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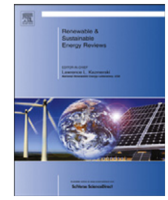

Finite-time thermodynamics optimization of absorption refrigeration systems: A review

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ABSTRACT

This paper presents a literature review of the optimization of absorption refrigeration systems based on finite-time thermodynamics. An overview of the various objective functions is presented.

Contents

1. Introduction

The absorption refrigeration systems are thermodynamic processes which produce cold thanks to thermal energy. Then, they exchange heat with at least three sources at different temperatures without receiving work. A three-heat-source reversible refrigerator operates between heat hot reservoir, heat cold reservoir and heat sink. When T_H, T_L and T_O denote the temperatures of heat hot reservoir, heat cold reservoir and heat sink respectively, the coefficient of performance for three-heat-source reversible refrigerators is expressed as: $\varepsilon_r=[(T_H-T_O)/T_H][T_L/(T_O-T_L)]$ [1]. This expression reveals the product of thermal efficiency of Carnot cycle for heat engines working between T_H and T_O and coefficient of performance of reversible Carnot refrigerator producing cold at T_L and rejecting heat at T_O: $\varepsilon_r=\eta_C\times\varepsilon_C$ with $\eta_C=(T_H-T_O)/T_H$ and $\varepsilon_C=T_L/T_O-T_L$. In classical thermodynamics, the efficiency of a cycle operating on reversibility principles proposed by Carnot [2] became the upper bound of thermal efficiency for heat engines that work between the same temperature limits. This equally applies to the coefficient of performance of refrigeration cycles that execute a reversed Carnot cycle (Carnot refrigerator). This implies that the coefficient of performance defined above is the maximum coefficient of performance for three-heat-source refrigerators from the point of view of classical thermodynamics.

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Nomenclature

A	total heat-transfer area (m^2)
A_A	heat-transfer area of absorber (m^2)
A_C	heat-transfer area of condenser (m^2)
A_L	heat-transfer area of evaporator (m^2)
A_H	heat-transfer area of generator (m^2)
A_O	A_A+A_C
$ECOP$	ecological coefficient of performance
I	internal irreversibility parameter
K_{LC}	heat leak coefficient (W K)
K_H	thermal conductance of heat source ($W K^{-1}$)
K_L	thermal conductance of cooled space ($W K^{-1}$)
K_O	thermal conductance of heat sink ($W K^{-1}$)
ncu	national currency unit
$\dot{Q}_A$	heat reject load from absorber to heat sink (W)
$\dot{Q}_C$	heat reject load from condenser to heat sink (W)
$\dot{Q}_L$	heat input load from cooled space to evaporator (W)
$\dot{Q}_H$	heat input load from heat source to generator (W)
$\dot{Q}_O$	$\dot{Q}_C+\dot{Q}_A$
R	cooling load (W)
R_m	cooling rate at maximum coefficient of performance (W)
r	specific cooling load ($W m^{-2}$)
r_m	specific cooling rate at maximum coefficient of performance ($W m^{-2}$)
T_1	temperature of working fluid in generator (K)
T_2	temperature of working fluid in evaporator (K)
T_3	temperature of working fluid in absorber and condenser (K)
T_3	temperature of working fluid in absorber and condenser (K)
T_A	temperature of the absorber-side heat sink (K)
T_C	temperature of the condenser-side heat sink (K)
T_{env}	temperature in environmental conditions
T_H	temperature of the heat source (K)
T_L	temperature of the cooled space (K)
T_O	$T_A=T_C$
U_H	overall heat-transfer coefficient of generator ($W K/m^2$)
U_L	overall heat-transfer coefficient of evaporator ($W K/m^2$)
U_O	overall heat-transfer coefficient of absorber and condenser ($W K/m^2$)
$\dot{W}$	power output (W)

Symbol

ε	coefficient of performance for absorption refrigerators
ε_C	coefficient of performance of reversible Carnot refrigerator
η_C	thermal efficiency of Carnot cycle
λ	Dissipation coefficient of cooling rate
σ	Entropy generation rate (W/K)
ε_I	coefficient of performance for three-heat-source refrigerator affected only by internal irreversibility
ε_r	coefficient of performance for reversible three-heat-source refrigerator
ε_m	coefficient of performance at maximum cooling rate

Subscripts

max	maximum

However, since the absorption refrigeration cycles are in direct contact with reservoirs and sink, the heat transfers during the isothermal processes are supposed to be carried out infinitely slowly. Therefore, duration of the processes will be infinitely long and hence it is not possible to obtain a certain amount of cooling load $\dot{Q}_L$ with heat exchangers having finite heat-transfer areas, i.e. $\dot{Q}=0$ for $0<A<\infty$. If we require certain amount of cooling load in an absorption refrigerator executing a reversible cycle, the necessary heat exchanger area would be infinitely large, i.e. $A\rightarrow\infty$ for $\dot{Q}>0$.

Thus in classical thermodynamics the real absorption refrigerators producing cold with a certain amount of cooling load are compared with the ideal absorption refrigerators developing no cooling load. In other words the performance of an absorption refrigerator of given size (in term of total heat-transfer area) is measured with an ideal absorption refrigerator which would require an infinite total heat-transfer area to produce the same amounts of cooling load. In practice, all absorption refrigeration processes take place in finite-size devices in finite-time; therefore, it is impossible to meet reversibility conditions between the absorption refrigeration system and the surroundings. For this reason, the reversible absorption cycle cannot be considered as a comparison standard for practical absorption refrigeration systems from the view of cooling load on size perspective, although it gives an upper bound for coefficient of performance. The performance bound of classical thermodynamics [3–6] is highly important in theory, but it is usually too rough to predict the coefficient of performance of practical absorption refrigerators. Therefore, it is necessary to establish the bound of finite-time thermodynamics [7].

The finite-time thermodynamics has been first proposed by Chambadal [8] and Novikov [9] independently on1957, then popularized in many works including Curzon and Ahlborn [10], De Vos [11], Sieniutycz et al. [12], Bejan [13–18], Wu [19],Chen [20], Stitou [21,22], Feidt [23,24], Leff and Teeters [25], Blanchard [26], Stitou and Feidt [27], Andresen [28], Sieniutycz and Salamon [29], De Vos [30], Bejan et al. [31], Bejan and Mamut [32], Berry et al. [33], Radcenco [34] and in many review articles including Sieniutycz and Shiner [35], Chen et al. [36], Hoffmann et al. [37] and Durmayaz et al. [38].

The finite-time thermodynamics tends to model the real systems in a way closer to reality and enable to distinguish the irreversibilities due to internal dissipation of the working fluid from those due to finite-rate heat transfer between the system and the external heat reservoirs and heat sink.

The objective of this paper is to review the present state of optimization of absorption refrigeration processes based on finite-time thermodynamics. The different performance optimization criteria are provided and discussed.

2. Optimization based on the coefficient of performance and cooling load criteria

2.1. Three-heat-source absorption refrigerator

An absorption refrigeration system (equivalent to three-heat-reservoir refrigeration system) affected by the irreversibility of finite rate heat transfer may be modeled as a combined cycle which consists of an endoreversible heat engine and an endoreversible