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Entropy generation analysis of fan-supplied gas cooler within the framework of two-stage CO_2 transcritical refrigeration cycle

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ABSTRACT

Optimization design of heat exchangers based on entropy generation analysis is critical to improve the efficiency of CO_2 refrigeration systems. However, most of the previous studies in this field were focused on individual components that are isolated from the entire system. Little has been done on the component analysis within the framework of entire refrigeration cycle. To probe this important issue, the entropy generation analysis is performed in this study on the fan-supplied gas cooler within a two-stage CO_2 refrigeration system. The optimum circuit length is found by investigating the influences of geometrical parameters such as tube spacing, fin density and tube diameter on the heat transfer rate and entropy generation number of the gas cooler. The designed fan-coil is then put into the frame of the two-stage refrigeration cycle and its energy performance and exergy performance are both evaluated on the system level. The results show that the analysis with isolated gas cooler can lead to overestimated or unrealistic predictions on the heat transfer performance compared to the analysis within the framework of entire cycle.

1. Introduction

Extensive research and development have been performed on the air conditioning and refrigeration system with carbon dioxide (CO_2) as the refrigerant due to its zero ozone depletion potential (ODP) and negligible global warming potential (GWP). However, due to its high pressure level, high power consumption, etc., the system with CO_2 shows poor thermodynamic characteristics compared to a traditional refrigeration cycle. Since the performance of most energy systems is dominated by heat exchanger design, the performance evaluation of the heat exchanger is indispensable to improve system efficiency and reduce cost.

Energy-based analysis (first law of thermodynamics) is the traditional method of assessing a thermal system. This approach is concerned on only with the conservation of energy, and provides no information on the degradation of energy or resources during a process. Entropy generation analysis, which is based on the second law of thermodynamics, provides a powerful tool for examining the influence of irreversibilities within a system on the required energy consumption. The analysis of heat exchangers based on this approach has been extensively studied, aiming to minimize the entropy generation [1–6]. With the maturity of modeling techniques in heat exchangers, the concept of entropy generation can be implemented more readily to guide the optimization design of heat exchangers.

However, most of the previous studies were focused on the isolated components, in which the refrigerant mass flow rate or heat transfer rate is fixed to mimic the system results [7]. In reality, the design change of a gas cooler affects the optimal discharge pressure, resulting in the variation of refrigerant mass flow rate and exit temperature of gas cooler. Although the variation of refrigerant mass flow rate compensates the change of discharge pressure, heat transfer rate of gas cooler is not kept constant with the optimization of gas cooler pressure due to the unique s-shape isotherms of CO_2. As a result, the isolated component analysis may not be able to generate meaningful guideline for the optimization design of heat exchangers. Therefore, an entropy generation model within the system frame is urgently needed since such model accounts for the feedbacks from the rest of the system.

In this work, an entropy generation approach is developed to guide the optimization design of heat exchangers within the framework of a two-stage CO_2 refrigeration system. The fan-supplied gas cooler is taken as the illustrative example. The design of the gas cooler is conducted by investigating interaction of geometrical parameters, such as tube spacing, fin density (number of fins per inch) and tube diameter, as well as operational parameters such as air flow rate supplied by a given fan. Both the energy-based and exergy-based approaches, which are based on the first law and second law of thermodynamics, respectively, are employed to determine the optimal combination of tube spacing, fin density, circuit

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Nomenclature

A_i, A_o	heat transfer area for tube inside and outside, m^2	η	compressor efficiency
C	heat capacity rate, W/°C		
C_{pa}	specific heat of air, J/(kg °C)	*Subscripts*	
COP	coefficient of performance	*a, pa*	index for air
dp	suction pressure drop, Pa	*c*	coil
e_x	exergy rate of fluid, W	*cal*	calculation
ED	exergy destruction, W	*cond*	gas cooler
h	enthalpy, kJ/kg	*disp*	displacement
k	thermal conductivity, W/(m °C)	*dis*	discharge
L_e	Lewis number	*evap*	evaporator
m	flow rate, kg/s	*ex*	exergy
NTU	number of transfer units	*exp*	experiment
Ns, EGN	entropy generation number	*i, o*	inlet, outlet
P_r	pressure ratio	*gc*	gas cooler
P	pressure, Pa	*is*	isentropic
Q	heat transfer rate, W	*r*	refrigerant
T	temperature, °C	*s*	space
S_{gen}	entropy generation rate, W	*sim*	simulation
s	entropy, J/(kg K)	*sub*	subcooling
U	overall heat transfer coefficient, W/(m^2 °C)	*tot*	total
W	power, W	*suc*	suction
		vol	volumetric
Greek symbols		*w*	wall
ε	heat exchanger effectiveness	ε	heat exchanger efficiency

length, etc. Particular attention is given to the fan-coil interaction within a system level instead of an isolated component level [8]. A detailed discussion of the demonstrative results is presented to identify the distinctions between the isolated component analysis and the analysis from the standpoint of entire system.

2. Descriptions of mathematical modeling

2.1. Two-stage CO_2 refrigeration cycle

To set up the boundary conditions for the gas cooler simulation, a cycle analysis of two-stage CO_2 refrigeration system is established, as shown in Fig. 1. The system consists of a gas cooler, an intercooler, an expansion valve, an evaporator and a two-stage compressor. The purpose of using intercooler is to reduce compressor power work and lower the discharge temperature of the compressor. In addition, a suction line heat exchanger (SLHX) is applied to increase the cooling capacity and COP by exchanging the heat between high pressure side fluid at the gas cooler exit and low pressure side gas after the evaporator. Both gas cooler and intercooler are cooled by air.

To emphasize the essential physics, the following assumptions are made on the refrigeration cycle in Fig. 1:

- Space temperature is maintained at 6 °C and ambient temperature is kept at 32.2 °C.
- Evaporating temperature and outlet superheat are maintained at −6.7 °C and 11.1 °C, respectively.
- Gas cooler, intercooler, evaporator and SLHX are assumed to be ideal and there is no pressure drop through the coils.
- The approach temperature at the gas cooler exit and intercooler exit are 3 °C and 5 °C, respectively.
- Effectiveness of liquid suction heat exchanger SLHX is 0.5.
- The speed of two-stage compressor is 3600 rpm. The total displacement V_{disp} for the compressor is 4.5 m^3/h. Three different configurations of the two-stage compressor are selected for analysis: $r = V_{disp1}/V_{disp2}$ = 1, 1.5 and 2. For each stage, the volumetric efficiency and isentropic efficiency are defined below [9]:

$$\eta_{vol} = 1.140 - 0.1224 \cdot P_r; \quad \dot{m}_{ref} = \rho_{suc}\eta_{vol}V_{disp} \tag{1}$$

$$\eta_{is} = 0.801 - 0.0477 \cdot P_r; \quad W_{comp} = \dot{m}_{ref}(h_{dis} - h_{suc}) = \dot{m}(h_{dis,is} - h_{suc})/\eta_{is} \tag{2}$$

To find the maximum COP, the compressor discharge pressure (i.e., gas cooling pressure) is optimized for the three different configurations of r = 1, 1.5 and 2, which are shown in Fig. 2. It is seen in Fig. 2 that, for the three configurations considered, the system COP reaches its maximum value when the compressor discharge pressure is about P_{dis} = 8750 kPa, which is consistent with the result in [10]. It is also observed in Fig. 2 that the configuration of r = 2 (V_{disp1} = 3 m^3/h and V_{disp2} = 1.5 m^3/h) has the best system performance. The calculated results with r = 2 are summarized in Table 1 as well as in the p–h diagram of Fig. 3. These results will be employed in the following simulation and analysis.

2.2. Energy and exergy analysis

From the point of view of the first law of thermodynamics, the system COP is defined as the net refrigeration effect produced per unit of work required:

$$COP_{tot} = \frac{\text{energy output}}{\text{energy input}} = \frac{Q_{evap} - Wfan, evap}{W_{tot}} \tag{3}$$

For a simple cycle analysis without consideration of fan power, the COP may be defined as

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{4}$$