THE POWER PARAMETERS OF A GAS TURBINE WITH REGENERATION

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The operating mode of a gas turbine depends not only upon the intake air parameters but also upon a degree of regeneration. The paper analyses the effect of a regeneration degree on the operation of a considered GT 750-6 gas turbine using a special software developed for this purpose. Based on case calculations, the effect of a degree of regeneration on the specific power and thermal efficiency of the actual GT 750-6 gas turbine is quantified.

Keywords: *degree of regeneration, gas turbine, specific power, thermal efficiency, thermal balance, heat exchanger*

1. Introduction

Cycle of a gas turbine with regeneration is formed from a simple Ericson-Bryton cycle of the gas turbine by introducing a regenerative heat exchanger, in which the compressed air from the air compressor is preheated by means of heat taken away from flue gases. In this way of heat recovery, a necessary input of heat from fuel added in the combustion chamber reduced, if a fixed temperature at the turbine inlet is assumed, thus increasing thermal efficiency of the gas turbine cycle. A temperature decline is needed for heat transfer in the heat exchanger so far the heat exchange surfaces are of finite dimensions. A degree of regeneration influences not only the thermal efficiency and specific work of the gas turbine cycle but also it influences the heat exchanger dimensions.

2. Efficiency of a regenerative heat exchanger

The analysis of the cooled gas turbine is based on an arrangement illustrated in Fig.1. The corresponding gas turbine cycle is shown in Fig. 2 on the T-s diagram. From this figure the numeric assignment of different points to which individual thermodynamic values refer is visible. Supposing that the gas mass flow is constant, the efficiency of regenerative heat exchanger is defined as a ratio of heat transferred to the air being heated to that which would be transferred by the ideal endlessly big exchanger without losses

$$\eta_\mathrm{R} = \frac{h_{03} - h_{02}}{h_{05} - h_{02}} \tag{1}$$

where: h_0 – total enthalpy (in points 2, 3, 5 – see Fig. 1).

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Provided that the isobaric specific heat capacity is constant, equation (1) is as follows

$$\eta_{\mathrm{R}} = \frac{T_{03} - T_{02}}{T_{05} - T_{02}} \tag{2}$$

where: T_0 – total temperature (in points 2, 3, 5 – see Fig. 1).

Ambient heat loss due to radiation being ignored, from the heat balance of a regenerative heat exchangers it follows

$$\frac{\dot{Q}_{5\mathrm{r}}}{\dot{m}_{\mathrm{K}}} = \frac{\dot{m}_{\mathrm{T}}}{\dot{m}_{\mathrm{K}}}\,\bar{c}_{\mathrm{p,sp,R}}\,(t_{05} - t_{06}) = \frac{\dot{m}_2}{\dot{m}_{\mathrm{K}}}\,\bar{c}_{\mathrm{p,vz,R}}\,(t_{03} - t_{02}) = \frac{\dot{Q}_{2\mathrm{r}}}{\dot{m}_{\mathrm{K}}} \tag{3}$$

where: $\dot{m}_{\mathrm{T}}$ – mass flow of flue gases at the regenerative heat exchanger inlet, $\dot{m}_{\mathrm{K}}$ – mass flow of flue air at the compressor inlet, $\dot{m}_2$ – mass flow of flue air at the regenerative heat exchanger inlet, $\bar{c}_{\mathrm{p,sp,R}}$ – mean value of the isobaric specific heat capacity of flue gases (for a temperature range of $t_{06} - t_{05}$), $\bar{c}_{\mathrm{p,vz,R}}$ – mean value of the isobaric specific heat capacity of air (for a temperature range of $t_{02} - t_{03}$), $\dot{Q}_{5\mathrm{r}}$ – heat recovered from flue gas, $\dot{Q}_{2\mathrm{r}}$ – heat absorbed by air.

Considering the ambient heat radiation, an unbalance ratio of thermal balance of a regenerative heat exchanger is calculated

$$\Delta = \frac{\dot{Q}_{5\mathrm{r}} - \dot{Q}_{2\mathrm{r}}}{\dot{Q}_{5\mathrm{r}}}\,100\ . \tag{4}$$

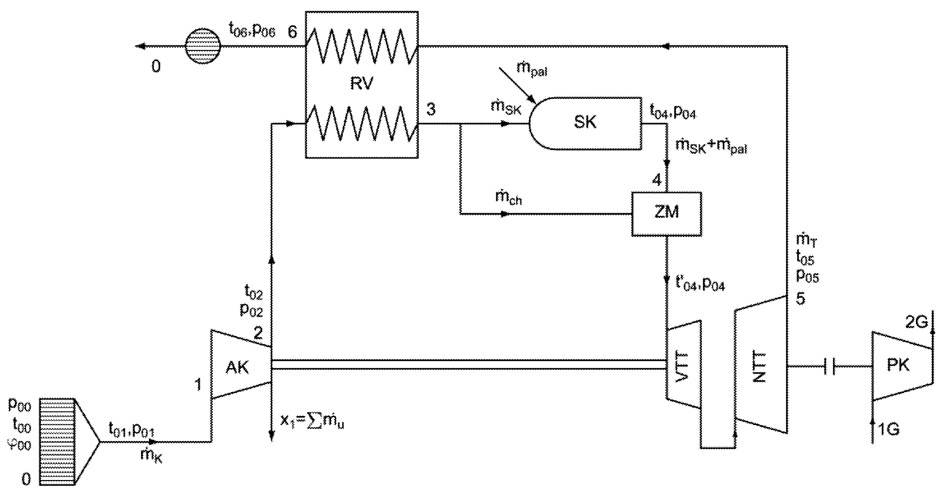

Fig. 1: A scheme of a gas turbine with regeneration (AK – air compressor, RV – regenerative exchanger, SK – combustion chamber, ZM – mixer, VTT – gas producer turbine, NTT – power turbine, PK – gas compressor)

3. Specific power output and specific cycle heat

The effective power of a gas turbine is defined as a sum of turbine power (VTT, NTT – see Fig. 1) and air compressor power – negative (AK – see Fig. 1)

$$P_{\mathrm{uz}} = \dot{m}_{\mathrm{T}}\,(h_{04} - h_{05})\,\eta_{\mathrm{m,T}} + \dot{m}_{\mathrm{K}}\,(h_{01} - h_{02})\,\frac{1}{\eta_{\mathrm{m,K}}}\ , \tag{5}$$

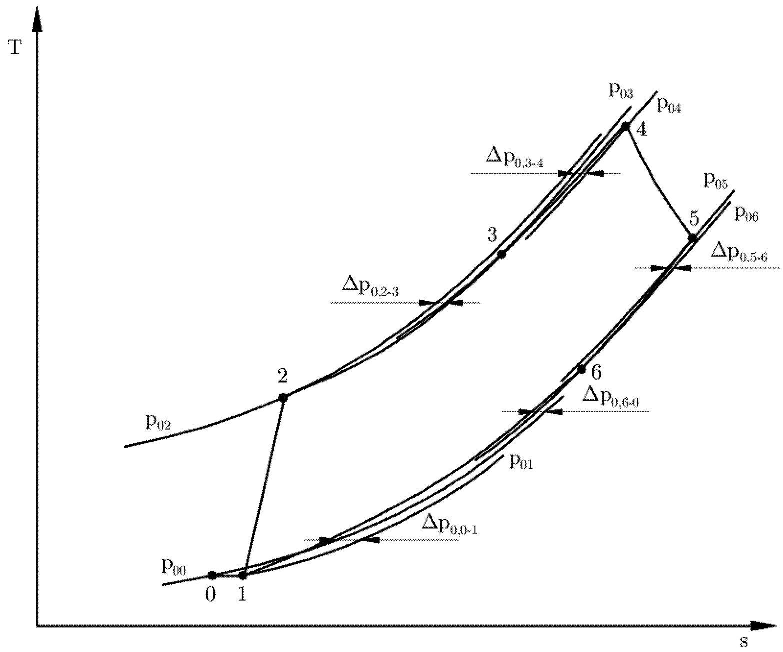

Fig.2: Gas turbine cycle with heat recovery on the T-s diagram

after rearranging

$$P_{\mathrm{uz}} = \dot{m}_{\mathrm{T}}\, \bar{c}_{\mathrm{p,T}}\, (T_{04} - T_{05})\, \eta_{\mathrm{m,T}} - \dot{m}_{\mathrm{K}}\, \bar{c}_{\mathrm{p,K}}\, (T_{02} - T_{01})\, \frac{1}{\eta_{\mathrm{m,K}}} \tag{6}$$

where: $\dot{m}_{\mathrm{T}}$ – mass flow of flue gases at the turbine inlet, $\eta_{\mathrm{m,T}}$ – mechanical efficiency of turbine, $\eta_{\mathrm{m,K}}$ – mechanical efficiency of air compressor, $\bar{c}_{\mathrm{p,T}}$ – mean value of the isobaric specific heat capacity of flue gases in turbine, $\bar{c}_{\mathrm{p,K}}$ – mean value of the isobaric specific heat capacity of air in compressor, T_0 – total temperature (in points 4, 5, 1, 2 – see Fig. 1), h_0 – total enthalpy (in points 4, 5, 1, 2 – see Fig. 1).

The specific power of a gas turbine is derived from a detail calculation of heat cycle, indirectly giving the compressor inlet-air mass flow necessary for the effective power required

$$P' = \frac{P_{\mathrm{uz}}}{\dot{m}_{\mathrm{K}}\, \bar{c}_{\mathrm{p,K}}\, T_{01}} = \left(\frac{\dot{m}_{\mathrm{T}}\, \bar{c}_{\mathrm{p,T}}}{\dot{m}_{\mathrm{K}}\, \bar{c}_{\mathrm{p,K}}}\right) A\, B\, \eta_{\mathrm{m,T}} - C\, \frac{1}{\eta_{\mathrm{m,K}}} \;, \tag{7}$$

$$A = \frac{T_{04} - T_{05}}{T_{04}} \;, \tag{8}$$

$$B = \frac{T_{04}}{T_{01}} \;, \tag{9}$$

$$C = \frac{T_{02} - T_{01}}{T_{01}} \;. \tag{10}$$

Instead of temperatures, an expansion ratio of turbine as a function of compressor pressure ratio and of pressure loss in different components of a gas turbine can be incorporated in this equation

$$P' = \frac{\dot{m}_{\mathrm{T}}\, \bar{c}_{\mathrm{p,T}}}{\dot{m}_{\mathrm{K}}\, \bar{c}_{\mathrm{p,K}}}\, \frac{T_{04}}{T_{01}} \left[1 - \left(\frac{p_{05}}{p_{04}}\right)^{\frac{\kappa_{\mathrm{sp}}-1}{\kappa_{\mathrm{sp}}}\, \eta_{\mathrm{pol,T}}}\right] \eta_{\mathrm{m,T}} - \left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\kappa_{\mathrm{vz}}-1}{\kappa_{\mathrm{vz}}\, \eta_{\mathrm{pol,K}}}} - 1\right] \frac{1}{\eta_{\mathrm{m,K}}} \;, \tag{11}$$

after rearranging

$$P' = \frac{\dot{m}_\mathrm{T}\,\bar{c}_\mathrm{p,T}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\left[1-\left\{\left[1-\sum\left(\frac{\Delta p_0}{p_0}\right)\right]\pi_\mathrm{K}\right\}^{-\frac{\kappa_\mathrm{sp}-1}{\kappa_\mathrm{sp}}\eta_\mathrm{pol,T}}\right]\frac{T_{04}}{T_{01}}\,\eta_\mathrm{m,T} - \\ -\left[\pi_\mathrm{K}^{\frac{\kappa_\mathrm{vz}-1}{\kappa_\mathrm{vz}\,\eta_\mathrm{pol,K}}}-1\right]\frac{1}{\eta_\mathrm{m,K}} \tag{12}$$

where: π_K – pressure ratio of air compressor, $\eta_\mathrm{pol,T}$ – polytropic efficiency of turbine, $\eta_\mathrm{pol,K}$ – polytropic efficiency of air compressor, κ_sp – adiabatic exponent of flue gases in turbine, κ_vz – adiabatic exponent of air in compressor, p_0 – total pressure (in points 4, 5, 1, 2 – see Fig. 1), $\Delta p_0/p_0$ – relative ratio of pressure loss.

In comparison with the ideal cycle of a gas turbine, analysis of the actual one is complicated because the working substance has nor constant composition neither constant thermodynamic properties during the whole cycle. There are pressure losses both in heat exchanger and connector pipes, which together with a different mass flow (an impact of taking air for cooling, flooding of air compressor seals and an intake of cooling air into flue gas flow) operating in different components of gas turbine often have a substantial impact on the characteristic parameters of cycle.

The specific external heat added into the cycle shows a ratio of the external heat brought into a combustion chamber SK (Fig. 1) to that of the heat of the air brought by compressor inlet air

$$\dot{Q}' = \frac{q}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}\,T_{01}}\ . \tag{13}$$

The external heat addition in a gas turbine cycle is given by the equation

$$q = \dot{m}_\mathrm{SK}\,(h_{04}-h_{03}) \tag{14}$$

after rearranging

$$q = \dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}\,(T_{04}-T_{03}) \tag{15}$$

where: $\dot{m}_\mathrm{SK}$ – air mass flow at the combustion chamber inlet, $\bar{c}_\mathrm{p,SK}$ – mean value of the isobaric specific heat capacity of flue gases in the combustion chamber.

Inserting equations (15) and (2) into equation (13)

$$\dot{Q}' = \frac{\dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\left[\frac{T_{04}}{T_{01}}-\eta_\mathrm{R}\left(\frac{T_{05}-T_{02}}{T_{01}}\right)-\frac{T_{02}}{T_{01}}\right] = \\ = \frac{\dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\left[\frac{T_{04}}{T_{01}}-\eta_\mathrm{R}\frac{T_{04}}{T_{01}}\left(\frac{T_{05}-T_{04}}{T_{04}}\right)-\eta_\mathrm{R}\frac{T_{04}}{T_{01}}+(\eta_\mathrm{R}-1)\frac{T_{02}}{T_{01}}\right] \tag{16}$$

after rearranging we obtain

$$\dot{Q}' = \frac{\dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\left\{B\left[1-\eta_\mathrm{R}\,(1-A)\right]-(1+C)\,(1-\eta_\mathrm{R})\right\} \tag{17}$$

where: A, B, C parameters – see equations (8), (9), (10), η_R – efficiency of a regenerative exchanger (RV – see Fig. 1).

4. Thermal efficiency of a gas turbine

An important indicator of the economy of the energy transfer into mechanical work is the thermal efficiency defined as a specific power – to – specific heat ratio

$$\eta_\mathrm{t} = \frac{P'}{\dot{Q}'} \ . \tag{18}$$

After substitution and modification we obtain

$$\eta_\mathrm{t} = \frac{\frac{\dot{m}_\mathrm{T}\,\bar{c}_\mathrm{p,T}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\frac{T_{04}}{T_{01}}\left[1-\left\{\left[1-\sum\left(\frac{\Delta p_0}{p_0}\right)\right]\pi_\mathrm{K}\right\}^{-\frac{\kappa_\mathrm{sp}-1}{\kappa_\mathrm{sp}}\,\eta_\mathrm{pol,T}}\right]\eta_\mathrm{m,T}-\left[\pi_\mathrm{K}^{\frac{\kappa_\mathrm{vz}-1}{\kappa_\mathrm{vz}\,\eta_\mathrm{pol,K}}}-1\right]\frac{1}{\eta_\mathrm{m,K}}}{\frac{\dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\left\{\frac{T_{04}}{T_{01}}\left[1-\eta_\mathrm{R}\left(1-\frac{T_{04}-T_{05}}{T_{04}}\right)\right]-\left(1+\frac{T_{02}-T_{01}}{T_{01}}\right)(1-\eta_\mathrm{R})\right\}} \ . \tag{19}$$

To simplify this equation following symbols skill be used

$$A = \frac{T_{04}-T_{05}}{T_{04}} = 1-\left\{\left[1-\sum\left(\frac{\Delta p_0}{p_0}\right)\right]\pi_\mathrm{K}\right\}^{-\frac{\kappa_\mathrm{sp}-1}{\kappa_\mathrm{sp}}\,\eta_\mathrm{pol,T}} \ , \tag{20}$$

$$C = \frac{T_{02}-T_{01}}{T_{01}} = \pi_\mathrm{K}^{\frac{\kappa_\mathrm{vz}-1}{\kappa_\mathrm{vz}\,\eta_\mathrm{pol,K}}} - 1 \ . \tag{21}$$

Now the equation (19) becomes

$$\eta_\mathrm{t} = \frac{\frac{\dot{m}_\mathrm{T}\,\bar{c}_\mathrm{p,T}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\,A\,B\,\eta_\mathrm{m,T} - C\,\frac{1}{\eta_\mathrm{m,K}}}{\frac{\dot{m}_\mathrm{SK}\,\bar{c}_\mathrm{p,SK}}{\dot{m}_\mathrm{K}\,\bar{c}_\mathrm{p,K}}\,\{B\,[1-\eta_\mathrm{R}\,(1-A)]-(1+C)\,(1-\eta_\mathrm{R})\}} \tag{22}$$

where the B parameter – see equation (9). The A parameter is a function of pressure losses and pressure ratio of the air compressor, the B parameter defines a ratio of the total temperature of flue gases at the turbine inlet and the total temperature of air at inlet into an air compressor and the C parameter is a function of the pressure ratio of the air compressor.

5. Results

Derived equations have been verified by analysing the real cycles of gas turbines in operation based on the results from technical measuring. A degree of unbalance in the heat balance of a regenerative heat exchanger calculated according to equation (4) showed a value of 1,44 %. In investigating an effect of a regeneration degree, defined by the efficiency of a regenerative heat exchanger, on the thermal efficiency and the specific power output an arrangement of a GT 750-6 gas turbine is considered. Figure 3 shows that regeneration has a greater effect on improving the thermal efficiency at lower pressure ratio. Pressure ratio at which the thermal efficiency amounts to its maximum shifts with an increasing degree of regeneration (η_R) towards lower values.

The influence of a regeneration degree on the specific power output (Fig. 4) is negligible on condition that due to a change in regeneration degree, there are no changes in pressure losses in the regeneration heat exchanger as well in other parts of a the turbine. The maximum value of the specific power output corresponds to various values of the thermal

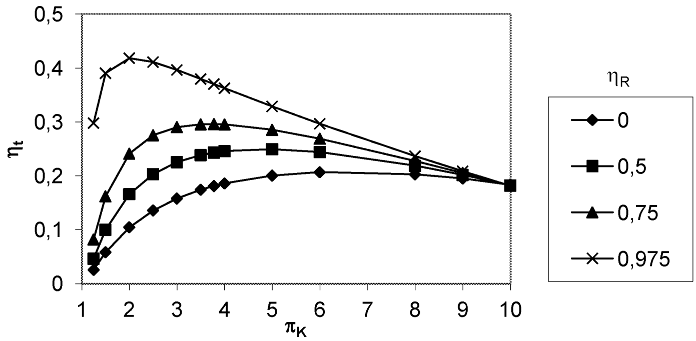

Fig.3: Thermal efficiency dependence on pressure ratio ($T_{04}/T_{01} = 3.5$)

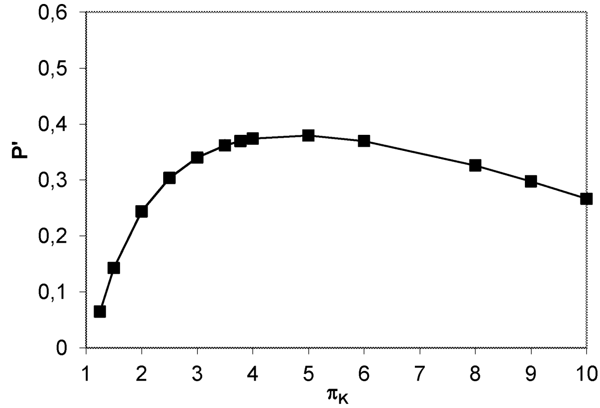

Fig.4: Specific power output dependence on pressure ratio ($T_{04}/T_{01} = 3.5$)

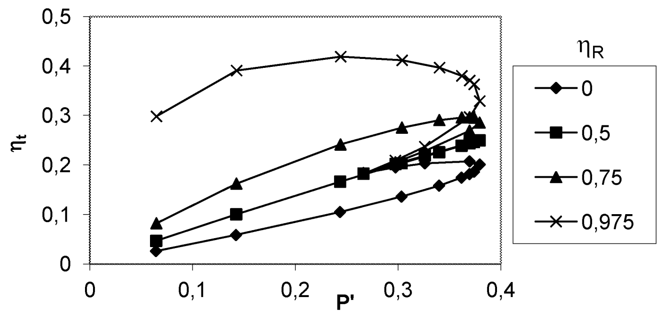

Fig.5: Dependence of thermal efficiency on specific power output ($T_{04}/T_{01} = 3.5$)

efficiency (Fig. 5) depending on a degree of regeneration. In case without regenerative heat exchanger ($\eta_R = 0$) the maximum specific power output is achieved at lower compression, as compared to that at which the maximum thermal efficiency is obtained (Figs. 3 and 4). Increasing the pressure ratio above the value at which the maximum thermal efficiency is achieved brings about an increase in a temperature of compressed air (and the needed of heat input into the combustion chamber reduces), thereby increasing the thermal efficiency until the compression limit. If compression continues to increase, the thermal efficiency shows a fall (Fig. 5), since there is a decrease in the adiabatic efficiency of air compressor and gas turbine as wheel, and the potential of thermal energy contained in flue gases at the gas turbine inlet is reduced.

If values of a degree of regeneration rise, the maximum thermal efficiency is achieved at lower pressure ratio and a subsequent increase in pressure ratio results in decreasing the thermal efficiency although the specific power output increases. It is caused by less usage of the potential of thermal energy contained in flue gases at the regeneration heat exchanger inlet.

6. Conclusions

The paper presents the derived computing relationship, which are derived in such a way that they are as universal as possible and suitable for software processing. Based on the developed software, in a case study an effect of a regeneration degree on the power parameters of the cycle of a gas turbine GT 750-6 is analysed. The computing results suggest that in designing a gas turbine it implement the optimization not only in terms of the maximum thermal efficiency but also the maximum specific power output.

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