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ASSESSMENT OF EJECTOR WASTE HEAT RECOVERY REFRIGERATION FOR PRE-COOLING GAS TURBINE INLET AIR

ANDREY RADCHENKO
National University of Shipbuilding named after admiral Makarov,
54025, Nikolaev, Ukraine
(Corresponding author: andrad69@mail.ru)

ABSTRACT: This paper presents an assessment of applying the ejector refrigeration system recovering the exhaust heat of gas turbines for pre-cooling compressor intake air. The influence of performance parameters of the ejector refrigeration system on the compressor intake air temperature drop was investigated. The efficiency of gas turbines with and without ejector waste heat recovery refrigeration was estimated.

INTRODUCTION

Gas turbines exhibit a great sensitivity to ambient conditions. Typically the effect of relative humidity on the gas turbine power output is no more than 1% within 0–100% relative humidity. However, ambient temperature has a strong influence upon the gas turbine performance, with power output dropping by about 0.5–0.9 % for every 1 °C of temperature rise [1, 2]. Aeroderivative gas turbines are affected by the change in ambient temperature more as compared to industrial engines. One way to prevent the loss in gas turbine power output, caused by high ambient temperature, is to pre-cool air at the inlet of gas turbine compressor.

There are two general methods to cool the compressor intake air: without refrigeration (by using water as a cooling medium) including first of all fogging – evaporative gas turbine compressor intake air cooling to saturated air flow at the wet bulb temperature, or overspray fogging – evaporative intake air cooling added by intercooling the air being compressed (wet compression), both by spraying demineralized water converted into a fog by atomizing nozzles operating at high pressure [1, 2, 4], and with refrigeration. The last group comprises two types of refrigeration: electrically or mechanically driven compressor chillers and waste heat recovery refrigeration such as absorption chillers [3], turboexpander or ejector chillers [5].

A comprehensive parametric study on the effects of inlet fogging (both evaporative and overspray) on the various existing gas turbines has been presented by Bhargava et al [1, 2]. Various inlet fogging techniques have certain limitations such as modifications in the plant lay out combined with increased investment and operating cost and low evaporative efficiency. Similarly, inlet fogging approach may loose its advantages at locations where shortage of a water source exits or ambient conditions are very humid. If the performance efficiency of evaporative air cooling is not high enough refrigeration could be used.

The ejector refrigeration systems consist of mainly heat exchangers and simple rotating machinery (pumps) and therefore they have a high reliability and availability required for any application. In the present study, the choice of the ejector waste heat recovery refrigeration system among the available inlet air cooling refrigeration techniques is justified by its extremely simplicity and high performance reliability. Due to this the ejector refrigeration system does not need any serious modifications in the plant lay out combined with increased investment and operating cost. However, it has certain limitations such as low coefficient of performance of ejector refrigeration cycle, nearly twice lower compared to absorption one [3, 5]. Therefore if the results for ejector refrigeration systems show quite moderate gas turbine performance improvement, the efficiency of waste heat recovery by other, complicated, alternatives (absorption or turboexpander systems) will be more optimistical.

For this investigation a set of 15 gas turbines with power output ranging from 5 MW to 260 MW is considered [1, 2]. The set chosen covers all the three categories of gas turbines into which all the machines from major gas turbine manufacturers worldwide can be classified, namely, traditional industrial engines (gas turbine inlet temperature TIT<1200ºC), advanced industrial gas turbines (TIT > 1200ºC) and aeroderivative machines.

In this paper an assessment of applying the ejector refrigeration system recovering the exhaust heat of gas turbines for pre-cooling compressor intake air has been undertaken. The influence of performance parameters of the ejector refrigeration system on the compressor intake air temperature drop was also investigated.

INTAKE AIR COOLING BY EJECTOR WASTE HEAT RECOVERY REFRIGERATION

The schematic diagram of ejector waste heat recovery refrigeration system (EWRS) utilizing the exhaust gas heat to pre-cool air at the inlet of gas turbine compressor is presented in Fig. 2.

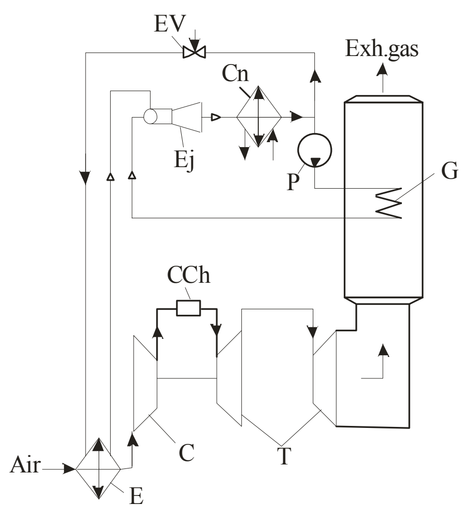


Fig. 1. Schematic diagram of the gas turbine with ejector waste heat recovery refrigeration system:
C − air compressor;
CCh – combustion chamber;
T – turbine;
G – generator;
E − evaporator;
Ej − ejector;
Cn − condenser;
P – pump;
EV – expansion valve

The EWRS in Fig. 1 runs in the following way. The heat of exhaust gas is used to generate a high pressure vapour of refrigerant in the generator G. The high pressure vapour is employed as a motive fluid in the ejector Ej to compress the low pressure refrigerant vapour, being sucked from the evaporator E, and to deliver refrigerant vapour mixture to the condenser Cn, where it is condensed. Liquid refrigerant after the condenser Cn is divided in two streams. The first liquid stream is throttled in the expansion valve EV and evaporated in the evaporator E at low pressure and temperature by extracting heat from the compressor intake air. The second liquid stream is pumped to the generator G where liquid refrigerant is heated and evaporated at high pressure by removing heat from the exhaust gas.

The application of EWRS provides decreasing the temperature of ambient air at the compressor inlet and reducing the compressor power input and growing the gas turbine power output as a result.

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 depends on the boiling temperatures of high pressure refrigerant in the generator t_G and low pressure refrigerant in the evaporator-intake air cooler t_0 (value of ζ increases with growing of t_G and t_0) and condensing temperature t_c (value of ζ drops with arising of t_c).

Variations of the coefficient of performance ζ of EWRS, specific heat load (related to the air mass flow through the compressor) on the generator (specific heat removed from the exhaust gas) $\bar{q}_G$ and refrigeration capacity $\bar{q}_0$ (specific heat removed from the compressor intake air), the compressor intake air temperature drop Δt_{air} versus the boiling temperature of high pressure refrigerant in the generator t_G are provided in Fig. 2. The calculations have been performed for the temperatures of exhaust gas at the inlet of generator t_{exh} = 500ºC and at the outlet of generator t_{exh2} = 100ºC 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.

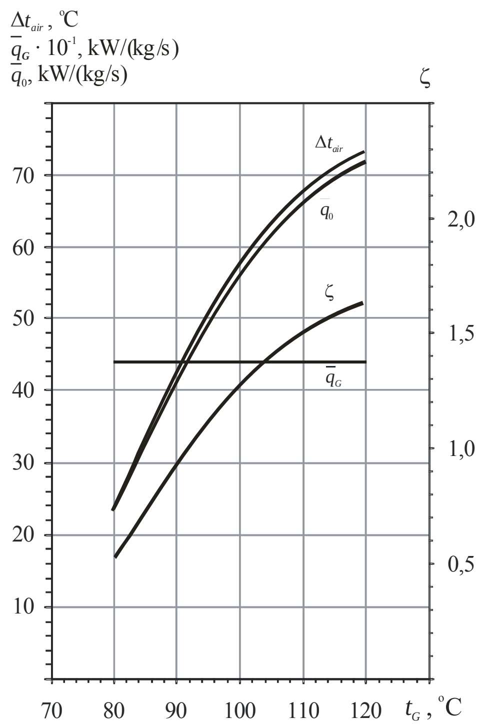
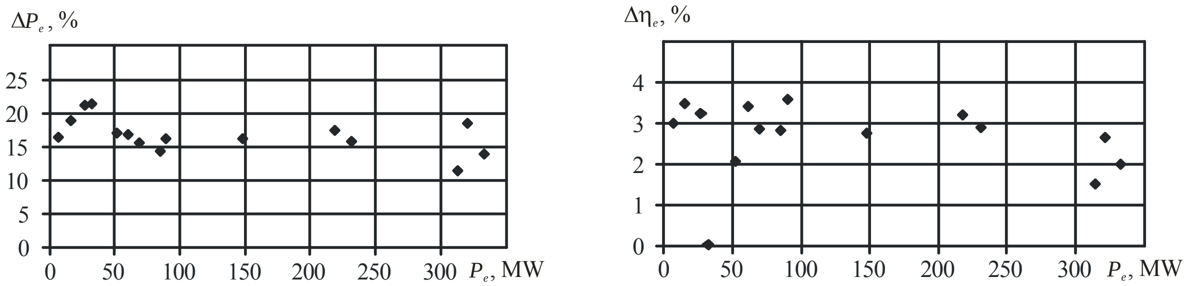


Fig. 2. Variation of the coefficient of performance ζ of EWRS, specific heat load on the generator (heat removed from the exhaust gas) $\bar{q}_G$ and refrigeration capacity $\bar{q}_0$ (heat removed from the compressor intake air), the compressor intake air temperature drop Δt_{air} versus the boiling temperature of high pressure refrigerant in the generator t_G for the temperatures of exhaust gas at the inlet of generator t_{exh} = 500ºC and at the outlet of generator t_{exh2} = 100ºC and boiling temperature of low pressure refrigerant in the evaporator t_0 = 0 °C

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. The maximum value Δt_{air} = 70ºC is achieved at the highest refrigerant boiling temperature in the generator t_G = 120 ºC. Thus, less than 30% of refrigeration capacity of EWRS, that is less than 30 % of the total gas turbine exhaust gas heat, is required to cool the ambient air at the compressor intake from initial ambient temperature of 40°C to the ISO temperature of 15 °C. The compressor intake air temperature drop of Δt_{air} = 25 °C results in the power output bust and increase in thermal efficiency of gas turbines referred to these parameters at the ambient temperature of 40°C (without refrigeration).

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 40°C to the ISO temperature of 15 °C, expressed in percent with respect to these parameters at the ambient temperature of 40°C (without refrigeration), against gas turbine power output P_e at ISO ambient temperature of 15 ºC for the set of 15 gas turbines considered are presented in Fig. 3(**a)** and 3(**b)** respectively.



(a) **(b)**

Figure 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 40 °C to the ISO temperature of 15 °C, referred to these parameters at the ambient temperature of 40 °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 30 % of the total exhaust gas heat to pre-cool the compressor intake air by 25 °C, provides the power output bust ΔP_e by about 15–20 % and increase in thermal efficiency $\Delta\eta_e$ of gas turbines by 2.0–3.5 %.

CONCLUSIONS

1. An assessment of applying the ejector refrigeration system recovering the exhaust heat of gas turbines for pre-cooling compressor intake air for a set of 15 gas turbines with power output ranging from 5 MW to 260 MW was undertaken. The application of EWRS, utilizing about 30 % of the total exhaust gas heat to pre-cool the compressor intake air by 25 °C, provides the power output bust ΔP_e by about 15–20 % and increase in thermal efficiency $\Delta\eta_e$ of gas turbines by 2.0–3.5 %.
2. The influence of performance parameters of EWRS on the compressor intake air temperature drop was investigated. 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. The maximum value Δt_{air} = 70 ºC is achieved at the highest refrigerant boiling temperature in the generator t_G = 120 ºC.

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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 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.
5. 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.