ISSN 2004-2965 Energy Proceedings, Vol. 18, 2021

International Conference on Applied Energy 2021
Nov. 29 - Dec. 5, 2021, Thailand/Virtual
Paper ID: 166

Simulation Study of a Heat Pump Dual Temperature Display Cabinet Using Two Natural Refrigerants

Qianlong Wang[1], Yao Wang[1], Tianxin Jiang[2], Bingqing Liu[2], Chongsheng Xu[2], Suxin Qian[1*]

1 Department of Refrigeration and Cryogenic Engineering, Xi'an Jiaotong University, Xi'an, China, 710049

2 Shandong New Beiyang Information Technology Co. Ltd., Weihai, China

ABSTRACT

The dual-temperature display cabinet stores hot and cold products in the same unit and requires a heat source in addition to the refrigeration system. The heat is provided by an electric heater in a baseline system and can be replaced by leveraging condensing-heat. Two natural fluids were numerically investigated in this study, R290, and R744. Experimental data from a baseline dual-temperature display cabinet validated the numerical model in this study. Among the geometric and operating parameters of the two heat exchangers, it was found that the most important parameter is the air volumetric flow rate of the condenser. In the medium and high temperature environment, the R290 has a better energy-saving effect, while the R744 is more suitable for low temperature conditions. When the ambient temperature is 25°C, the power consumption of the heat pump system using the R290 and R744 has 49% and 43% energy saving benefits over the baseline, respectively.

Keywords: Open display cabinet; Condensing heat recovery; Steady state simulation

1. INTRODUCTION

Open refrigerated display cabinet (ORDC) is widely used in supermarkets and retail stores. Considering the conventional ORDCs only provide chilled products, we have developed the dual-temperature open display cabinet (DTODC) that stores both chilled and heated products [1].

To address the heating requirement in a DTOTC, the otherwise wasted condensing heat could be a viable solution. Considering the application of natural refrigerants, propane and carbon dioxide are selected as two potential refrigerants for DTODC. In fact, heat pump systems with carbon dioxide as the refrigerant have been widely used. Song et al. [2] show that the heat pump system for the electric bus can deliver satisfactory heating performance by using CO_2 (R744) as the refrigerant. Deniz et al. [3] examined the influence of phase change material (PCM) in a closed display cabinet cooled by a transcritical CO_2 refrigeration system. The goal of this paper is to demonstrate the energy-saving potential by recovering condensing heat with the above mentioned two natural refrigerants in a DTODC, which serves as a guideline for this kind of product.

2. DUAL-TEMPERATURE DISPLAY CABINET

Figure 1 is the schematic diagram for two kinds of DTOTC. The left one is the baseline system that uses a PTC electric heater, while the right one recovers the condensing heat. Both have two modes: (1) The cooling and heating mode, which preserves the 40-50°C products in the top two layers and 1-10°C in other layers simultaneously; (2) The full refrigeration model, which only preserves the 1-10°C products. The DTODC delivers the condensing heat by a heated air ducts, as shown in Fig. 1.

The cooling and heating load distributions of the heating mode (5°C ambient temperature) and the full refrigeration mode (25°C ambient temperature, 60%RH working condition) are 773 W and 272 W, respectively. The main heat loss comes from the air curtain (69%), followed by radiation heat transfer (14%), and heat conduction through each wall (13%). These results are consistent with the experimental data from Gaspar et al. [4] and Faramarzi et al. [5].

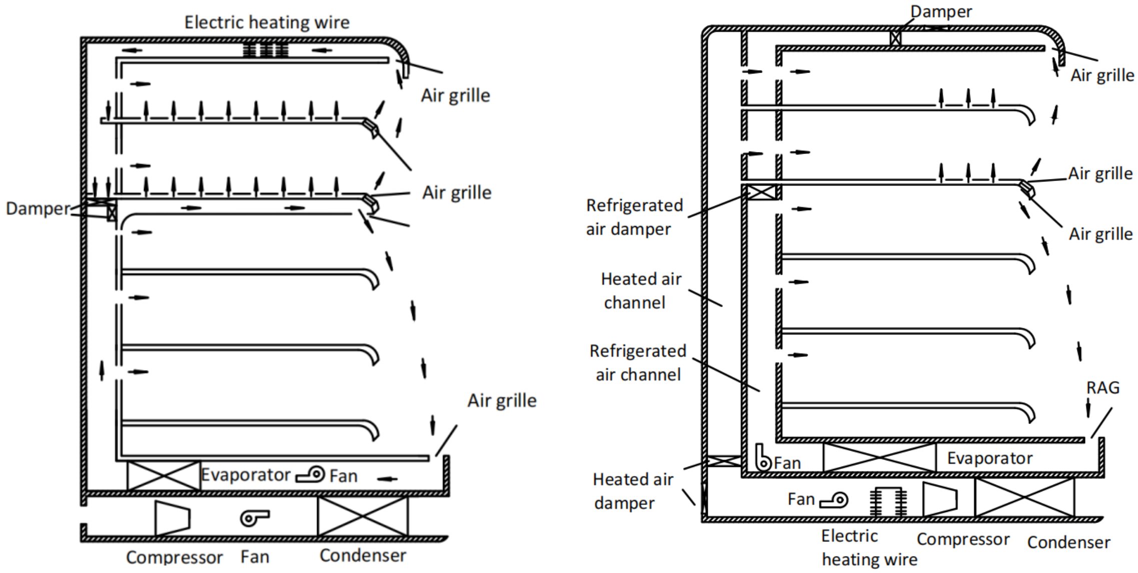

Fig. 1. Schematic diagram of traditional dual-temperature display cabinet with electric heater (baseline, left) and condensing heat recovery (right).

3. METHODS

3.1 Simulation model

VapCyc [6] is used to simulate the steady-state performance of the vapor compression refrigeration system. In the refrigeration system model, the evaporator and the condenser adopt CoilDesigner [7] models, which simulate the steady-state performance of the three-dimensional tube-fin heat exchangers.

3.2 Prototype

A baseline prototype dual temperature display cabinet was developed without recovering the condensing heat, in which the top two layers are the heating cabinets, and the bottom three layers are the refrigeration cabinets.

Using the data from the prototype, the condensing pressure, evaporating pressure, compressor power, and heating coefficient of performance (COP) are compared against the simulation results and are shown in Table 1.

Tab.1. Experimental verification results.

	Experiment	Simulation
Condensing pressure (MPa)	1.91(±0.02)	1.82
Evaporating pressure (MPa)	0.27(±0.02)	0.36
Compressor power (W)	751(±3)	783
COP (heating)	2.48(±0.03)	2.59

4. RESULTS AND DISCUSSIONS

4.1 Parametric optimization of R290 system

Taking R290 as refrigerant, the system can be optimized from the following aspects: the tube length, the fin density of the condenser, and the air volumetric flow rate of the condenser. The parameters of the baseline system and modifications are shown in Table 2. The ambient temperature is 25°C. In the heating area, the stored products need to be heated to and maintained at 42°C, and thus the air higher than 42°C is effective heating power, which is different from the condensing heat in the broad sense. It is defined in Eq.(1) as follows.

$$Q = \dot{m}c(T - 42) \tag{1}$$

where $\dot{m}$ is the condenser air mass flow rate, c is the specific heat of air, T is the outlet air temperature of the condenser, and Q is the effective heating power.

Tab.2. Parameters of different design modifications.

	Compressor volume [mL]	Tube length [m]	Fin density [cm^{-1}]	Air flow rate [m^3·s^{-1}]
Baseline	22.0	0.2	1.97	0.07
Mod1	22.4	0.2	1.97	0.07
Mod2	22.4	0.3	1.97	0.07
Mod3	22.4	0.3	3.94	0.07
Mod4	22.4	0.3	3.94	0.06

Taking the heating power as the index, the parametric optimization result of the system is shown in Figure 2. With the increase of the displacement of the compressor, both the refrigerant flow rate and the condensing temperature increase, which increases the heating power defined in Eq. (1). When the tube length and fin density increase, the heat exchange rate between the air and the refrigerant increases, and the air outlet temperature rises, leading to the increase of the heating power.

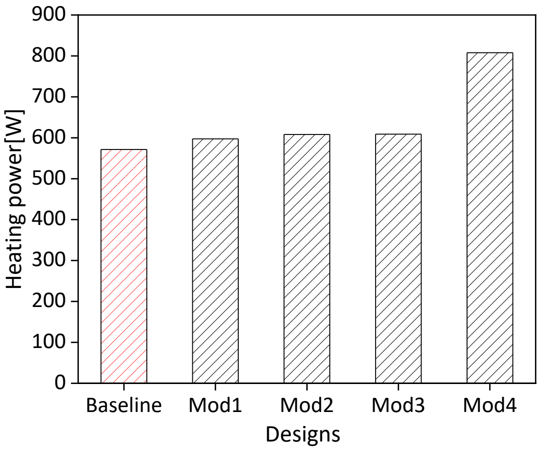

Fig.2. Effect of different designs on available heating power.

With the decrease of air flow rate, the outlet air temperature increases, and so does the heating power, according to Eq. (1). However, the thermal resistance on the air side increases, and thus the heat transfer becomes relatively insufficient, which leads to the rise of the condensing temperature and power consumption of the compressor. Therefore, it is important to choose a proper air flow rate that balances the benefit of increased heating power and the side effect of increased compressor power.

The optimization of the condensing air flow rate is discussed below in terms of four different ambient temperatures, and other geometric parameters are set in accordance with Modification 4. The power consumption of different air flow rate at different ambient temperatures is obtained, as shown in Figure 3.

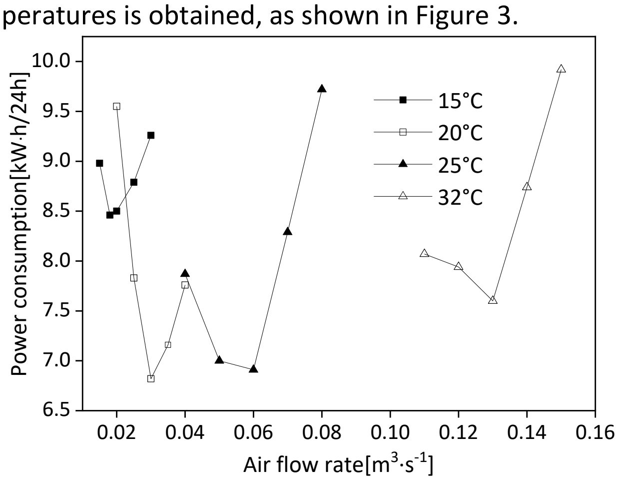

Fig.3. Power consumption of different air flow rates at different ambient temperatures in the R290 system.

The power consumption is determined as follows: the system first meets the demand of the cooling load, and if the required heat load cannot be fully provided by the condensing heat, a PTC heater will compensate for the difference. When the air volumetric flow rate decreases, heating power from the condenser increases, at the cost of increasing the condensing temperature and the compressor power. Consequently, the power by PTC heater decreases and the total power consumption decreases significantly. When the cooling air flow rate becomes too low, the compressor power consumption dominates, and the refrigeration required power consumption increases. As a result, the total power consumption rises. When the ambient temperature is 15°C, 20°C, 25°C, 32°C (matches the experimental value), respectively, the optimal air volumetric flow rate is 0.018 $m^3 \cdot s^{-1}$, 0.03 $m^3 \cdot s^{-1}$, 0.06 $m^3 \cdot s^{-1}$, 0.13 $m^3 \cdot s^{-1}$, and the corresponding total power consumption is 8.46 kW·h/24h, 6.82 kW·h/24h, 6.91 kW·h/24h, 7.60 kW·h/24h, respectively. Therefore, a variable speed fan is needed to reduce the air volumetric flow rate at low temperature and increase the air volumetric flow rate appropriately under high temperature conditions.

4.2 Parametric optimization of R744 system

When switching from R290 to R744, it is important to optimize the heat exchanger configuration to better accommodate the refrigerant temperature profile of R744 in the gas cooler. Though both heat exchangers use quasi-counter flow configuration to achieve higher air outlet temperature, it is found that R744 prefers more tube banks (12 columns) than R290 (5 columns), since 70% of R290 condenser exchanges heat isothermally and the margin effect by further increasing tube banks becomes decremental.

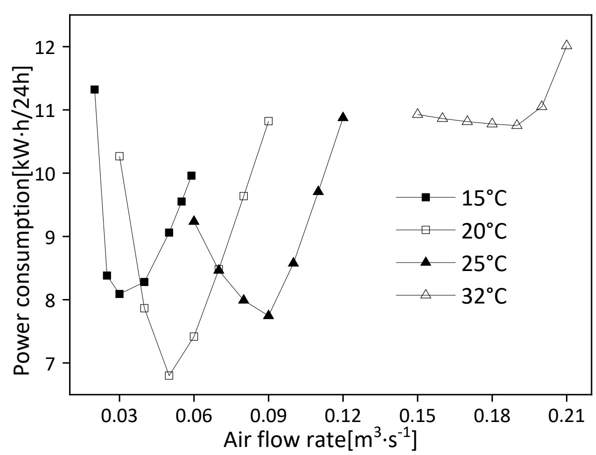

Fig.4. Power consumption of different air flow rates at different ambient temperatures.

As shown in Fig. 4, when the ambient temperature is 15°C and 20°C, the discharge pressure is set to 11 MPa and 10 MPa, respectively. When the ambient temperature is 25°C or 32°C, the discharge pressure is set to 9 MPa. Under these four ambient temperatures, the optimal air volumetric flow rate is 0.03$m^3 \cdot s^{-1}$, 0.05 $m^3 \cdot s^{-1}$, 0.09 $m^3 \cdot s^{-1}$, and 0.19 $m^3 \cdot s^{-1}$, respectively; the corresponding power consumption is 8.09 kW·h/24h, 6.80 kW·h/24h, 7.74 kW·h/24h, 10.75 kW·h/24h, respectively.

4.3 Comparison of R290 and R744 systems

Compared with R290, when the ambient temperature is below 20°C, R744 has better energy-saving potential, while R290 is superior when the ambient temperature exceeds 25°C. This observation is intuitive since low ambient temperature requires further lifting of the air temperature, which better matches R744 with larger temperature difference along with the gas cooler. At medium to high ambient temperatures, the power consumption of the R744 compressor outbalances the cooling power, and it becomes the main constraining factor for power consumption. To mitigate R744's disadvantage at high temperature but inherit its advantage at low temperature, zeotropic mixtures may be an option and should be investigated in future studies.

As shown in Figure 5, when the ambient temperature is 25°C, the optimal power consumptions of the system using R290 and R744 are compared. The power consumption of the baseline system is 13.59 kW·h/24h, in which the PTC heater provides necessary heat, and it accounts for 52.4% of the total power consumed. Recovering the condensing heat can reduce power consumption by 6.67 kW·h/24h (49%) and 5.83 kW·h/24h (43%) for R290 and R744, respectively.

The condensing pressure of the baseline system with PTC heater is 1.81 MPa, while that of the R290 heat pump system is slightly higher at 1.82 MPa. This is the tradeoff one needs to consider when recovering the condensing heat. If the condenser is not properly designed, the increment of condensing pressure and compressor work may outweigh the benefit of the recovered heat.

5. CONCLUSIONS

This paper demonstrates that recovering condensing heat is beneficial for a standalone dual-temperature display cabinet for two natural refrigerants with GWP less than 3. Compared with R290, using R744 as refrigerant is more conducive to energy saving when the ambient temperature is below 20°C, while R290 performs better at ambient temperature above 25°C. At 25°C, the power consumption of R290 system and R744 system is 49% and 43% lower than the baseline system, respectively. When R744 is used as refrigerant, the discharge pressure should be 10-11 MPa for better performance. In addition, the air volumetric flow rate of the condenser is the key parameter when recovering condensing heat. When the ambient temperature decreases, the reduction of air volumetric flow rate can significantly improve the effective heating power and reduce the power consumption, and thus variable speed fan should be implemented.

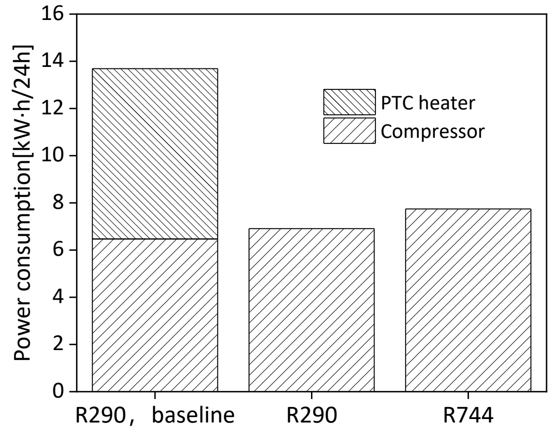

Fig.5. Optimal power consumptions and their breakdown chart at 25°C

ACKNOWLEDGEMENT

This study is supported by the Young Elite Scientists Sponsorship Program by CAST under the grant No. 2019QNRC001.

REFERENCE

[1] Wang Y, Qian SX, Xu JB, Li LY, Dou XN, Li XY, Yan G, Yu JL. Numerical study on the air curtain characteristics of a dual-temperature open display cabinet. Int J Refrig 2021;126:23–24.

[2] Song X, Lu DX, Qiang L, Yu C, Wang DD, Shi JY, Chen JP. Experimental study on heating performance of a CO2 heat pump system for an electric bus. Appl Therm Eng 2021;190:116789.

[3] Deniz Y, Ebru M, Bariş Y. Experimental Investigation of PCM Location in a Commercial Display Cabinet Cooled by a Transcritical CO2 System. Int J Refrig 2020;120:396–405.

[4] Gaspar PD, Carrilho GLC, Pitarma RA. Experimental analysis of the thermal entrainment factor of air curtains in vertical open display cabinets for different ambient air conditions. Appl Therm Eng 2011;31(5):961–969.

[5] Faramarzi RT, Woodworth ML. Effects of low-emissivity shields on the performance and poweruse of refrigerated display cabinet. ASHRAE Trans 1999;105(1): 533–540.

[6] Aute V, 2015 Vapcyc [EB/OL]. (2021-9-12) https://optimizedthermalsystems.com/index.php/products/vapcyc.

[7] Jiang H, Aute V, Radermacher R. CoilDesigner: a general-purpose simulation and design tool for air-to-refrigerant heat exchangers. Int J Refrig 2005;29(4):601–610.

