Finite-time exergoeconomic performance of a real intercooled regenerated gas turbine cogeneration plant. Part 2: heat conductance distribution and pressure ratio optimization

Bo Yang, Lingen Chen*, Yanlin Ge and Fengrui Sun
College of Naval Architecture and Power, Naval University of Engineering, Wuhan 430033, P. R. China

Abstract

Finite-time thermodynamics is applied to establish a real intercooled regenerated gas turbine cogeneration plant model in Part 1 of this article and the profit rate performance of the plant is researched using finite-time exergoeconomic analysis. In this part, the optimization of finite-time exergoeconomic performance of the plant is performed by optimizing the intercooling pressure ratio and the heat conductance distributions among the hot-, cold- and consumer-side heat exchangers, the intercooler and the regenerator together, and it is found that the optimal heat conductance distribution of the regenerator is zero. When the heat conductance distribution of the regenerator is fixed, the results show that there exists an optimal intercooling pressure ratio and a group of optimal heat conductance distributions among the other four heat exchangers, which leads to a maximum dimensionless profit rate for the fixed total pressure ratio. When the total pressure ratio is variable, a double-maximum dimensionless profit rate is obtained. The characteristics of the double-maximum dimensionless profit rate, the corresponding exergetic efficiency, heat conductance distributions and total pressure ratio versus some main design parameters are analyzed in detail. Finally, it is found that the dimensionless profit rate has a thrice-maximum value with respect to the thermal capacitance rate matching between the working fluid and the heat reservoir.

Keywords: irreversible intercooled regenerative gas turbine cogeneration plant; finite-time thermodynamics; profit rate; heat conductance distribution; pressure ratio; optimization

Received 27 January 2012; revised 6 September 2012; accepted 27 September 2012

1 INTRODUCTION

A cogeneration plant model composed of an irreversible intercooled regenerated gas turbine closed cycle coupled to variable-temperature heat reservoirs is provided using finite-time thermodynamic in Part 1 [1] of this article based on the Refs. [2–14]. The analytical relationships among the dimensionless profit rate, exergetic efficiency and the effectivenesses of the five heat exchangers, the intercooling pressure ratio and the total pressure ratio are deduced. In this part, the optimization of dimensionless profit rate will be carried out by using the similar principle and method used for performance optimization of various intercooled regenerative Brayton cycles [15–24] in which the power, efficiency and power density were taken as the optimization objectives, respectively.

By means of numerical calculations, the finite-time exergoeconomic performance is optimized by searching for the optimal intercooling pressure ratio and the optimal heat conductance distributions among the hot-, cold- and consumer-side heat exchangers, the intercooler and the regenerator for the fixed total pressure ratio and fixed total heat conductance. When the total pressure ratio is variable, the performance optimization is performed further by searching the optimal total pressure ratio. The effects of the ratio of the hot-side heat reservoir inlet temperature to environment temperature, the total heat conductance, the price ratio, the efficiencies of the compressors and the...
turbine, the pressure recovery coefficients and the consumer-side temperature on the dimensionless profit rate, as well as the exergetic efficiency, heat conductance distributions and total pressure ratio at the dimensionless profit rate are investigated in detail. At last, the characteristic of the dimensionless profit rate versus the thermal capacitance rate matching between the working fluid and the heat reservoir is studied.

2 NUMERICAL EXAMPLES

The T-s diagram of the cogeneration plant model is shown in Figure 1. Processes 1-2 and 3-4 are nonisentropic adiabatic compression in the low- and high-pressure compressors, whereas the process 5-6 is nonisentropic adiabatic expansion in the turbine. Process 2-3 is an isobaric intercooling process in the intercooler. Processes 4-7 and 6-8 are absorbed and evolved heat processes in the regenerator, respectively. Process 7-5 is an absorbed heat process in the hot-side heat exchanger. Processes 8-9 and 9-1 are evolved heat processes in the consumer-side and cold-side heat exchangers, respectively. Processes 1-2s, 3-4s and 5-6s are isentropic adiabatic processes corresponding to the nonisentropic adiabatic processes 1-2, 3-4 and 5-6, respectively. Assuming that the working fluid used in the cycle is an ideal gas with thermal capacity rate (mass flow rate and specific heat product) $C_{\text{st}}$. The hot-side heat reservoir is considered to have a thermal capacity rate $C_{\text{H}}$ and the inlet and the outlet temperatures of the heating fluid are $T_{\text{Hin}}$ and $T_{\text{Hout}}$, respectively. The cold-side heat reservoir is considered to have a thermal capacity rate $C_{\text{L}}$ and the inlet and the outlet temperatures of the cooling fluid are $T_{\text{Lin}}$ and $T_{\text{Lout}}$, respectively. The consumer-side heat reservoir is considered to have a thermal capacity rate $C_{\text{K}}$ and the inlet and the outlet temperatures of the heating fluid are $T_{\text{Kin}}$ and $T_{\text{Kout}}$, respectively. The pressure recovery coefficients and the consumer-side heat exchangers, the intercooler and the regenerator are $U_{\text{H}}, U_{\text{T}}, U_{\text{K}}, U_{\text{L}}$ and $U_{\text{R}}$, respectively.

It can be seen from equation (25) in Part 1 of this article [1] that the dimensionless profit rate $(\Pi)$ of the cogeneration plant is the function of the intercooling pressure ratio $(\pi_1)$, the total pressure ratio $(\pi)$ and the five heat conductance $(U_{\text{H}}, U_{\text{T}}, U_{\text{K}}, U_{\text{L}}$ and $U_{\text{R}})$ when the other boundary condition parameters $(a, b, T_{\text{Hin}}, T_{\text{Lin}}, T_{\text{K}}, C_{\text{H}}, C_{\text{L}}, C_{\text{K}}, \eta_{\text{K}}, \eta_{\text{H}}, D_1$ and $D_2$) are given. In practical design, $\pi_1, \pi, U_{\text{H}}, U_{\text{T}}, U_{\text{K}}, U_{\text{L}}$ and $U_{\text{R}}$ are changeable. To simplify the problem, the constraint on total heat exchanger conductance is used for the performance optimization of various Brayton cycles [15–24]. Assuming that the total heat exchanger conductance $(U_1 = U_{\text{H}} + U_{\text{T}} + U_{\text{K}} + U_{\text{L}} + U_{\text{R}})$ is fixed, a group of heat conductance distributions are defined as:

$$u_h = \frac{U_{\text{H}}}{U_{\text{T}}}, u_l = \frac{U_{\text{L}}}{U_{\text{T}}}, u_k = \frac{U_{\text{K}}}{U_{\text{T}}}, u_i = \frac{U_{\text{I}}}{U_{\text{T}}}, u_r = \frac{U_{\text{R}}}{U_{\text{T}}}$$

(1)

Obviously, one has the following constraints:

$$0 < u_h < 1, 0 < u_l < 1, 0 < u_k < 1, 0 < u_i < 1, 0 < u_r < 1, u_h + u_l + u_k + u_i + u_r = 1$$

(2)

For the fixed $\pi$ and $\pi_1$, the optimization can be performed by searching the optimal heat conductance distributions $\left(u_h, u_l, u_k, u_i, u_r\right)$, which lead to an optimal dimensionless profit rate $(\Pi_{\text{opt}})$. One can always obtain $(u_i)_{\Pi_{\text{opt}}} = 0$. The reason is that for the gas turbine cogeneration cycle discussed herein, the regeneration has little effect on the optimal dimensionless profit rate compared with the other factors. Another method is adopted herein: when the heat conductance distribution of the regenerator $u_i$ is fixed, the optimization can be performed by searching the other four optimal heat conductance distributions.

To search the optimal values of $u_h$, $u_l$, $u_k$ and $u_r$, the numerical calculations are provided by using the optimization toolbox of Matlab 7.1. In the calculations, without special illustration, the values of some correlative parameters are selected.

Figure 1. T-s diagram for the cogeneration plant model.

Figure 2. Effect of $\pi_1$ on the characteristic of $\Pi_{\text{opt}}$ versus $\pi$. 

International Journal of Low-Carbon Technologies 2014, 9, 262–267 263
according to previous literature [25–31] about gas turbine real power plants or cogeneration plants, and are set as follows: $u_t = 0.1$, $k = 1.4$, $a = 10$, $b = 6$, $C_{oef} = 1.0 \text{ kW/K}$, $C_{tt} = C_l = C_i = 1.2 \text{ kW/K}$, $\tau_1 = 5.0$, $\tau_2 = \tau_3 = 1$, $\tau_4 = 1.2$, $\eta_c(= \eta_l) = 0.85$, $D_1(= D_2) = 0.95$ and $U_T = 10 \text{ kW/K}$.

2.1 Optimal dimensionless profit rate
Presuming that $\pi = 18$ ($1 \leq \pi_1 \leq 18$). Figure 2 shows the characteristic of the optimal dimensionless profit rate ($\Pi_{\text{opt}}$) versus $\pi_1$ for different $\tau_1$. Figure 3 shows the characteristic of the corresponding heat conductance distributions ($\eta_{\text{ex}}/\Pi_{\text{opt}}$) with respect to $\pi_1$, and the intercooling pressure ratio corresponding to $\Pi_{\text{max}}$ is ($\pi_1/\Pi_{\text{max}}$). $\Pi_{\text{opt}}$ increases equally with the increase of $\pi_1$. It can be seen from Figure 4 that $\Pi_{\text{opt}}$ has a maximum value ($\Pi_{\text{max}}$) with respect to $\pi_1$, and the intercooling pressure ratio corresponding to $\Pi_{\text{max}}$ is ($\pi_1/\Pi_{\text{max}}$). $\Pi_{\text{opt}}$ increases first and then increase, ($\eta_{\text{ex}}/\Pi_{\text{opt}}$) increases first and then decreases. The heat conductance distributions corresponding to $\Pi_{\text{max}}$ are ($\eta_{\text{ex}}/\Pi_{\text{max}}$) and the corresponding exergetic efficiency ($\eta_{\text{ex}}/\Pi_{\text{max}}$) decreases, but the value of ($\eta_{\text{ex}}/\Pi_{\text{max}}$) decreases, but the value of ($\eta_{\text{ex}}/\Pi_{\text{max}}$) decreases. It can be seen from Figures 6 and 7 that with the increase of $b$, $\Pi_{\text{max},2}$ increases, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases first and then increases, but the value of ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases, but the value of ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases, but the value of ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases. It can be seen from Figures 8 and 9 that with the increase of $\eta_{\text{c}} (= \eta_l)$, $\Pi_{\text{max},2}$ and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) increase monotonously, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) and ($\pi_1/\Pi_{\text{max},2}$) increase, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decreases, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) changes slightly and the value of ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) is about 0.1. It can be seen from Figures 10 and 11 that with the increase of $\pi_1$, $\Pi_{\text{max},2}$ has a thrice-maximum value ($\Pi_{\text{max},3}$) and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) also has an extremum when $\tau_1$ is large, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decrease first and then increase appreciably, ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) decrease and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) increases.

The numerical calculation also illustrates that the double-maximum dimensionless profit rate increases with the total heat conductance ($U_T$) and price ratio ($a$), and decreases with the pressure recovery coefficients ($D_1(= D_2)$).

3 EFFECTS OF SOME MAJOR PARAMETERS

Figures 6–11 show the characteristics of the double-maximum dimensionless profit rate ($\Pi_{\text{max},2}$) as well as the corresponding exergetic efficiency ($\eta_{\text{ex}}/\Pi_{\text{max},2}$), the heat conductance distributions ($\Pi_{\text{opt}}$) versus the price ratio ($b$), the compressors and turbine efficiencies ($\eta_{\text{c}} (= \eta_l)$) and consumer-side temperature ($\tau_1$), respectively.

4 THERMAL CAPACITY RATES MATCHING BETWEEN THE WORKING FLUID AND THE HEAT RESERVOIR

For variable-temperature heat reservoirs, the changes of thermal capacity rates of the heat reservoirs and the working

![Figure 3. Characteristics of ($\eta_{\text{ex}}/\Pi_{\text{opt}}$), ($\eta_{\text{ex}}/\Pi_{\text{max}}$), ($\eta_{\text{ex}}/\Pi_{\text{min}}$), ($\eta_{\text{ex}}/\Pi_{\text{opt},2}$) and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) versus $\pi_1$.](image)

![Figure 4. Characteristics of $\Pi_{\text{max}}$ and ($\eta_{\text{ex}}/\Pi_{\text{max},2}$) versus $\pi$.](image)
Figure 5. Characteristics of $\left( u_j \right)_{\Pi_{\text{max}}}$, $j = h, l, k, i$ and $\left( \pi_i \right)_{\Pi_{\text{max}}}$ versus $\pi$.

Figure 6. Characteristics of $\Pi_{\text{max}, 2}$ and $\left( \eta_{\text{ex}} \right)_{\Pi_{\text{max}}}$ versus $b$.

Figure 7. Characteristics of $\left( u_j \right)_{\Pi_{\text{max}}}$, $j = h, l, k, i$ and $\left( \pi_i \right)_{\Pi_{\text{max}}}$ versus $b$.

Figure 8. Characteristics of $\Pi_{\text{max}}, 2$ and $\left( \eta_{\text{ex}} \right)_{\Pi_{\text{max}, 2}}$ versus $\eta_\text{e}(=\eta_\text{h})$.

Figure 9. Characteristics of $\left( u_j \right)_{\Pi_{\text{max}, 2}}$, $j = h, l, k, i$ and $\pi_{\text{ex}, 2}$ versus $\eta_\text{e}(=\eta_\text{h})$.

Figure 10. Characteristics of $\Pi_{\text{max}, 2}$ and $\left( \eta_{\text{ex}} \right)_{\Pi_{\text{max}, 2}}$ versus $\tau_\text{d}$.
The heat conductance distribution of the regenerator corresponding to the optimal dimensionless profit rate is always zero. For the fixed total pressure ratio and the heat conductance distribution of the regenerator, there exists an optimal intercooling pressure ratio and a group of optimal heat conductance distributions of the hot-, cold- and consumer-side heat exchangers and the intercooler which lead to a maximum dimensionless profit rate. When the total pressure ratio is variable, there exists an optimal total pressure ratio which leads to a double-maximum dimensionless profit rate. The double-maximum dimensionless profit rate increases monotonously with the increases of total heat conductance, price ratio, efficiencies of the compressors and the turbine and pressure recovery coefficients, meanwhile the exergy efficiency, total pressure ratio and heat conductance distributions corresponding to the double-maximum dimensionless profit rate are obtained. There exists an optimal consumer-side temperature and an optimal thermal capacitance rates matching between the heat reservoir and the working fluid which lead to a thrice-maximum dimensionless profit rate, respectively.

This article mainly investigates the design work condition of the cogeneration plant. The next work will be the research about the real operation work condition and the comparison between the theoretical and real cogeneration plants. The results obtained in this article may be helpful to search for the optimal design parameters of practical intercooled regenerative gas turbine cogeneration plant.

5 CONCLUSION

Finite-time exergoeconomic performance of a real gas turbine cogeneration plant model is optimized by detailed numerical examples. The main conclusions are as follows:

ACKNOWLEDGEMENTS

This manuscript is supported by the National Natural Science Foundation of P. R. China (Project No. 10905093). The authors thank the reviewers for their careful, unbiased and constructive suggestions, which led to this revised manuscript.

REFERENCES

[1] Chen L, Yang B, Sun F. Finite time exergoeconomic performance of a real intercooled regenerated gas turbine cogeneration plant. Part 1: model description and parametric analyses. Int J Low-Carbon Technol 2012; 10.1093/ijlct/cts031.
[2] Yilmaz T. Performance optimization of a gas turbine-based cogeneration system. J Phys D: Appl Phys 2006; 39:2454–8.
[3] Hao X, Zhang G. Maximum useful energy-rate analysis of an endoreversible Joule-Brayton cogeneration cycle. Appl Energy 2007; 84:1092–101.
[4] Hao X, Zhang G. Exergy optimisation of a Brayton cycle-based cogeneration plant. Int J Energy 2009; 6:34–48.
[5] Ust Y, Sahin B, Yilmaz T. Optimization of a regenerative gas-turbine cogeneration system based on a new exergetic performance criterion: exergetic performance coefficient. Proc IMech E Part A: J Power Energy 2007; 221:447–58.
[6] Tao G, Chen L, Sun F, et al. Exergoeconomic performance optimization for an endoreversible simple gas turbine closed-cycle cogeneration plant. Int J Ambient Energy 2009; 30:115–24.
Finite-time exergoeconomic performance of a real intercooled regenerated gas turbine cogeneration plant

[7] Tao G, Chen L, Sun F. Exergoeconomic performance optimization for an endoreversible regenerative gas turbine closed-cycle cogeneration plant. Rev Mex Fis 2009;55:192–200.

[8] Chen L, Tao G, Sun F. Finite time exergoeconomic optimal performance for an irreversible gas turbine closed-cycle cogeneration plant. Int J Sustainable Energy 2012;31:43–58.

[9] Chen L, Yang B, Sun F, et al. Exergetic performance optimisation of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 1: thermodynamic model and parametric analysis. Int J Ambient Energy 2011;32:116–23.

[10] Chen L, Yang B, Sun F. Exergoeconomic performance optimization of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 1: thermodynamic model and parameter analyses. Int J Energy Environ 2011;2:199–210.

[11] Yang B, Chen L, Sun F, et al. Exergetic performance optimisation of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 2: exergy output rate and exergy efficiency optimisation. Int J Ambient Energy 2012;33:98–104.

[12] Yang B, Chen L, Sun F. Exergoeconomic performance optimization of an endoreversible intercooled regenerated Brayton cogeneration plant. Part 2: heat conductance allocation and pressure ratio optimization. Int J Energy Environ 2011;2:211–8.

[13] Yang B, Chen L, Sun F. Exergoeconomic performance analyses of an endoreversible intercooled regenerative Brayton cogeneration type model. Int J Sustainable Energy 2011;30:65–81.

[14] Yang B, Chen L, Sun F. Finite time exergoeconomic performance of an irreversible intercooled regenerative Brayton cogeneration plant. J Energy Inst 2011;84:5–12.

[15] Wang W, Chen L, Sun F, et al. Power optimization of an endoreversible intercooled regenerated Brayton cycle. Int J Therm Sci 2005;44:89–94.

[16] Wang W, Chen L, Sun F, et al. Power optimization of an endoreversible closed intercooled regenerated Brayton cycle coupled to variable-temperature heat reservoirs. Appl Energy 2005;82:181–95.

[17] Wang W, Chen L, Sun F, et al. Optimal heat conductance distribution and optimal intercooling pressure ratio for power optimization of an irreversible closed intercooled regenerated Brayton cycle. J Energy Inst 2006;792:116–9.

[18] Wang W, Chen L, Sun F, et al. Power optimization of an irreversible closed intercooled regenerated Brayton cycle coupled to variable-temperature heat reservoirs. Appl Therm Eng 2005;25:1097–113.

[19] Wang W, Chen L, Sun F, et al. Efficiency optimization of an irreversible closed intercooled regenerated gas-turbine cycle. Proc IMechE Part A: J Power Energy 2006;220:551–8.

[20] Chen L, Wang J, Sun F. Power density optimisation of an endoreversible closed intercooled regenerated Brayton cycle. J Energy Inst 2007;80:105–9.

[21] Chen L, Wang J, Sun F, et al. Power density optimization of an endoreversible closed variable-temperature heat reservoir intercooled regenerated Brayton cycle. Int J Ambient Energy 2006;27:99–112.

[22] Chen L, Wang J, Sun F. Power density analysis and optimization of an irreversible closed intercooled regenerated Brayton cycle. Math Comput Model 2008;48:527–40.

[23] Chen L, Wang J, Sun F, et al. Power density optimization of an irreversible variable-temperature heat reservoir closed intercooled regenerated Brayton cycle. Int J Ambient Energy 2009;30:9–26.

[24] Wang W, Chen L, Sun F. Ecological optimisation of an irreversible ICR gas turbine cycle. Int J Exergy 2011;9:66–79.

[25] Khaliqu A, Khan TA. Energetic and exergetic efficiency analysis of an indirect fired air-turbine combined heat and power system. Int J Exergy 2007;4:38–53.

[26] Reddy B V, Butcher C. Second law analysis of a natural gas-fired gas turbine cogeneration system. Int J Energy Res 2009;33:39:728–36.

[27] Sanaye S, Ziaabsharhagh M, Ghazinejad M. Optimal design of gas turbine CHP plant with preheater and HRSG. Int. J Energy Res 2009;33:766–77.

[28] Khaliqu A, Choudhary K, Dincer I. Exergy analysis of a gas turbine trigeneration system using the Brayton refrigeration cycle for inlet air cooling. Proc IMechE Part A: J Power Energy 2010;224:449–61.

[29] Qin J, Zhou W, Bao W, et al. Thermodynamic analysis and parametric study of a closed Brayton cycle thermal management system for scramjet. Int J Hydrogen Energy 2010;35:356–64.

[30] Ahmadi P, Dincer I. Thermodynamic and exergoenvironmental analyses, and multi-objective optimization of a gas turbine power plant. Appl Therm Eng 2011;31:2529–40.

[31] Khaliqu A, Dincer I. Energetic and exergetic performance analyses of a combined heat and power plant with absorption inlet cooling and evaporative aftercooling. Energy 2011;36:2662–70.