EFFECTIVE EVAPORATOR CIRCUITS OF WASTE HEAT RECOVERY REFRIGERATION SYSTEMS FOR PRE-COOLING GAS TURBINE INLET AIR

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ABSTRACT: This paper presents an assessment of applying the recirculation of liquid refrigerant in evaporators of refrigeration system recovering the exhaust heat of gas turbines for pre-cooling gas turbine compressor intake air. The influence of refrigerant mass velocity on thermal efficiency of evaporators was considered and the results of calculation of its optimum value providing the maximum heat flux in the evaporators were presented.

INTRODUCTION

A performance of gas turbines is very sensitive to ambient temperature. Thus, every 10 °C of ambient temperature arise causes gas turbine power output dropping by about 5–9 % [1, 2]. To pre-cool air at the gas turbine compressor inlet at high ambient temperature the waste heat recovery refrigeration systems (WHRS) can be used [3]. Such systems consist of mainly heat exchangers with phase change of refrigerant: a generator where the exhaust heat of gas turbine is used to evaporate liquid refrigerant at high pressure; a condenser where high pressure refrigerant vapour is condensed and an evaporator where liquid refrigerant is evaporated at low pressure by taking a heat from the compressor intake air. To avoid a possible refrigerant leakage from the evaporator in the air suction line of compressor an intermediate water circuit can be applied as shown in Fig. 1. In this case the evaporator is applied to chill water which is then used for extracting the heat from the compressor intake air in air cooler.

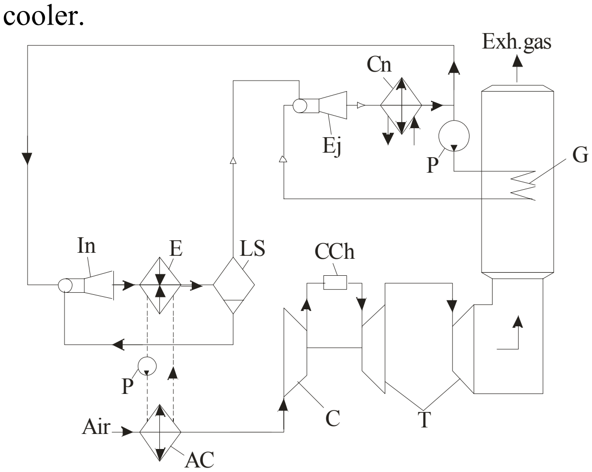


Fig. 1. Schematic diagram of the gas turbine with ejector waste heat recovery refrigeration system: AC – air cooler; C − air compressor; CCh – combustion chamber; T – turbine; G – refrigerant generator; E − evaporator; Ej − vapour ejector; In – liquid injector; LS – liquid separator; P – pump; Cn − condenser; - - - – intermediate water cooling circuit

In the evaporator a boiling refrigerant can flow inside tubes or through the channels outside tubes. At high velocity of refrigerant the performance of such evaporator is followed by sharp reduction in the intensity of heat transfer at the final stage of refrigerant phase change. A dry out of the wall surface of the tubes (channels) with transition of the boiling refrigerant flow from annular regime to disperse regime when so called "burnout" takes place is the reason for this [4, 5]. It should be noted that a "burnout" effect takes place for most of refrigerants.

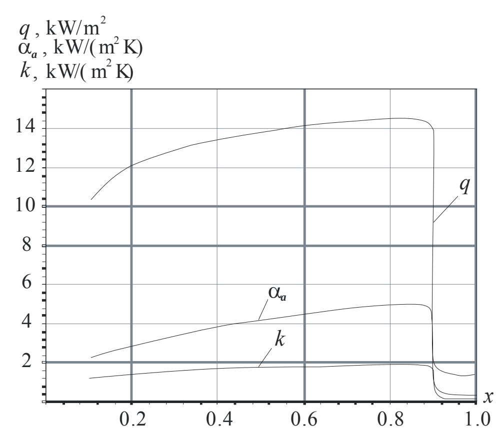
A drop in the intensity of heat transfer to boiling refrigerant at the final stage of phase change causes a decrease in the total heat transfer intensity that results in lowering the efficiency of WHRS performance. To avoid this incomplete evaporation of refrigerant by recirculating a liquid refrigerant in the evaporator might be used.

This paper presents an assessment of incomplete evaporation of refrigerant on thermal efficiency of evaporators for WHRS.

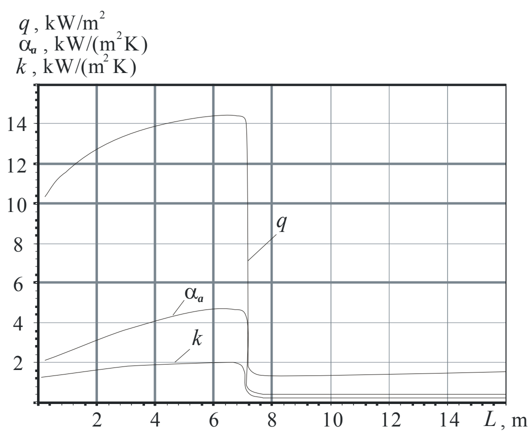
REFRIGERANT RECIRCULATION EVAPORATOR CIRCUITS FOR WASTE HEAT RECOVERY REFRIGERATION SYSTEMS

The incomplete phase change of refrigerant could be performed by employing the evaporator with injector liquid recirculation circuit as shown Fig. 1. The liquid separator has to be used to separate unevaporated liquid refrigerant from the vapour.

Intensification of heat transfer to boiling refrigerant results in an increase in the heat flux q, i.e. reduction in the evaporator surface, or in a decrease in the temperature difference θ between water being cooled and boiling refrigerant in the evaporator of the same dimensions, i.e. a reduction in the energy losses caused by an external irreversibility in the refrigeration cycle. To estimate the thermal efficiency of evaporators the variation of local thermal characteristics with the refrigerant vapour mass fraction x (Fig. 2(a)) and the refrigerant channel (tube) length L (Fig. 2(b)) has been considered. Calculations have been performed for the evaporator with water temperature at the inlet t_{w1} = 8 °C and outlet t_{w2} = 5 °C, refrigerant boiling temperature at the exit t_{02} = 0 °C and refrigerant R142B.



(a)



(b)

Fig. 2. Variation of the heat flux q, heat transfer coefficient to refrigerant α_a and total heat transfer coefficient k with the refrigerant vapour mass fraction (***a***) and channel length L (b)

As Fig. 2(a) shows a sharp decrease in the heat flux q occurs at burnout refrigerant mass vapour fraction $x_b \approx 0{,}9$ corresponding to drying the channel wall surface with the transition from annular to disperse flow of refrigerant. The reduction in q is due to subsequent lowering the heat transfer coefficient to refrigerant α_a which causes a decrease in the total heat transfer coefficient k.

Fig. 2(b) shows that the channel length L corresponding to evaporation of dispersed liquid droplets can be of about half a channel length.

A comparison of the thermal efficiency of the evaporators is usually carried out at maximum heat flux q_{max} for each case. The correlation for heat flux is

$$q = k\,\theta = \frac{1}{\dfrac{1}{\alpha_a} + \dfrac{1}{\alpha_w}} \cdot \frac{t_{w1} - t_{w2} + \Delta t_0}{\ln \dfrac{t_{w1} - t_{02}}{t_{w2} - \left(t_{02} + \Delta t_0\right)}}$$

with logarithmic temperature difference θ, heat transfer coefficients to refrigerant α_a and water α_w.

The drop in the boiling temperature Δt_0 caused by the pressure drop for two phase flow ΔP is obtained from Clausius−Clapeyron relationship

$$\frac{dP}{dt_0} = \frac{r}{T\left(\upsilon_v - \upsilon_l\right)} \approx \frac{\Delta P}{\Delta t_0}$$

The existence of maximum heat flux q_{max} is caused by the following. With increasing mass velocity of refrigerant ρw the heat transfer coefficient to refrigerant α_a and total heat transfer coefficient k increases. But the refrigerant pressure drop ΔP and corresponding refrigerant boiling temperature drop Δt_0 increases also. In conventional practice of optimum evaporator designing the value of refrigerant boiling temperature t_{02} at the evaporator exit is fixed to keep the other points of refrigerant cycle invariable [6]. With fixed t_{02} the increase in Δt_0 causes the increase in refrigerant boiling temperature t_{01} at the evaporator inlet and decrease in logarithmic temperature difference θ between water to be cooled and boiling refrigerant as a result. Such opposite influence of the refrigerant mass velocity ρw upon k and θ causes the existence of maximum of function $q = k\theta$ at quite definite value of ρw. This value is considered as optimum $(\rho w)_{opt}$.

The results of calculation of optimum value of refrigerant mass velocity ρw providing the maximum heat flux q in the evaporator are presented in Fig. 3.

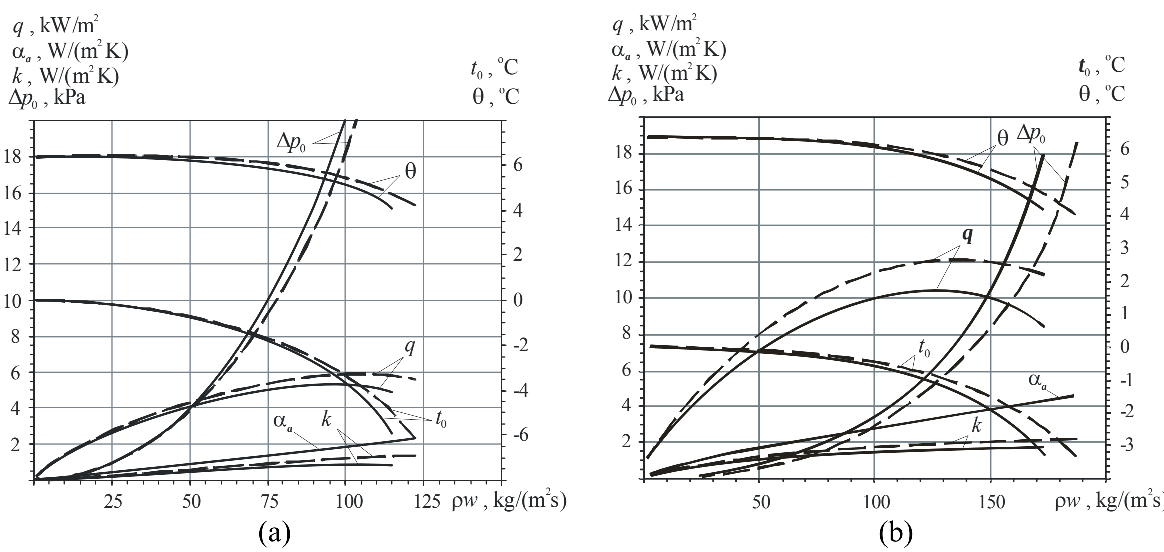


Fig. 3. Variation of heat transfer coefficient to refrigerant α_a, total heat transfer coefficient k, refrigerant pressure drop ΔP, boiling temperature t_0 and logarithmic temperature difference θ versus refrigerant mass velocity ρw in evaporator: (a) – without liquid refrigerant recirculation; (b) – with recirculation; t_{w1} = 8 °C; t_{w1} = 5 °C; —— – w_w = 1.0 m/s; - - - – w_w = 1.5 m/s

As shown with increasing ρw the values of heat transfer coefficient to refrigerant α_a and total heat transfer coefficient k increase. But arising the refrigerant pressure drop ΔP and corresponding refrigerant boiling temperature drop Δt_0 causes a reduction of logarithmic temperature difference θ.

The optimum value of refrigerant mass velocity $(\rho w)_{opt} \approx 110$ kg/(m^2 s) corresponds to the maximum heat flux $q_{max} = 6$ kW/m^2 (at water velocity $w_w = 1.5$ m/s) in the conventional evaporator with complete evaporation of refrigerant (Fig. 3(a)) and $(\rho w)_{opt} \approx 140$ kg/(m^2 s) refers to $q_{max} = 12$ kW/m^2 for evaporator circuit with liquid refrigerant recirculation (Fig. 3(b)). Thus, the intensification of heat transfer in evaporator due to elimination of dry wall regime by liquid refrigerant recirculation results about twice increasing heat flux.

CONCLUSIONS

1. The analyses of variation in thermal parameters of the evaporators with inside channel refrigerant boiling exposes a significant influence of refrigerant mass velocity and the intensity of heat transfer to boiling refrigerant at the final stage of phase change upon the thermal efficiency of evaporators.
2. The application of evaporator circuit with recirculation of liquid refrigerant providing intensive heat transfer on the whole evaporator surface has been proposed for the refrigeration systems recovering the exhaust heat of gas turbines for pre-cooling gas turbine compressor intake air.

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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. Radchenko, R.N. [2008], Increasing the efficiency of marine diesel engines by intake air cooling in waste heat recovery refrigeration systems, *Proceedings, 12th International Symposium on Heat Transfer and Renewable Sources of Energy "HTRSE−2008"*, Szczecin, Poland, pp. 553−560.
4. Chaddock, J.B., Varma, H.K. [1979], An experimental investigation on dry-out with R22 evaporating in a horizontal tube, *ASHRAE Transactions*, Vol. 85, pp. 105–121.
5. Hewitt, G. F., Govan, A.H. [1990], Phenomenological modelling of non-equilibrium flows with phase change, *Int. J. Heat mass transfer*, Vol. 33, pp. 243−252.
6. Slipcevic, B. [1972], Wärmeübergang beim sieden von R-Kältemitteln in horizontalen rohren, *Kältetechnik–Klimatisierung*, No. 12, s. 345–351.