Numerical Simulation of an Aerothermopressor with Incomplete Evaporation for Intercooling of the Gas Turbine Engine

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Abstract. Complex cycles with cyclic air intercooling are used to increase the energy efficiency of gas turbines. A modern and widespread way to improve the cooling process is to humidify the working fluid (cyclic air). The efficiency of wet compression primarily depends on the intensity of evaporation and heat exchange of droplets with the air flow, which begins to increase sharply when the effective diameter of droplet spraying decreases to 20 µm. It is proposed to use a contact heat exchanger to obtain a finely dispersed flow of water in the flow path of a gas turbine. The operation of such contact heat exchanger called aerothermopressor was investigated in this paper. CFD simulation of the water droplet evaporation process in the aerothermopressor airflow was carried out. Calculations were carried out for three variants of evaporation of water injected into the air flow: complete evaporation of water droplets in the evaporation chamber, additional evaporation of water droplets in the diffuser and incomplete evaporation, with obtaining smaller droplets at the outlet of the aerothermopressor diffuser. Efficiency of the aerothermopressor application in the gas turbine circuit for contact cooling of cyclic air is analyzed. It has been revealed that the aerothermopressor allows increasing the cyclic air pressure between the compressor stages by 2–10%, which will lead to a decrease in the compression work in the compressor stages and makes it possible to increase the gas turbine engine efficiency by 1–2%.

Keywords: Droplet diameter · Two-phase flow · Thermogasdynamic compression

1 Introduction

One of the most perspective ways to increase the power of modern power plants based on gas turbine engines (GTE) is to increase the flow rate of the working fluid (air) and the useful work of expanding combustion products (exhaust gases) simultaneously with a decrease in costs for compression in the compressor stage. This is achieved by

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cooling the working fluid (air) before or during the compression process, which reduces the specific fuel consumption. One of the most common ways to implement this principle is to use the contact air cooling with water injection. At the same time, there is a significant length of the evaporation section. This is not always constructively possible to implement and, if the workflow is ineffective, it can lead to droplets getting into the GTE compressors.

A promising development of the contact cooling principle in the gas turbine engine is to use the energy potential of the working fluid to accelerate the air flow to transonic speeds and for instant evaporation of water injected into the air flow, that is, the implementation of the thermogasdynamic compression effect.

2 Literature Review

A modern and widespread way to increase the efficiency of gas turbine plants is to humidify the working fluid (cyclic air). For this, the following technologies with water (steam) injection are used:

- air cooling at the compressor inlet [1–3];
- wet compression (contact air cooling during compression in a compressor stage) [2, 4];
- additional cooling at the compressor outlet with subsequent recuperation [5];
- utilization of the exhaust gases heat of the gas turbine plant (GTP) in the steam-turbine heat recovery circuit (gas-steam turbine technology "Vodoley") [6];
- steam injection (technology Steam Injected Gas Turbine (STIG), Cheng cycle) [2, 7];
- water injection into the air flow in the humidifying tower (Humidified Air Turbine (HAT) and Evaporative Gas Turbines (EvGT) technologies) [8, 9];
- superheated water injection [6].

To increase the energy efficiency of gas turbines, complex cycles with cyclic air intercooling in the units are used [2]. Intercooling can be carried out in the following way – contact cooling of the cyclic air with injection of the finely dispersed water flow, which was proposed in the 60s by Jones and Hawkins [10].

As a rule, water injection at the compressor inlet or between compressor stages is used to increase the power and efficiency of the gas turbine plant at high outside temperatures. Injection of steam generated in the heat recovery circuits increases the power and efficiency of the gas turbine plant by increasing the amount of the working fluid in the cycle [11, 12].

As a result of cyclic air intercooling during the compression process, the compressor efficiency is increased by isothermal air compression process (approaching the final compression temperature to the initial one) [2, 6, 12, 13]. Hence, without loss of the gas turbine performance, the gas temperature at the turbine inlet decreases. It favorably affects the gas turbine resource [14–16].

The efficiency of wet compression primarily depends on the intensity of evaporation and heat exchange of droplets with the air flow, which begins to increase sharply when the effective diameter of droplet spraying decreases to 20 μm. This occurs as a result of a significant increase in the total surface area of the droplets, which, in turn, is inversely proportional to its average diameter. At the same time, the separation of droplets of such a small size on the surface of the blades is almost absent. Accordingly, the associated mechanical losses are sharply reduced [3]. However, even the most modern mechanical and pneumatic nozzles do not allow water atomization with an effective droplet diameter less than 30 μm [2, 6].

For example, to supply water into the cyclic air flow, nozzles of a special design were developed (the outlet diameter is 0.1–0.4 mm), in which spraying is realized due to the impact action: water under high pressure (7–14 MPa) is fed to the nozzle head, after which it breaks down due to the "pin" into small droplets, the diameter of which does not exceed 20 μm [3, 17]. According to Mee Industries Inc, an increase in the efficiency of gas turbine plants using a contact cooling system in the summer season reaches 2–4%. Despite the relatively widespread use, such systems require rather complex pumping equipment for water supply and create difficulties in operating the system under high pressure [17].

A promising method for realizing more efficient atomization is to use a liquid superheated relative to the saturation temperature. In the process of the outflow of such a liquid through the nozzle of the spraying device, its explosive boiling occurs, as a result of which the liquid is crushed into small droplets [18]. In addition, during experimental studies it was found that if the diameter of a water droplet in the flow path of the compressor does not exceed 20 μm, then the effect of centrifugal forces on the droplet becomes almost imperceptible and droplets are directed along the air flow [19]. From this point of view, ideally nozzles installed at the compressor inlet should spray a liquid with a droplet diameter of no more than 20 μm.

A promising method for humidifying the working fluid of a gas turbine plant is to use a two-phase jet apparatus – an aerothermopressor [20, 21]. The operation of this apparatus is based on the process of thermogasdynamic compression. A feature of this process is the air pressure increase as a result of instantaneous evaporation of the water injected into the air flow accelerated to the speed of sound. In this case, the air pressure and air cooling are increased by removing heat from the air flow [22]. It should be noted that the aerothermopressor can have one more function – providing effective fine dispersion of liquid due to incomplete evaporation [22].

There are experimental studies of high- and low-flow aerothermopressors in which the authors have shown the effectiveness of the aerothermopressor using as an effective high-speed compact contact heat exchanger [20, 23]. For example, experimental studies [23] were carried out with the following data (a low-flow aerothermopressors): maximum gas flow rate of 1.63 kg/s; inlet temperatures 355–365 °C; the length of the evaporation chamber was regulated in the range (l/D) = 2.15–11.30; the inlet Mach number varied within the range M = 0.32–0.89; the relative water flow rate was 0–15%. The paper concludes that it is advisable to narrow the evaporation chamber. A conclusion is given on the possibility of reducing the back pressure in the gas turbine plant to 5 kPa at gas flow rates by 20 kg/s and M = 0.7–0.8. In addition, the possibility of increasing the total flow pressure in the aerothermopressor flow path to 20% was shown due to the appearance of the thermogasdynamic compression effect [20].

The following advantages of the aerothermopressor using are:

– increasing the pressure and cooling the working fluid will reduce the work on compression in the compressor;
– it also will provide effective atomization and humidification of water between compressor stages;
– it will reduce the additional compressor work when water droplets evaporate in the flow path during compression, and increase the amount of working fluid in the cycle.

The rational organization of thermophysical processes in the flow path of the aerothermopressor in the presence of the thermogasdynamic compression effect and incomplete evaporation is possible when the optimal geometric parameters are selected. These parameters include: the evaporation chamber diameter, the relative length of the evaporation chamber, confuser convergent angle, diffuser divergent angle, distance between the water injection point and the evaporation chamber inlet. The correct choice will ensure the evaporation of the amount of water (80–85%) in the evaporation chamber and the diffuser, and the additional evaporation of remaining water (15–20%) in the flow path of the high-pressure compressor. In this case, the water droplets diameter entering the compressor will not exceed 20 μm.

The choice of such optimal geometric parameters of the aerothermopressor, as well as the determination of the characteristics and injection mode (flow velocity; average, maximum and minimum droplet diameters; inlet air temperature; relative water flow rate, air pressure and air flow rate) should be carried out according to the results of an experimental study of working processes and in numerical modeling.

Based on the above analysis, the following research hypothesis can be formulated: to use an aerothermopressor for cooling the working fluid (air) of a gas turbine engine will provide effective fine dispersion of liquid due to intense heat and mass exchange processes. These processes are associated with incomplete evaporation in the flow path of the apparatus, hence a more effective isothermal compression process in the compressor will be provided. This, in turn, will make it possible to compensate for hydraulic pressure losses along the air path with a corresponding decrease in the compression work due to an increase in total pressure by 5–10%. As a result, it will increase the efficiency of the GTE with a corresponding decrease in specific fuel consumption by 1.0–1.5%.

3 Research Methodology

An aerothermopressor was designed for cyclic air intercooling in the cycle of a WR-21 Rolls Royce gas turbine engine (N_e = 25,250 kW). For the numerical simulation of the aerothermopressor operation an experimental model of the apparatus [22, 23] was used, which was designed and manufactured for carrying out experimental studies. The model has the following geometrical characteristics (Fig. 1): the aerothermopressor length L_{atp} = 1324 mm; the evaporation chamber diameter D_{ch} = 68 mm; air flow rate G_{air} = 1.56 kg/s. The confuser convergent angle (α_c = 35°) and the diffuser divergent angle (β_d = 5°) were chosen taking into account the obtaining of minimum losses when overcoming the friction forces and local resistances in the narrowing-expansion sections of the apparatus [24, 25].

The finite volume method was chosen as a numerical method. This method is implemented using computer CFD modeling in the ANSYS Fluent software package [26]. It is a reliable, feature-rich and easy to use tool for designing, modeling and analyzing fluid flows of varying complexity. The computational domain is divided by means of a grid into a set of finite volumes. The nodes are in the center of these volumes for which the calculation is carried out. For each control volume, the laws of conservation of mass, momentum and energy must be satisfied [24].

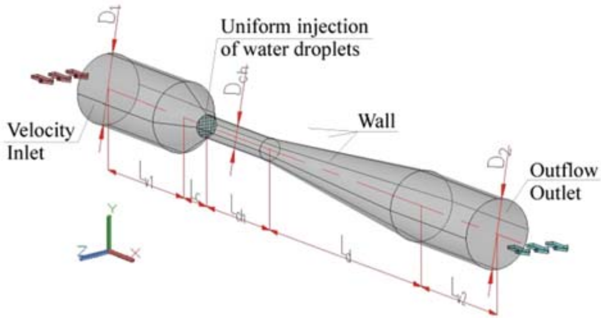


Fig. 1. Geometry model of the aerothermopressor.

The Eulerian-Lagrangian approach was used to simulate the interaction of injectable droplets and the air flow. Among existing turbulence models RANS, a two-parameter k-ε Realizable turbulence model was chosen [27]. Discrete Phase Model (DPM) has been used to model the motion of water droplets [28, 29]. A step-by-step grid solution with a simple algorithm for the relationship between velocity and pressure is carried out. The calculation was carried out taking into account the convergence of the results. The processing and visualization of the output data in the postprocessor was carried out, in the form of graphs, fields and streamlines for the main parameters of the working processes in the flow path of the aerothermopressor.

Air with the initial parameters of pressure, temperature and relative humidity, which correspond to the parameters of the GTP cyclic air after the first stage of the compressor, is considered as a working fluid: inlet air pressure P_1 = 301325 Pa; inlet air temperature T_{air1} = 473 K; inlet air velocity w_{air1} = 55 m/s; air mass flow G_{air} = 1.56 kg/s; Mach number at the evaporation chamber inlet M = 0.55.

The output parameters for numerical computer modeling are the main parameters of the two-phase flow (air-water) at the aerothermopressor outlet (at the inlet to the next stage of the gas turbine compressor): total air pressure P_{atp}, air velocity w_{air}, water velocity w_w, temperature T_{atp}, water droplet diameter in the air flow δ_p, air density ρ_{atp}, mass concentration of water m_{H2O}.

When analyzing the gas turbine scheme (Fig. 2), the well-known methods of calculating the cycles of gas turbines [6], as well as calculating the process of thermogasdynamic compression [20] were used. On the basis of these methods the corresponding software package was developed. The calculation of the GTP cycles was carried out for the degrees of pressure increase p_c = 12–42 and for the air parameters corresponding to ISO: t_0 = 15 °C, φ_0 = 30%. At the same time, instead of an air cooler (surface or contact with nozzle injection), it is proposed to use the aerothermopressor in the circuits.

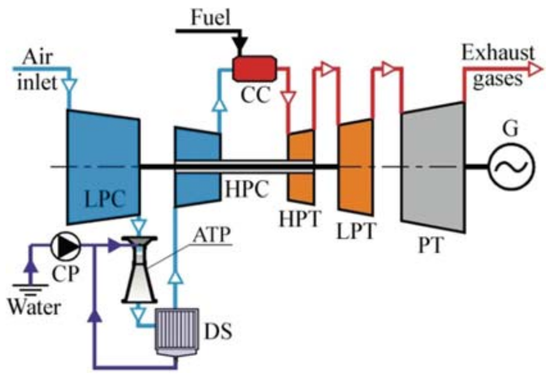


Fig. 2. A complex cycle with intercooling of air by using an aerothermopressor: LPC – low pressure compressor; HPC – high pressure compressor; CC – combustor; HPT – high pressure turbine; LPT – low pressure turbine; PT – power turbine; ATP – aerothermopressor; CP – circulation pump; DS – droplet separator; G – generator.

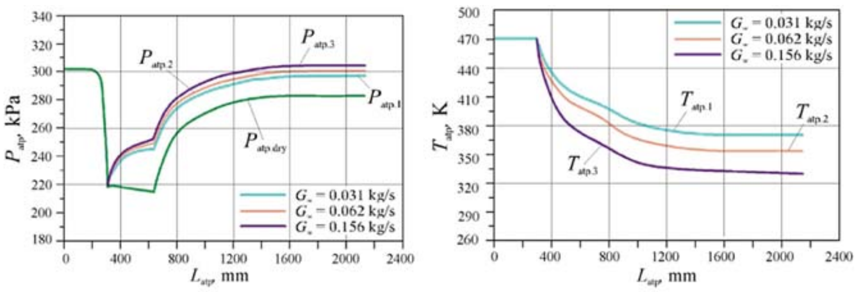
4 Results

The simulation of a "dry" aerothermopressor (without injection of water droplets into the evaporation chamber) was carried out to determine the pressure loss through the aerodynamic resistance in the flow path of the apparatus. It was found that the decrease in the air flow pressure (Fig. 3) due to friction losses is $\Delta P_{atp.dry}$ = 17 kPa (6%).

A discrete phase model was used when simulating the injection of water droplets into the flow path of the aerothermopressor (at the inlet to the evaporation chamber). Calculations were carried out for three variants of evaporation of water injected into the air flow: complete evaporation of water droplets in the evaporation chamber (δ_p = 3–30 μm, G_w = 0.031 kg/s); additional evaporation of water droplets in the diffuser (δ_p = 3–30 μm, G_w = 0.062 kg/s) and incomplete evaporation, with obtaining smaller droplets at the diffuser outlet of the aerothermopressor (δ_p = 3–30 μm, G_w = 0.156 kg/s).

A comparative analysis of the results obtained when simulating the aerothermopressor operation with and without water injection is carried out. Comparison of the pressure change P_{atp} along the length of the flow path with and without injection (Fig. 3a) shows that the pressure in the evaporation chamber due to losses decreases from 220 kPa to 215 kPa, that is, the pressure loss is up to 5 kPa (1.7%). However, the presence of thermogasdynamic compression increases the pressure and compensates for these losses. In this case, the total pressure in the evaporation chamber of the aerothermopressor increases by ΔP_{atp} = 30 kPa (10%).

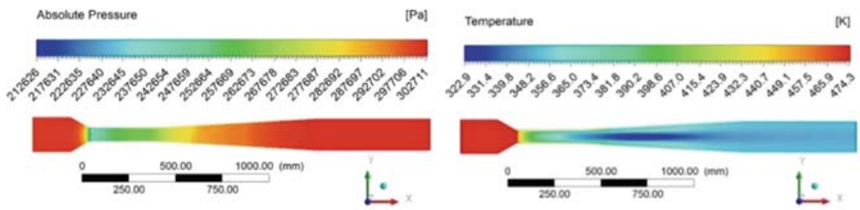
The magnitude of the cyclic air cooling in the aerothermopressor is: with complete evaporation of water droplets ΔT_{atp} = 106 K; with additional evaporation of water droplets in the diffuser ΔT_{atp} = 118 K; with incomplete evaporation ΔT_{atp} = 133 K (Fig. 3b).



(a) (b)

Fig. 3. Dependences of the total pressure P_{atp}, (a) and two-phase flow temperature T_{atp} (b) on the length L_{atp} of the aerothermopressor: $P_{atp.dry}$ – without injection of water droplets into the evaporation chamber.

With complete evaporation of water droplets in the evaporation chamber, the increase in total pressure (Fig. 3a) as a result of thermogasdynamic compression is ΔP_{atp} = –2.3 kPa (–0.76%) relative to the inlet pressure. With complete evaporation of water droplets in the diffuser, the increase in total pressure is already a positive value and reaches ΔP_{atp} = 0.25 kPa (0.08%). The greatest effect from thermogasdynamic compression is observed with incomplete evaporation, with obtaining smaller droplets at the outlet from the diffuser part of the apparatus, while the increase in total pressure (Fig. 4a) is ΔP_{atp} = 1.4 kPa (0.5%).



(a) (b)

Fig. 4. The fields distribution of total pressure P_{atp} (a) and flow temperature T_{atp} (b) on the length L_{atp} of the aerothermopressor at incomplete evaporation, with obtaining smaller droplets at the outlet of the diffuser part of the apparatus (δ_p = 3–30 μm, G_w = 0.156 kg/s).

Thus, the inlet temperature T_{atp1} = 473 K (200 °C) decreases to the outlet temperature (Fig. 4b) by T_{atp2} = 340–367 K (67–94 °C).

The dispersion of water droplets at the evaporation chamber inlet is δ_p = 3–30 µm. The distribution of sprayed water droplets in the flowing part of the aerothermopressor is given: for complete evaporation of water droplets (G_w = 0.031 kg/s) (Fig. 5a); additional evaporation of water droplets in the diffuser (G_w = 0.062 kg/s) (Fig. 5b); and incomplete evaporation, with obtaining smaller droplets at the outlet of the diffuser part of the aerothermopressor (G_w = 0.156 kg/s) (Fig. 5c).

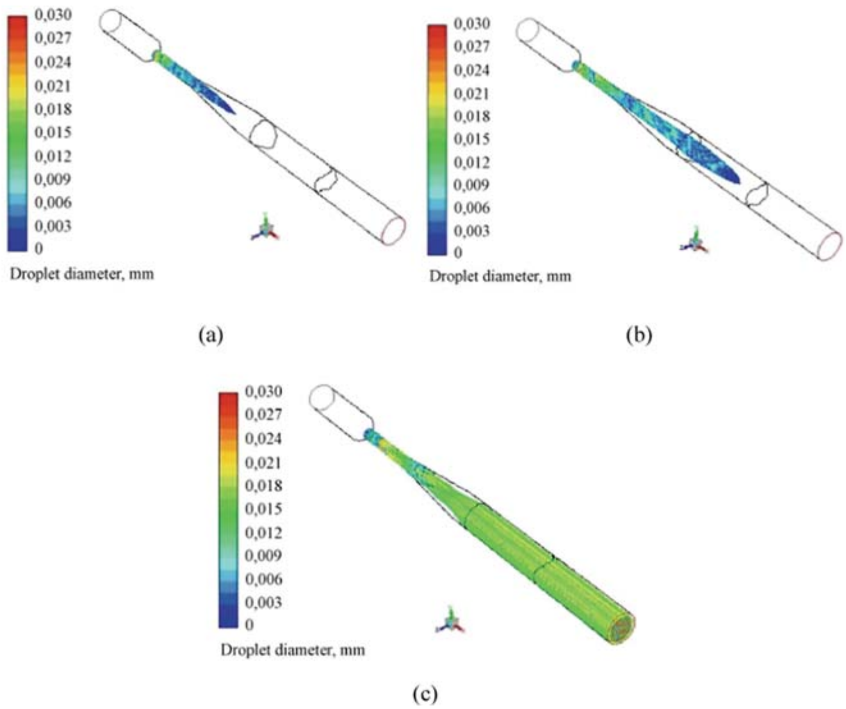


Fig. 5. Distribution of dispersion δ_p of sprayed water in the flow path of the aerothermopressor: a – complete evaporation of water droplets in the evaporation chamber (g_w = 2%); b – additional evaporation of water droplets in the diffuser (g_w = 4%); c – incomplete evaporation, with obtaining smaller droplets at the outlet of the diffuser part of the apparatus (g_w = 10%).

With incomplete evaporation, the droplet diameter decreased, and at the outlet from the diffuser of the aerothermopressor for non-evaporated water, the dispersion averaged δ_{p2} = 18 µm (Fig. 5c). In this case, the concentration of water droplets (dispersed two-phase flow) at the outlet is evenly distributed over the cross section. The obtained diameter of water droplets at the outlet of the aerothermopressor (18 µm) meets the requirements for the dispersion of the two-phase flow before the high-pressure compressor stage. Excess water will evaporate already in the flow path of the compressor. This amount is up to 4%.

Comparing the efficiency of the aerothermopressor operating regime with different relative flow rates of injected water (Fig. 6), it can be seen that an increase in the liquid amount to $g_w = 10\%$, leads to an increase in the relative increase in the total pressure by ΔP_{atp} to 1% and a decrease in the outlet temperature by $\Delta T_{atp} = 130$ K. This is due to the water injection in excess of the amount required for evaporation reduces the friction pressure loss and, accordingly, increases the total outlet pressure to practically the initial one (there is compensation to hydraulic pressure losses in the aerothermopressor). With an appropriate rational organization of the workflow associated with water injection, it is possible to obtain an increase in pressure relative to the initial one (more than 2–5%).

The average diameter of a dispersed flow droplet at the outlet is $\delta_{p2} = 18$ μm, with an excess amount of water injected to 10%. In this case, an increase in water flow rate leads to an increase in the average droplet diameter. With an increase in water flow rate g_w of more than 10%, it can be expected that the pressure loss due to the drag of the droplet in the flow will exceed all other losses. The positive effect of the pressure increase as a result of the thermogasdynamic compression effect will be significantly reduced. Therefore, the water injection into the aerothermopressor nozzle should be determined by the maximum pressure increase ΔP_{atp} and a decrease in the outlet temperature $T_2 < 343$ K.

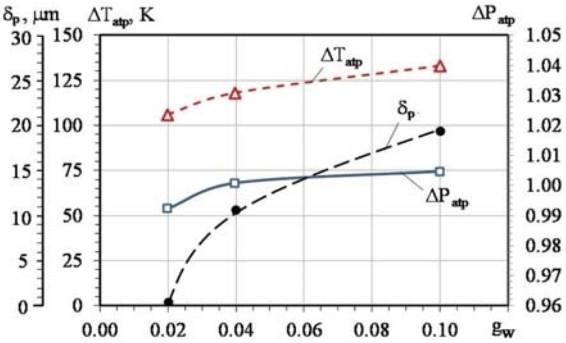


Fig. 6. Dependences of the temperature decrease ΔT_{atp}, the relative pressure increase ΔP_{atp}, the average droplet diameter at the diffuser outlet δ_p on the relative amount of water injected g_w into the aerothermopressor.

The aerothermopressor using in the GTP cycle for air intercooling allows reducing the cycle air temperature between compressor stages by 60–130 °C. Such a temperature decrease under thermogasdynamic compression conditions allows increasing the pressure by P_2–P_1 = 5–35 kPa, that is, by Δp_{atp} = 2–10% (Fig. 7a).

The simulation of gas turbine plant operation by using the developed software package to calculate of the GTP cycles showed the following efficiency from the aerothermopressor using to provide intercooling of cyclic air. Injected water after evaporation is an additional working fluid, the increase of which, in turn, makes it possible to increase the GTP specific power. A decrease in the compressor operation and a simultaneous increase in the amount of the working fluid in the cycle makes it possible to increase the efficiency GTP by $\Delta\eta_e$ = 0.01–0.02 (1–2%) in comparison with intercooling of air by using a surface air cooler. In this case, the specific fuel consumption will decrease by Δg_e = 5–10 g/(kW•h) (Fig. 7b). At the same time, the GTP specific power is increased by ΔN_s = 5–30 kW/(kg/s), which is 1.5–7.0% (Fig. 7b). The simulation of the GTP operation was carried out for the range of degrees of pressure increase in compressor stages of the gas turbine plant p_c = 12–42, which are typical for the operation mode according to the classical cycle (Fig. 2).

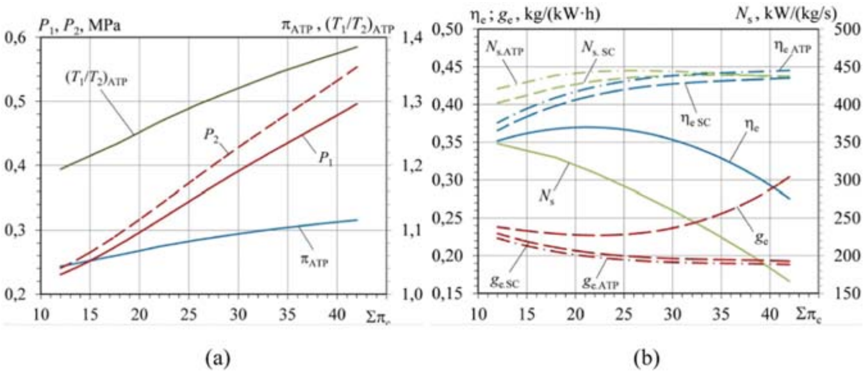


Fig. 7. Dependences of inlet pressure (P_1) and outlet pressure (P_2) of the aerothermopressor, relative air temperature $(T_1/T_2)_{atp}$, specific power output (N_s), specific fuel consumption (g_e) and efficiency (η_e) on the total compression ratio ($\Sigma\pi_c$) in compressors for the simple cycle, the complex cycle with a surface air cooler (SC) and the complex cycle with an aerothermopressor (ATP).

5 Conclusions

CFD simulation of the water droplet evaporation process in the airflow in the aerothermopressor was carried out. It has been established that the aerothermopressor provides effective fine atomization of water (the droplets diameter at the outlet from the apparatus does not exceed 18 μm), and, therefore, a more effective isotherm of the compression process in the high-pressure compressor occurs. In this case, the additional amount of water injected can be up to 10% relative to the flow rate of the cycle air.

The efficiency of the aerothermopressor application in the gas turbine cycle for contact cooling of cyclic air is analyzed. It has been revealed that the aerothermopressor allows increasing the pressure of the cyclic air between the compressor stages by Δp_{atp} = 2–10%, which will lead to a decrease in the compression work in the compressor stages. A decrease in the compressor operation and a simultaneous increase in the amount of the working fluid in the cycle makes it possible to increase the GTP efficiency by $\Delta\eta_e$ = 0.01–0.02 (in comparison with intercooling of air by using a surface air cooler). In this case, the specific fuel consumption decreases by Δg_e = 5–10 g/(kW•h). At the same time, the gas turbine engine specific power is increased by ΔN_s = 5–30 kW/(kg/s) (1.5–7.0%).

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