A parameter optimization method for actual thermal system

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ABSTRACT

The objective of the parameter optimization of thermal system is to increase the system efficiency and decrease the cost of unit output power under the condition which the total output power can meet the actual requirements in energy applications. In this paper, a new parameter optimization method for actual thermal system was proposed, which was called the Equivalent Transformation Constant State Finite-time Thermodynamic Analysis Method and had two calculation steps. The first step was choosing the optimum efficiency under the principle of taking into account both output power and efficiency weights, and the second step was calculating the optimum total heat transfer coefficient ratio and other concerned parameters with the optimum efficiency fixed. The related calculation principles and formulas for parameter optimizations of this method were given. That provided the guiding theoretical basis for actual thermal system efficiency increase and cost decrease of unit output power.

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1. Introduction

Thermal system is a device which outputs work and discharges waste heat to the environment by the cyclic heat absorption of working medium from heat source. It mainly includes high and low temperature heat exchangers, expansion engine, liquid circulating pump. The main parameters of thermal system are heat source temperatures, heat transfer temperature difference and rates of heat exchangers, output power and system efficiency. The objective of the parameter optimization of thermal system is to increase the system efficiency and decrease the cost of unit output power under the condition which the total output power can meet the actual requirements in energy applications.

The efficiency objective of thermal system given by classical thermodynamics is the Carnot engine efficiency, which is $\eta_c = 1 - T_L/T_H$. T_L and T_H are the temperatures of high and low temperature heat sources, respectively. The Carnot engine efficiency is obtained under three specific conditions, which are: the heat exchanger has no heat transfer temperature difference, the heat exchange area of heat exchanger or the heat exchange time between working medium and heat source is infinite, and the total output power is equal to 0. But for the actual thermal system, all of the three specific conditions above are invalid. In 1975, Curzon and Ahlborn gave the relationship between efficiency and output power of the heat engine with finite rate and cycle time based on the endo-reversible Carnot cycle model, and gave the

endo-reversible Carnot cycle heat engine efficiency under the maximum output power (η_{ca}) [1]. Then some other researchers made lots of efforts on the finite-time thermodynamics analysis method and related research work [2–9]. For instance, a method for the optimization of thermal systems, based on finite time thermodynamics, was proposed by Stitou and Feidt [10], and some finite time optimization analyses of refrigerators were carried out [11,12]. As the alternating finite-time analysis method simplifies many practical problems, some of the corresponding analysis results are unacceptable in engineering. For now, the relationship between efficiency and output power of actual thermal system is still obtained by experiments.

Thus a parameter optimization theory is imperative, which must be convenient for actual thermal system applications. In this paper, a new parameter optimization method for actual thermal system was proposed, which was named the Equivalent Transformation Constant State Finite-time Thermodynamic Analysis Method. The related principles, operating steps and calculation formulas for this method were given, aiming at providing the guiding theoretical basis for system efficiency increase and cost decrease of unit output power from the viewpoints of thermodynamics and heat transfer.

2. Equivalent thermodynamic transformation of thermal system

Equivalent thermodynamics transformation is the basis of equivalent transformation finite-time thermodynamics analysis method [13,14]. A superheated steam cycle thermal system was

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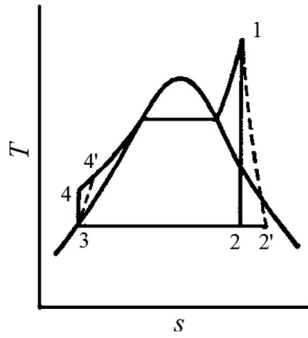   E-mail address: kinggang009@163.com (G. Wang).
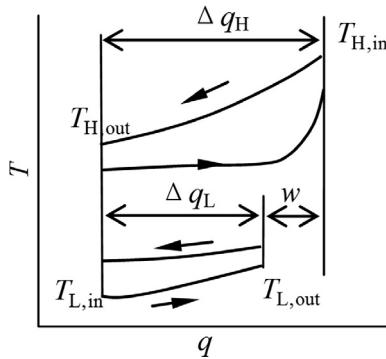
Fig. 1. *T-s* curve of superheated Rankine cycle.

Fig. 2. *T-q* relationship corresponding to Fig. 1.

chosen as the instance here. The temperature-entropy (*T-s*) curve and heat transfer processes (*T-q*) are presented in Figs. 1 and 2.

The purpose of equivalent thermodynamics transformation was to give the relationships of heat quantity, entropy change and temperature of different processes in Figs. 1 and 2 according to the thermodynamics definition equation of entropy ($dq = Tds$). For a reversible process from the initial State a to final State b, the equivalent thermodynamics temperature $T_{R,\,ab}$ of a-b process was:

$$T_{R,ab} = \Delta q_{ab}/\Delta s_{ab} \tag{1a}$$

where Δq_{ab} was the heat change of unit mass of working medium from State a to State b, including the heat quantities transferred with outside system and generated by endo-irreversible process. Δs_{ab} was the entropy change of unit mass of working medium from State a to State b, which was a function of state and had nothing to do with the process. For a constant state, assuming the heat flux rate transferred in a-b process was ϕ and the theoretical cycle working medium mass flow rate was Q_m, then the equivalent thermodynamics temperature of a-b process was:

$$T_{R,ab} = \frac{\phi}{Q_m \Delta s_{ab}} \tag{1b}$$

where the units of $T_{R,\,ab}$, ϕ, Q_m and Δs_{ab} were K, W, kg/s and J/(K kg), respectively.

2.1. Equivalent temperature of heat source

For Fig. 2, it was assumed that the mass flow rates of fluid in high and low temperature heat sources were $Q_{m,H}$ and $Q_{m,L}$, the inlet and outlet temperatures of fluid in the two heat sources were $T_{H,in}$, $T_{H,out}$, $T_{L,in}$ and $T_{L,out}$, and the heat transfer flux rates of high and low temperature heat exchangers were ϕ_H and ϕ_L. By the transformation of Δq_{ab} in Eqs. (1), the equivalent temperatures of the two heat sources of thermal system were as follows:

$$T_{R,H} = \frac{\phi_H}{Q_{m,H}\Delta s_H} = \frac{T_{H,in} - T_{H,out}}{\ln(T_{H,in}/T_{H,out})} \tag{2}$$

$$T_{R,L} = \frac{\phi_L}{Q_{m,L}\Delta s_L} = \frac{T_{L,out} - T_{L,in}}{\ln(T_{L,out}/T_{L,in})} \tag{3}$$

2.2. High and low temperatures of equivalent Carnot cycle

The typical cycle of thermal system is presented in Fig. 1, which included high temperature heat absorption, gas expansion, low temperature heat release and liquid squeeze processes. 1-2-3-4 and 1-2′-3-4′ were theoretical and actual cycles of superheated steam, respectively. The theoretical cycle was comprised of constant pressure high temperature heat absorption, constant entropy gas expansion, constant pressure low temperature heat release and constant entropy liquid squeeze processes. There was $\Delta s_{14} = \Delta s_{13} = \Delta s_{23}$ in the theoretical cycle, so the Rankine cycle could be equivalently transformed to the Carnot cycle comprised of two constant temperature processes and two constant pressure processes just by transforming the temperatures of high and low temperature constant pressure processes of theoretical cycle to equivalent high and low temperatures. The transforms of the two equivalent temperatures were:

$$T_{R,1} = \frac{\Delta q_H}{\Delta s_{14}} = \frac{\Delta q_{14}}{s_1 - s_3} = \frac{h_1 - h_4}{s_1 - s_3} = \frac{\phi_H}{Q_m(s_1 - s_3)} \tag{4}$$

$$T_{R,2} = \frac{\Delta q_L}{\Delta s_{23}} = \frac{\Delta q_{23}}{s_1 - s_3} = \frac{h_2 - h_3}{s_1 - s_3} = \frac{\phi_L}{Q_m(s_1 - s_3)} \tag{5}$$

As the cycle of actual heat engine was irreversible, the expansion and squeeze processes were both not constant pressure processes. Normally, we defined constant entropy expansion coefficient ($\eta_T = w'_{exp}/w_{exp}$) and constant entropy squeeze coefficient ($\eta_c = w'_{sq}/w_{sq}$) to express the expansion and squeeze processes, where w' was the actual work quantity. When the Rankine cycle was equivalently transformed to the Carnot cycle, 1-2′-3-4′ could be divided to two recombination processes. 3-4′-1 process included high temperature heat absorption and liquid squeeze processes, and 1-2′-3 was comprised of low temperature heat exchange and gas expansion processes. The entropy values at the initial and final points of the two recombination processes had nothing to do with the processes and the absolute values of the two entropy differences were the same ($\Delta s_{13} = -\Delta s_{31}$). By the two complex processes, the equivalent high and low temperatures of the equivalent Carnot cycle could be given as:

$$T'_{R,1} = \frac{\Delta q'_H}{\Delta s_{13}} = \frac{h_1 - h_4}{s_1 - s_3} = \frac{h_1 - h_3 - (h_4 - h_3)/\eta_c}{s_1 - s_3} = \frac{\phi_H}{Q'_m(s_1 - s_3)} \tag{6}$$

$$T'_{R,2} = \frac{\Delta q'_L}{\Delta s_{13}} = \frac{h_1 - h_3 - (h_1 - h_2)\eta_T}{s_1 - s_3} = \frac{\phi_L}{Q'_m(s_1 - s_3)} \tag{7}$$

where Q'_m was the mass flow rate of actual cycle working medium ($Q'_m/Q_m = \Delta q_H/\Delta q'_H$). The equivalent transformation of actual thermal system was completed by Eqs. (2)–(7).

3. Efficiency and output power of equivalent endo-reversible Carnot cycle heat engine

3.1. Constant state finite-time analysis method

According to the thermodynamics transformation introduced in Section 2, the actual thermal system presented in Figs. 1 and 2 could be equivalent to the endo-reversible Carnot cycle model [15], which is presented in Fig. 3.