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STEAM TURBINE, GAS TURBINE, STEAM-GAS PLANTS  
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## Study of Cycling Air-Cooling System with a Cold Accumulator for Micro Gas-Turbine Installations

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**Abstract**—Using the cycling air-cooling systems of the CTIC type (Combustion Turbine Inlet Cooling) with a cold accumulator in a micro gas-turbine installation (micro-GTI) to preserve its capacity under the seasonal temperature rise of outside air is described. Water ice is used as the body-storage in the accumulators, and ice water (water at 0.5–1.0°C) is used as the body that cools air. The ice water circulates between the accumulator and the air-water heat exchanger. The cold accumulator model with renewable ice resources is considered. The model contains the heat-exchanging tube lattice-evaporator covered with ice. The lattice is cross-flowed with water. The criterion heat exchange equation that describes the process in the cold accumulator under consideration is presented. The calculations of duration of its active operation were performed. The dependence of cold accumulator service life on water circulation rate was evaluated. The adequacy of the design model was confirmed experimentally in the mock-up of the cold accumulator with a refrigerating machine periodically creating a 200 kg ice reserve in the reservoir-storage. The design model makes it possible to determine the weight of ice reserve of the discharged cold accumulator for cooling the cycle air in the operation of a C-30 type micro-GTI produced by the Capstone Company or micro-GTIs of other capacities. Recommendations for increasing the working capacity of cold accumulators of CTIC-systems of a micro-GTI were made.

**Keywords:** micro gas-turbine installation, cycle air cooling system, cold accumulator, ice water, cold accumulator service life

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The efficiency of gas-turbine installations (GTI) depends on the parameters of air coming into a turbine compressor. Since the air density determines the mass flow rate of gas in the installation under constancy of geometrical dimensions of the turbine setting of turbomachines, the output capacity depends directly on the density of incoming air. The seasonal temperature rise of outside air, which decreases its density and reduces the installation output capacity negatively influences the basic characteristics of GTI. The capacity of microturbine installations produced by the Capstone Company at that period of time decreases by approximately 20%, and the efficiency is reduced by 4% [1, 2].

The design air temperature maintenance in the GTI inlet (15°C according to ISO standard) and design capacity is achieved using a cycling air cooling system of the CTIC type (Fig. 1) consisting of a refrigeration steam-compressor unit and an ice cold accumulator. In the daytime, the cold accumulated with ice is used, and the accumulator is charged by a reduced rate for electricity at nighttime. The disadvantage of this system is the necessity to consume the electrical energy to operate the compressor of the refrigeration unit [3].

Figure 2 presents the construction arrangement of an air-water heat exchanger, in which the cooled air is fed to the inlet of the micro-GTI compressor using a fan.

The charging and discharging time of the cold accumulator were determined experimentally and these data underlay the subsequent calculation analysis of the ice-melting rate in the accumulator. During the experiment, the temperatures of air and ice water at the inlet and outlet of the heat exchanger—air cooler were recorded. Figure 3 shows the layout of the temperature sensors.

### EXPERIMENTAL STUDY OF MOCK-UP OF THE CTIC SYSTEM OF A CONSUMED TYPE WITH A COLD ACCUMULATOR

The main objective of the experimental study was in extension of the system operating time in the discharging regime of a cold accumulator while ensuring the air temperature at the inlet to the turbine compressor of the micro-GTI at the level of 15°C. After turning on the circulating pump and the fan of the air-cooler after approximately 10 min, the stationary mode of the system was achieved. In the stationary mode, the tem-

perature of cooled air at the inlet into the micro-GTI was stabilized. In 1 h during the discharging regime of the accumulator, the average air temperature at the outlet from the air-cooler was 15°C and reached the environment temperature in 6.5 h after the accumulator discharge cycle started (Fig. 4a).

Decrease of ice-water supply by reducing the motor supply voltage frequency of the circulating pump made it possible to extend the cold accumulator operation in the stationary mode by two times up to 6 h when the temperature of air incoming into the compressor amounted 15°C (Fig. 4b).

CALCULATION OF THE MELTING ICE PROCESS IN THE COLD ACCUMULATOR

The known expression for the Nusselt number [4–6] was taken as a basis for building a simplified mathematical model of the melting ice process in the heat-insulated tank of the cold accumulator, which made it possible to further calculate the melting ice rate.

The constant parameters of the system are the following:

- (1) air temperature at the outlet of the heat exchanger–air-cooler equal to 15°C;
- (2) air consumption through the heat exchanger–air-cooler equal to air consumption through the compressor of the C-30 micro-GTI at maximum design regime;
- (3) water temperature at the inlet of the heat exchanger–air-cooler; and
- (4) geometry of the ice block of the accumulator.

The following relations were used to describe the mathematical model of the installation [5]:

$$Re = \frac{w_w L_{i.b.}}{\nu_{av}},$$

where Re is the Reynolds number;  $w_w$  is the water velocity in the tank, m/s;  $L_{i.b.}$  is the characteristic dimension of the ice block, m;  $\nu_{av}$  is the kinematic coefficient of viscosity of circulating water, m<sup>2</sup>/s;

$$Nu = 0.52Re^{0.5}Pr^{0.36},$$

where Nu is the Nusselt number; Pr is the Prandtl number;

$$\alpha = \frac{Nu\lambda_w}{L_{i.b.}},$$

where  $\alpha$  is the heat transfer coefficient, W/(m<sup>2</sup> K);  $\lambda_w$  is the coefficient of heat conductivity of the circulating water, W/(m K);

$$\alpha(t_w - t_{i.b.}) = \lambda_i \rho_i \frac{d}{d\tau} w(\tau),$$

where  $t_w, t_{i.s.}$  is the temperature of water and the ice surface, which define the heat balance when ice melts, °C;  $\lambda_i$  is the heat conductivity coefficient of ice, W/(m K);

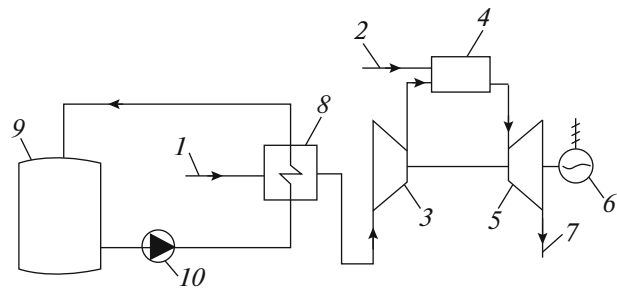


Fig. 1. Schematic diagram of the cooling system of inlet air of a GTI based on a refrigeration steam-compressive unit and an ice cold accumulator. 1, 2—Air and fuel supply; 3, 4—compressor and combustion chamber of the GTI; 5—turbine; 6—generator; 7—exit gases; 8—heat exchanger–air cooler; 9—cold accumulator; 10—circulating pump for ice-water supply.

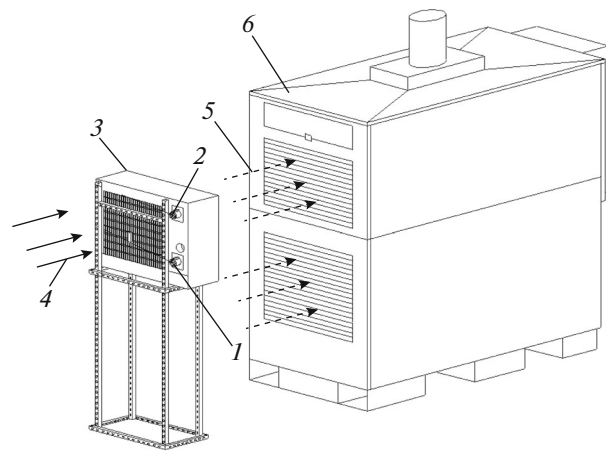


Fig. 2. Construction arrangement of the air-water heat exchanger. 1, 2—Delivery and return pipeline of coolant; 3—heat exchanger–air cooler; 4—air with temperature higher than 15°C; 5—air with temperature less than 15°C; 6—microturbine of the Capstone model.

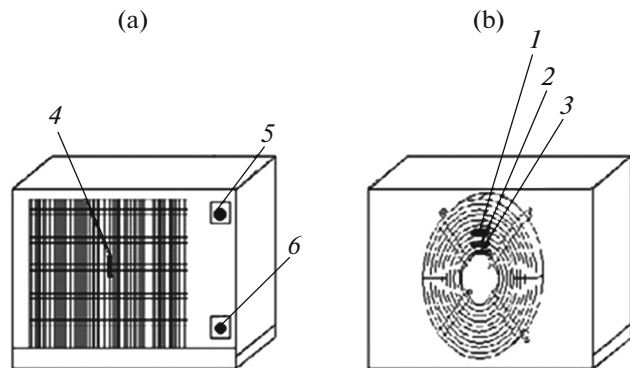
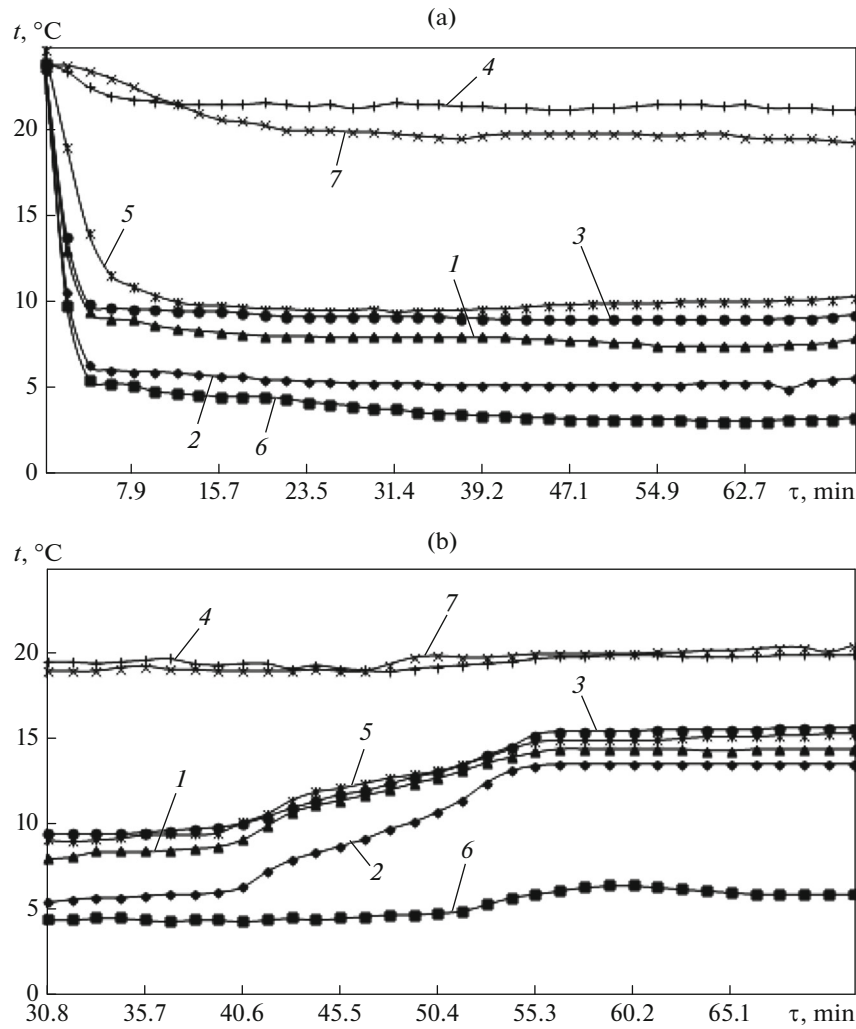


Fig. 3. Arrangement of the air cooler's temperature sensors. (a) Front view and (b) rear view. 1–3—Air at the heat exchanger outlet at various places of the lattice; 4—air at the heat exchanger inlet; 5–6—water at the heat exchanger outlet and inlet.



**Fig. 4.** Dependence of temperature,  $t$ , on time,  $\tau$ , when the frequency of the supply voltage of the motor of the circulation pump is (a) 50 Hz and (b) 25 Hz. 1–3—temperature of air at the heat exchanger outlet in various places of the lattice, °C; 4—temperature of air at the heat exchanger inlet, °C; 5, 6—temperature of water at the heat exchanger outlet and inlet, °C; 7—outdoor air temperature, °C.

$\rho_i$  is the ice density,  $\text{kg/m}^3$ ;  $w_{i.m.}(\tau)$  is a function of the ice melting rate depending on time  $\tau$ ;

$$w_{i.m.} = \frac{\alpha(t_w - t_{i.s.})F_{i.b.}}{\lambda_i \rho_i},$$

where  $w_{i.m.}$  is the ice melting rate,  $\text{m}^3/\text{s}$ ;  $F_{i.b.}$  is the surface area of the ice block washed by water,  $\text{m}^2$ ;

$$\tau_{i.m.} = \frac{V_{i.b.}}{w_{i.m.}},$$

where  $\tau_{i.m.}$  is the duration of melting ice, s;  $V_{i.b.}$  is the volume of the ice block,  $\text{m}^3$ .

The design characteristics of the ice-melting process are the following:  $L_{i.b.} = 0.095 \text{ m}$ ,  $F_{i.b.} = 9.2 \text{ m}^2$ ,  $V_{i.b.} = 0.218 \text{ m}^3$ .

The ice-melting rate in the accumulator determining its operating resource was the desired quantity.

The ice-water velocity in the accumulator tank calculated as the average between the velocity in the start time of the accumulator being turned on (when it is full charged and the tank is filled with ice) and the velocity after the complete ice melting is approximately  $3 \times 10^{-4} \text{ m/s}$ . During the experiment on the mock-up of the CTIC system when the ice with a weight of 200 kg melted for 6.5 h, the ice-melting rate in a mock-up installation with a tubular heat exchanger–evaporator was  $w_{i.m.} = 3.4 \times 10^{-2} \text{ m}^3/\text{h}$ , which is close to the design value.

The difference in obtained results is caused by calculation errors according to simplified mathematical models as well as by the fact that the heat supplied into

the accumulator from the environment influenced the experimental data.

### CONCLUSIONS

(1) The application of a CTIC system under seasonal temperature rise of the environment for cooling the cycle air at the inlet into the GTI up to 15°C ensures the maximum designed capacity of installations and it is appropriate even for climatic conditions of Russia in the spring–summer period.

(2) Adjusting the frequency of the supply voltage of the circulation pump motor for changing the ice-water supply into the heat exchanger of the cycle air cooling, one can increase the discharge time of the cold accumulator and, on that basis, create a digital control system.

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