topic "Creating high-tech production of reaction hydro-steam turbines for renewable energy sources in heating plants".

References

[1] Volkov, A. V., Ryzhenkov, A. V., Vikhlyantsev, A. A., Druzhinin, A. A., Cherepanov, S. P., Šoukal, J., Sedlař, M., Komárek, M., Pochylý, F., Rudolf, P., Fialová, S.: *The influence of wetting of flow passage surfaces in pumps-as-turbines on their energetic characteristics*. International Journal of Engineering Trends and Technology 68(11), 16–24 (2020).

[2] Schrock, V. E., Starkman, E. S., Brown, R. A.: *Vskipanie nedogretoi vody pri istechenii cherez sopla Lavalia* [Flashing flow of initially subcooled water in convergent-divergent nozzles]. Teploperedacha 99(2), 113–120 (1977).

[3] Korotkiy, I. A., Neverov, E. N., Korotkaya, E. V.: *New cryoscopic temperature determination technology for plant-based products*. Nexo Revista Científica 35(03), 771–776 (2022) https://doi.org/10.5377/nexo.v35i03.15006

[4] Goldin, A. S.: *Issledovanie sopl gidroparovoi turbiny pri istechenii zhidkosti s bolshim nedogrevom do temperatury nasyshcheniia* [A study of hydro-steam turbine nozzles at the outflow of liquid with a large underheating to the saturation temperature]. Candidate degree dissertation in technical sciences, Kaluga State Pedagogical University named after K.E. Tsiolkovsky, Kaluga, pp. 147 (2004).

[5] Avdeev, A. A., Maidannik, V. N., Shanin, V. K.: *Metodika rascheta vskipaiushchikh adiabatnykh potokov* [Calculation methodology for adiabatic boiling flows]. Teploenergetika 8, 67–69 (1977).