COMPLEX RECOVERING OF GAS TURBINE WASTE HEAT FOR COOLING OF TURBINE CYCLIC AIR BY EJECTOR REFRIGERATION SYSTEM

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ABSTRACT. The paper considers the use of ejector refrigeration system recovering the heat of exhaust gas and compressed air after the low pressure compressor of gas turbine to cool the turbine cyclic air. The effects of performance parameters of the ejector refrigeration system on the cyclic air temperature drop have been investigated. The efficiency of gas turbines with and without ejector waste heat recovery refrigeration was compared.

INTRODUCTION

The power output and thermal efficiency of gas turbines are strongly influenced by ambient conditions. Thus, every 10 °C of ambient air temperature rise causes about 5–10% power output drop and 1.0-1.5% decrease in thermal efficiency and increase in specific fuel consumption by 1.0-2.0% [1–3]. One of the approaches to reduce the gas turbine performance degradation due to an increase in the ambient temperature is to pre-cool air at the inlet of gas turbine compressor by spraying demineralized water converted into a fog by atomizing nozzles operating at high pressure [1, 2]. A systematic parametric study on the effects of inlet fogging on the various existing gas turbines has been carried out by authors [1, 2, 4, 5]. However, in spite of the minimum investment and operating cost, the efficiency of fogging techniques drops at high relative humidity ambient conditions. Therefore the indirect cooling by refrigeration cycle would achieve a larger improvement in gas turbine output power augmentation. To increase the thermal efficiency of turbine engines their exhaust heat might be recovered to drive the refrigeration system.

In the present study, the ejector waste heat recovery refrigeration system [7] has been chosen due to its extremely simplicity and high performance reliability compared to the other available inlet air refrigeration techniques (absorption [3, 6] or turboexpander systems). The ejector refrigeration systems consist of mainly heat exchangers and pumps.

An assessment of applying the ejector refrigeration system to cool a cyclic air of gas turbines (compressor intake air cooling and intercooling of compressed air) by complex recovering the heat of exhaust gas and compressed air has been considered. The influence of performance parameters of the ejector refrigeration system on the temperature drop in the gas turbine cyclic air was also evaluated.

GAS TURBINE CYCLIC AIR COOLING BY COMPLEX WASTE HEAT RECOVERING IN EJECTOR REFRIGERATION SYSTEM

The schematic diagrams of ejector waste heat recovery refrigeration systems (EWRS) utilizing the heat of exhaust gas and compressed air after the low pressure compressor to cool the cyclic air of gas turbine are presented in Fig. 1. The ejector refrigeration systems consist of the main heat exchangers: generator of high pressure refrigerant vapour G, evaporator-air cooler E and condenser

Cn. Generally the generator consists of economizer section G_{EC}, where liquid refrigerant is heated from condensing temperature t_c to boiling temperature t_G, and evaporative section G_{EV}, where liquid refrigerant is evaporated at temperature t_G. In the case of utilizing the heat of exhaust gas the generator G, incorporating both sections, is located in the exhaust gas chimney (Fig. 1(a)). If the EWRS uses adequate heat of compressed air the economizer section G_{EC} is placed after the low pressure compressor C_{LP} (Fig. 1(b)).

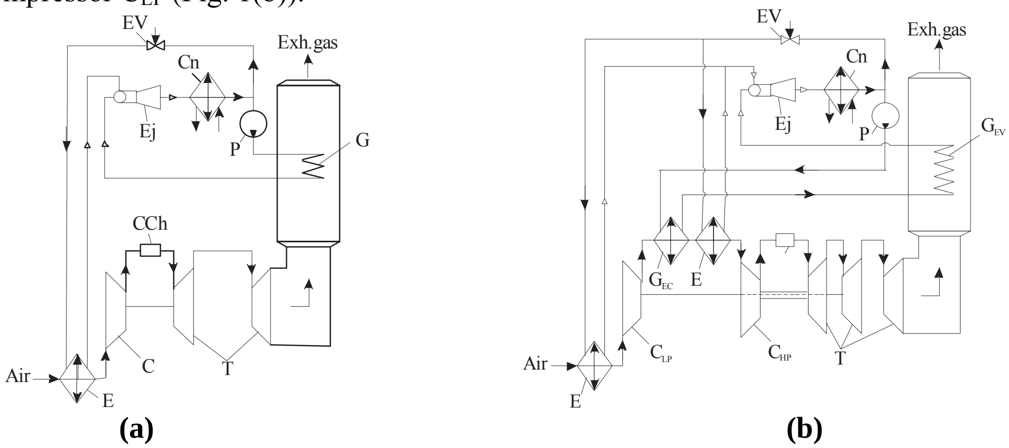


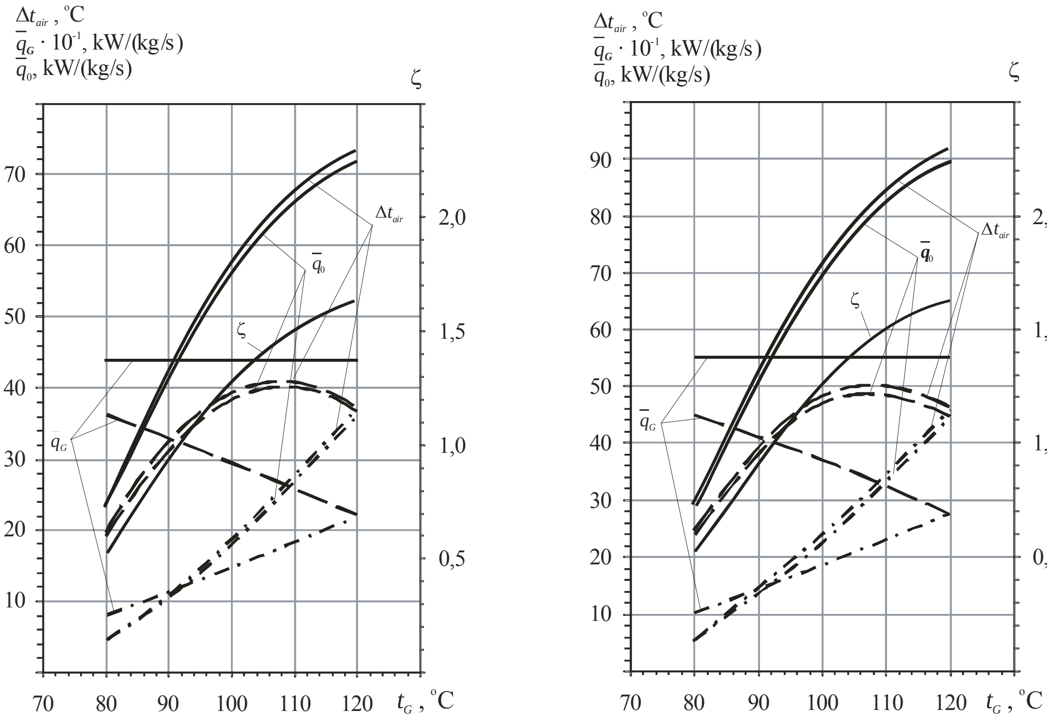
Fig. 1. Schematic diagrams of gas turbines with ejector waste heat recovery refrigeration systems: C − air compressor; C_{LP} and C_{HP} – low and high pressure air compressors; CCh – combustion chamber; T – turbine; G – generator; G_{EC} and G_{EV} – economizer and evaporative sections of generator; E − evaporator; Ej − ejector; Cn − condenser; P – pump; EV – expansion valve

The efficiency of EWRS is estimated by coefficient of performance ζ which is the ratio of amount of a cold produced (refrigeration capacity) to the quantity of heat spent (the heat removed from the exhaust gas). The value of ζ increases with arising the boiling temperatures of high pressure refrigerant in the generator t_G and low pressure refrigerant in the evaporator-air cooler t_0 and decreasing the condensing temperature t_c.

Variations of the coefficient of performance ζ, specific heat load (related to the air mass flow through the compressor) on the generator $\bar{q}_G$ and refrigeration capacity $\bar{q}_0$ (specific heat removed from the compressor intake air), the compressor intake air temperature drop Δt_{air} with the boiling temperature of high pressure refrigerant in the generator t_G are presented in Fig. 2.

The calculations have been performed for the temperatures of exhaust gas at the inlet of generator t_{exh} = 500 ºC and at the outlet of generator t_{exh2} = 100 ºC and boiling temperature of low pressure refrigerant in the evaporator t_0 = 0 °C. A refrigerant R142B is used as a working fluid in EWRS. For the case in Fig. 1(b) and 2(b) the compressed air is cooled in the economizer section by 100 ºC.

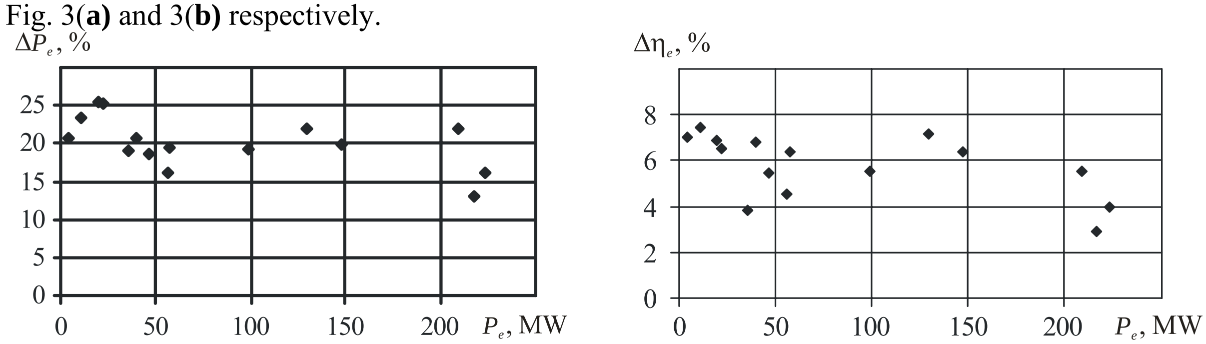
As one can see the variation of boiling temperature of high pressure refrigerant in the generator t_G has a significant influence upon the compressor intake air temperature drop Δt_{air} in the evaporator-air cooler. For both cases the maximum values Δt_{air} = 70–90 ºC achieved in EWRS twice higher than it is required to cool the ambient air at the compressor intake from the highest initial ambient temperature of 55 °C to the ISO temperature of 15 °C. Thus, about half a refrigeration capacity of EWRS might be used for adequate cooling of gas turbine cyclic air, for example deep intercooling of compressed air at the inlet of high pressure compressor. For this purpose the evaporator-air cooler has to include two sections: the first one – to pre-cool the ambient air at the suction of low pressure compressor and the second one – for intercooling the compressed air at the inlet of high pressure compressor, by lowering the input power of both compressors, that results in the gas turbine power output bust and increase in thermal efficiency of turbine engine.



(a) **(b)**

Fig. 2. Coefficient of performance ζ, specific heat load on the generator $\overline{q}_G$ and refrigeration capacity $\overline{q}_0$, cyclic air temperature drop Δt_{air} versus the boiling temperature of refrigerant in the generator t_G : - - - – evaporative section; -·-·- – economizer section

Values of power output bust ΔP_e and increase in thermal efficiency $\Delta\eta_e$ of gas turbines due to pre-cooling the compressor intake air from the ambient temperature of 55 °C to the ISO temperature of 15 °C, expressed in percent with respect to these parameters at the ambient temperature of 55 °C (without refrigeration), against gas turbine power output P_e at ISO ambient temperature of 15 ºC for a set of 15 gas turbines with power output ranging from 5 MW to 260 MW [1, 2] are presented in Fig. 3(**a**) and 3(**b**) respectively.



(a) **(b)**

Fig. 3. Power output bust ΔP_e (**a**) and increase in thermal efficiency $\Delta\eta_e$ (**b**) of gas turbines due to pre-cooling the compressor intake air from the ambient temperature of 55 °C to the ISO temperature of 15 °C, referred to these parameters at the ambient temperature of 55 °C, against gas turbine power output P_e at the ISO ambient temperature of 15 ºC

As seen in the Fig. 3, the application of EWRS, utilizing about a half the total exhaust gas heat to pre-cool the compressor intake air by 40 °C, provides the power output bust ΔP_e by about 15–25 % and increase in thermal efficiency $\Delta\eta_e$ of gas turbines by 4–7 %.

CONCLUSIONS

1. An assessment of application of the ejector refrigeration system utilizing the heat of exhaust gas and compressed air after the low pressure compressor for cooling a cyclic air of gas turbines was considered. The use of EWRS, utilizing about a half the total exhaust gas heat to pre-cool the compressor intake air by 40 °C, provides the power output bust ΔP_e by about 15–25 % and increase in thermal efficiency $\Delta\eta_e$ of gas turbines by 4–7 %.
2. The variation of boiling temperature of high pressure refrigerant in the generator t_G exposes a greate influence on the cyclic air temperature drop Δt_{air} in the evaporator-air cooler. The maximum values Δt_{air} = 70–90 ºC is achievable by complex utilization of waste heat of gas turbines.

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REFERENCES

1. Bhargava, R., Bianchi, M., Melino, F., Peretto, A. [2003], Parametric analysis of combined cycles equipped with inlet fogging, *ASME Paper* No. GT-2003-38187.-11 p.
2. Bhargava, R., Meher-Homji, C.B. [2002], Parametric analysis of existing gas turbines with inlet evaporative and overspray fogging, *ASME Paper* No. GT-2002-30560.-15 p.
3. Bortmany, J.N. [2002], Assesstment of aqua-ammonia refrigeration for pre-cooling gas turbine inlet air, *ASME Paper* No. GT-2002-30657.-12 p.
4. Cataldi, G., Guntner, G., Matz, C., McKay, T., Hoffmann, J., Nemet, A., Lecheler, S., Braun, J. [2004], Influence of high fogging on gas turbine engine operation and performance, *ASME Paper* No. GT-2004-53788.-11 p.
5. Chaker, M., Meher-Homji, C.B., Mee, III, T.R., [2002], Inlet fogging of gas turbine engines-Part A: Fog droplet thermodynamics, heat transfer and practical considerations, *ASME Paper* No. GT-2002-30562.-16 p.
6. Nixdorf, M., Prelipceanu, A., Hein, D. [2002], Thermo-economic analysis of inlet air conditioning methods of a cogeneration gas turbine plant, *ASME Paper* No. GT-2002-30561.-10 p.
7. Radchenko, A.N. [2008], Cooling of ventilation air in marine electric engines by ejector waste heat recovery refrigeration systems, *Proceedings, 12th International Symposium on Heat Transfer and Renewable Sources of Energy "HTRSE–2008"*, Szczecin, Poland., pp. 521-528.