

## Calculation Of Refrigeration Capacity Of GTU Intake Air Cooling Systems For Climatic Conditions With Saving Fuel Cost For Libyan Tripoli City

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### ABSTRACT:

A potential values of gas turbine (GT) intake air temperature decrease due to its cooling by exhaust gas heat recovery chillers and corresponding specific fuel savings have been estimated for climatic conditions of Libya. Absorption lithium-bromide chiller (ABC) and refrigerant ejector chiller (REC) providing the GT intake air temperature decrease down to 15°C and 10°C respectively are discussed. The Calculate of refrigeration capacity of waste heat recovery chillers to provide the maximum it was shown that gas turbine intake air cooling down to the temperatures of 10°C and 7°C by absorption aqua-ammonia and refrigerant ejector chillers provides annular fuel saving and the cost fuel saving (1.5 - 2.0) times larger as compared with absorption lithium--bromide chiller providing a gas turbine intake air temperature decrease down to 15°C. The higher efficiency of gas turbine intake air deep cooling down to the temperatures of 10°C and 7°C by absorption aqua-ammonia and refrigerant ejector chillers has been proved as a result. The annular fuel saving due to air cooling at the inlet of gas turbine generators by thermo transformers of different types has been evaluated for city (Tripoli) of Libya.

**Key words:** gas turbine generator, fuel saving,refrigeration capacity,absorption lithium-bromide chiller, refrigerant ejector chiller, absorption aqua-ammonia chiller, climatic conditions, using the corresponding power sales as weights,intake air cooling, exhaust gas waste heat recovery.

### 1. INTRODUCTION

Efficiency of gas turbines (GT) significantly depends on ambient air temperature  $t_{amb}$  at the inlet and sharply decreases with its increase [1–2]. So, the air temperature increase at the inlet of General Electric GT LM2500+ ( $N_e = 27$  MW at  $t_{amb} = 15$  °C) on 10 °C causes decrease in GT efficiency by 2% with corresponding increase in specific fuel consumption  $b_e$ , and for GT LM1600( $N_e = 15$  MW) – approximately by 1.6% [1]. Because of the intake air temperature raising

the electrical power output of GT are by 15–20% lower than nominal value at  $t_{amb} = 15\text{ °C}$  [2]. Therefore the problem of GT intake air cooling is particularly actual in energetic of Libya. A deficit of fresh water makes difficult the application of evaporative cooling of GT intake air [3] in the regions of Libya with arid hot climate where the effect from inlet fogging could be very essential. In such conditions pre-cooling GT intake air by waste heat recovery chillers (WHRC) utilizing exhaust gas heat can be a real alternative. As a working fluid an ammonia and water boiling under vacuum are used for aqua-ammonia chiller (AAC) and absorption lithium-bromide chiller (ABC) respectively and ozone-safe refrigerants R142B and R600 for refrigerant ejector chillers (REC) [4–5]. As ambient air parameters are characterized not only seasonal, but also daily fluctuations of temperature  $t_{amb}$  and relative humidity  $\varphi$ , it is necessary to solve a problem of choice of the specified (designed) refrigeration capacity of WHRC which, on one hand, has to provide whenever possible maximum decrease in GT intake air temperature and, respectively, maximum fuel saving, and on the other hand, minimizing operation of WHRC at lowered refrigeration capacity and, therefore, not proved increased capital expenditure for WHRC, ammonia in absorption aqua-ammonia thermo transformer (REC) the gas turbine intake air can be chilled to the temperature  $t_{a2} = 10\text{ °C}$  and even down to  $7\text{ °C}$  (with  $t_x = 2...3\text{ °C}$ ) [1, 7], but in the case of water applied as a coolant in absorption lithium-bromide thermo transformers (ABC) the temperature of chilled air is higher:  $t_{a2} = 15...17\text{ °C}$  ( $t_x \approx 7\text{ °C}$ ) [7, 8, 9, 10].

**The goal of the analysis** is estimation the efficiency of GT intake air cooling and. The Calculate of refrigeration capacity of WHRC providing maximum fuel saving due to cooling of air according to climatic conditions of operation.

## 2. RESULTS OF INVESTIGATION

As mentioned above the performance conditions of GT are characterized by daily fluctuations of intake ambient air temperature  $t_{amb}$  and relative humidity  $\varphi$  that influences thermodynamic efficiency of GT and, respectively, specific fuel consumption  $b_e$  [1–2]. Changes in ambient air parameters within July 2009 for Tripoli (the southern region of Libya with arid tropical climate) are presented in Figure 1.

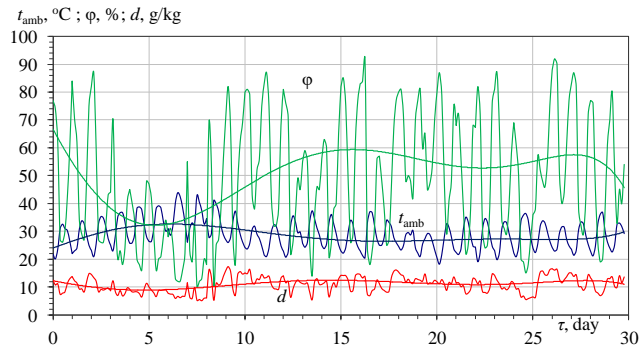
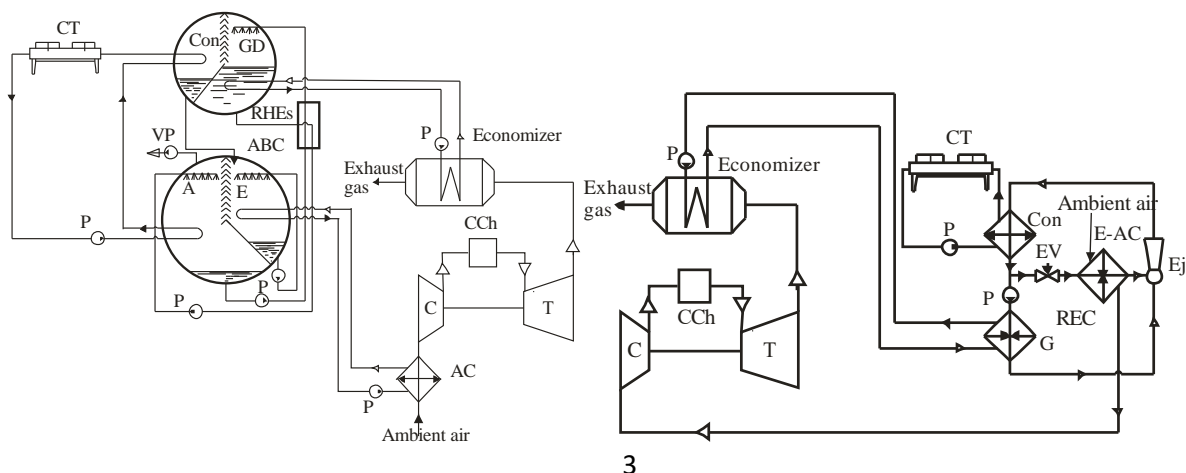


Fig. 1. Change of temperature  $t_{amb}$ , relative humidity  $\phi$  and absolute humidity  $d$  of ambient air within July, 2009 – Tripoli

Apparently, daily considerable fluctuations of ambient air temperature  $t_{amb}$  of 10–15 °C and relative humidity  $\phi$  are characterized by existence of day and night opposite directed maximum and minimum values of  $t_{amb}$  and  $\phi$ : to maxima of temperatures  $t_{amb}$  correspond minima of humidity  $\phi$  in the afternoon and vice versa at night. Their existence creates favorable conditions for bigger decrease in GT intake air temperature in the afternoon when considerable reduction of thermodynamic efficiency of GT takes place caused by the raised temperatures  $t_{amb}$ , but corresponding lowered humidity  $\phi$  causes some decrease in thermal load on WHRC.

Depth of air cooling in the heat exchanger is defined by temperature of a coolant and intensity of heat transfer. So, at the intensive heat exchange providing the minimum difference of temperatures between the cooled air and a coolant (5–8 °C), cooling of air down to the temperature about 15 °C in the air cooler by water (AC) of the absorption lithium bromide chiller (ABC) and to 10 °C in the refrigerant evaporator-air cooler (E-AC) of the refrigerant ejector chiller (REC) is possible. Schemes of cooling systems of GT intake air in ABC and REC with utilization of exhaust gas heat are given in Figure 2.



(a)

(b)

Fig. 2. Schemes of cooling systems of GT intake air in ABC (a) and REC (b) with utilization of exhaust gas heat: C – compressor; T – turbine; CCh – combustion chamber; Economizer – a water-heating utilization boiler; P – pump; ABC: GD – generator-desorber of water vapour; Con – condenser; A – absorber; E – evaporator; RHEs – regenerative heat exchanger of solutions, VP – vacuum pump; REC: Ej – ejector; G – generator of refrigerant vapour; Con – condenser; E-AC – evaporator-air cooler; EV – expansion valve; CT – cooling tower

The WHRC consists of high-temperature (waste heat utilization) and low-temperature (refrigeration) contours. So, in the high-temperature contour of ABC the heat of GT exhaust gas is used for evaporation of water from water lithium-bromide mixture in the generator-desorber GD (Figure 2(a)) with next condensation of water vapour by extracting its heat into surroundings in cooling tower. In the low-temperature contour the water is evaporated under vacuum with extracting the heat of evaporation from water as a coolant and thus producing refrigeration capacity spent for GT intake air cooling.

The heat of GT exhaust gas can also be used in generator G of a high-temperature contour of REC for generation of high pressure refrigerant vapour as a motive fluid for ejector Ej to compress the low pressure refrigerant vapour (at refrigerant boiling temperature  $t_0 = 2-5\text{ }^\circ\text{C}$ ), sucked from refrigerant evaporator-air cooler E-AC of refrigeration contour with increasing its pressure up to the pressure in the condenser Con (at refrigerant condensing temperature  $t_c$ ).

The air temperature decrease  $\Delta t = t_{amb} - t_{a2}$  and, respectively, the gained effect of GT intake air cooling depends on the current ambient air temperature  $t_{amb}$  and air cooled temperature  $t_{a2}$  which, in its turn, depends on the temperature of a coolant – cold water  $t_{cw}$  or boiling refrigerant  $t_0$ , i.e. on the type of WHRC: in ABC the air could be cooled to  $t_{a2} = 15-20\text{ }^\circ\text{C}$  (cold water  $t_{cw} = 7-10\text{ }^\circ\text{C}$ ), but in REC – to lower temperature  $t_{a2} = 7-10\text{ }^\circ\text{C}$  (boiling refrigerant  $t_0 = 2-4\text{ }^\circ\text{C}$ ) [5].

Values of decrease in air temperature  $\Delta t_{15}$  when cooling GT intake air from the current ambient temperature  $t_{amb}$  to temperature  $t_{a2} = 15\text{ }^\circ\text{C}$  in ABC and values  $\Delta t_{10}$  for  $t_{a2} = 10\text{ }^\circ\text{C}$

for cooling GT intake air in REC within July 2009 for Tripoli are presented in Figure 3. With decrease of GT intake air cooled temperature  $t_{a2}$  in REC not only the temperature differences  $\Delta t_{10}$  increase (as compared with  $\Delta t_{15}$  in ABC), but also duration of GT operation at the lowered temperature, and consequently, total effect from cooling.

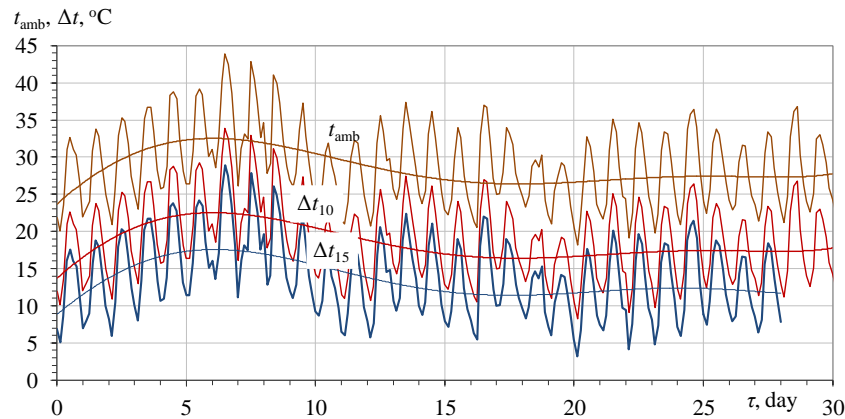


Fig. 3. Values of decrease in air temperature  $\Delta t_{15}$  when cooling GT intake air from the current ambient temperature  $t_{amb}$  to temperature  $t_{a2} = 15^{\circ}\text{C}$  (in ABC) and  $\Delta t_{10}$  for  $t_{a2} = 10^{\circ}\text{C}$  (in REC) within July 2009 for Tripoli

The refrigeration capacity  $Q_0$  spent for GT intake air cooling depends not only on the value of temperature decrease  $\Delta t = t_{amb} - t_{a2}$ , i.e. on the sensible heat which is taken away from ambient air, but also on the heat of condensation of water vapor from damp air when decreasing its temperature below wet bulb temperature  $t_{wb}$ .

It is more convenient to present the required refrigeration capacities of thermo transformers in the form of the specific refrigeration capacities relating on a single consumption of air  $G_a = 1 \text{ kg/s}$ :  $q_0 = \xi \cdot c_{wa} \cdot (t_{amb} - t_{a2})$ , kW/(kg/s) or kJ/kg, proceeding from which it is possible to calculate easily full established refrigeration capacity  $Q_0 = q_0 \cdot G_a$  for GT consumption of air  $G_a$ , where  $c_{wa}$  – a thermal capacity of damp air;  $\xi$  – total-to-sensible airheat decrease relation (total heat decrease of air related to sensible heat decrease or inversely proportional value of sensible heat rate) calculated as ratio of total heat extracted from the wet air during cooling (an air enthalpy decrease including the heat of water vapor condensation) and sensible heat extracted depending on temperature difference of air on dry thermometer  $\Delta t = t_{amb} - t_{a2}$ .

for GTU with the same impact of intake air temperature depression  $\Delta t$  on the fuel efficiency, i.e. the same decrease in specific fuel consumption  $\Delta b_e$  for  $1^{\circ}\text{C}$  depression of intake air temperature:  $\Delta b_{e1^{\circ}\text{C}} = \Delta b_e / \Delta t$ , it is quite convenient to use as parameter the specific fuel consumption saving – for 1 kW of GT electric power output:  $B_{f,y1} = B_{\tau} / N_e$ , kg/kW, where  $B_{\tau}$  – the total fuel consumption saving for GT with electric power output  $N_e$ , kW, for any time interval  $\tau$ ; for the estimation of annual specific fuel consumption saving as  $B_{f,y1} = \Sigma[(\Delta t \tau)] \cdot (\Delta b_e / \Delta t)$ , where  $\tau$  – a time interval, within which the temperature depression  $\Delta t$  could be assumed as constant:  $\tau = 1 \text{ h}$  [5, 6].

Dependence of GT specific electric power output for 1 kW, an annual fuel economy of  $B_{f,y1} = B_f / N_e$ , the kg/kw, gained as  $B_{f,y1} = \Sigma[(\Delta t_B \tau)] \cdot (\Delta b_e / \Delta t)$ , is resulted on fig. 4 .

Thus recognised that at decrease in temperature of air on an entry on 1 °C a specific fuel rate decreases for magnitude  $\Delta b_{e1^\circ C} = \Delta b_e / \Delta t = 0,35 \text{ g}/(\text{KW}_T \cdot \text{h})$ .

Values of the annual specific fuel saving  $B_{f,1kW}$ , relating to 1 kW of GT power output, due to GT intake air cooling from changing ambient air temperature  $t_{amb}$  to various temperature against the temperature of GT intake air cooled  $t_{a2}$  :  $t_{a2} = 10 \text{ }^\circ\text{C}$  in ETT and to  $t_{a2} = 15 \text{ }^\circ\text{C}$  in ALBTT,

To estimate the impact of cooling technologies considered the annual specific fuel consumption savings  $B_{f,y1}$  (related to 1 kW of GT electric power output) due to GT intake air cooling from actual changing ambient temperature  $t_a$  to various cooled air temperature  $t_{a2}$  by thermo transformers of different types have been calculated for ambient conditions at the location Tripoli during 2009. The results of this analysis are presented in Fig.4.

With this a specific fuel consumption reduction of 0.35 g/(kW·h) for every 1°C drop in gas turbine intake air temperature has been considered [1, 2].

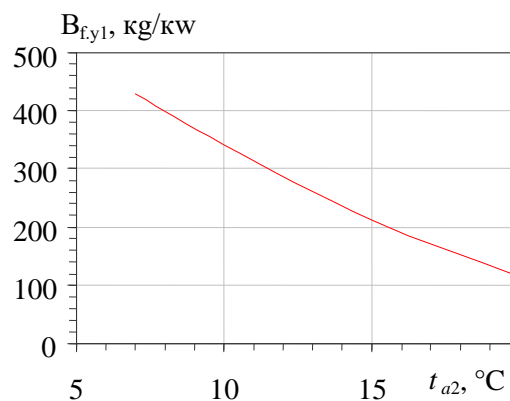


Fig. 4. Annual specific fuel consumption saving  $B_{f,y1}$  (related to 1 kW of GT electric power output) due to GT intake air cooling from actual ambient temperature  $t_a$  to various cooled air temperature  $t_{a2}$  by thermo transformers of different types:  $t_{a2} = 7-10 \text{ }^\circ\text{C}$  in RETT and ALBTT;  $t_{a2} = 15-20 \text{ }^\circ\text{C}$  in ALBTT (Tripoli, 2009) .

Knowing the annual specific fuel consumption saving  $B_{f,y1}$  (related to 1 kW of GT electric power output) due to GT intake air cooling, the total annual fuel consumption saving  $B_T$  for GT of any electric power output  $N_e$  may be calculated easily as  $B_f = B_{f,y1} N_e$  . The results of the impact of

cooling technologies considered on the total annual fuel consumption saving  $B_f$  for GT of electric power output  $N_e = 10\text{MW}$ .

Annual fuel saving value of  $B_{f,10\text{MW},T}$ , due to cooling the air at the gas turbine inlet with an electric power of 10MW, depending on the corresponding costs of cooling power  $Q_{0.10\text{MW}} = q_0 \cdot G_{a,10\text{MW}}$  at different final temperatures of cooled air  $t_{a2}$  (Tripoli 2009)  $B_{f,7...20}$  at  $t_{a2} = 7; 10; 15$  and  $20^\circ\text{C}$  are presented in Fig.5. at the same time, it was believed that for a gas turbine unit with an electric power of 10 MW the air flow rate  $G_{a,10\text{MW}} = 40 \text{ kg/s}$  (taken by analogy with LM1600)[1], based on which the cost of refrigeration power for air cooling with a flow rate of  $G_{a,10\text{MW}} = 40 \text{ kg/s}$  was calculated as  $Q_{0.10\text{MW}} = q_0 \cdot G_{a,10\text{MW}} = \zeta \cdot c_{wa} \cdot (t_{amb} - t_{a2}) \cdot G_{a,10\text{MW}}$ , kW

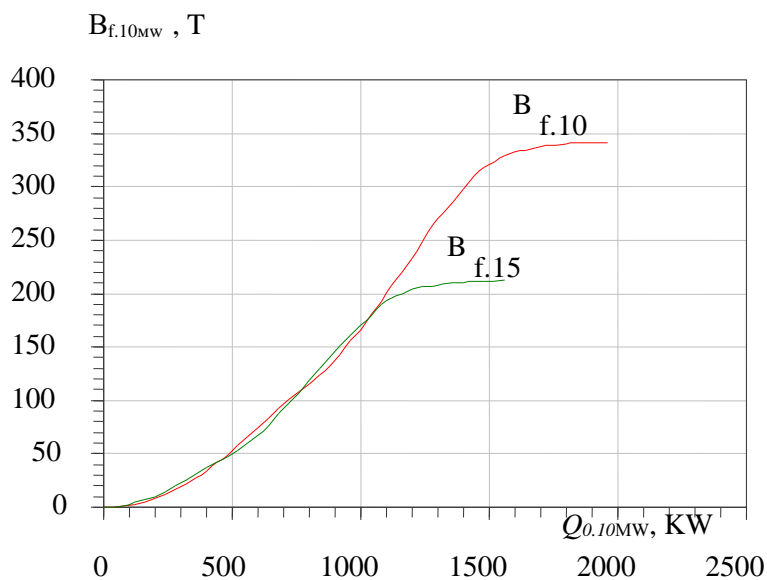


Fig.5. values of the annual specific fuel saving  $B_{f,10\text{MW}, T}$ , due to cooling at the gas turbine inlet with an electric power of 10 MW depending on the corresponding costs of cooling power  $Q_{0.10\text{MW}}$ , KW, at different final temperatures of cooled air  $t_{a2}$  (Tripoli, 2009) :  $B_{f,7...20}$  at  $t_{a2} = 7; 10; 15$  and  $20^\circ\text{C}$ .

Apparently, when cooling the air at the gas turbine inlet with an electric power of 10 MW in REC ( $t_{a2} = 10^\circ\text{C}$ ) the fuel saving  $B_{f,10} = 320\text{T}$ , for which it is necessary to use REC with an installed refrigeration capacity  $Q_{0.10\text{MW}} = 1520 \text{ kW}$ . the use of a more powerful (REC) does not lead to a noticeable reduction in fuel consumption: when it is increased to  $Q_{0.10\text{MW}} = 2000 \text{ kW}$ , by 30%, the  $B_{f,10\text{M}}$  fuel economy increases from 320T to 340T, a total of 6%. Therefore, the refrigeration capacity  $Q_{0.10\text{MW}} = 1520 \text{ kW}$  is a rational value. In the case of air cooling in a gas turbine unit in

an ABC ( $t_{a2}=15^{\circ}\text{C}$ ), the fuel savings are  $B_{f,15}=210\text{T}$ , for which it is necessary to use an ABC with an installed refrigeration capacity  $Q_{0.10\text{MW}}=1240\text{kW}$ .

Accordance with Fig. 5. When cooling the air at the gas turbine inlet with an electric power of 10 MW in REC (up to a temperature  $t_{a2} = 10^{\circ}\text{C}$ ), the rational installed cooling power  $Q_{0.10\text{MW}} = 1520$  kW, which provides annual fuel saving  $B_{f,10} = 320\text{T}$ , or  $320\text{T} / 0.8 \cdot 10^{-3} \text{ T/M}^3 = 400$  thousand.  $\text{M}^3$ , which at a cost of 500 \$/thousand.  $\text{M}^3$  is 200 thousand. \$/year, annual income  $I_{\text{REC}}=200$  thousand. \$ for electric gas turbine unit 10 MW.

In the case of cooling at the inlet of the gas turbine unit in the ABC ( $t_{a2}=15^{\circ}\text{C}$ ), the fuel saving  $B_{f,15} = 210\text{T}$ , for which it is necessary to use the ABC with an installed refrigeration capacity  $Q_{0.10\text{MW}}=1240$  kW. Then the annual income  $I_{\text{ABC}} = 131$  thousand. \$ for a gas turbine unit with an electrical capacity of 10 MW.

### 3. CONCLUSIONS

On the basis of the analysis of potentially possible decrease in temperature of GT intake air due to its cooling in WHRC by utilizing the exhaust gas heat the corresponding fuel savings are defined. The ABC and REC as WHRC providing cooling of GT intake air respectively to  $15^{\circ}\text{C}$  and  $10^{\circ}\text{C}$  are considered. Calculate of refrigeration capacity of WHRC providing the maximum effect from cooling in the form of annual fuel saving It is shown that deeper cooling of air on entry GT to temperature  $t_{a2} 10^{\circ}\text{C}$  and  $7^{\circ}\text{C}$  in REC provides in (1.5-2.0) times the big annual the cost of fuel economy in comparison with cooling of air to temperature  $t_{a2} = 15^{\circ}\text{C}$  in ABC The estimation of effect from air cooling on entry GTU in the form of an annual fuel economy for city ( Tripoli ) of Libya in which manufacture of electric energy by gas-turbine power stations is concentrated is resulted.

### REFERENCES

- [1] Bortmany J.N.: Assesment of aqua-ammonia refrigeration for pre-cooling gas turbine inlet air, Proceedings of ASMETURBOEXPO 2002 (Paper GT-2002-30657), 2002, 12 p.
- [2] Planning Electrical Power System Studies for Libya (Demand Forecasting & Generation Expansion Planning until 2025). Final Report, General People's Committee of Electricity, Water & Gas (GPCoEWG), ICEPCO, 2008.



[3] Bhargava R., Meher-Homji C.B.: Parametric analysis of existing gas turbines with inlet evaporative and overspray fogging, Proceedings of ASME TURBO EXPO 2002 (Paper GT-2002-30560), 2002, 15 p.

[4] Radchenko A.N., Kantor S.A.: Effektivnost sposobov ohlazhdeniya vozduha na vhode GTU kompressornyh stanciy v zavisimosti ot klimaticeskikh usloviy [The efficiency of ways for GTU inlet air cooling at the compressor stations, depending on climatic conditions], Aviacionno-kosmicheskaya tehnika i tehnologiya [Aerospace technics and technology], No. 1 (2015), pp. 95–98.

[5] Radchenko A.N.: Termoeconomicheskii metod analiza effektivnosti ohlazhdeniya vozduha na vhode dvigatelei teploispolzuyushchimi kholodilnymi mashinami [Thermoeconomical method of analysis of the efficiency of engine intake air cooling by waste heat recovery chillers], Cholodylna tehnika ta tehnologiya [Refrigeration engineering and technology], No. 5 (2014), pp. 30–36.

[6] Radchenko A.N., Kantor S.A.: Ocenka potenciala ohlazhdeniya vozduha na vhode gazoturbinnih ustanovok transformaciy teploty otrabotannyh gazov v teploispolzuyushchih kholodilnyh mashinah [Evaluation of the potential of gas turbine unite inlet air cooling by transforming the heat of exhaust gases in the waste heat recovery chillers], Aviacionno-kosmicheskayatehnikaitehnologiya[Aerospacetechnicsandtechnology], No. 4 (2014), pp. 56–59.

[7] Radchenko N.I., Rami El Gerbi: Effekt ot ohlazhdeniya vozduha na vhode gazoturbinnih electrostanciy v raznyh regionah Livii [Effect of air cooling at the inlet of gas turbine electricalstations in various regions of libya], Aviacionno-kosmicheskaya tehnika i tehnologiya[Aerospace technics and technology], No. 1 (2013), pp. 76–79.

[8] Radchenko N.I., Rami El Gerbi: Vybor sposoba ohlazhdeniya vozduha na vhode GTU IN zavisimosty ot regionalnyh klimaticeskikh usloviy [A choice of method for GTU intake air cooling according to region climatic conditions], Aviacionno-kosmicheskaya tehnika itehnologiya [Aerospace technics and technology], No. 2 (2013), pp. 97–102.

[9] Radchenko, R.N., Improved cogeneration gas piston module of a trigeneration plant for autonomous power supply [Text] / R.N. Radchenko, A.V. Ostapenko, A.A. Lekhmus // Aviation and space technology. – 2015. – No. 2(119). – pp. 104–107

[10] Radchenko, A.M. Heat transformation in an autonomous energy supply installation with an absorption refrigerating machine [Text] / A.M. Radchenko, O.V. Ostapenko // Refrigeration equipment and technology. – 2015. – T. 51, issue 2.– P. 32–37.