Applied Energy 113 (2014) 982–989

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

An optimization method for gas refrigeration cycle based on the combination of both thermodynamics and entransy theory

Qun Chen *, Yun-Chao Xu, Jun-Hong Hao

Key Laboratory for Thermal Science and Power Engineering of Ministry of Education, Department of Engineering Mechanics, Tsinghua University, Beijing 100084, China

HIGHLIGHTS

- An optimization method for practical thermodynamic cycle is developed.
- The entransy-based heat transfer analysis and thermodynamic analysis are combined.
- Theoretical relation between system requirements and design parameters is derived.
- The optimization problem can be converted into conditional extremum problem.
- The proposed method provides several useful optimization criteria.

ARTICLE INFO

Article history:
Received 8 March 2013
Received in revised form 17 August 2013
Accepted 17 August 2013

Keywords:
Gas refrigeration system
Optimization
Thermodynamics
Entransy

ABSTRACT

A thermodynamic cycle usually consists of heat transfer processes in heat exchangers and heat-work conversion processes in compressors, expanders and/or turbines. This paper presents a new optimization method for effective improvement of thermodynamic cycle performance with the combination of entransy theory and thermodynamics. The heat transfer processes in a gas refrigeration cycle are analyzed by entransy theory and the heat-work conversion processes are analyzed by thermodynamics. The combination of these two analysis yields a mathematical relation directly connecting system requirements, e.g. cooling capacity rate and power consumption rate, with design parameters, e.g. heat transfer area of each heat exchanger and heat capacity rate of each working fluid, without introducing any intermediate variable. Based on this relation together with the conditional extremum method, we theoretically derive an optimization equation group. Simultaneously solving this equation group offers the optimal structural and operating parameters for every single gas refrigeration cycle and furthermore provides several useful optimization criteria for all the cycles. Finally, a practical gas refrigeration cycle is taken as an example to show the application and validity of the newly proposed optimization method.

1. Introduction

Refrigeration cycle, a typical thermodynamic system, has been widely utilized in every corner of the world to provide specific environmental conditions. Effective improvement of refrigeration cycle performance has been an attractive but tough issue in both research and engineering fields, which on one hand has huge potential for energy conservation, but on the other hand is a multi-parameter coupling problem including several different physical phenomena.

During the past several decades, a large number of approaches for thermodynamic cycle performance improvement have been explored and employed in engineering, which can be classified into two categories. One category is to find a better cycle with higher theoretical conversion efficiency between thermal energy and power including gas–steam combined cycles [1], integrated gasification combined cycles (IGCC) [2], and combined cooling, heating and power (CCHP) systems [3]. The other category is to improve the existing system's performance, where researchers usually list several possible combinations of such structural and operating parameters as pressures, temperatures and mass flow rates, estimate their influences on the system performance, and finally find a better solution by experiments or some simulation-based optimization methods including gradient-based algorithm, genetic algorithm and neural network algorithm [4–6]. These methods can great improve the system performance, especially very complex systems, and have successfully reduced not only the energy consumption but also the equipment cost. However, compared with the theoretical optimization methods, these methods share weaker physical analyses and rely more on the computer to empirically choose a better solution among huge amounts of possible parameter configurations.

* Corresponding author. Tel.: +86 10 62796332.
E-mail address: chenqun@tsinghua.edu.cn (Q. Chen).

0306-2619/$ - see front matter © 2013 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.apenergy.2013.08.039

Nomenclature

A	area, m^2
c_p	constant pressure specific heat, J kg^{-1} K^{-1}
G	entransy flow rate, W K
K	heat transfer coefficient, W m^{-2} K^{-1}
KA	thermal conductance, W K^{-1}
m	mass flow rate, kg s^{-1}
n	polytropic index
p	pressure, Pa
q	heat flux, W m^{-2}
Q	heat flow rate, W
T	temperature, K
W_0	power consumption rate, W
Π	Lagrange function
Φ_g	entransy dissipation rate, W K
λ	Lagrange multiplier
ξ	fluid arrangement factor of heat exchanger

Subscripts

a	air
C	compression
E	expansion
h	hot fluid, hot end of thermodynamic cycle
in	inlet
l	cold fluid, cold end of thermodynamic cycle
out	outlet
γ	adiabatic exponent

In parallel, many researchers optimize thermodynamic cycle performance from the viewpoint of irreversibility. Curzon and Ahlborn [7] and Salamon et al. [8,9] focused on the power output maximization of Carnot cycles and mentioned that when the area of each heat exchanger is specified, the finite-time irreversibility of heat transfer should be taken into account. Afterwards, Bejan et al. [10–13] optimized several thermodynamic cycles including both power and refrigeration plants by allocating the areas of each heat exchanger based on the combination of finite-time thermodynamics and entropy theory. Thenceforth, this method attracted attention of many researchers. They advocated meaningful efforts on the optimization problems with different objectives through the area allocation of heat exchangers in different thermodynamic cycles. For instance, Grazzini and Rinaldi [14], Sahin and Kodal [15], Klein [16], and Ait-Ali [17] focused on the maximization of the coefficient of performance and the cooling capacity rates of refrigerators. Chen et al. [18] studied the total area variation of all heat exchangers versus their area allocation at the hot and cold ends to minimize the total area. Sarkar and Bhattacharyya [19] derived a brief expression of the total heat exchanger area in terms of working fluid temperatures for an irreversible refrigeration cycle and obtained the optimal fluid temperatures to minimize the total area.

All the aforementioned developments effectively improve the thermodynamic system performance. However, many studies share a common hypothesis that the thermodynamic cycle is divided into three parts, i.e. two heat exchangers at hot and cold ends with irreversible heat transfer processes and a thermodynamic cycle between them simplified as ideal Carnot cycle, which cannot offer any information about such components as working fluids, turbines and compressors inside the thermodynamic cycle. As a result, they can only optimize heat exchanger performance without considering the influence of other design parameters, such as the physical properties and heat capacity rates of working fluids and the performance of turbines, compressors and expanders. Therefore, it is highly desired to develop a new method for thermodynamic cycle optimization with comprehensive considerations of both heat-work conversion and heat transfer performance.

Chen et al. [20] applied the finite-time thermodynamics to analyze an air refrigeration cycle and maximized the COP and the cooling load of the cycle. Thereafter, Chen et al. [21] optimized the cycle with the aim of maximizing the exergetic efficiency. Tu et al. [22] maximized the COP of a real air-refrigerator by finding optimal allocation of heat exchanger inventory and considered the influencing factors, e.g. the total heat exchanger inventory and the efficiency of compressor and expander. Liu et al. [23] discussed the relation between the optimal COP and pressure ratio. They have made some contributions for the theoretical optimization of the air refrigeration cycle. However, in order to make the complex optimization simpler, they always reduce the degrees of freedom of the system to be single degree via fixing other design parameters, which can lead to good local optimization results but may miss the global optimal configuration of all the design parameters.

For heat transfer, Guo et al. [24] recently introduced the physical quantities of entransy and entransy dissipation to respectively represent the heat transfer ability of an object during a time period and describe the irreversibility of a heat transfer process. Furthermore, they proposed an entransy dissipation-based method for optimization of heat transfer elements [25–29], heat exchangers [30–34], and heat exchanger networks [35–37] in practical engineering applications. Besides, Chen et al. extended entransy theory to analyze and optimize mass transfer [38] and coupled heat and mass transfer processes [39–41] based on the analogy between heat and mass transfer.

In order to consider the combined influence of working fluids, heat exchangers, turbines and compressors on thermodynamic cycle optimization, this paper proposes an optimization method based on the combination of both thermodynamics and entransy theory. Gas refrigeration cycle is taken as an example and analyzed through not only thermodynamic analysis for heat-work conversion processes in a compressor and an expander, but also entransy analysis for heat transfer processes in heat exchangers. Both analyses are combined to develop a new method for the structural and operation parameter optimization. Finally, a practical gas refrigeration cycle is taken as an example to show the application and superiority of the newly proposed method.

2. Analysis of a gas refrigeration cycle

Fig. 1 shows the sketch of a typical gas refrigeration cycle consisting of a compressor, an expander and two counter-flow heat exchangers at the hot and cold ends, HX_h and HX_l. At the cold end, the fluid from low-temperature environment flows into the HX_l and heats the gas, i.e. the working fluid in the refrigeration cycle, from the temperature T_4 to T_1. The heated gas leaves the HX_l and enters the compressor, C, where gas is compressed from the pressure p_1 to p_2 and its temperature correspondingly rises to T_2. After compression, the gas enters the HX_h and is cooled to T_3 by the fluid from high-temperature environment. Finally, the gas enters the expander, E, and expands from the pressure p_3 to p_4 and its temperature drops to T_4. In Fig. 1, W_1 and W_2 are the input and output works of the compressor and the expander, respectively, m is the mass flow rate, and the subscripts a, l, h, in and