Analysis and Calculation Of Refrigeration Capacity Of GTU intake Air Cooling Systems For Climatic Conditions with Saving Fuel Cost For Libyan Hon City

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ABSRACT:

The paper is devoted to increasing the fuel efficiency of gas turbine unites (GTU) of a simple cycle by using the heat of exhaust gases in the thermo transformers to cool the intake air . The type and rational parameters of thermo transformers according to climatic conditions are determined. A fuel saving due to cooling of the air at the inlet of GTU in 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 (Hon) of Libya.

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

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 °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 φ, 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 ºC and even down to 7 ºC (with t_x = 2…3 ºC) [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 ºC (t_x ≈ 7 ºC) [7,9,10, 11].

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 and the Saving Fuel Cost .

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 φ that influences thermodynamic efficiency of GT and, respectively, specific fuel consumption b_e [1– 6]. Changes in ambient air parameters within July 2009 for Hon (the southern region of Libya with arid tropical climate) are presented in Figure 1.

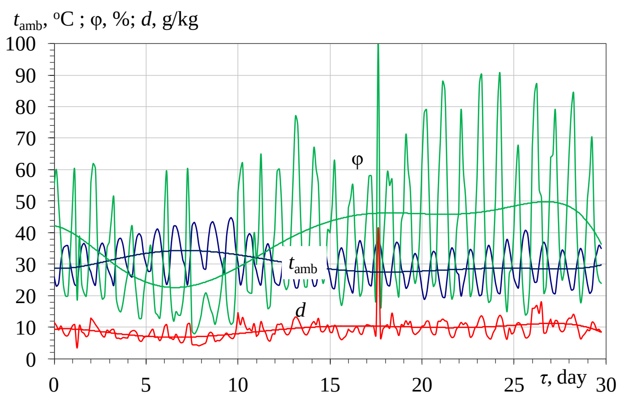
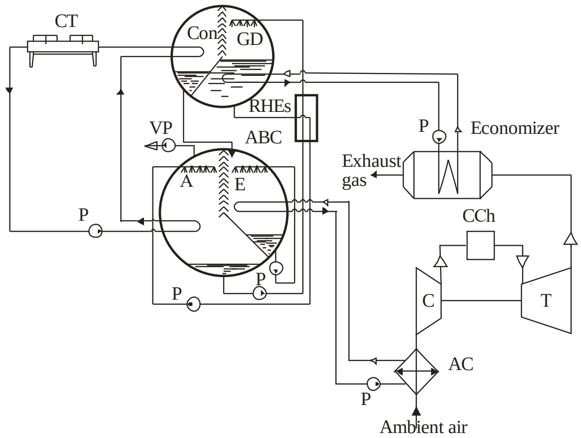


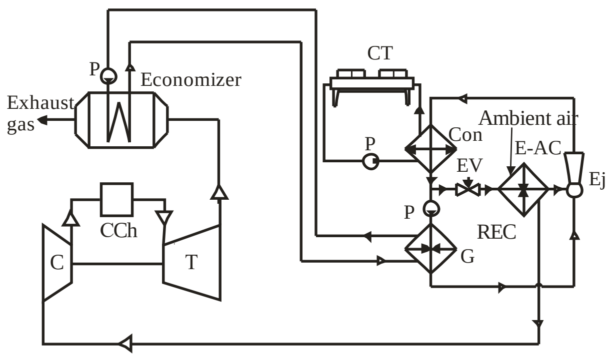
Fig. 1. Change of temperature t_{amb}, relative humidity φ and absolute humidity d of ambient air within July, 2009 - Hon

Apparently, daily considerable fluctuations of ambient air temperature t_{amb} of 10 –15 °C and relative humidity φ are characterized by existence of day and night opposite directed maximum and minimum values of t_{amb} and φ: to maxima of temperatures t_{amb} correspond minima of humidity φ 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 φ 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 heath 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 another water as a coolant and thus producing refrigeration capacity spent for GT intake air cooling.

The heath 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 °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°C (cold water t_{cw} = 7–10 °C), but in REC – to lower temperature t_{a2} = 7–10 °C (boiling refrigerant t_0 = 2–4 °C) [5].

Values of decrease in air temperature Δt_{15} when cooling GT intake air from the current ambient temperature t_{amb} to temperature t_{a2} = 15 °C in ABC and values Δt_{10} for t_{a2} = 10 °C for cooling GT intake air in REC within July 2009 for Hon are presented in Figure 3. With decrease of GT intake air cooled temperature t_{a2} in REC not only the temperature differences Δt_{10} increase (as compared with Δt_{15} in ABC), but also duration of GT operation at the lowered temperature, and consequently, total effect from cooling.

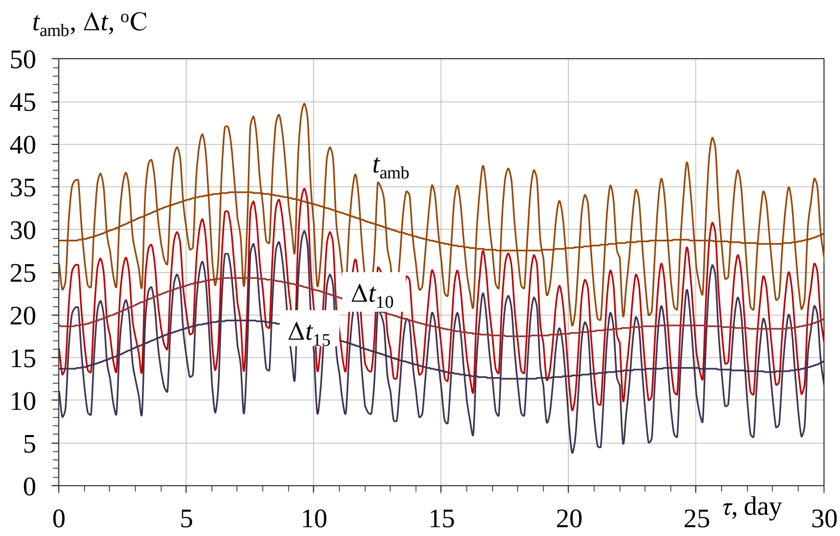
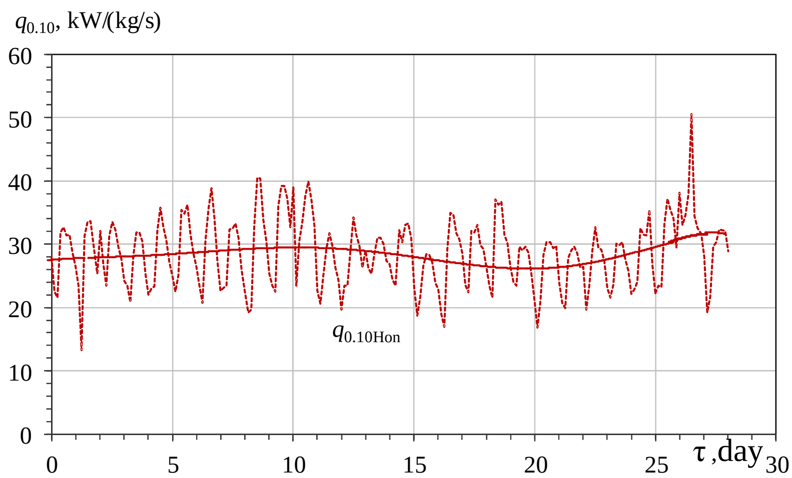


Fig. 3. Values of decrease in air temperature Δt_{15} when cooling GT intake air from the current ambient temperature t_{amb} to temperature t_{a2} = 15°C (in ABC) and Δt_{10} for t_{a2} = 10 °C (in REC) within July 2009 for Hon

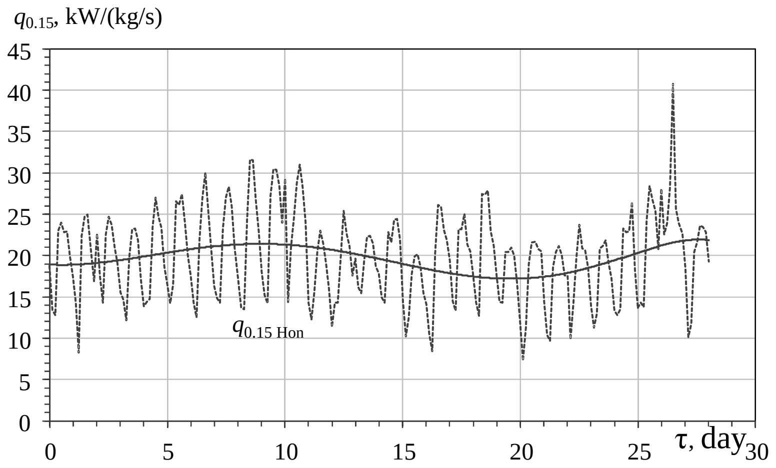
The refrigeration capacity Q_0 spent for GT intake air cooling depends not only on the value of temperature decrease $\Delta t = t_{amb} - t_{a2}$, i.e. on the sensible heat which is taken away from ambient air, but also on the heat of condensation of water vapor from damp air when decreasing its temperature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

It is more convenient to present the required refrigeration capacities of thermotransformers in the form of the specific refrigeration capacities relating on a single consumption of air $G_a = 1$ 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; ξ – total-to-sensible air heat 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 $\Delta t = t_{amb} - t_{a2}$.

Current values of specific refrigeration capacity q_0 (for Hon) when cooling GT intake air from temperature of ambient air t_{amb} to t_{a2} = 10 °C in REC and to t_{a2} = 15 °C in ABC in July 2009 are shown in Figure 4.



(a)



(b)

Fig. 4. Current values of specific refrigeration capacity q_0 for Tripoli and for Hon when cooling GT intake air from ambient air temperature t_{amb} air to temperature t_{a2} = 10 °C in REC (a) and t_{a2} = 15 °C in ABC (b) in July 2009: Hon

for GTU with the same impact of intake air temperature depression Δt on the fuel efficiency, i.e. the same decrease in specific fuel consumption Δb_e for 1 °C depression of intake air temperature : $\Delta b_{e1°C} = \Delta b_e / \Delta t$, it is quite convenient to use as parameter the specific fuel consumption saving – for 1 kW of GT electric power output: $B_{f.y1} = B_T / N_e$, kg/kW, where B_T – the total fuel consumption saving for GT with electric power output N_e , kW, for any time interval τ ; for the estimation of annual specific fuel consumption saving as $B_{f.y1} = \Sigma[(\Delta t\ \tau)]\cdot(\Delta b_e / \Delta t)$, where τ – a time interval, within which the temperature depression Δt could be assumed as constant: τ = 1 h [5, 7].

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 $\Delta b_{e1°C} = \Delta b_e / \Delta t$ =·0,35 g/(KW$_T$·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 airtemperature t_{amb} to various temperature against the temperature of GT intake air cooled t_{a2} : t_{a2} = 10 °C in ETT and to t_{a2} = 15 °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 thermotransformers of different types have been calculated for ambient conditions at the location hon during 2009. The results of this analysis are presented in Fig.5.

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].

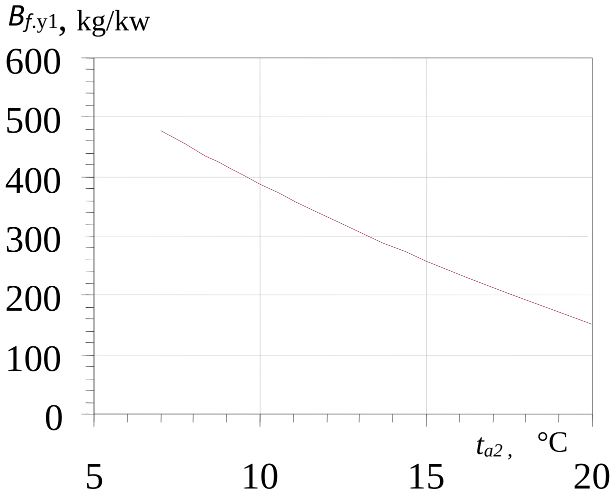


Fig. 5. Annual specific fuel consumption saving $B_{f.y1}$ (related to 1 kW of GT electric power output) due to GT intake air cooling from actual ambient temperature t_a to various cooled air temperature t_{a2} by thermo transformers of different types: t_{a2} = 7-10 °C in RETT and ALBTT; t_{a2} = 15-20 °C in ALBTT (Hon, 2009) .

Knowing the annual specific fuel consumption saving $B_{f.y}$ (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 = 10MW.

Annual fuel saving value of $B_{f.10MW,T}$, due to cooling the air at the gas turbine inlet with an electric power of 10MW, depending on the corresponding costs of cooling power $Q_{0.10MW}=q_0 \cdot G_{a.10MW}$ at different final temperatures of cooled air t_{a2} (Hon 2009) $B_{f.7...20}$ at t_{a2} = 7; 10; 15 and 20°Care 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 $G_{a.10MW}$ = 40 kg/s (taken by analogy with LM1600) [1] , based on which the cost of refrigeration power for air cooling with a flow rate of $G_{f.10MW}$= 40 kg/s was calculated as $Q_{0.10MW} = q_0 \cdot G_{a.10MW} = \xi \cdot c_{wa} \cdot (t_{amb} - t_{a2}) \cdot G_{f.10MW}$, Kw

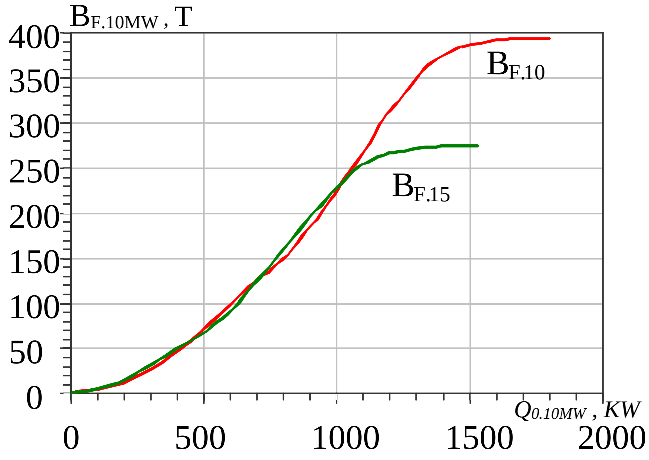


Fig.6. values of the annual specific fuel saving $B_{f.10MW}$, T, due to cooling at the gas turbine inlet with an electric power of 10 MW depending on the corresponding costs of cooling power $Q_{0.10MW}$, KW , at different final temperatures of cooled air t_{a2} (Hon, 2009) : $B_{f.7...20}$ at t_{a2} = 7 ; 10 ; 15 and 20°C.

Apparently , for climatic conditions of Hon when cooling the gas turbine intake air with an electric power of 10 MW in REC (t_{a2} = 10°C) temperature the fuel saving $B_{f.10}$ = 390 T, for which it is necessary to use REC with an installed refrigeration capacity $Q_{0.10MW}$ = 1440 kW. In the case of air cooling in a gas turbine unit in an ABC (t_{a2}=15°C), the fuel savings are $B_{f.15}$=260 T, for which it is necessary to use an ABC with an installed refrigeration capacity $Q_{0.10MW}$=1160 KW.

Accordance with Fig. 6.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°C), the rational installed cooling power $Q_{0.10MW}$ = 1440 kW, which provides annual fuel saving $B_{f.10}$ = 390T, or 390T / 0.8 · 10^{-3} T/M^3 = 488 thousand.M^3, which at a cost of 500 \$/thousand.M^3 is 244 thousand. \$Per year, annual income I_{REC}=244 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°C), the fuel saving $B_{f.15}$ = 260T , for which it is necessary to use the ABC with an installed refrigeration capacity $Q_{0.10\mathrm{MW}}$=1160 kW. Then the annual income I_{ABC} = 163 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°C and 10 °C are considered. the 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°C and 7°C in REC provides in (1.5- 2) times the big annual the cost of fuel economy in comparison with cooling of air to temperature t_{a2} = 15°C in ABC The estimation of effect from air cooling on entry GTU in the form of an annual fuel economy for city (Hon) of Libya in which manufacture of electric energy by gas-turbine power stations is concentrated is resulted.

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