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];
- Type H (from “Hydrogen”): the working fluids as being the $\text{O}_2$-$\text{H}_2\text{O}$ combustion products of syngas derived from coal by $\text{O}_2$-$\text{H}_2\text{O}$ 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 $\text{O}_2$-$\text{CO}_2$ combustion products of syngas derived from coal by $\text{O}_2$-$\text{CO}_2$ 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 ($\eta_{id}^C = \eta_{id}^T = 1$) useful work $l_0$ is determined by the equation (1):

$$l_0 = (1 + b) \cdot l_T - l_C \ [\text{kJ/kg}]$$

(1)

$l_T$, $l_C$ are gas turbine and compressor works, kJ/kg:

$$l_T = c_{pm,3} \cdot T_3 \cdot (1 - \pi_{GT}^{-m_3}) \ [\text{kJ/kg}]$$

$$l_C = c_{pm,1} \cdot T_1 \cdot (\pi_K^{m_1} - 1) \ [\text{kJ/kg}]$$

(2)

$$m_{1,3} = \frac{k_{1,3} - 1}{k_{1,3}} = \frac{R_H}{c_{pm,1,3}} < 1$$

$\pi_C$, $\pi_T$ are compressor compression ratio and gas turbine expansion ratio. In the present work we assumed that $\pi_K = \pi_T = \pi$.

The heat and mass balance analysis of gas turbine revealed that the equation $c_{pm,3}/c_{pm,1} = 1$ 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:

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

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

$$l_0 = c_{pm} \cdot T_1 \cdot [\xi \cdot (1 + b) \cdot (1 - \pi^{-m_3}) - (\pi^{m_1} - 1)] \ [\text{kJ/kg}]$$

(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, $\text{H}_2\text{O}$ or $\text{CO}_2$. The fuel coefficient $b$ depends on the fuel gas composition, thermal regime for gas fuel and oxidizer preparation and the gas turbine class ($T_3$):
\( b = \frac{1}{\alpha} \cdot \frac{1}{l^0} = \frac{G_F}{G_A} \)  

(4)

\( \alpha \) 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:
\[
\Delta Q = Q_F + (G_A + G_F) = G_F \cdot Q_F \cdot \eta_{CC}
\]

(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;
\( \eta_{CC} \) 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:
\[
b = \frac{T_1}{c_{pm} \cdot T_1 \cdot (\xi - \pi_{m1})}
\]

(6)

The coefficient \( b \) depending on parameters \( Q_F \) and \( \xi \) 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).

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 an 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 \( \eta_t \) of the ideal \( (\eta_{CC} = 1) \) gas turbine cycle is determined by the equation:
\[
\eta_t = \frac{\eta_{0t}}{q_s} = \frac{c_{pm} \cdot T_1 \cdot \left[ \frac{1}{(1 + b \cdot (1 - \pi_{m3} - (\pi_{m1} - 1))} \right]}{c_{pm} \cdot T_1 \cdot \eta_{0t}}
\]

(7)

While determining the actual gas turbine cycle energy parameters (actual useful work \( l_0^{act} \) and net efficiency \( \eta_l \)) the corresponding corrections are introduced by the above equations:
\[
l_0^{act} = (1 + b) \cdot l_1 \cdot \eta_{0t} \cdot \frac{l_c}{\eta_{0t}}; \quad \eta_l = \frac{l_0^{act}}{q_s}
\]

(8)

3. Results

The optimal compression ratio \( \pi_{opt} \), corresponding to the maximum useful work of the unclosed gas turbine cycle \( (l_0^{act})_{max} \), is found according to the standard procedure to determining the one-variable function extremum position (9):
\[
\frac{d(l_0^{act})}{d\pi} \bigg|_{\pi = \pi_{opt}} = c_{pm} \cdot T_1 \cdot \frac{d}{d\pi} \left( \xi \cdot (1 + b) \cdot \left( 1 - \frac{1}{\pi_{m3}} \right) \cdot \eta_{0t} - (\pi_{m1} - 1) / \eta_{0t} \right) = 0
\]

(9)

The solution of equation (9) is:
In equation (10) it is assumed that the compressor inlet air temperature is $t_1 = 15^\circ C \text{ (} T_1 = 288.15 \text{ K)}$, then the inlet air molar specific heat is $c_{p,1} = 29.085 \text{ kJ/(kmol} \cdot \text{K)}$ 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 $\pi_{\text{opt}}$ depending of the working fluids thermodynamic property $\xi$ and the fuel coefficient $b$ at $\eta_{0i}^{\text{T}} = 0.88$ and $\eta_{0i}^{\text{C}} = 0.86$ takes the form (11) by substituting $m_3 = 0.286$ and $m_3 = 1$ into the equation (10):

$$\pi_{\text{opt}} = [2.646 \cdot \xi \cdot (1+b)]^{0.778}$$  (11)

Figure 3 shows the optimal compression ratio $\pi_{\text{opt}}$ depending of the working fluid thermodynamic property $\xi$ and the fuel coefficient $b$ at $m_3 = 1, \eta_{0i}^{\text{T}} = 0.88$ and $\eta_{0i}^{\text{C}} = 0.86$.

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 $1000\div1600^\circ C$ ($\xi = 4.418 \div 6.500$) the parameter $m_3$ is about $0.23 \div 0.26$.

Figure 4 shows the optimal compression ratio $\pi_{\text{opt}}$ depending of the working fluid thermophysical $m_3$ and thermodynamic $\xi$ properties and the fuel coefficient $b$ at $\eta_{0i}^{\text{T}} = 0.88$ and $\eta_{0i}^{\text{C}} = 0.86$.

Figure 3. The optimal compression ratio $\pi_{\text{opt}}$ depending of the working fluid thermodynamic property $\xi$ and the fuel coefficient $b$ at $m_3 = 1, \eta_{0i}^{\text{T}} = 0.88$ and $\eta_{0i}^{\text{C}} = 0.86$.

Figure 4. The optimal compression ratio $\pi_{\text{opt}}$ depending of the thermophysical $m_3$ and thermodynamic $\xi$ properties of the working fluids and the fuel coefficient $b$ at $\eta_{0i}^{\text{T}} = 0.88$ and $\eta_{0i}^{\text{C}} = 0.86$: triangles – natural gas-fired gas turbine [8].
4. Conclusions

The optimal combination of $\pi$ and $\xi$ depends on the working fluid thermophysical property $m_3$. The optimal compression ratio $\pi_{\text{opt}}$ corresponding to the maximum useful work of the gas turbine cycle $(l_0^\text{act})_{\text{max}}$ depends not only on the gas turbine working fluid heat ratio $\xi$ (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 affects the working fluid thermophysical property $m_3$.

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

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

The influence of the fuel coefficient $b$ on optimal compression ratio $\pi_{\text{opt}}$ decreases and tends to 0 while increasing $Q_F$ as like for a closed gas turbine cycle.

References

[1] Ryzhkov A, Bogatova T, Gordeev S 2018 Fuel 214 63-72
[2] Ryzhkov A, Levin E, Filippov P, Abaimov N, Gordeev S 2016 Metallurgist 60 19-30
[3] Romano M, Lozza G 2010 Int. J. Greenh. Gas Con. 4 459-468
[4] Klimenko A, Milman O, Shifrin B 2015 Thermal Engineering 62 807-816
[5] Romano M, Lozza G 2010 Int. J. Greenh. Gas Con. 4 469-477
[6] Allam R et al. 2017 Energy Procedia 114 5948-5966
[7] Gauthier J, Brinkmann G, Copsey B, Lecomte M 2006 Nucl. Eng. Des. 236 526-533
[8] Breeze P 2019 Power Generation Technologies 3 71-97