Experimental Research of the Excessive Water Injection Effect on Resistances in the Flow Part of a Low-Flow Aerothermopressor

Dmytro Konovalov[1(✉)], Halina Kobalava[1], Vitaliy Maksymov[1], Roman Radchenko[1], and Mykola Avdeev[2]

[1] Admiral Makarov National University of Shipbuilding, 9, Heroes of Ukraine Avenue, Mykolayiv 54025, Ukraine
dimitriyko79@gmail.com

[2] Kherson Shipbuilding Company, 32b Torhova Street, Kherson 73009, Ukraine

Abstract. Water injection to the compressor channel is one of the effective ways to increase the power and efficiency of gas turbine plants. A promising method of water spraying is to use a jet apparatus – an aerothermopressor. Experimental studies of the excessive water injection effect on resistances in the flow part of a low-flow aerothermopressor are presented in this paper. To conduct an experimental study, an experimental setup was developed. An analysis of the obtained experimental data was carried out. A decrease in pressure losses by 15–20% relative to pressure losses in a "dry" aerothermopressor is stated. Checking the calculated equation for adequacy with experimental data is shown in a discrepancy in a range from 40% to −20%. An empirical equation is obtained to determine the pressure losses for the low-flow aerothermopressor (checking the calculated empirical equation for adequacy with experimental data is shown a discrepancy in a range from +15% to −15%). It was found that the pressure loss becomes equal to or exceeds the losses for the dry aerothermopressor when the flow rate water amounts more than 0.2 (20%).

Keywords: Thermogasdynamic compression · Injection · Relative pressure losses

1 Introduction

Complex schemes with cyclic air cooling are usually used in modern gas turbine plants (GTP) to increase fuel and energy efficiency. They are used to bring the process of compressing the working fluid in gas turbine compressors closer to isothermal. It leads to an increase in the thermal efficiency of the cycle and a decrease in specific fuel consumption [1–3].

One of the promising areas is the contact cooling of cyclic air by using a two-phase jet apparatus – an aerothermopressor. This device provides efficient evaporative cooling of the gas turbine without loss of total pressure. Pressure losses due to the friction are reduced with the rational organization of work processes and the corresponding development of the apparatus flow part design. By increasing the total air pressure at the outlet, the efficiency of the aerothermopressor is increased up to 30% [4, 5].

V. Ivanov et al. (Eds.): DSMIE 2020, LNME, pp. 292–301, 2020.
https://doi.org/10.1007/978-3-030-50491-5_28

2 Literature Review

One of the most common ways to increase the efficiency of gas turbines is to use the cyclic air intercooling in the compression process. Thus, the compressor efficiency increase is achieved by approximating the final compression temperature to the initial temperature (the compression process becomes close to isothermal) [3, 6]. The water injection to the compressor channel is one of the effective ways to increase the power and efficiency of gas turbine plants [3, 7, 8]. The water injection fulfills two functions: it removes the heat from the airflow and returns the heat to the gas turbine cycle.

One of the current directions in increasing the efficiency of power plants is to use the aerothermopressor as a means of efficient water spraying while increasing the mass flow rate of the working fluid in the gas turbine plant [5]. Due to evaporative cooling in the aerothermopressor, an effect of the thermodynamic compression takes place – gas pressure is increased in the process of instantaneous evaporation of water injected into the gas (air) flow accelerated to a speed close to sound [9]. This apparatus combines two working processes: contact cooling of cyclic air and cyclic air pressure increase [4, 9, 10]. The difference of this cooling cyclic air method is that due to the intensive heat and mass transfer processes and a high flow rate, sufficient liquid thin atomization occurs. When injecting more water than is necessary for a complete evaporation, an additional evaporation of the remaining droplets in the airflow compressed in the gas turbine compressor is done. Due to this, the necessary conditions are created to ensure the process of an isothermal compression.

A significant influence on the working processes in the aerothermopressor is carried out by design factors. These factors affect energy consumption to overcome the friction forces and local resistance in the aerothermopressor flow part. To ensure a positive increase in pressure, the gas velocity in the aerothermopressor evaporation chamber should be near the speed of sound. The relative velocity, in this case, is in the range M = 0.5–0.9 Mach number. However, at such gas velocities, the aerodynamic resistance of the aerothermopressor [4, 10] is significantly increased.

The positive effect of using the aerothermopressor is greater the smaller the pressure loss through aerodynamic resistance [5] is. One of the ways to increase the efficiency of the aerothermopressor (relative increase in pressure) is to inject water into the flow part in an amount greater than necessary. This method ensures a dispersed mode of flow along the entire length of the aerothermopressor [11]. To determine the aerodynamic resistance losses in the low-flow aerothermopressor, it is necessary to clarify the methodology for determining the total aerodynamic resistance.

3 Research Methodology

The main objective of the research is to study the effect of the water injection (in excess of the amount necessary for evaporation) into the aerothermopressor flow part on the pressure losses due to aerodynamic resistance and to determine the dependences for calculating such losses.

The calculation model developed by the authors was used considering the calculation features: of gas turbine cycles with the cyclic air intercooler; of the thermodynamic compression effect; of contact cooling processes of moist air. Based on the computational model, a software complex has been developed. It allows determining parameters at the characteristic points of the cycle, GTP efficiency parameters (efficiency, power, specific fuel consumption, etc.), working process parameters in the aerothermopressor, and its geometric characteristics.

The calculation of GTP cycles was carried out according to well-known methods [6, 12] considering the features of processes in moist air [13]. The effect of thermodynamic compression was calculated considering the dependences [4, 9, 10]. The calculation of the evaporation of finely dispersed water was carried out considering contact heat transfer in gas turbine and combined-cycle plants [8, 14, 15].

The reliability of the work results is ensured by the tasks correct formulation of the theoretical and experimental research, the confirmation of the adequacy of the mathematical model with satisfactory agreement between the calculated and experimental data, using modern methods of experimental study and analytical modeling.

To conduct an experimental study, an experimental setup was developed (Fig. 1). The experimental setup is designed to simulate the aerothermopressor operation in the conditions of GTP cyclic air cooling.

The experimental aerothermopressor consists of the following elements: a receiving chamber with a nozzle for injecting water into the flow; a confuser; an evaporation chamber; a diffuser; nozzles for installing temperature and pressure sensors. All elements of the aerothermopressor are removable allowing carrying out studies for the different geometric characteristics: a confuser – inlet diameter D_{c1} = 65 mm; a convergent angle $\alpha = 30°$; an evaporation chamber – diameter D_{ch} = 25 mm; length L_{ch} = 125; 175 mm; a diffuser – length L_{ch} = 192 mm; a divergence angle $\beta = 6°$; a nozzle – the distance between the water exit point to the receiving chamber inlet L_f = 5; 65; 125 mm.

In the course of the study, the airflow rate was 6.0 m^3/h, at the pressure of 0.32 MPa, with the velocity in the evaporation chamber of 0.4–0.9 M.

All temperature, pressure, and airflow sensors were connected to a developed computerized monitoring system. The parameters were measured with an interval $\tau = 1$ s. To record the readings of measuring instruments, eight-channel meters I8-TS (temperature measurement) and I8-AT (pressure measurement) were used by RegMik. To collect and organize the information about the data, a PI485/USB RS485, a communication interface converter was used.

The error of the experimental results was determined by the error of the measuring instruments, methodological, and systematic errors.

The gas expansion process occurs adiabatically in the aerothermopressor subsonic nozzle. A liquid is supplied to the receiving chamber located up to the confuser, which is mechanically finely atomized with a special nozzle. In the evaporation section, these flows interact, and, as a result, droplets are accelerated, crushed, heated, and evaporated, and gas is also cooled [4]. There are three main operating modes of the aerothermopressor [4]:

1) the influence of the water droplets resistance prevails over the positive effect of evaporation and determines the gas flow behavior. The Mach number is increased, the static pressure of the flow is decreased, and the water temperature is risen approaching the saturation temperature;
2) the water evaporation is predominated. The Mach number is decreased, and the total and static pressure is increased;
3) the surface friction (it was relatively insignificant in the first two modes) becomes the predominant factor. This mode takes place when the liquid is evaporated completely.

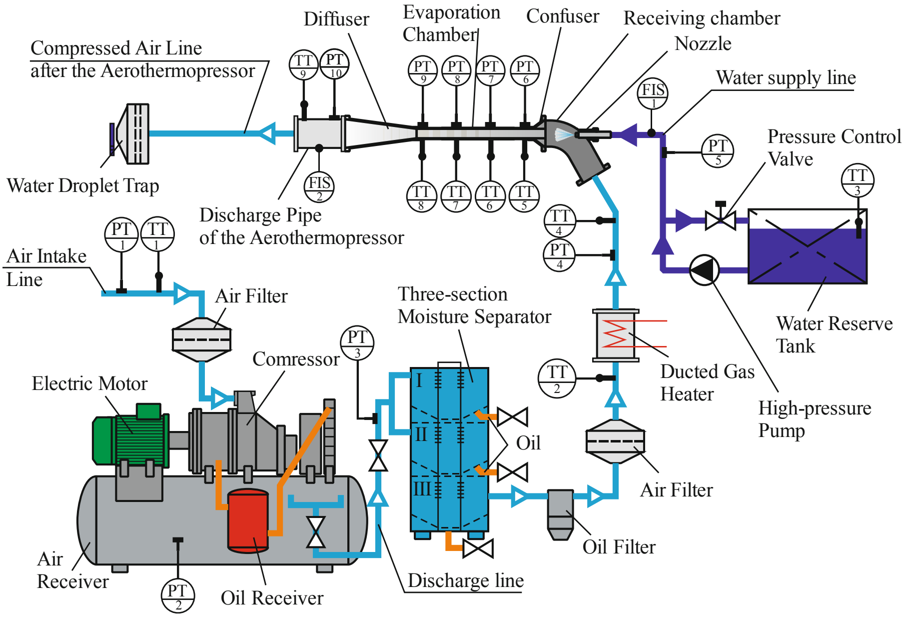

Fig. 1. The scheme of the experimental setup.

This mode takes place when the liquid evaporates completely. When providing the injection of water in an amount more than necessary for the evaporation, the third mode will be absent. This can positively affect the increase in the total pressure of the flow as a result of the thermodynamic effect.

Pressure losses due to the resistance (acceleration or deceleration of the droplet, depending on the initial velocity at the nozzle exit) can be determined by the aerodynamic resistance coefficient of the droplet in the airflow [8, 15]:

$$\zeta_d = \left(\frac{16}{Re_d} + \frac{2.2}{Re_d^{0.5}} + 0.32\right) \cdot \left(\frac{1.5\mu_w + \mu_{air}}{\mu_w + \mu_{air}}\right), \quad (1)$$

where μ_{air}, μ_w – dynamic viscosity of air and water.

Reynolds number for a drop moving in the airflow:

$$Re_{\mathrm{d}} = \frac{\delta_{\mathrm{w}} \Delta w_{\mathrm{w}} \rho_{\mathrm{air}}}{\mu_{\mathrm{air}}}, \tag{2}$$

where Δw_{w} – droplet velocity relative to air velocity, δ_{w} – water droplet diameter.

The local resistance coefficients also determine pressure losses due to aerodynamic resistance in the aerothermopressor flow part: the confuser (nozzle) – ζ_{c}; the working chamber (evaporation chamber) – ζ_{ch}; the diffuser – ζ_{dif}.

For a dispersed flow, an equation can be given to determine the hydrodynamic pressure loss. It is used in the phase slip model [17]. The mass flow rates occurring in the evaporation chamber of the low-flow aerothermopressor (with the diameter of the evaporation chamber $D_{\mathrm{ch}} < 30$ mm, $G_{\mathrm{g}} < 1.5$ kg/s) is (ρw) > 100 [10].

The hydraulic resistance of two-phase flow with phase slip is calculated by the Lockhart-Martinelli method [11, 16]. According to this method, the pressure loss due to the friction of a two-phase flow dP/dZ can be expressed in terms of their value for one phase:

$$\frac{dP}{dZ} = \Phi_{\mathrm{w}}^2 \left(\frac{dP}{dZ}\right)_{\mathrm{w}} \text{ or } \frac{dP}{dZ} = \Phi_{\mathrm{air}}^2 \left(\frac{dP}{dZ}\right)_{\mathrm{air}}, \tag{3}$$

where Martinelli parameter Φ [17]:

$$\Phi_{\mathrm{w}} = \frac{\Phi_{\mathrm{air}}}{X_{\mathrm{tt}}} = X_{\mathrm{tt}}^{-1} + 2.85 \cdot X_{\mathrm{tt}}^{-0.48}; \Phi_{\mathrm{air}} = 1 + 2.85 \cdot X_{\mathrm{tt}}^{0.523}, \tag{4}$$

Martinelli-Nelson parameter X_{tt} [18]:

$$X_{\mathrm{tt}} = \left(\frac{\mu_{\mathrm{w}}}{\mu_{\mathrm{air}}}\right)^{0{,}1} \left(\frac{1-x}{x}\right)^{0{,}9} \left(\frac{\rho_{\mathrm{air}}}{\rho_{\mathrm{w}}}\right)^{0{,}5}, \tag{5}$$

where $x = G_{\mathrm{air}}/(G_{\mathrm{air}} + G_{\mathrm{w}})$ – dryness of the vapor.

Pressure friction losses of the water phase and the gas phase:

$$\left(\frac{dP}{dZ}\right)_{\mathrm{w}} = \frac{\zeta_{\mathrm{ch.w}} \cdot (\rho_{\mathrm{air}} w_{\mathrm{air}})^2 (1-x)^2}{2 D_{\mathrm{ch}} \rho_{\mathrm{w}}} \text{ or } \left(\frac{dP}{dZ}\right)_{\mathrm{air}} = \frac{\zeta_{\mathrm{ch.air}} \cdot (\rho_{\mathrm{air}} w_{\mathrm{air}})^2 (1-x)^2}{2 D_{\mathrm{ch}} \rho_{\mathrm{air}}}. \tag{6}$$

The friction coefficient for turbulent flow in the evaporation chamber is found by the Blasius equation [19]:

$$\zeta_{\mathrm{ch}} = 0.3164 \cdot Re^{-0.25}. \tag{7}$$

The phase with the determining pressure drop must be taken as the one whose pressure gradient is more significant importance for a given vapor content [11], while the total pressure loss is:

$$\Delta P_{\mathrm{ch}} = \bar{l}_{\mathrm{ch}} \cdot \frac{\Phi_{\mathrm{w}}^2 \cdot \zeta_{\mathrm{ch \cdot w}} \cdot (\rho_{\mathrm{air}} \cdot w_{\mathrm{air}})^2 \cdot (1-x)}{2 \cdot \rho_{\mathrm{w}}}, \tag{8}$$

$$\Delta P_{\mathrm{ch}} = \bar{l}_{\mathrm{ch}} \cdot \frac{\Phi_{\mathrm{air}}^2 \cdot \zeta_{\mathrm{ch \cdot air}} \cdot (\rho_{\mathrm{air}} \cdot w_{\mathrm{air}})^2 \cdot (1-x)}{2 \cdot \rho_{\mathrm{air}}}, \tag{9}$$

where $l_{\mathrm{ch}} = (L_{\mathrm{ch}}/D_{\mathrm{ch}})$ – relative length of the evaporation chamber.

To calculate losses from the total resistance for the confuser ζ_{c} and the diffuser ζ_{dif} using the equation for the local loss coefficient [19]. For aerothermopressor calculated values: ζ_{c} = 0.005–0.010; ζ_{dif} = 0.01–0.20.

Thus, combining Eqs. (1), (7)–(9) to determine the pressure losses in the aerothermopressor flow part:

$$\Delta P_{\mathrm{tr}} = \Delta P_{\mathrm{ch}} + \frac{(\zeta_{\mathrm{d}}(g_{\mathrm{w}} + \Delta g_{\mathrm{w}}) + \zeta_{\mathrm{c}} + \zeta_{\mathrm{dif}}) \cdot (a_{\mathrm{df}} \cdot M)^2 \rho_{\mathrm{air}}}{2}, \tag{10}$$

where a_{df} – sound speed in the two-phase flow; g_{w} – amount of water required for complete evaporation; Δg_{w} – excess water.

Relative pressure losses in the aerothermopressor flow part:

$$\delta_{\mathrm{tr}} = \frac{\Delta P_{\mathrm{tr}}}{P_1} = \frac{P_1 - P_2}{P_1}, \tag{11}$$

where P_1, P_2 – aerothermopressor inlet and outlet pressure.

4 Results

The experimental determination of pressure losses in the aerothermopressor flow part was carried out for a number of mass flow rates of water and it was compared with the values of pressure loss without water injection (dry aerothermopressor). The dependences plotted from experimental points are shown in Fig. 2.

The maximum pressure losses are made for the aerothermopressor without the water injection: at the inlet pressure P_1 = 150–300 kPa – ΔP_{tr} = 20–70 kPa (δ_{tr} = 10–27%). At the water injection of G_{w} = 0.0175 kg/s (8%), pressure losses is decreased – ΔP_{tr} = 20–55 kPa (δ_{tr} = 6–22%), but with an increase in the amount of the injected water, the pressure losses is increased: at G_{w} = 0.0407 kg/s (15%) – ΔP_{tr} = 20–60 kPa (δ_{tr} = 7–24%), at G_{w} = 0.0487 kg/s (17%) – ΔP_{tr} = 20–65 kPa (δ_{tr} = 9–26%). This is explained by the fact that, with a gradual increase in the water flow rate, the pressure losses also increase due to the droplets resistance in the flow. When the flow rate amounts are $g_{\mathrm{w}} > 0.2$ (20%), the pressure loss becomes equal to or exceeds the losses for the dry aerothermopressor. A comparison of the experimental and calculated dependences (Fig. 3) shows that the calculated line of pressure loss at pressures P_1 = 250–300 kPa lies below the line constructed from the experimental data. Moreover, the calculated data are 8–12 kPa less (10–20%).

Equation checking (10) for adequacy with experimental data on the pressure loss ΔP_{tr} shows a discrepancy from + 40% to −20% (Fig. 4), which can be considered satisfactory. However, the discrepancy is quite large, and it might explain the difficulty in determining the initial velocity of the droplet, and therefore underestimated pressure losses due to the droplet's aerodynamic resistance in the airflow.

To obtain more accurate values of the relative pressure losses δ_{tr} in the flow part of the low-flow aerothermopressor, it was proposed to obtain an empirical equation depending on the relative water flow rate g_w and the inlet air pressure P_1. The equation was determined by the approximation method:

$$\delta_{tr} = 0.0457 - 7.3581 \cdot 10^{-8} \cdot P_1 + 0.4682 \cdot g_w + 1.7814 \cdot 10^{-12} \cdot P_1^2 - 1.0768 \cdot g_w^2. \tag{12}$$

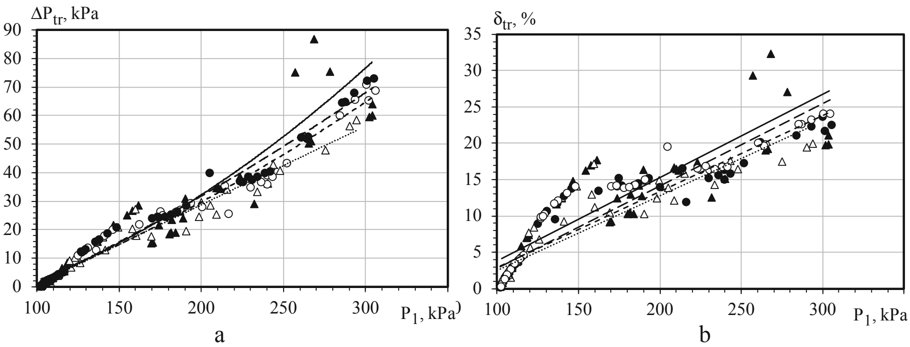

Fig. 2. The experimental data on the measurement of pressure losses ΔP_{tr} (a) and relative pressure losses δ_{tr} (b) in the aerothermopressor flow part on the air pressure at the inlet P_1 at various injected water rates G_w: ▲; _______ – 0 kg/s; Δ; .... – 0.0175 kg/s; ○; _ _ _ _ _ _ – 0.0407 kg/s; ●; __ __ __ __ – 0.0487 kg/s.

This equation (regression coefficient R = 0.9260) was obtained for the following characteristics of the aerothermopressor: G_{air} = 0.10–0.52 kg/s; M = 0.4–0.8; P_1 = 125–300 kPa; g_w = 0–0.4; t_1 = 50–180 °C. The deviation of the calculated $\delta_{tr.C}$ values from the experimental $\delta_{tr.E}$ is ±15% (Fig. 5), which is more accurate. However, such equation does not make it possible to calculate aerothermopressors with high airflow.

Thus, at the injected water in an amount greater than necessary for evaporation, the pressure losses in the aerothermopressor is decreased by 15–20% relative to the losses in the dry aerothermopressor. This makes it possible to increase the relative increase in the total pressure in the aerothermopressor $(P_2/P_1)_{atp}$. Thus, when simulating the aerothermopressor operation to cooling cyclic air of the gas turbine WR-21 (Fig. 6) from Rolls Royce (N_e = 25250 kW, g_e = 0.190 kg/(kW•h), η_e = 41.2%) without the injection with the excess water – $(P_2/P_1)_{atp}$ = 1.025–1.060; the injection of the excess water g_w = 0.1 (10%) – $(P_2/P_1)_{atp}$ = 1,045–0,080, so the total pressure is increased by 2%.

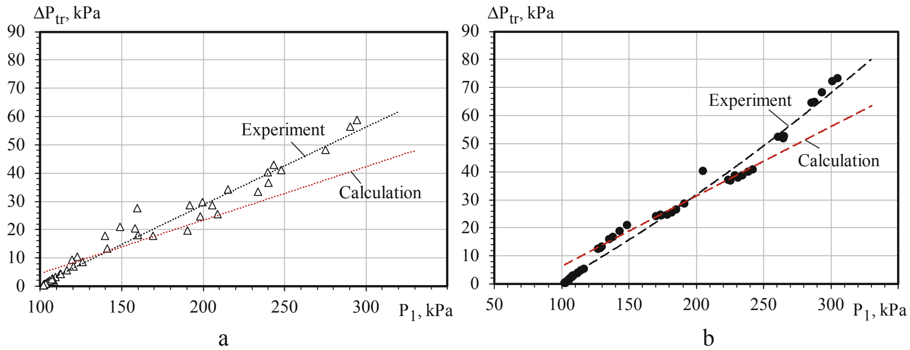

Fig. 3. Dependences of experimental and calculated data on the measurement of pressure losses ΔP_{tr} in the aerothermopressor on the air pressure at the inlet P_1 at various injected water rates G_w: a – 0.0175 kg/s; b – 0.0487 kg/s.

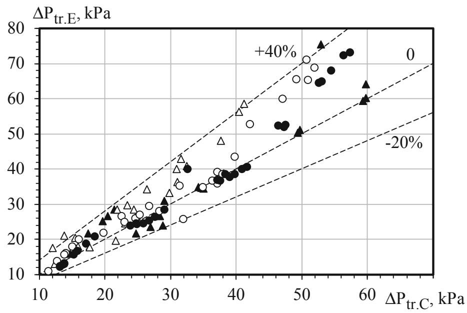

Fig. 4. Comparison of the experimental data on the measurement of pressure losses $\Delta P_{tr.E}$ in the aerothermopressor flow part with the calculated data $\Delta P_{tr.C}$ (Eq. (10)) at various injected water rates G_w: ▲ – 0 kg/s; Δ – 0.0175 kg/s; ○ – 0.0407 kg/s; ● – 0.0487 kg/s.

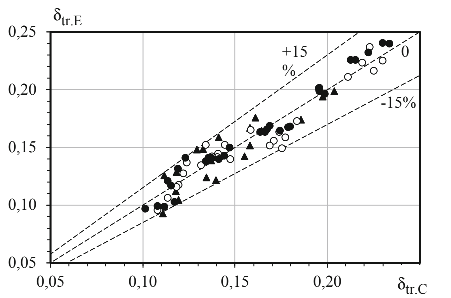

Fig. 5. Comparison of the experimental data on the relative pressure losses in the aerothermopressor flow part $\delta_{tr.E}$ with the calculated data $\delta_{tr.C}$ (Eq. (12)) at various injected water rates G_w: ▲ – 0 kg/s; Δ – 0.0175 kg/s; ○ – 0.0407 kg/s; ● – 0.0487 kg/s.

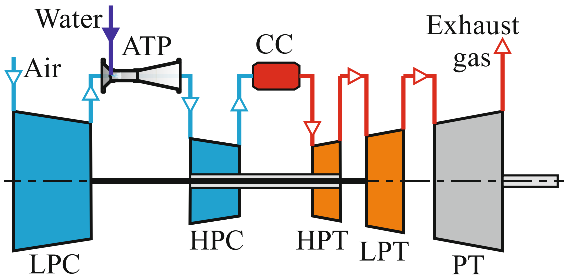

Fig. 6. The scheme of the gas turbine with cyclic air cooling by the aerothermopressor: ATP – aerothermopressor; LPC, HPC – low and high pressure compressors; CC – combustion chamber; HPT, LPT – low and high-pressure turbines; PT – power turbine.

5 Conclusions

An analysis of the obtained experimental data on pressure losses in the aerothermopressor flow part shows that the water injection in an amount g_w in excess of necessary for the evaporation provides a decrease in these losses by 15–20% relative to pressure losses in a dry aerothermopressor. A calculation equation to determine the pressure losses in the aerothermopressor flow part is presented; it takes into account the influence of the droplet aerodynamic resistance and the hydrodynamic resistance of the two-phase flow. Checking the calculated equation for adequacy with experimental data shows a discrepancy in a range from 40% to –20%.

An empirical equation is obtained to determine the pressure losses for the low-flow aerothermopressor at the airflow rate of G_{air} = 0.10–0.52 kg/s. Checking the calculated equation for adequacy with experimental data shows a discrepancy of ±15%.

The application of excessive water injection into the aerothermopressor to cooling cyclic air of the gas turbine allows increasing the relative pressure $(P_2/P_1)_{atp}$ in the aerothermopressor by 2%.

The study results can be used in the methodology of aerothermopressors rational design to determine the optimal geometric characteristics of the apparatus flow part. The finely dispersed two-phase flow obtained in the aerothermopressor can be further vaporized to the high-pressure gas turbine compressor. This fact will ensure that the compression process is close to isothermal. The application of the principle of the excess water injection in the aerothermopressor is relevant for using at the intercooling of cyclic air of gas turbine plants and the charge air cooling of internal combustion engines.

References

1. Romanovskyy, H.F., Vashchylenko, M.V., Serbin, S.I.: Theoretical Bases of the Ship Gas-Turbine Aggregates Designing. UDMTU, Mykolayiv (2003)
2. Romanovskyy, H.F., Serbin, S.I., Patlaychuk, V.M.: Modern Gas Turbine Units. NUOS, Mykolayiv (2005)

3. Jonsson, M., Yan, J.: Humidified gas turbines – a review of proposed and implemented cycles. Energy **30**, 1013–1078 (2005)
4. Stepanov, I.R., Chudinov, V.I.: Some Problems of Gas and Liquid Motion in the Channels and Pipelines of Power Plants. The Science Publisher, Leningrad (1977)
5. Konovalov, D., Kobalava, H.: Efficiency analysis of gas turbine plant cycles with water injection by the aerothermopressor. In: Ivanov, V., et al. (eds.) Advances in Design, Simulation and Manufacturing II. DSMIE 2019. LNME, pp. 581–591. Springer, Cham (2020)
6. Radchenko, A., Radchenko, M., Konovalov, A., Zubarev, A.: Increasing electrical power output and fuel efficiency of gas engines in integrated energy system by absorption chiller scavenge air cooling on the base of monitoring data treatment. In: HTRSE-2018, 6 p. E3S Web of Conferences 70, 03011 (2018)
7. Chaker, M.A., Meher-Homji, C.B., Mee III, T.: Inlet fogging of gas turbine engines: experimental and analytical investigations on impaction pin fog nozzle behavior. J. Eng. Gas Turbines Power **128**(4), 826–839 (2003)
8. Dykyy, M.O., Solomakha, A.S.: Features of the contact cooling process of cyclic air in gas turbine plants and the way of its realization. Energy: Econ. Technol. Ecol. **1**, 22–27 (2012)
9. Vulis, L.A.: Gas Flow Thermodynamics. Gosenergoizdat, Moscow (1950)
10. Fowle, A.: An Experimental Investigation of an Aerothermopressor Having a Gas Flow Capacity of 25 Pounds Per Second. Massachusetts Institute of Technology, Cambridge (1972)
11. Radchenko, N.: A concept of the design and operation of heat exchangers with change of phase. Arch. Thermodyn.: Polish Acad. Sci. **4**(25), 3–19 (2004)
12. Perelshtein, B.H.: To the entropy analysis of the thermal resistance in the combustion chamber of the gas turbine engine and the “aerothermopression effect”. Aerosp. Tech. **1**, 77–80 (2003)
13. Wang, S.K.: Handbook of Air Conditioning and Refrigeration, 2nd edn. McGraw-Hill Publisher, New York (2000)
14. Dykyy, M.O.: Marine Gas and Steam Turbine Plants. Shipbuilding, Leningrad (1978)
15. Dolinsky, A.A., Ivanitsky, G.K.: Heat and Mass Transfer and Hydrodynamics in Vapor-Liquid Dispersed Flow. Naukova Dumka, Kyiv (2008)
16. Lockhart, R.W., Martinelli, R.C.: Proposed correlation of data for isothermal two-phase, two-component flow in pipes. Chem. Eng. Prog. **45**(1), 39–48 (1949)
17. Soliman, M., Schuster, J.R., Berenson, P.J.: A general heat transfer correlation for annular flow condensation. Trans. ASME: Ser. C **90**(2), 267–276 (1968)
18. Martinelli, R.C., Nelson, D.B.: Prediction of pressure drop during forced – circulation boiling of water. Trans. ASME **70**, 695–702 (1948)
19. Idelchik, I.E.: Handbook of Hydraulic Resistances. Mashinostroenie Publisher, Moscow (1992)