THERMAL CALCULATIONS AND NO_x EMISSION ANALYSIS OF A MICRO GAS TURBINE SYSTEM WITHOUT A RECUPERATOR

by

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A thermal calculation based on a table of thermal properties of gas was carried out for a micro gas turbine system without a recuperator. The performance parameters of the micro gas turbine system were obtained. The results of the thermal calculations were verified using ASPEN PLUS, and it shows that the thermal calculations fit well with the ASPEN simulation results. Based on this thermal calculation method, the variation of the performance parameters of the micro gas turbine system under different pressure and temperature ratios was analyzed. The results show that there is no optimum pressure ratio within the general design parameters of micro gas turbines, which leads to extreme values of thermal efficiency. The NO_x generation in the combustion chamber of the micro gas turbine based on the Zeldovich mechanism was modeled and analyzed by coupling the 1-D thermal calculation model with the NO_x emission model. The relationship between NO_x generation rate, molar fuel factor, the characteristic pressure, and the characteristic temperature was obtained. The results of the analysis show that, in terms of controlling NO_x emissions from a gas turbine, the use of an increased pressure ratio has a significant advantage over an increased temperature ratio to improve the thermal efficiency of the micro gas turbine.

Key words*: micro gas turbine, thermal calculations, NO_x emissions, characteristic pressure, characteristic temperature*

Introduction

The micro gas turbine is a small Brayton cycle heat engine, typically with a power of tens to hundreds of kilowatts. In recent years, distributed energy systems based on small heat engines have developed rapidly in response to changes in the structure of energy and power demand worldwide, concerns about environmental protection, and the frequency of localized power crises [1]. At the same time, there is a growing concern about the harmful effects of NO_x on human health, forest crops, and the atmosphere [2].

Due to the higher thermal efficiency, most of the current research on micro gas turbines is based on systems with recuperators. These include experimental analysis of micro gas turbine performance and component characteristics based on detailed measurements of various parameters [3]. An experimental and modelling study of the effect of three types of internal flow leakage on the performance of micro gas turbines [4], experimental investigation of the effect of

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pressure loss between the compressor outlet and turbine inlet of a micro gas turbine-solid oxide fuel cell (SOFC/MGT) system on performance, including surging [5], comprehensive thermodynamic modelling of a SOFC/MGT system for multi-effect desalination [6], integration of a micro gas turbine system with thermally activated cooling technology and discussion of its performance parameters [7], experimental and simulation studies of the performance parameters of a biogas-fuelled micro-gas turbine system, including surge [8], *etc.* The subjects of the aforementioned studies are characterized by integrated systems with recuperators and even with components such as fuel cells. In contrast, less research has been carried out in recent years on micro gas turbines without recuperators. However, micro gas turbines without recuperators have the advantages of high power-to-weight ratio, low cost, and high environmental adaptability [9, 10]. In some special cases, such as emergency situations, it has certain advantages.

There is also a large body of research on NO_x emissions from micro gas turbines. These include experimental and numerical simulations of the combustion and emissions of micro gas turbines fuelled by pure ammonia or methane-ammonia [11-13], experimental studies of the combustion and emissions of micro gas turbines fuelled by biofuels [14-16], numerical simulations of the effect of pressure and fuel-air unmixedness on NO_x emissions from gas turbines [17], experimental studies of a new combustion chamber structure to reduce NO_x emissions [18], numerical simulations and experimental studies of the emissions of micro gas turbines fuelled by syngas [19, 20], numerical simulations of the emissions of micro gas turbines fuelled by vegetable oil or vegetable oil mixtures [21, 22], *etc*. Most of the aforementioned studies have predicted or measured NO_x generation from micro gas turbines based on the specific combustion conditions in complex flow fields in combustion chambers. However, research has shown that the turbulent field structure in the premixed lean combustion of a gas turbine combustion chamber does not have a significant effect on NO_x emissions from the gas turbine [23]. Studying NO_x emissions in a combustion chamber based on specific complex combustion conditions is a detailed but tedious task. A simplification of the model can be considered by coupling a 1-D thermal calculation model with a NO_x emission model.

In this paper, thermal calculations based on a table of thermal properties of gas were carried out for a micro gas turbine system without a recuperator, and the results were verified using ASPEN PLUS. Based on this thermal calculation method, the performance parameters of the micro gas turbine system at different pressure and temperature ratios were analyzed and compared. The performance of the system based on the range of micro gas turbine design parameters was discussed. The generation of NO_x in the combustion chamber of a micro gas turbine was modeled and analyzed. The relationship between the generation of nitrogen oxides in the combustion chamber of the micro gas turbine and the design parameters of the micro gas turbine was discussed. A suggestion was given for improving NO_x emissions from micro gas turbines.

Models and methodology

Thermal calculation model

For the thermal calculations of micro gas turbines, an isentropic model incorporating isentropic efficiency was used for the compressor and turbine. For the complex chemical combustion process in the combustion chamber, a table of thermal properties of the gas was used to calculate the thermal properties at each operating point. For the heat losses in the combustion chamber, the combustion efficiency was introduced. For the calculation of the pressure losses of the components, the total pressure recovery coefficient of the inlet tube, the combustion chamber, and the outlet tube were introduced. For the losses in the flow rate of the components, the ratio of the inlet and outlet flow rates of compressor and turbine were introduced. For the friction losses of the components, the mechanical efficiency of the compressor, turbine, and shaft system was introduced. In addition, the efficiency of the reduction gear was introduced.

During the calculation, the specific work consumed by the compressor can be found:

$$W_{e_C} = \frac{i_2^* - i_1^*}{\eta_{m_C}} \tag{1}$$

Then there is an intermediate enthalpy in the turbine that removes the portion of work done for the compressor:

$$i_{3\text{mid}}^* = i_3^* - \frac{W_{e_C}}{(1+f) g_C \eta_{m_T}} \tag{2}$$

From this, the net specific work of the micro gas turbine can be obtained:

$$W_e = \left[i_3^* - i_4^* - \frac{i_2^* - i_1^*}{(1+f) g_C \eta_{m_T} \eta_{m_C}} \right] (1+f) g_C g_T \eta_{m_T} \eta_{rg} \eta_{sh} \tag{3}$$

The compressor inlet flow is determined by the selected micro gas turbine power:

$$G_C = \frac{W_e}{N_e} \tag{4}$$

From this, the fuel consumption rate and thermal efficiency of the micro gas turbine can be obtained:

$$b_e = \frac{3600 f G_C}{N_e} = \frac{3600 f}{W_e} \tag{5}$$

$$\eta_e = \frac{3600}{b_e H_u} \tag{6}$$

As for the calculation of the thermal properties of the gas, the standard fuel C_8H_{16} is used. Its gas properties table can be correctly used as a table of the thermal properties of the gas from $(CH_1)_n$ to $(CH_3)_n$ fuel composition [24]. For the molar fuel factor, β, the chemical equilibrium equation:

$$\begin{gathered} \beta C_8H_{16} + 12O_2 + 44.610N_2 + 0.560A \rightarrow \\ \rightarrow \beta(8CO_2 + 8H_2O) + 12(1-\beta)O_2 + 44.610N_2 + 0.560A \end{gathered} \tag{7}$$

Writing 1 mol of air as (AIR) and 1 mol of gas with a molar fuel factor equal to 1 as $(GAS)_{\beta=1}$, we have:

$$\beta C_8H_{16} + 57.170(\text{AIR}) \rightarrow 61.170\beta(\text{GAS})_{\beta=1} + 57.170(1-\beta)(\text{AIR}) \tag{8}$$

Equation (8) shows that a number of (AIR)'s and a number of $(GAS)_{\beta=1}$'s can be used to represent the combustion chemistry equation for an arbitrary molar fuel factor, β. The molar fuel factor, β, can be determined from the heat balance equation based on the combustion chamber inlet and outlet temperatures. The main selected parameters are given in tab. 1. After the parameters had been selected, according to the gas thermal properties table, the thermal calculations of the gas turbine were carried out. The main calculation results are given in tab. 2.

Table 1. Main selected parameters for thermal calculation of the micro gas turbine

Parameter	Value
Total pressure recovery coefficient of the inlet pipe, σ_{in}	0.985
Compressor pressure ratio, π	3.6
Compressor isentropic efficiency, η_C	0.82
Outlet temperature of the combustion chamber, T_3^* [K]	1253.15
Combustion efficiency, η_b	0.975
Total pressure recovery coefficient of the combustion chamber, σ_b	0.958
Turbine isentropic efficiency, η_T	0.88
Reduction gear efficiency, η_{rg}	0.98
Shaft system efficiency, η_{sh}	0.98
Micro gas turbine power, N_e [kW]	300

Table 2. Main results of thermal calculations for the micro gas turbine

Parameter	Value
Actual compressor outlet enthalpy, i_2^* [kJkg^{-1}]	464.8
Actual compressor outlet temperature, T_2^* [K]	459.1
Compressor specific work consumption, W_{e_C} [Jkg^{-1}]	163.95
Molar fuel factor, β	0.3286
Combustion chamber outlet enthalpy, i_3^* [kJkg^{-1}]	1372
Turbine inlet pressure, p_3^* [Pa]	344123
Actual turbine outlet enthalpy, i_4^* [kJkg–1]	1026.4
Actual turbine outlet temperature, T_4^* [K]	966
Gas turbine specific net work, W_e [kJkg–1]	177.4
Compressor inlet air-flow, G_C [kgs–1]	1.69
Gas turbine fuel consumption rate, b_e [kg/(kWh)]	0.4517
Thermal efficiency of gas turbines, η_e	0.1849

Verification of thermal calculation results

The current main method for thermodynamic simulation of gas turbines is to use MATLAB/Simulink, which is a tedious workload. The ASPEN PLUS can reduce the workload more significantly [25]. To verify the accuracy of the thermal calculation results, the micro gas turbine system was simulated using ASPEN PLUS. In the gas processing of ASPEN PLUS, the compressor and turbine were simulated using the Compr module, and the combustion chamber was simulated using the RGibbs module. The isentropic efficiency and mechanical efficiency of the compressor and turbine can be set directly under the Compr module. It is noted that the RGibbs module cannot set the combustion efficiency and the simulation results need to be corrected later.

The parameters entered in ASPEN PLUS are consistent with tab. 1. Table 3 indicates the results of thermal calculations and ASPEN PLUS simulations for some key parameters. The principle of the RGibbs reactor in ASPEN PLUS is similar to that of the gas properties table. They both calculate the composition and thermal equilibrium of the system when chemical and phase equilibria are reached. The C_8H_{16} fuel is available in the ASPEN PLUS database. The thermal calculation results fit well with the ASPEN PLUS simulation results, and the relative errors between them are within 1%.

Table 3. Results of thermal calculations and ASPEN PLUS simulations for partial parameters of the micro gas turbine

Parameter	Thermal calculation result	Aspen Plus simulation result	Relative error
T_2^* [K]	459.11	461.38	0.49%
W_{e_C} [kJkg^{-1}]	163.95	164.18	0.14%
T_4^* [K]	966.00	966.13	0.01%
Heat duty [kW]	1581.94	1596.73	0.93%
η_e	0.1849	0.1832	0.93%

Meanwhile, based on this ASPEN PLUS model, the pressure ratio, π, and combustion chamber outlet temperature, T_3^*, were varied to verify the results of the thermal calculations under different operating conditions. The π was taken as 2.8, 3.6, and 4.4, and T_3^* was taken as 1053.15 K, 1253.15 K, and 1453.15 K. Thermal calculations and ASPEN simulations were carried out for each of the nine operating points to obtain the thermal efficiencies of the cycle, as shown in fig. 1. As can be seen from fig. 1, the efficiencies obtained from the thermal calculations are all greater than those obtained from the ASPEN PLUS simulations. However, the difference is small and the fit is relatively good. The maximum relative error occurs at (π, T_3^*) = (4.4, 1453.15 K) with a value of 1.42%.

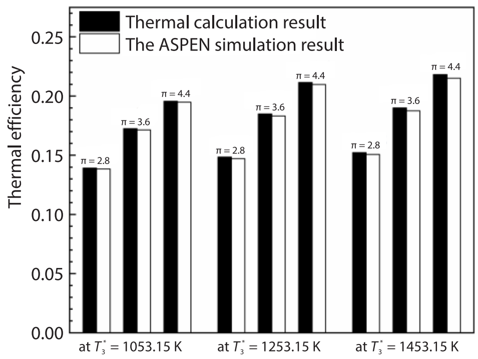

Figure 1. Thermal efficiencies of the micro gas turbine at different temperature and pressure ratios using thermal calculations and ASPEN PLUS simulations

The NO_x emission model

The 1-D thermodynamic model of the micro gas turbine needs to be coupled with a model of NO_x emissions in the combustion chamber. At combustion with clean fuels, while the excess air coefficient is greater than 1, the NO_x are mainly thermal NO_x formed by the extended Zeldovich mechanism [26]. According to the extended Zeldovich mechanism, the gas reacts:

$$N_2 + O \underset{k_{-1}}{\overset{k_1}{\rightleftarrows}} N + NO \tag{9a}$$

$$O_2 + N \underset{k_{-2}}{\overset{k_2}{\rightleftarrows}} NO + O \tag{9b}$$

$$N + OH \underset{k_{-3}}{\overset{k_3}{\rightleftarrows}} NO + H \tag{9c}$$

However, for a lean premixed combustion flame with higher oxygen concentration and lower fuel concentration, the chemical reaction model can be simplified by removing the chemical eq. (9c). The calculated results still match the experiment [27]. According to the chemical reaction kinetics, there is:

$$\frac{\mathrm{d}[\mathrm{NO}]}{\mathrm{d}t}=k_1[\mathrm{N}_2][\mathrm{O}]-k_{-1}[\mathrm{NO}][\mathrm{N}]+k_2[\mathrm{N}][\mathrm{O}_2]-k_{-2}[\mathrm{NO}][\mathrm{O}] \tag{10}$$

According to Zeldovich's experimental results, bringing in k_1, k_{-1}, k_2, and k_{-2} and simplifying, the rate of NO production can be obtained:

$$\frac{\mathrm{d}[\mathrm{NO}]}{\mathrm{d}t}=3\cdot 10^{14}[\mathrm{N}_2][\mathrm{O}_2]^{1/2}\exp\left(\frac{-542000}{\mathrm{R}T}\right) \tag{11}$$

The molar volume of the gas can be obtained by using the ideal gas equation of state, based on the temperature and pressure of the gas. The percentage of each component in the gas can be determined from the chemical eq. (7), and they are the variables of the molar fuel factor β. From this, the concentration of nitrogen and oxygen in the gas can be obtained:

$$[\mathrm{N}_2]=\frac{44.610\cdot 10^{-6}}{8\beta+8\beta+12(1-\beta)+44.610+0.560}\frac{n}{V}=\frac{44.61\cdot 10^{-6}}{4\beta+57.17}\frac{p_\mathrm{c}}{\mathrm{R}T_\mathrm{c}} \tag{12}$$

$$[\mathrm{O}_2]=\frac{12(1-\beta)\cdot 10^{-6}}{8\beta+8\beta+12(1-\beta)+44.610+0.560}\frac{n}{V}=\frac{12(1-\beta)\cdot 10^{-6}}{4\beta+57.17}\frac{p_\mathrm{c}}{\mathrm{R}T_\mathrm{c}} \tag{13}$$

There is a pressure loss during the actual combustion process, which is related to the selected total pressure recovery coefficient of the combustion chamber. Also, from eqs. (11)-(13), the NO_x generation rate is proportional to the 1.5$^{\mathrm{th}}$ power of pressure. Therefore, the power average pressure in the combustion chamber can be calculated to approximate the characteristic pressure of NO_x emission:

$$p_\mathrm{c}\approx\left[\frac{\left(p_2^*\right)^{1.5}+\left(\sigma_\mathrm{b}p_2^*\right)^{1.5}}{2}\right]^{1/1.5} \tag{14}$$

The temperature distribution in the actual combustion process is not uniform, and the rate of NO_x generation is related to the –1.5 power of temperature and the exponent of the inverse of the temperature from eqs. (11)-(13). Therefore, the characteristic temperature of NO_x emission can be theoretically solved:

$$T_\mathrm{c}^{-1.5}\exp\left(\frac{-542000}{\mathrm{R}T_\mathrm{c}}\right)=\frac{\sum_{i=1}^{j}T_i^{-1.5}\exp\left(\frac{-542000}{\mathrm{R}T_i}\right)}{j} \tag{15}$$

where T_i is the sampling temperatures at each point in the combustion chamber. However, the thermal calculation as a 1-D model cannot sample the temperature at each point in the combustion chamber. Moreover, combustion chambers with different geometries and configurations have different temperature distributions. Heitor *et al.* [28] experimentally investigated the isothermal and combustion flow characteristics in a can-type combustion chamber of a gas turbine. The combustion chamber model from their experiments is used here. Its temperature field is sampled at several points using the spatial averaging method. After sampling, the characteristic temperature of the combustion chamber can be obtained by eq. (15). From this, the ratio of the characteristic temperature in the combustion chamber to the outlet temperature of the combustion chamber can be calculated:

$$\frac{T_\mathrm{c}}{T_3^*}=1.3376 \tag{16}$$

It is emphasized that this ratio is based on the combustion chamber model from the experiments of Heitor *et al.* [28]. With this ratio, it is relatively easy to use the combustion chamber outlet temperature to approximately calculate the NO_x emissions of a micro gas turbine based on this combustion chamber model.

Results and discussion

Overall performance

The temperature ratio, τ, and pressure ratio are important parameters to measure the performance of a gas turbine. Based on this verified thermal calculation method for the micro gas turbine system, keeping other selected parameters the same, the performance parameters of the micro gas turbine with different pressure ratios and temperature ratios can be calculated, as shown in fig. 2. Figure 2 represents the variation of specific net work, fuel consumption rate, compressor inlet air-flow, and turbine outlet temperature of the micro gas turbine with pressure ratios at different temperature ratios. The increase of temperature ratio and/or pressure ratio will lead to the increase of the specific net work of the gas turbine, and the effect of temperature ratio is more significant than the effect of pressure ratio, as shown in fig. 2(a). The gradual increase of temperature ratio and/or pressure ratio will lead to the decrease of fuel consumption rate of gas turbine, as shown in fig. 2(b). A higher pressure ratio and/or temperature ratio results in a higher specific net work of the gas turbine, which means a higher energy per unit mass. Therefore, the lower the required inlet flow rate of the gas turbine for the same power, as shown in fig. 2(c). Figure 2(d) shows the exhaust temperature of the micro gas turbine with different temperature ratios and pressure ratios.

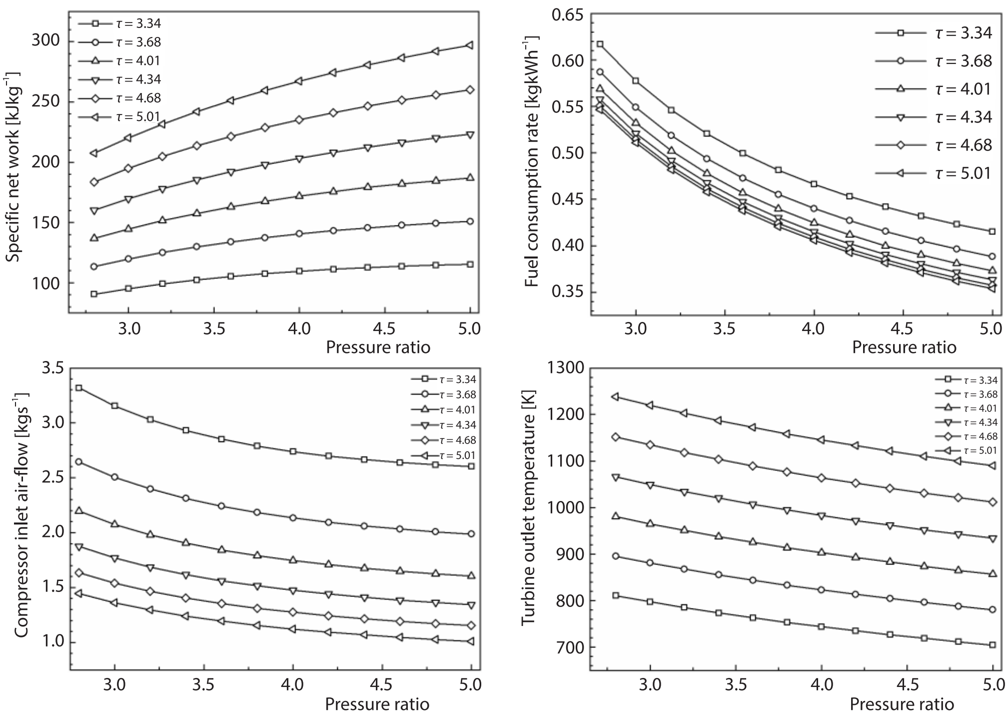

Figure 2. Variation of specific net work, fuel consumption rate, compressor inlet air-flow, and turbine outlet temperature of the micro gas turbine with pressure ratio at different temperature ratio

Thermal efficiency

Figure 3 represents the thermal efficiency of the micro gas turbine at different temperature and pressure ratios, where the horizontal axis indicates the pressure ratio of the gas turbine and the vertical axis indicates the thermal efficiency of the gas turbine. From fig. 3, it can be seen that the thermal efficiency of the micro gas turbine does not show extreme values as the pressure ratio increases in the selected range of pressure ratio and temperature ratio. It has been shown that the optimal pressure ratio that maximizes the cycle efficiency occurs in the interval of higher π and lower τ for the actual Brayton cycle. However, for micro gas turbines without recuperators, due to their special structure, their π is generally below 5, while the combustion chamber outlet temperature is often above 1000 K. Therefore, the optimum pressure ratio does not generally exist for normal micro gas turbines without recuperators.

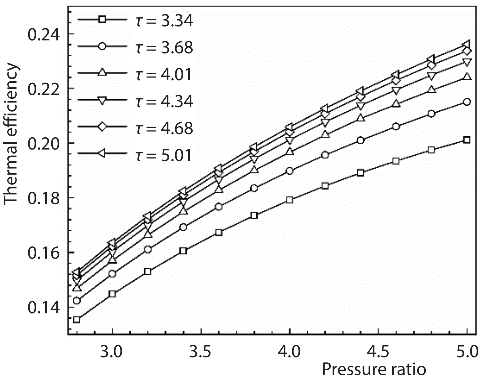

Figure 3. Variation of thermal efficiency of the micro gas turbine with pressure ratio at different temperature ratios

At a certain π, the increment of efficiency decreases as τ increases. To increase the same efficiency at a higher τ will require more τ than at a lower τ, as shown in fig. 4(a). Similarly, at a certain τ, the increment of efficiency decreases as π increases, as shown in fig. 4(b). The horizontal axis of figs. 4(a) and 4(b) represent the temperature ratio and pressure ratio of the gas turbine, respectively, and the vertical axis represents the thermal efficiency of the gas turbine. As can be seen from fig. 4(a), the set of curves tends to contract with the positive direction of the horizontal axis, which indicates that the effect of different π on the efficiency is more obvious in the case of smaller τ than in the case of larger τ. Similarly, the set of curves in fig. 4(b) has an expanding trend with the positive direction of the horizontal axis, which indicates that the effect of different τ on the efficiency is more pronounced in the case of larger π than in the case of smaller π.

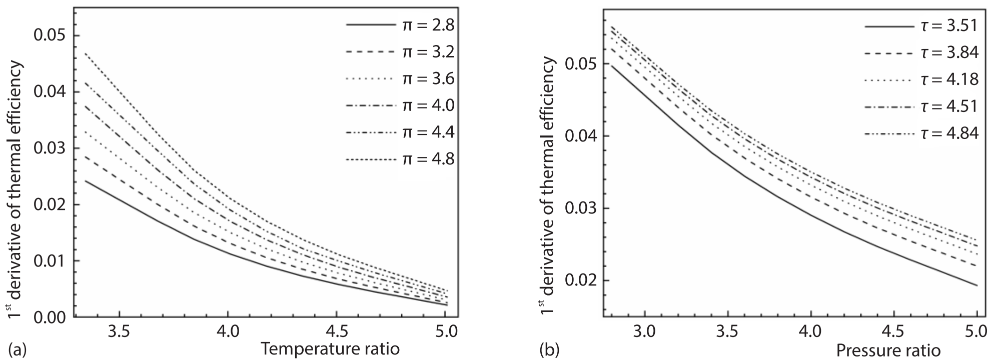

Figure 4. Variation of increment of thermal efficiency of the micro gas turbine with temperature ratio and pressure ratio

The NO_x generation rate

As mentioned earlier the prerequisite for simplifying the extended Zeldovich mechanism from eqs. (9) and (10) is that the combustion flame is a lean premixed combustion flame with higher oxygen concentration and lower fuel concentration. Micro gas turbines are compact and commonly use premixed combustion. Meanwhile, fig. 5 represents the variation of the molar fuel factor, β, and fuel-air mass ratio with the combustion chamber outlet temperature at different pressure ratios, where the horizontal axis is the gas turbine combustion chamber outlet temperature and the two vertical axes are the fuel molar factor and the fuel-air mass ratio in the combustion chamber, respectively. From fig. 5, it can be seen that the molar fuel factor and fuel-air mass ratio are rising as the combustion chamber outlet temperature rises and the pressure ratio decreases. However, there is a significant excess of air regardless of the operating conditions, $\beta < 1$. The minimum pressure ratio and the maximum combustion chamber outlet temperature are considered in the calculation process, *i.e.*, (π, T_3^*) = (2.8, 1503.15 K), when the molar fuel factor is 0.4648 and the fuel-to-air mass ratio is 0.03149. Therefore, the gas turbine model calculated in this paper applies to the simplified prerequisites.

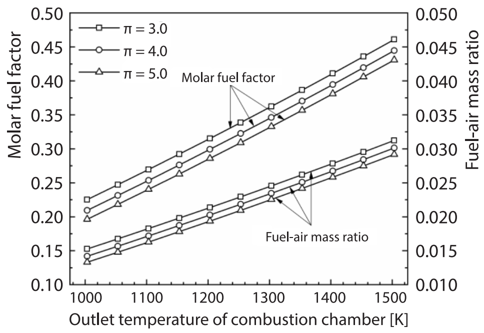

Figure 5. Variation of molar fuel factor and fuel-air mass ratio with combustion chamber outlet temperature at different pressure ratios

The rate of NO_x generation in the combustion chamber of the micro gas turbine based on the Zeldovich mechanism for different combustion chamber outlet temperatures and pressure ratios can be calculated using the ratio of the characteristic temperature to the combustion chamber outlet temperature, as shown in fig. 6. The horizontal axis of the large graph in fig. 6(a) is the inverse of the temperature ratio of the gas turbine, and the logarithmic vertical axis is the generation rate of NO_x in the combustion chamber. The horizontal axis of the small graph in fig. 6(a) is the combustion chamber outlet temperature of the gas turbine, and the linear vertical axis is the generation rate of NO_x in the combustion chamber. The horizontal co-ordinate in fig. 6(b) is the pressure ratio of the gas turbine, and the three vertical axes are the generation rates of NO_x in the combustion chamber at different combustion chamber outlet temperatures of the gas turbine.

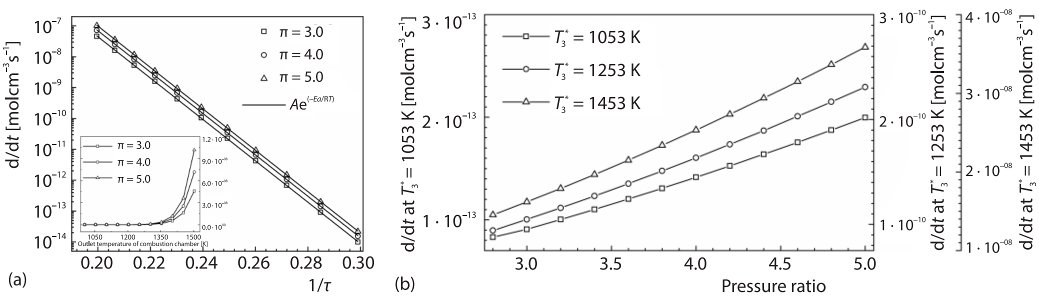

Figure 6. Variation of NO_x generation rate in the combustion chamber of the micro gas turbine with the inverse of temperature ratio, combustion chamber outlet temperature, and pressure ratio

From figs. 6(a) and fig. 6(b), it can be seen that the effect of temperature on NO_x generation in the combustion chamber of the micro gas turbine is huge and can be fitted as an Arrhenius function, while the effect of pressure ratio on NO_x generation in the combustion chamber of the micro gas turbine is relatively much smaller. Increasing the combustion chamber outlet temperature from 1003.15-1503.15 K at a pressure ratio equal to 3.6 increases the NO_x generation rate in the gas turbine combustion chamber by a factor of $5.18 \cdot 10^5$, increasing the pressure ratio from 2.8-5.0 at a combustion chamber outlet temperature equal to 1253.15 K increases the NO_x generation rate in the gas turbine combustion chamber by a factor of only 1.45. Based on the calculation results in part 3.2, increasing the combustion chamber outlet temperature from 1003.15-1503.15 K at a pressure ratio equal to 3.6 increases the gas turbine thermal efficiency by 14.05%, increasing the pressure ratio from 2.8-5.0 at a combustion chamber outlet temperature equal to 1253.15 K increases the gas turbine thermal efficiency by 53.06%. It can be seen that from the point of view of controlling NO_x emissions from gas turbines, to improve the thermal efficiency of gas turbines, using the method of increasing the pressure ratio has a great advantage compared to using the method of increasing the temperature ratio.

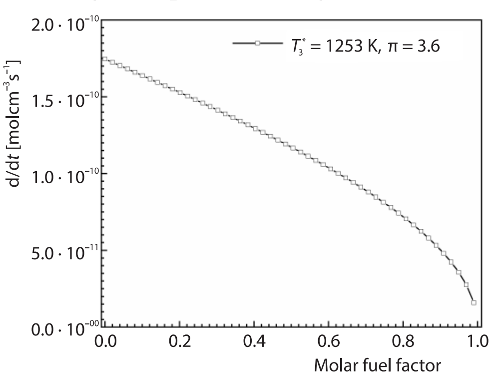

Figure7. Variation of NO_x generation rate with molar fuel factor in the combustion chamber of the micro gas turbine

At the same time, a higher combustion chamber outlet temperature requires a higher molar fuel factor to achieve, fig. 5. In turn, the variation of the molar fuel factor affects the component share of gas in the combustion chamber, which affects the molar concentration of oxygen and nitrogen in the combustion chamber, and thus the rate of NO_x generation. For this reason, the variation of the NO_x generation rate in the combustion chamber with the molar fuel factor for a micro gas turbine at (π, T_3^*) = (3.6, 1253.15 K) was calculated, as shown in fig. 7. The horizontal axis of fig. 7 is the molar fuel factor in the combustion chamber of the gas turbine, and the vertical axis is the rate of NO_x generation in the combustion chamber.

From fig. 7, it can be seen that the rate of NO_x generation in the combustion chamber of the gas turbine decreases gradually with the increase of the molar fuel factor. However, it should be noted that fig. 7 is only an assumption of the ideal condition. In practice, the molar fuel factor is a calculated result based on design parameters such as temperature ratio and combustion chamber outlet temperature:

$$\beta = f\left(\pi, T_3^*, \ldots\right) \tag{17}$$

Therefore, the molar fuel factor cannot be changed arbitrarily. However, the effect of the molar fuel factor on the rate of NO_x production in the combustion chamber of a gas turbine cannot be ignored. As the outlet temperature of the gas turbine combustion chamber increases and the molar fuel factor decreases, the molar volume of the gas in the combustion chamber increases, and not only does the component ratio of the gas change, but also the molar concentration of each reactant changes. Therefore, the generation of NO_x in the combustion chamber of a gas turbine is a complex series of correspondence. The molar fuel factor, the characteristic pressure determined by the pressure ratio and the total pressure recovery coefficient of the combustion chamber, and the characteristic temperature determined by the temperature ratio and the combustion chamber structure, together determine the rate of NO_x generation in the combustion chamber of a micro gas turbine:

$$\frac{\mathrm{d}[\mathrm{NO}]}{\mathrm{d}t} = \mathrm{g}\left(p_\mathrm{c}, T_\mathrm{c}, \beta\right) \tag{18}$$

Conclusions

The thermal calculations of the micro gas turbine thermal system without a recuperator were carried out. The results were verified using ASPEN, and the verification results showed a good fit. The Zeldovich mechanism was also coupled with the thermal calculations to analyze the generation of NO_x in the combustion chamber of the micro gas turbine. The results show as follows.

- Within the selected range of design parameters for a typical micro gas turbine without a recuperator ($2.8 < \pi < 5.0$, $3.34 < \tau < 5.01$), the thermal efficiency of the gas turbine does not show extreme values. There is no optimal pressure ratio and the efficiency always increases with increasing pressure ratio and/or temperature ratio.
- As the pressure ratio and/or temperature ratio gradually increases, the increment of thermal efficiency decreases. To further increase the efficiency at a higher pressure ratio and/or temperature ratio would require a more costly investment. The performance parameters of micro gas turbines without recuperators need to be improved by taking into account their economic and durability advantages.
- The increase in NO_x generation rate in the gas turbine combustion chamber brought about by increasing the gas turbine pressure ratio is not in the same order of magnitude as the increase in NO_x generation rate in the gas turbine combustion chamber brought about by increasing the gas turbine temperature ratio, which is much greater than the former. From the point of view of controlling NO_x emissions from gas turbines, to improve the thermal efficiency of gas turbines, using the method of increasing the pressure ratio has a great advantage compared to using the method of increasing the temperature ratio.
- The rate of NO_x generation in the combustion chamber of a micro gas turbine is determined by several factors:
  - the design parameters (or operating conditions) of the gas turbine, such as the temperature ratio and pressure ratio, determine the molar fuel factor in the combustion chamber,
  - the pressure ratio and total pressure recovery coefficient determine the characteristic pressure in the combustion chamber, and
  - the temperature ratio and the combustion chamber structure determine the characteristic temperature in the combustion chamber.

  Then the molar fuel factor, the characteristic temperature, and the characteristic pressure together determine the rate of NO_x generation in the combustion chamber of a micro gas turbine.

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Nomenclature

A – other components of air dominated by argon
(AIR) – 1 mol of air
b_e – fuel consumption rate of the micro gas turbine, [kg(kWh)$^{-1}$]
f – fuel-air mass ratio
G_C – compressor inlet flow, [kgs^{-1}]
$(\mathrm{GAS})_{\beta=1}$ – 1 mol of gas with a molar fuel factor equal to 1
g_C – compressor inlet and outlet flow ratio

g_T – turbine inlet and outlet flow ratio
H_u – low calorific value of the fuel, [kJkg^{-1}]
i_1^* – actual enthalpy of the compressor inlet, [kJkg^{-1}]
i_2^* – actual enthalpy of the compressor outlet, [kJkg^{-1}]
i_3^* – turbine inlet actual enthalpy, [kJkg^{-1}]
i_{3mid}^* – turbine intermediate enthalpy, [kJkg^{-1}]
i_4^* – actual enthalpy of the turbine outlet, [kJkg^{-1}]
j – number of sampling points
N_e – power of the micro gas turbine, [kW]
[N_2], [O_2], [N], [O], [NO] – corresponding component concentrations, [molcm^{-3}]
n/V – inverse of the molar volume of the gas, [molm^{-3}]
p_2^* – compressor outlet pressure, [Pa]
p_3^* – turbine inlet pressure, [Pa]
p_c – characteristic pressure regarding the NO_x generation, [Pa]
R – molar gas constant, [J(molK)$^{-1}$]
T_2^* – actual compressor outlet temperature, [K]
T_3^* – outlet temperature of combustion chamber, [K]
T_4^* – actual turbine outlet temperature, [K]
T_c – characteristic temperature regarding the NO_x generation, [K]
T_i – sampling temperatures at each point in the combustion chamber, [K]
W_e – net specific work of the micro gas turbine, [kJkg^{-1}]
W_{e_C} – compressor specific work consumption, [kJkg^{-1}]

Greek symbols

β – molar fuel factor
η_b – combustion efficiency
η_C – compressor isentropic efficiency
η_e – thermal efficiency of the micro gas turbine
η_{m_C} – mechanical efficiency of the compressor
η_{m_T} – mechanical efficiency of the turbine
η_{rg} – reduction gear efficiency
η_{sh} – shaft system efficiency
η_T – turbine isentropic efficiency
π – compressor pressure ratio
σ_b – total pressure recovery coefficient of the combustion chamber
σ_{in} – total pressure recovery coefficient of the inlet pipe

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