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Thermoeconomic optimization and exergy analysis of CO_2/NH_3 cascade refrigeration systems

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In this paper, thermoeconomic optimization and exergy analysis are applied to a CO_2/NH_3 cascade refrigeration cycle. Cooling capacity, ambient temperature and cold space temperature are constraints of the optimization procedure. Four parameters including condensing temperature of ammonia, evaporating temperature of carbon dioxide, condensing temperature of carbon dioxide and temperature difference in the cascade condenser are chosen as decision variables. The objective function is the total annual cost of the system which includes costs of input exergy to the system and annualized capital cost of the system. Input exergy to the system is the electricity consumption of compressors and fans, and the capital cost includes purchase costs of components. Results show that, optimum values of decision variables may be found by trade-off between the input exergy cost and capital cost. Results of the exergy analysis for each of the system components in the optimum state are also given.

1. Introduction

In low temperature applications including rapid freezing systems and storage of frozen food where the evaporation temperature of the evaporator lies within a limit of −30 °C and −55 °C, as in industrial refrigeration where there is a high temperature difference between the heat source and the heat sink, it is not economical to use a single stage refrigeration system, since a high pressure ratio, high output pressure and temperature of the oil will result in a low volumetric efficiency and a low coefficient of performance of the system. Moreover, using a refrigerant in a wide temperature interval, results in reduction of evaporator pressure and higher suction volume and condenser pressure. Instead, for these applications, two-stage compression and cascade refrigeration systems can be used. Two-stage compression systems contain the same refrigerant for both high temperature and low temperature circuits, while cascade systems have different refrigerants for these two stages. Cascade systems can reach lower temperatures in comparison to two-stage systems.

Nowadays, many manufacturers tend to use natural refrigerants due to environmental problems and harmful effects of fluorocarbons. CO_2/NH_3 cascade system uses ammonia in the high temperature cycle and carbon dioxide in the low temperature cycle. Ammonia is a natural refrigerant with a pungent smell and it is toxic and to some extent, flammable. At a temperature below −35 °C, it has a vapor pressure lower than atmosphere pressure which may cause air leakage into the system. This is why it cannot be used in the low temperature circuit [1]. In contrast, carbon dioxide is a non-toxic, non-flammable gas with a positive vapor pressure at temperatures below −35 °C which makes it a suitable choice for low temperature cascade cycle [2].

So far, numerous methods have been developed for thermodynamic optimization of cascade refrigeration cycles. Ratts and Brown [3] used the entropy generation method to analyze the cascade cycle. In their work, relationships were developed for the specific heat and temperature ratio terms and the results were investigated for a cascade system in two reduced temperatures of 0.684 and 0.681, with the refrigerant R-134a and finally, the optimum temperature distribution was found. Bhattacharyya et al. [4] demonstrated optimization results for a C_3H_8/CO_2 cascade system. Lee et al. [2] determined the optimum temperature in a cascade condenser, maximized the coefficient of performance and minimized the exergy destruction of the system, using the evaporation temperature of carbon dioxide, condensation temperature of ammonia and temperature difference in the cascade condenser as decision variables. In another analysis, Bhattacharyya examined the internally reversible two stage cascade cycle in order to determine the optimum intermediate temperature regarding the exergy and

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Nomenclature

A_i	Inner heat transfer area (m^2)
$A_{c,o}$	Free flow area (m^2)
A_{fr}	Frontal surface area (m^2)
A_o	Outer heat transfer area (m^2)
c	Unit cost of exergy ($\$\,kW^{-1}$)
C	Cost ($)
C_{el}	Unit cost of input exergy ($\$\,kW^{-1}\,h^{-1}$)
C_{total}	Total annual cost of plant ($)
d_i	Inner diameter (m)
d_o	Outer diameter (m)
$\dot{E}x$	Rate of exergy (kW)
F_P	Constant coefficient
G	Mass velocity ($kg\,m^{-2}\,s^{-1}$)
H	Period of operation per year (hour)
$\dot{m}$	Mass flow rate ($kg\,s^{-1}$)
$\dot{Q}$	Heat transfer rate (kW)
R_p	Pressure ratio
R_w	Wall thermal resistance (kW^{-1})
s	Entropy ($kJ\,kg^{-1}\,K^{-1}$)
T	Temperature (°C or K)
U	Overall heat transfer coefficient ($Wm^{-2}\,K^{-1}$)
$\dot{W}$	Power (kW)

Subscripts

0	Ambient
1	Saturated vapor state of carbon dioxide
2	Superheated vapor state of carbon dioxide
3	Saturated liquid state of carbon dioxide
4	Saturated state of carbon dioxide
5	Saturated vapor state of ammonia
6	Superheated vapor state of ammonia
7	Saturated liquid state of ammonia
8	Saturated state of ammonia
C	Condensing
Cas	Cascade
cas.cond	Cascade condenser
Comp	Compressor
Cond	Condenser
CL	Cold refrigerated space
D	Destruction
E	Evaporating
Exp	Expansion valve
H	High temperature circuit
L	Low temperature circuit
MC	Condensing temperature of LTC
ME	Evaporating temperature of HTC
Tot	Total

Greek symbol

η_m	Mechanical efficiency
η_e	Electrical efficiency
η_S	Isentropic efficiency
ρ	Density ($kg\,m^{-3}$)
μ	Viscosity (Pa.s)
ε	Rational efficiency

maximum refrigeration load [5]. Mafi et al. [6] analyzed the cascade refrigeration system in olefin plants from the exergy and thermodynamic points of view. In their analysis, the two methods of pinch analysis and exergy analysis were used to improve the total exergetic efficiency of the system. Getu and Basnal [7] analyzed the CO_2/NH_3 cascade refrigeration system to optimize its operational and design parameters. Subcooling, superheating and evaporation temperatures in ammonia and carbon dioxide circuits were the parameters they considered in their analysis. In this work the multi-linear regression method was used to develop the mathematical relationships for the optimum coefficient of performance, optimum evaporation temperature of ammonia, and optimum mass ratio of ammonia to carbon dioxide in the cascade cycle.

In all the abovementioned papers the objective function is a thermodynamic one which results in maximizing the coefficient of performance or minimizing entropy generation or exergy destruction. Although this method results in a higher efficiency, it also may lead to an excessive increase of costs of the system. Thermoeconomic optimization is a method in which decision variables are found by trade-off between the capital cost and energy cost. This method has successfully been applied to single stage refrigeration cycles [8–11] and for the first time in this paper is applied to a cascade refrigeration cycle.

In this present work, first thermodynamic and exergy analyses are carried out and then, heat exchangers are designed. Later, decision variables are changed simultaneously using direct search method and the thermoeconomic objective function (annual cost of the system) which includes the cost of electricity consumption in compressors and fans and also the capital cost (purchase equipment cost) of the system, is minimized. Finally, optimum values for geometrical design and thermal parameters including saturation temperature of condenser and evaporator, saturation temperatures in cascade condenser, temperature difference in cascade condenser, powers of compressors and fans and surface areas of heat exchangers are obtained through minimizing the objective function.

2. Thermodynamic and exergetic analysis

A schematic diagram of CO_2/NH_3 refrigeration cycle is shown in Fig. 1. This refrigeration system includes two separate circuits, one high temperature circuit (HTC) with ammonia as the refrigerant and the other one, low temperature circuit (LTC) with carbon dioxide as the refrigerant. These two circuits are thermally coupled using a heat exchanger called cascade condenser which acts as an evaporator for the HTC and a condenser for the LTC. The condenser rejects the heat $\dot{Q}_H$ at the condensing temperature T_C to the ambient which is at the temperature T_0. The evaporator of the system absorbs a refrigerated load $\dot{Q}_L$ from the cold space which has the temperature T_{CL}. T_{ME} and T_{MC} are the evaporation and condensation temperatures of ammonia and carbon dioxide, respectively. $\Delta T_{cas} = T_{MC} - T_{ME}$ is the cascade condenser temperature difference. The T–s diagram of the cycle is presented in Fig. 2.

The governing equations including energy and exergy balance for components may be written as follows:

- For evaporator:

$$\dot{Q}_L = \dot{m}_L(h_1 - h_4) \tag{1}$$

$$\dot{E}x_{D,evap} = \left(1 - \frac{T_0}{T_{CL}}\right)\dot{Q}_L + \dot{m}_L(ex_4 - ex_1) + \dot{W}_{fan,evap} \tag{2}$$

- For LTC compressor:

$$\dot{W}_{LTC,comp} = \frac{\dot{m}_L(h_{2S} - h_1)}{\eta_S\eta_m\eta_e} = \frac{\dot{m}_L(h_2 - h_1)}{\eta_m\eta_e} \tag{3}$$