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Optimization of the thermodynamic and thermophysical properties of the gas turbine cycle working fluid

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Abstract. A gas turbine cycle working fluids group based on their production technology and application is presented. The energy parameters analysis procedure of unclosed and closed gas turbine cycles is described. The comparison of the effect of the working fluids thermophysical and thermodynamic properties variations on the useful work l_0 of both unclosed and closed gas turbine cycles is presented. The effect of the fuel gas composition variation on the useful cycle work l_0 is revealed. The effect of the working fluid thermophysical properties variation on the gas turbine cycle optimal combination of thermodynamic properties (T_3 and P_3) is presented.

1. Introduction

Artificial gases is an air-blown or oxygen-blown syngas, blast-furnace gas, coke-oven gas, etc. Energy efficiency of the unclosed gas turbine cycle in an artificial gas-fired CCPP is more determined by the gas turbine work l_{T} than by the compressor work l_{C}. This is due to not only the possibility to vary the thermodynamic properties of working fluids (T_3 and P_3), but also possibility to correct the gas turbine working fluid thermophysical properties by varying the fuel gas properties.

The methods of correction of the gas turbine working fluid composition, identified during the analysis of flow diagram and operating regimes production and developed artificial gas-fired CCPP, are listed below:

1. gas fuel or oxidizer (air or O_2) dilution by nitrogen, water or CO_2;
2. enrichment of gas fuel (by coke-oven or natural gas) and oxidizer (by O_2);
3. removal of the least energy-intensive component CO_2 by pre-combustion CCS;
4. changing the thermal regime for gas fuel and oxidizer preparation.

Working fluid composition correction leads to the working fluid thermophysical properties variations. The working fluid thermophysical properties variations influence to the gas turbine work l_{T} and the cycle useful work l_0.

In this paper a comparison of the effect of the working fluids thermophysical and thermodynamic properties variation on the unclosed and closed gas turbine cycle useful work l_0.

2. Method

A variety of working fluid compositions formed on the basis of known fluids enlarged can be divided into four types, allocated according to the technology of their production and/or application:

- Type A (from "Air"): the working fluids as being the air combustion products of natural and alternative gases. It is used in unclosed gas turbines cycle both for production and development of natural gas-fired CCPP, IGCC and artificial gas-fired CCPP [1, 2];

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- Type H (from "Hydrogen"): the working fluids as being the O_2-H_2O combustion products of syngas derived from coal by O_2-H_2O conversion and pre-combustion of CCS. It is used in the developments of semi-closed gas turbine and steam-turbine cycles [3, 4];
- Type C (from "Carbon"): working fluids as being the O_2-CO_2 combustion products of syngas derived from coal by O_2-CO_2 conversion. It is used in advanced developments of a semi-closed gas turbine cycle as part of Oxy-fuel power plants [5, 6];
- Type G (from "Gas"): working fluids as being special fluids (He/Xe/N_2/Ar mixtures) used in low-power closed gas turbine cycle also known as High Temperature Reactor Helium Gas Turbine (HTR-GT) [7]

The gas turbine cycle is presented in figure 1:

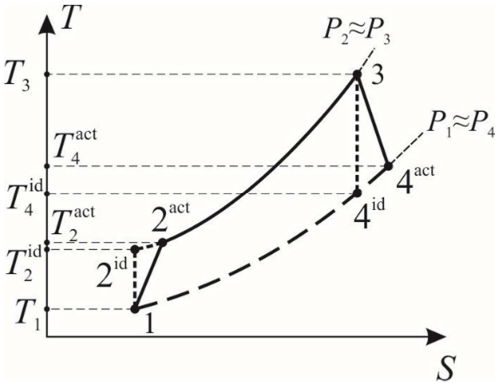

Figure 1. Gas turbine cycle: id – ideal cycle; act – actual cycle.

The ideal gas turbine cycle ($\eta_{oi}^{\mathrm{C}} = \eta_{oi}^{\mathrm{T}} = 1$) useful work l_0 is determined by the equation (1):

$$l_0 = (1 + b) \cdot l_{\mathrm{T}} - l_{\mathrm{C}} \quad [\mathrm{kJ/kg}] \tag{1}$$

l_{T}, l_{C} are gas turbine and compressor works, kJ/kg:

$$\begin{gathered} l_{\mathrm{T}} = \mathrm{c}_{pm,3} \cdot T_3 \cdot (1 - \pi_{GT}{}^{-m_3}) \quad [\mathrm{kJ/kg}] \\ l_{\mathrm{C}} = \mathrm{c}_{pm,1} \cdot T_1 \cdot (\pi_K{}^{m_1} - 1) \quad [\mathrm{kJ/kg}] \\ m_{1,3} = \frac{k_{1,3} - 1}{k_{1,3}} = \frac{R_\mu}{c_{p\mu,1,3}} < 1 \end{gathered} \tag{2}$$

π_{C}, π_{T} are compressor compression ratio and gas turbine expansion ratio. In the present work we assumed that $\pi_{\mathrm{K}} = \pi_{\mathrm{T}} = \pi$;

The heat and mass balance analysis of gas turbine revealed that the equation $\mathrm{c}_{pm,3}/\mathrm{c}_{pm,1} = 1$ is maintained with good accuracy while gas turbine operated at G- type working fluid in a closed cycle or H-type and C-type in a semi-closed cycle.

When gas turbine operated at A-type working fluid, the divergence of mass specific heat capacities for advanced gas turbine was in range of:

$$\frac{\mathrm{c}_{pm,3}}{\mathrm{c}_{pm,1}} = 1 + (0{,}1 \div 0{,}19)$$

Thus, in the first approximation, we accept that the ratio of mass specific heat $\mathrm{c}_{pm,3}/\mathrm{c}_{pm,1}$ equal to 1, i.e. $\mathrm{c}_{pm,3} = \mathrm{c}_{pm,1} = \mathrm{c}_{pm}$. Then the equation to determine the gas turbine cycle useful work l_0 takes the form:

$$l_0 = \mathrm{c}_{pm} \cdot T_1 \cdot [\xi \cdot (1 + b) \cdot (1 - \pi^{-m_3}) - (\pi^{m_1} - 1)] \quad [\mathrm{kJ/kg}] \tag{3}$$

$\xi = T_3/T_1$ is working fluid heat ratio in the gas turbine cycle;

b is fuel coefficient characterizing the difference in mass flow rates of the compressor and the gas turbine working fluids during fuel gas supply into the cycle in order to heat 1 kg of cycle air, H_2O or CO_2. The fuel coefficient b depends on the fuel gas composition, thermal regime for gas fuel and oxidizer preparation and the gas turbine class (T_3):

$$b = \frac{1}{\alpha \cdot L^0} = \frac{1}{L} = \frac{G_\mathrm{F}}{G_\mathrm{A}} \quad (4)$$

α is excess air coefficient;
L^0 is theoretical air mass flow required for complete combustion of 1 kg artificial fuel gas, kg/kg;
G_F, G_A are artificial fuel gas and air mass flow rates, kg/s.
The heat balance of the gas turbine combustion chamber is as follows:

$$\Delta t \cdot \mathrm{c}_{pm} \cdot (G_\mathrm{A} + G_\mathrm{F}) = G_\mathrm{F} \cdot Q_\mathrm{F} \cdot \eta_{CC} \quad (5)$$

$\Delta t = T_3 - T_2$ is working fluid heat rate in the combustion chamber, °C;
Q_F is fuel gas lower calorific value, MJ/kg;
η_{CC} is combustion chamber efficiency, for modern GTU it is about 0.97÷0.99. Therefore, we assume that $\eta_{CC} = 1$.
From the equation (5) it follows that:

$$b = \frac{T_1 \cdot (\xi - \pi^{m_1})}{\frac{Q_\mathrm{F}}{c_{pm}} - T_1 \cdot (\xi - \pi^{m_1})} \quad (6)$$

The coefficient b depending on parameters Q_F and ξ is presented in figure 2 at $T_1 = 288.15$ K, $m_1 = 0.286$ and $c_{pm,1} = c_{pm,3} = 1$ kJ/(kg·K).

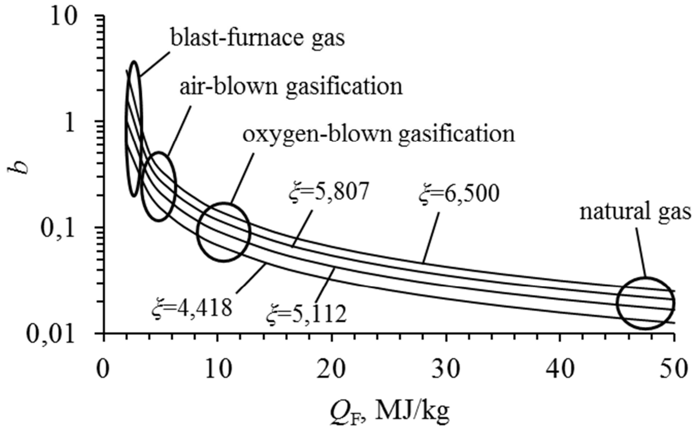

Figure 2. The coefficient b depending of the parametersQ_F and ξ at $T_1 = 288{,}15$ K, $m_1 = 0.286$ and $c_{pm,1} = c_{pm,3} = 1$ kJ/(kg·K).

The fuel coefficient b for G-type working fluids in a closed cycle is $b = 0$. The fuel coefficient b for A-type working fluids in a unclosed cycle, H-type and C-type in semi-closed cycles is $b > 0$. The coefficient b for natural gas-fired gas turbine cycles is close to 0.
The thermal efficiency η_t of the ideal ($\eta_{oi}^\mathrm{C} = \eta_{oi}^\mathrm{T} = 1$) gas turbine cycle is determined by the equation:

$$\eta_t = \frac{l_0}{q_1} = \frac{\mathrm{c}_{pm} \cdot T_1 \cdot [\xi \cdot (1+b) \cdot (1 - \pi^{-m_3}) - (\pi^{m_1} - 1)]}{\mathrm{c}_{pm} \cdot \mathrm{T}_1 \cdot {\pi_\mathrm{K}}^{m_1} \cdot (\xi - 1)} \quad (7)$$

While determining the actual gas turbine cycle energy parameters (actual useful work l_0^act and net efficiency η_i) the corresponding corrections are introduced by the above equations:

$$l_0^\mathrm{act} = (1 + b) \cdot l_\mathrm{T} \cdot \eta_{oi}^\mathrm{T} - \frac{l_\mathrm{C}}{\eta_{oi}^\mathrm{C}}; \qquad \eta_i = \frac{l_0^\mathrm{act}}{q_1} \quad (8)$$

3. Results

The optimal compression ratio π_opt, corresponding to the maximum useful work of the unclose gas turbine cycle $(l_0^\mathrm{act})_\mathrm{max}$, is found according to the standard procedure to determining the one-variable function extremum position (9):

$$\left.\frac{dl_0^\mathrm{act}}{d\pi}\right|_{\pi = \pi_\mathrm{opt}} = \mathrm{c}_{pm} \cdot T_1 \cdot \frac{d}{d\pi}\left(\xi \cdot (1 + b) \cdot \left(1 - \frac{1}{\pi^{m_3}}\right) \cdot \eta_{oi}^\mathrm{T} - (\pi^{m_1} - 1)/\eta_{oi}^\mathrm{C}\right) = 0 \quad (9)$$

The solution of equation (9) is:

$$\pi_{\text{opt}} = \left[\xi \cdot \frac{m_3}{m_1} \cdot (1+b) \cdot \eta_{oi}^{\text{T}} \cdot \eta_{oi}^{\text{C}}\right]^{\frac{1}{m_1+m_3}} \tag{10}$$

In equation (10) it is assumed that the compressor inlet air temperature is $t_1 = 15$°C ($T_1 = 288.15$ K), then the inlet air molar specific heat is $c_{p\mu,1} = 29.085$ kJ/(kmol·K) and $m_1 = 0.286$. According to equation (2), as $m_3 \to 1$ the gas turbine cycle useful work tends to the maximum value $l_0^{\text{act}} \to (l_0^{\text{act}})_{\text{max}}$.

The optimal compression ratio π_{opt} depending of the working fluids thermodynamic property ξ and the fuel coefficient b at $\eta_{oi}^{\text{T}} = 0.88$ and $\eta_{oi}^{\text{C}} = 0.86$ takes the form (11) by substituting $m_1 = 0.286$ and $m_3 = 1$ into the equation (10):

$$\pi_{\text{opt}} = [2{,}646 \cdot \xi \cdot (1+b)]^{0{,}778} \tag{11}$$

Figure 3 shows the optimal compression ratio π_{opt} depending of the working fluid thermodynamic property ξ and the fuel coefficient b at $m_3 = 1$, $\eta_{oi}^{\text{T}} = 0.88$ and $\eta_{oi}^{\text{C}} = 0.86$.

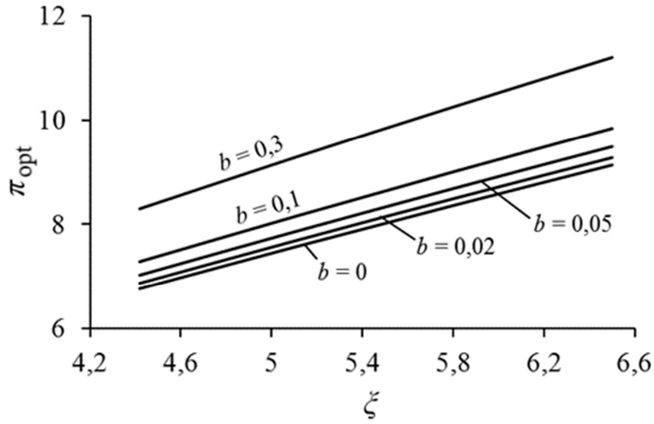

Figure 3. The optimal compression ratio π_{opt} depending of the working fluid thermodynamic property ξ and the fuel coefficient b at $m_3 = 1$, $\eta_{oi}^{\text{T}} = 0.88$ and $\eta_{oi}^{\text{C}} = 0.86$.

The analysis of calculated composition of the gas turbine working fluids of the natural and artificial gas-fired CCPP is showed that for gas turbines of class 1000÷1600°C ($\xi = 4.418 \div 6.500$) the parameter m_3 is about $0.23 \div 0.26$.

Figure 4 shows the optimal compression ratio π_{opt} depending of the working fluid thermophysical m_3 and thermodynamic ξ properties and the fuel coefficient b at $\eta_{oi}^{\text{T}} = 0.88$ and $\eta_{oi}^{\text{C}} = 0.86$

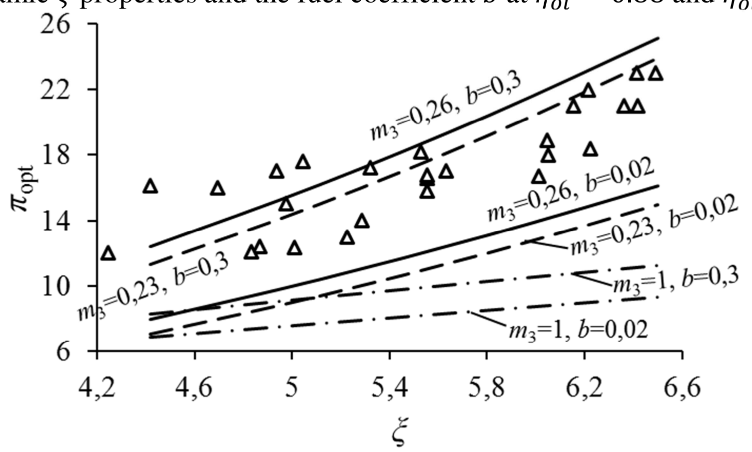

Figure 4. The optimal compression ratio π_{opt} depending of the thermophysical m_3 and thermodynamic ξ properties of the working fluids and the fuel coefficient b at $\eta_{oi}^{\text{T}} = 0.88$ and $\eta_{oi}^{\text{C}} = 0.86$: triangles – natural gas-fired gas turbine [8].

4. Conclusions

The optimal combination of π and ξ depends on the working fluid thermophysical property m_3. The optimal compression ratio π_{opt} corresponding to the maximum useful work of the gas turbine cycle $(l_0^{\mathrm{act}})_{\mathrm{max}}$ depends not only on the gas turbine working fluid heat ratio ξ (gas turbine class T_3), but also on the fuel gas composition. The fuel gas composition determines the amount of fuel gas supplied (b) into the cycle and effects the working fluid thermophysical propertym_3.

The fuel gas lower calorific value Q_{F} affects π_{opt} through the fuel coefficient b. A decrease in Q_{F} leads to an increase in π_{opt} while $\xi = \mathrm{const}$.

A decrease in Q_{F} leads to an increase in the gas turbine cycle useful work l_0 while working fluid heat ratio $\xi = \mathrm{const}$ and optimal compression ratio $\pi_{\mathrm{opt}} = \mathrm{const}$.

The influence of the fuel coefficient b on optimal compression ratio π_{opt} decreases and tends to 0 while increasing Q_{F} as like for a closed gas turbine cycle.

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