

Parametric Study on Thermoelectric Power Generator and Microturbine Combined Power Generation System

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Microturbine is an important power equipment for distributed energy system with the advantages of compact structure, light weight, low noise, and so on. However, its power generation efficiency is still too low while its cost is too high to be widely applied. Thermoelectric material is a solid-state energy converter which can directly convert thermal energy into electrical energy without affecting the original thermodynamic cycle. It is expected to significantly improve the system's efficiency and reduce the power generation cost by integrating the thermoelectric generators into the microturbine system. In this study, a thermodynamic model is developed for the thermoelectric power generator and microturbine combined power generation system. The numerical results showed that the addition of thermoelectric power generators can increase the power output of microturbine system by 1-2 times. Increasing the figure of merit value of TEGs and turbine inlet temperature improve the power output and efficiency of the combined system, but there is an optimal compressor pressure ratio for the combined system.

1. Introduction

Microturbine is a small power generation equipment with power output of around 25 ~ 300 kW (Zhu and Tomsovic, 2002). It is suitable for distributed energy system due to its compact structure, flexible fuels, light weight, low noise and so on. However, its power generation efficiency is still too low while its power generation cost is too high to be wide application. In order to improve the efficiency, the hybrid cycles are usually used, such as CCHP (Combined cooling, heating and power) system, and solid oxide fuel cell and microturbine hybrid power system.

Recently, the thermoelectric generator (TEG) has received considerable attention because the thermoelectric material is a solid-state energy converter which can directly convert thermal energy into electrical energy. The advantages of TEG include high reliability and convenient installation without affecting the original thermodynamic cycle. However, the TEG is hardly to be used as the main power generation component in many power and energy systems because the figure of merit of thermoelectric material is too small for practical application. But if the TEG is added into the power and energy system, it may result in better performance. Di Bella (2011) added TEGs into a Rolls-Royce MT-30 gas turbine system. The analytical result showed that the addition of TEGs could improve the net power by 5 %. Sarnacki et al. (2010) added TEGs into an SR30 microturbine. The test showed that the addition of TEGs could recover the exhaust heat to extract a resistive load of 192 W at power of 18 A. Yazawa et al. (2013) found that adding TEGs into a steam turbine system could minimize the payback of initial cost as well as fuel consumption.

The power output of microturbine system could also be increased by integrating with TEGs. Yazawa et al. (2017) proposed a concept of TEG and microturbine combined power generation system. Based on a typical Brayton cycle, three TEGs are embedded in the combustion chamber, recuperator and flue gas waste heat recovery component. In the combustion chamber, the flame temperature can reach 1700 °C, while the compressed air temperature of the shell is around 600 °C, so that the maximum temperature difference for the first TEG can reach 1100 °C. In the recuperator, the hot-side inlet temperature is usually closed to 700 °C, while the cold-side inlet temperature is around 180 °C, so that the maximum temperature difference of the second TEG is 520 °C. In the flue gas waste heat recovery component, the temperature of exhaust gas is about 230 °C, which may result in a temperature different around 200 °C for the third TEG. The analysis result showed that the efficiency

of TEG and microturbine combined power generation system could reach 42 % (Yazawa et al., 2017). In this paper, the effects of key component parameters on the performance of TEG and microturbine combined power generation system are conducted.

2. Thermodynamic model of TEG and microturbine combined power generation system

The thermodynamic process of the TEG and microturbine combined power generation system is shown in the schematic diagram and solid lines of T-S diagram in Figure 1, which is similar to a conventional Brayton cycle, but there are three TEGs providing additional power output. The complete cycle is as follows. The air is compressed in the compressor in process 1-2, then the compressed air enters into the recuperator for heating in process 2-5. In process 5-3, the fuel with total heat $Q_{L,TE}$ is mixed and burned with a small amount of air in the combustion chamber, and the generated high temperature flue gas is used as the heat source of the first TEG. The remaining air is used as the cold source of the first TEG, and then mixed with the high temperature flue gas. The power output of the first TEG is Q_{TEG-1} . The mixed high-temperature gas expands and works in the turbine in process 3-4. The discharged gas from turbine is used as the heat source for the second TEG in process 4-6, while the compressed air in process 2-5 is used as the cold source for the second TEG. The power output of the second TEG is Q_{TEG-2} . Finally, the waste heat of the discharged gas Q_{out} is recovered in process 6-1 in other ways like hot water, which is also used as the heat source of the third TEG. The cooling water is usually used as the cold source of the third TEG. The power output of the third TEG is Q_{TEG-3} .

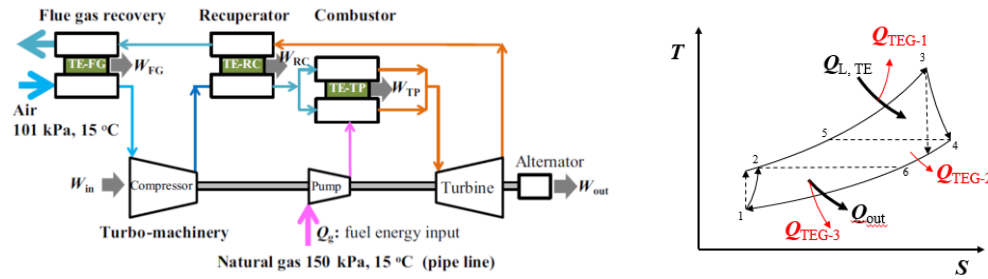


Figure 1: Schematic and T-S diagrams of TEG and microturbine combined power generation system: (a) Schematic diagram (Yazawa et al., 2017), (b) T-S diagram

3. Thermodynamic model of combined power generation system

3.1 Total power output and efficiency of system

The efficiency of system η_{sys} is defined as follows (Yazawa et al., 2017):

$$\eta_{sys} = \frac{w_{sys}}{q_f} \quad (1)$$

where q_f is the chemical energy of input fuel. w_{sys} is the power output of system, which is calculated as follows:

$$w_{sys} = w_{TEG-1} + w_{TEG-2} + w_{TEG-3} + \eta_{mech} (w_t - w_c) \quad (2)$$

where w_t is the power output of turbine, w_c is the consumed power of compressor, and η_{mech} is the efficiency of motor. The power outputs of the three TEGs w_{TEG-1} , w_{TEG-2} and w_{TEG-3} are calculated as follows:

$$w_{TEG-1} = \eta_{TEG-1} \eta_r q_f \quad (3)$$

$$w_{TEG-2} = \eta_{TEG-2} (1 - \eta_{TD}) (1 - \eta_{TEG-1}) \eta_r q_f \quad (4)$$

$$w_{TEG-3} = \eta_{TEG-3} (1 - \eta_{TEG-2}) (1 - \eta_{TD}) (1 - \eta_{TEG-1}) \eta_r q_f \quad (5)$$

where, η_{TEG-1} , η_{TEG-2} , η_{TEG-3} are the thermal conversion efficiency of the three TEGs, respectively. η_r , η_{TD} are the efficiency of combustion chamber and turbine, respectively.

3.2 Maximum power output and thermal conversion efficiency of TEG

The maximum power output of TEG w_{\max} is calculated by (Yazawa et al., 2017):

$$w_{\max} = \frac{mZ}{\alpha^2(1+m)^2} \frac{\beta F}{d_{opt}} (T_s - T_a)^2 \quad (6)$$

where, T_s and T_a are the temperatures of heat source and cold source, respectively. m is the ratio of load resistance and internal resistance, and $m = \sqrt{1+ZT}$ for the maximum power output. α is the ratio of thermal resistances on both sides, here, $\alpha=2$. β is the thermal conductivity. F is the feature size factor. d_{opt} is the characteristic length of the thermoelectric leg, which is defined as:

$$d_{opt} = m\beta F \sum \psi \quad (7)$$

$$\sum \psi = (T_s - T_c) / q \quad (8)$$

where $\sum \psi$ is the total heat resistance of TEG between hot side and cold sides. q is the specific heat flux.

According to Eqs(6)-(8), for a given heat transfer Q_h , the maximum thermoelectric conversion efficiency η_{TEG} and the maximum power output w_{TE} can be calculated by:

$$\eta_{TEG} = \frac{w_{\max}}{q} = \frac{Z}{4(1+m)^2} (T_s - T_a) \quad (9)$$

$$w_{TEG} = \frac{Z \cdot Q_h}{4(1+m)^2} (T_s - T_a) \quad (10)$$

3.3 Thermodynamic calculation of each component

The thermodynamic calculation of each component in the combined power generation system are similar to the conventional Brayton cycle.

(1) Compressor

The calculations of each parameter during air compression are described as follows:

$$p_1 = p_a (1 - \beta_c) \quad (11)$$

$$p_2 = p_1 r_c \quad (12)$$

$$s_c = s(p_1, T_a) \quad (13)$$

$$w_c = [h(p_2, s_c) - h(p_1, T_a)] / \eta_c \quad (14)$$

$$h_2 = w_c + h_1 \quad (15)$$

where p_1 and p_2 are the inlet and outlet pressures of compressor, respectively. h_1 and h_2 are the enthalpy of the inlet and outlet of compressor, respectively. s_c is the specific enthalpy of the inlet air of compressor. r_c is the pressure ratio. β_c is the inlet pressure loss rate. w_c is the compression ratio work. η_c is entropic compression efficiency. p_a and T_a are atmospheric pressure and temperature, respectively.

(2) Recuperator

The compressed cold air in the recuperator is preheated to increase its specific enthalpy. The effectiveness has a significant effect on the thermal efficiency of microturbine system. The enthalpy difference of recuperator Δh is calculated by:

$$\Delta h = h_4 - h(p_4, T_2) \quad (16)$$

On the cold side, the pressure and enthalpy are calculated by:

$$p_5 = p_2 (1 - \beta_{\text{rec_cold}}) \quad (17)$$

$$h_5 = h_2 + \varepsilon \cdot \Delta h \quad (18)$$

On the hot side, the pressure and enthalpy are calculated by:

$$p_6 = p_4 (1 - \beta_{\text{rec_hot}}) \quad (19)$$

$$h_6 = h_4 - \varepsilon \cdot \Delta h \quad (20)$$

where h_2 and h_5 are the specific enthalpy of the cold-side inlet and outlet, respectively, while h_4 and h_6 are the specific enthalpy of the hot-side inlet and outlet, respectively. p_2 and p_5 are the pressure of cold-side inlet and outlet, respectively, while p_4 and p_6 are the pressure of hot-side inlet and outlet, respectively. $\beta_{\text{rec_cold}}$ and $\beta_{\text{rec_hot}}$ are the pressure loss rates on the cold side and hot side, respectively. T_2 is the outlet temperature of compressor. ε is the effectiveness of recuperator.

(3) Combustion chamber

The heating process parameters are calculated as follows:

$$p_3 = p_5 (1 - \beta_r) \quad (21)$$

$$h_3 = \eta_r q_r + h_5 \quad (22)$$

where h_5 and h_3 are the specific enthalpy of combustion chamber inlet and outlet, respectively, while p_5 and p_3 are the pressure of combustion chamber inlet and outlet, respectively. β_r is the pressure loss of the combustion chamber. η_r is the combustion efficiency.

(4) Turbine

The turbine expansion process is calculated as follows:

$$s_t = s(p_3, T_3) \quad (23)$$

$$w_t = \eta_t [h(p_3, T_3) - h(p_4, s_t)] \quad (24)$$

$$h_4 = w_t + h_3 \quad (25)$$

where h_3 and h_4 are the enthalpy of turbine inlet and outlet, respectively. p_3 and p_4 are the pressure of turbine inlet and outlet, respectively. T_3 and s_t are the inlet temperature and specific entropy of turbine, respectively. η_t is the isentropic expansion efficiency. w_t is output work of turbine.

(5) Exhaust gas

The exhaust gas maintains the positive pressure, and the outlet pressure p_6 is calculated as follows:

$$p_6 = p_a / (1 - \beta_{\text{ex}}) \quad (26)$$

where β_{ex} is pressure loss of the exhaust gas.

4. Model validation

The parameters of a 2 ~ 3 kW recuperated microturbine system from Visser et al. (2011) are used in this study. Based on the parameters and the mentioned method summarized in Section 3, an in-house thermodynamic code is developed in MATLAB. The result is compared with the result of Yazawa et al. (2017), which was calculated by Engineering Equations Solver (EES) software. The maximum deviation is less than 3.4 % and the average deviation is 0.6 %, compared with the result of Yazawa et al. (2017). Therefore, the present thermodynamic model is valid.

5. Results and discussion

5.1 Effect of turbine inlet temperature

The figure of merit (ZT) of TEGs varies greatly with temperature. Two cases are calculated to study the effect of ZT values of TEGs. Case 1 uses commercially available thermoelectric materials, whose ZT values for the

three TEGs are 0.8, 1.2 and 1.2. Case 2 uses recently developed thermoelectric materials, whose ZT values for the three TEGs are 1.0, 1.8 and 1.8.

Increasing the inlet temperature of turbine can significantly increase the power output of turbine. As shown in Figure 2a, the power outputs of the single microturbine system and the combined system increase with the increase of turbine inlet temperature (1,100 K~1,350 K). Compared with the single microturbine system, TEGs can significantly increase the power output of system. The power output of the combined system is 2-3 times the power output of the conventionally single microturbine system. Figure 2b shows the power generation efficiency of the single microturbine system and combined system with different ZT values. It can be seen that as the turbine inlet temperature (1,100 K~1,350 K) increases, the power generation efficiency of the combined system increases. Among them, the power generation efficiency of the single microturbine system increases from 11.7 % to 18.2 %, but the power generation efficiency of the combined system efficiency increases from 23.2 % to 29.5 % under the condition of Case 1, while it increases from 26.4 % to 32.8 % under the condition of Case 2. It indicates that improving the ZT value of TEG can increase the efficiency of the combined system.

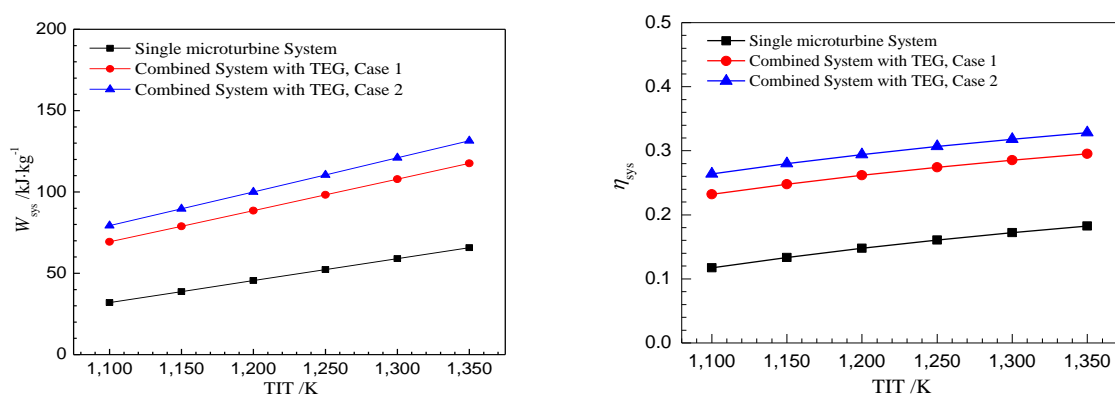


Figure 2: Effects of turbine inlet temperature on power output and efficiency: (a) Power output, (b) Efficiency

5.2 Effect of compressor pressure ratio

Figure 3a shows the trend of the power output varying with the compressor pressure ratio (2.0-3.6) for the single microturbine system and combined system with different ZT values. The power output of the single microturbine system increases from 49.14 kJ/kg to 70.17 kJ/kg. The power output of the combined system increases from 98.75 kJ/kg to 117.7 kJ/kg under the condition of Case 1, while it increases from 112.1 kJ/kg to 130.2 kJ/kg under the condition of Case 2. As can be seen from the figure, with the increase of compressor pressure ratio, the growth rate of power output of all the systems gradually slow down. Figure 3b shows the trend of the power generation efficiency varying with the compressor pressure ratio. The efficiency of the single microturbine system increases from 15.4 % to 17.9 %. The efficiency of combined system under the condition of Case 1 rises from 27.8 % to 28.6 % and then decreases to 27.8 %. Under the condition of Case 2, the efficiency of combined system rises from 31.4 % to 31.8 % and then decreases to 30.6 %. Compared with the single microturbine system, there is an optimal compressor pressure ratio for the combined system. In the studied operating condition, the optimal compressor pressure ratio of Case 1 is about 2.8, while the optimal compressor pressure ratio of Case 2 is around 2.4. With the increases of ZT value, the optimal compressor pressure ratio of the combined system reduces, which is beneficial to reducing the operating pressure and cost of the system.

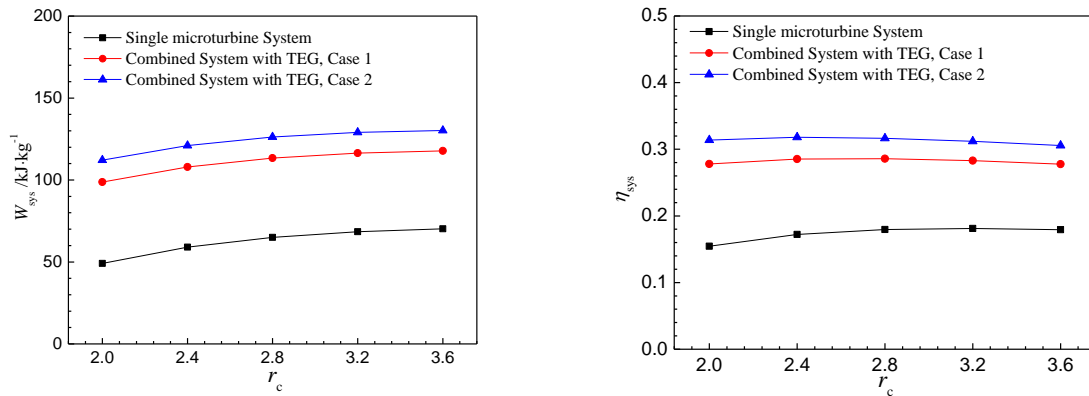


Figure 3: Effects of compressor pressure ratio on power output and efficiency: (a) Power output, (b) Efficiency

6. Conclusions

In this work, a thermodynamic model is developed to study the effects of inlet temperature and compressor pressure ratio on the performance of thermoelectric power generator and microturbine combined power generation system. The main conclusions are summarized as below:

- 1) Increasing the ZT value of TEGs and turbine inlet temperature improve the power output and efficiency of system. The power output of the combined system is 2-3 times the power output of the conventionally single microturbine system
- 2) With the increase of compressor pressure ratio, the growth rate of power output of all the systems gradually slow down. There is an optimal compressor pressure ratio for the combined system. In the studied operating condition, the optimal compressor pressure ratio of Case 1 is about 2.8, while the optimal compressor pressure ratio of Case 2 is around 2.4. With the increases of ZT value, the optimal compressor pressure ratio of the combined system reduces, which is beneficial to reducing the operating pressure and cost of the system.

Acknowledgments

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References

- Di Bella F.A., 2011, Gas turbine engine exhaust waste heat recovery navy shipboard module development, Supercritical CO₂ Power Cycle Symposium, Boulder, USA, May 24-25.
- Sarnacki W.P., Wallace T.T., Wlodkowski P.A., 2010, Increasing the diesel and Brayton cycle efficiency with thermoelectric materials, proceedings of ASME 2010 International Mechanical Engineering Congress & Exposition (IMECE2010), Vancouver, Canada, November 12-18.
- Visser W., Shakariyants S., Oostveen M., 2011, Development of a 3 kW microturbine for CHP applications. Journal of Engineering for Gas Turbines and Power, 133, 042301.
- Yazawa K., Koh Y.R., Shakouri A., 2013, Optimization of thermoelectric topping combined steam turbine cycles for energy economy, Applied Energy, 109, 1-9
- Yazawa K, Fisher T.S., Groll E.A., Shakouri A, 2017, High exergetic modified Brayton cycle with thermoelectric energy conversion, Applied Thermal Engineering, 114, 1366-1371
- Zhu Y., Tomsovic K., 2002, Development of models for analysing the load-following performance of microturbines and fuel cells, Electric Power Systems Research, 62, 1-11.