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Thermodynamic optimization of irreversible refrigerators

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ABSTRACT

An irreversible inverse cycle, operating at steady state conditions with finite thermal capacity heat sources, is analyzed in order to obtain an expression for the coefficient of performance accounting for the Second Law. Some dimensionless parameters are proposed to link the entropy variation rate and the temperature differences at the heat exchangers to the cycle efficiency. A maximum for efficiency appears when a parameter depending only on the temperature of the inlet streams at each heat exchanger is used. The influence of dimensionless parameters and irreversibilities on the maximum cycle efficiency is analyzed. A graphical analysis, based on data from literature, is presented to show the use of this thermodynamic optimization criteria in design and verification process of refrigerators.

1. Introduction

The analysis of thermal cycles was often accompanied by the formulation of thermodynamic optimization criteria useful to improve their performance. Finite-time thermodynamic has certainly played an important role, introducing the finite duration of each thermodynamic transformation in a cycle and hence moving the attention toward non-equilibrium concepts and new irreversibility sources. Moreover general analytical models were proposed to modeling endoreversible or irreversible thermal machines. A comprehensive list of works and current trends on finite time thermodynamic can be found in [1]. Feidt [2] proposed a extensive review focused on thermodynamic models of reverse cycle machines and Huleihil et al. [3] present a finite-time general model of real refrigerators applied to various types of refrigeration/heat pump systems as the thermoelectric refrigerator, the reverse Brayton cycle and the reverse Rankine cycle. In [4] and [5] the authors used an optimization model of irreversible Carnot refrigerators based on finite mass flow rate and finite size to investigate the maximum COP and obtain analytical expressions for the relevant optimum performance parameters. Optimization criteria based on thermodynamic parameters were the most used as cycle efficiency, power input and specific cooling load. Other criteria have been proposed as exergy optimization [6], called "thermo ecological criterion ECOP", which was defined as the ratio of power output to the loss rate of availability. Recently, Xu et al. [7] used a modified ecological optimization criterion, substituting the power output with the exergy output rate, defined as the ratio of availability to cycle period. The authors applied this optimization criterion to an irreversible Carnot refrigerator. In [8,9] the authors proposed an available power exchange optimization based on finite-time exergy model for irreversible cycles. Thermo-economic criteria were also used and Le et al. [10] proposed their profit rate performance optimization criterion by searching the optimum COP at maximum profit, which is termed as the finite-time exergoeconomic performance bound, applied to an endoreversible Carnot refrigerator and heat pump. In [11] energetic, exergetic, economic and exergoeconomic analyses of a transcritical CO_2 refrigeration was presented.

Moreover, Acikkalp [12] used the entrancy, a parameter recently introduced in thermodynamics [13], to analyze the performance of an irreversible refrigeration cycle referred to dimensionless temperature ratio and dimensionless heat conductance ratio. In our opinion, entransy does not contain any new information in comparison with a classical analysis of systems using entropy [14,15].

A persistent limit, in all the criteria above indicated, is the Carnot-shape assigned to the refrigeration cycle, with constant temperature or mean temperature at the heat transfer interfaces. This condition was avoided applying the thermodynamic optimization to air (Brayton) refrigeration cycles, because Newton's law and ideal gas behavior of the refrigerant in thermal exchanges were admitted [16–19].

Based on general thermodynamic analytical models for a vapor compression reverse cycle refrigerator, this paper aims to overcome the aforementioned Carnot-shape limit, introducing actual temperatures variation at both sides of the heat exchangers of refrigerator through proper dimensionless parameters. A performance optimization method is presented and discussed and a

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Nomenclature

COP	coefficient of performance
c_{pl}	specific heat of the refrigerant in liquid phase (J K^{-1} kg^{-1})
c_{pv}	specific heat of the refrigerant in vapor phase (J K^{-1} kg^{-1})
G	dimensionless parameter defined as $\Delta S'_F/(k_C\,\Theta_{FC})$
h	specific enthalpy (J kg^{-1})
k	thermal power over temperature difference or heat exchanger inventory (W K^{-1})
m	flow rate (kg s^{-1})
p	pressure (Pa)
Q	thermal power (W)
r	latent heat (J kg^{-1})
R	perfect gas constant (J K^{-1} kg^{-1})
s	specific entropy (J K^{-1} kg^{-1})
S'	entropy rate (W K^{-1})
t	dimensionless parameter depending on temperatures at the heat exchangers
T	temperature (K)
v	specific volume (m^3 kg^{-1})
x	quality
W	mechanical power (W)
Greek	
β	volumetric expansion coefficient
Δ	difference
ε	heat exchanger efficiency
η_{II}	second law efficiency
ϕ_t	dimensionless parameter $(k_H\,T_{Hi})/(k_C\,T_{Ci})$
ϕ_Θ	dimensionless parameter $(k_H\,\Theta_{FH})/(k_C\,\Theta_{FC})$
Θ	dimensionless parameter defined in Eqs. (9) and (10)
Subscripts	
C	relative to lower T
F	relative to refrigerant
H	relative to higher T
i	inlet
max COP	related to the maximum value of COP
o	outlet
sat	relative to saturation condition

maximum COP_F as a function of a parameter depending on the temperature difference at both heat exchangers is shown. Some experimental data from literature are used to compare the results and to explain how this approach could be useful in diagnostic and design of refrigerators.

2. Thermodynamic analysis

2.1. The steady state balance

Let us consider a steady–state refrigerator working between two fluids as an open system. The system, using external work, extracts heat from a cold fluid stream and assigns it to a higher temperature stream (Fig. 1).

If pressure losses, kinetic energy and gravitational potential are neglected, first law states

$$W_F = +Q_C - Q_H \tag{1}$$

where Q_H and Q_C represent the thermal power exchanged.

The cycle efficiency COP_F is defined as

$$COP_F = \frac{|Q_C|}{|W_F|} \tag{2}$$

The entropy rate variation for a fluid is:

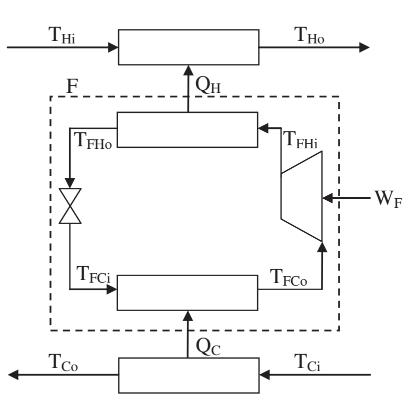

Fig. 1. Scheme of vapor compression refrigerator.

$$\Delta S' = m\Delta s = m\left(\int_i^o \frac{dh}{T} - \int_i^o \left(\frac{\partial v}{\partial T}\right)_p dp\right) \tag{3}$$

In general, it is possible to state

$$\int_i^o \left(\frac{\partial v}{\partial T}\right)_p dp = l\Delta p \tag{4}$$

where $l = \beta v$ for a liquid, and $l = R/p$ for a perfect gas [20]. Eq. (4) includes terms linked to pressure change, i.e. pressure drop contribution, and they can be considered within internal irreversibilities. In a closed cycle the entropy variation of the refrigerant is null, hence the internal irreversibilities should be transferred outside the cycle through heat exchangers, when other thermal exchanges are neglected. Therefore the rate of entropy transfer is evaluated as

$$\Delta S'_F = \Delta S'_{FH} - \Delta S'_{FC} \tag{5}$$

where entropy rate variations refer to inlet and outlet state of the refrigerant fluid side of the heat exchangers. Combining Eqs. (3) and (5) with the thermal power exchanged by fluid stream of the heat exchanger and neglecting pressure losses

$$Q = m\Delta h \tag{6}$$

Than we obtain:

$$\Delta S'_F = Q_H \frac{\left(\int_i^o \frac{dh}{T} - l\Delta P\right)_{FH}}{\Delta h_{FH}} - Q_C \frac{\left(\int_i^o \frac{dh}{T} - l\Delta P\right)_{FC}}{\Delta h_{FC}} \tag{7}$$

Entities within the fractions will be evaluated in relation to the physical state of the fluid and the heat exchanger characteristics.

2.2. Vapor compression refrigeration cycle

We consider the traditional vapor compression refrigerator scheme in Fig. 2 where polytropic curves represent the irreversible branches of the cycle and the cold and hot temperature changes of the external streams crossing the heat exchangers are indicated.

In a first approach, the pressure losses in the refrigerant side of each heat exchanger are neglected, even if they may have a significant impact on the entropy rate change [21], then constant temperatures within the phase changes are considered. The fluid is considered as an ideal gas with constant specific heat c_{pv} in superheating and de-superheating and as an incompressible liquid