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Characterization of Energy Efficient Vapor Compression Cycle Prototype with a Linear Compressor

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Abstract

In this paper, an experimental vapour compression cycle (VCC) prototype is developed. The VCC system composes of a high performance 300W linear compressor with variable capacity control, expansion valve, air-cooled condenser, and two microchannel plate heat exchangers acting as evaporator and recuperator. Wide range of experimental characterization is performed to investigate the influence of changing the compressor capacity, evaporating temperature and expansion valve opening position on the VCC performance. The prototype unit is able to achieve a high coefficient of performance (COP) of 4.5. Component and refrigeration cycle models are developed and validated with the experimental data. The proposed cycle model provides the insight to guide energy-efficient compact cooling system design and operation.

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Keywords: Vapor compression cycly, Recuperator, Linear Compressor, Characterization

1. Introduction

Vapor compression cycles (VCC) are widely used in refrigeration and air conditioning systems, which are considered as one of the most energy-consuming domestic appliances. Refrigerators and air conditioning systems represent 13.7% and 16% respectively of all residential electricity consumption in USA for 2001 [1]. Also, air conditioning systems consume up to 70% of the UAE total energy consumption [2]. Hence, better designs of cooling systems are required to minimize the energy consumption. The trend in electronic industry is towards more compact systems. These systems required more effective cooling as they produce a lot of heat. Therefore, compact vapor compression cycles are sought for portable high-end electronic systems. A study has been done by Trutassanawin et al. [3] to develop a compact refrigeration system prototype to demonstrate its feasibility in electronic cooling

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applications. The system cooling capacity varied from 121 to 268 W, with a COP of 2.8 to 4.7, and with a second-law efficiency ranging from 33% and 52%. A test-bed has been developed by Chang et al. [4] to examine the thermal performance of a miniature VCC for electronic cooling purposes. The effect of the compressor speed and the expansion valve opening on the COP of the cycle has been studied. The study concluded that the condensation phenomena were improved when the compressor speed was reduced from 4200 to 3840 RPM.

A detailed study of a VCC test-bed has been carried out by Arora and Kaushik [5] to analyze the cycle COP and exergetic efficiency for different refrigerant, where experiment were performed with evaporator and condenser temperature range from -50 ^{O}C to 0 ^{O}C and 40 ^{O}C to 55 ^{O}C respectively. This study concluded that when the evaporator temperature increases, the pressure ratio across the compressor decreases which lead to a lower compression work and then a higher COP. In order to enhance the COP of the refrigeration systems, the flow within the evaporator should remain in two phase region. Heat transfer coefficient deteriorated in superheated zone. Therefore, two phase flow is preferable to achieve higher heat transfer coefficient [6]. An analysis has been conducted by Torrella et al. [7] described a general methodology for analyzing six possible configuration of VCC. They found that configuration with higher subcooling degree has higher COP, due to the fact that the degree of subcooling increases the difference in specific enthalpy across the evaporator. The study by Jensen and Skogestad [8] focused on the optimal operation for a simple VCC. It discussed the effect of having sub-cooling and super-heating degrees on the compressor power consumption. The study found that super-heating degree should be minimized whereas some sub-cooling is optimal. The work in Aprea et al. [9] discussed the advantages of using suction-liquid heat exchanger on the performance of refrigeration systems from a thermodynamic point of view. It showed the influence of using the recuperator on the COP enhancement by adding some sub-cooled and super-heated degrees on the refrigerant.

In the above studies, there are limited discussions on developing recuperator-based VCCs with linear compressors. Linear compressor has the inherent advantage for refrigeration applications since it can be easily controlled to provide variable refrigeration cooling capacities upon demand. This work highlights the effect of installing a fluid-to-fluid recuperator on the system COP. Wide range of parametric studies are performed to analyze the influence of changing component operating conditions on the system performance. Additionally, components and cycle level characterizations are developed to predict the cycle performance at different operation conditions. This study contributes to the development of energy-efficient compact vapor compression cycle. It also shows the potential of the compact VCC systems for portable cooling applications such as electronics cooling.

2. Experimental Test-bed

The basic VCC consists of compressor, condenser, evaporator and expansion device. For better heat transfer in the evaporator an accumulator is usually added after the evaporator so the evaporator exit is maintained in the two-phase flow region [6]. In addition, an internal heat exchanger can be added to utilize the temperature difference in the loop by exchanging the heat between the condenser and evaporator exits.

Figure 1 shows the schematic diagram and photo of the VCC prototype we have developed. All the sensors are connected to data acquisition systems. The accuracy of the temperature, pressure, and mass flow rate are ±0.7 ^{O}C, ±0.15% of full scale, and ±0.08% of measurement respectively. The data are recorded into a computer station using LabVIEW software. More details may be found in [10]. All components are assembled on an aluminum frame and connected using copper pipe and brass fittings as shown in Figure 1(b). Once the loop is assembled, it is tested for leakage, insulated and charged with R134a refrigerant.

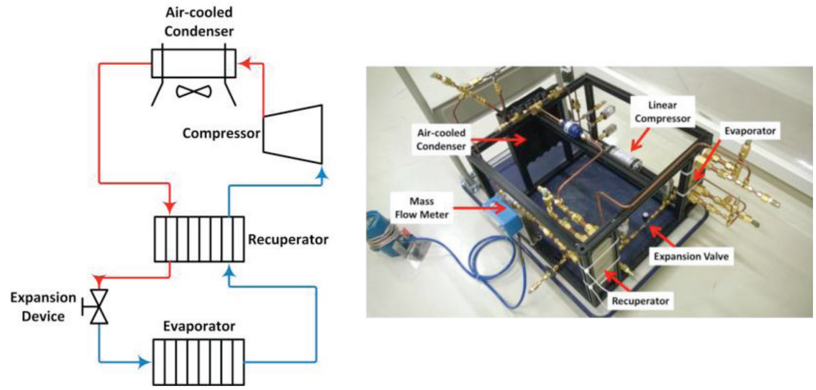

Fig. 1. Schematic diagram of VCC test-bed setup (left) and prototype photo before insulation (right)

3. Experimental Characterization

The controllable parameters in the VCC are the compressor capacity, heat source temperature, and expansion valve opening position. Changing these parameters result in different compressor and evaporator energy input, recuperator heat exchange rate and condenser heat rejection rate. The compressor capacity has been changed from 50-100% with 10% step and the expansion valve opening position has been changed from open number 4 into open number 6.

Figure 2-a shows the effect of changing the compressor capacity on the coefficient of performance (COP) of the cycle at different expansion valve opening positions. With increasing the compressor capacity, the length of the piston stroke in the compressor increases which leads to a higher power consumption as shown in Figure 2-d. The increase of the piston stroke length means more refrigerant mass flow rate which increases the evaporator cooling capacity as shown in Figure 2-b. However, the increase in the power consumption is higher than the increase in the evaporator cooling capacity. Therefore, the cycle COP decreases as compressor capacity increases. Figure 2-c shows the effect of changing the compressor capacity at the pressure ratio across the compressor. With increasing the compressor capacity, the compressor outlet pressure increases [11] which lead to a higher pressure ratio across the compressor at fix valve opening position.

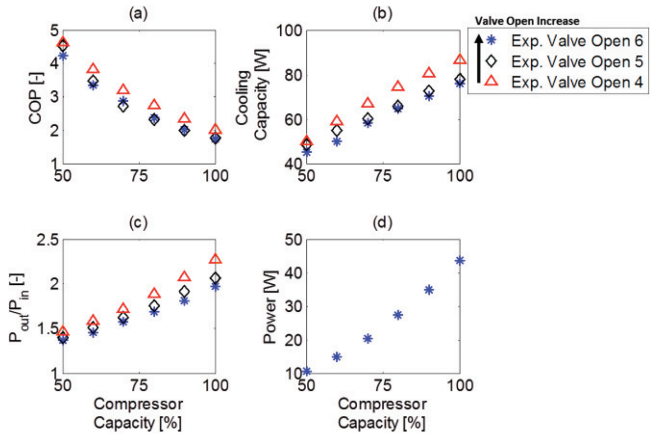

Fig. 2. Effect of changing compressor capacity on (a) coefficient of performance (b) cooling capacity (c) pressure ratio across the compressor and (d) compressor power consumption

4. VCC Modeling

4.1. Plate Heat Exchanger (Evaporator and Recuperator) and Fin-Tube Air-Cooled Heat Exchanger (Condenser)

In the evaporator, the refrigerant gains heat from heat source fluid. The refrigerant enters and exits the evaporator as a two-phase fluid. Therefore, the evaporator has only one flow zone as shown in Figure 3-a. In order to model the heat transfer in the evaporator, heat balance between the refrigerant, plate's wall, and oil should be solved to predict the evaporator outlet condition as below:

$$\left[\alpha_{evap}(T_{sat}-T_w)p_w L_p\right]_{evap} = \left[\alpha_{oil}(\bar{T}_{oil}-T_w)p_w L_p\right]_{oil} + \left[(\alpha_{nc}\{A_s+\pi D_h\})(T_{amb}-T_w)\right]_{amb} \tag{1}$$

The recuperator transfers the heat from refrigerant at the condenser outlet into refrigerant at the evaporator outlet due to the temperature difference between these two points. In the condenser side, refrigerant enters the recuperator as a two-phase fluid and exits as a sub-cooled liquid. On the other side, the refrigerant exits the evaporator and enters recuperator as a two-phase fluid and exits as super-heated gas. Since, the scenario of the interaction between these two fluids are unknown, an assumption of dividing the recuperator into three flow zones has been made as shown in Figure 3-b. In order to model the heat transfer in the recuperator, heat balance heat balance equations between condenser-refrigerant, heat exchanger wall, and evaporator-refrigerant should be solved for each zone:

$$\begin{aligned}
\left[\alpha_{evap_{TP}} L1(T_w-T_{sat})\right]_{evap} &= \left[\alpha_{cond_f} L1\left(\bar{T}_{cond_f}-T_w\right)\right]_{cond} \\
\left[\alpha_{evap_{TP}} L2(T_w-T_{sat})\right]_{evap} &= \left[\alpha_{cond_{TP}} L2(T_{sat}-T_w)\right]_{cond} \\
\left[\alpha_{evap_g} L3\left(T_w-\bar{T}_{evap_g}\right)\right]_{evap} &= \left[\alpha_{cond_{TP}} L3(T_{sat}-T_w)\right]_{cond}
\end{aligned} \tag{2}$$

In order to predict the outlet condition of the condenser, heat balances between refrigerant, pipe wall, and air flow across the condenser should be solved for each zone. The flow inside the condenser is divided into two main regions as shown in Figure 3-c.

$$\begin{aligned}
\left[\alpha_{cond_{TP}} L1\left(T_{cond_{TP}}-T_w\right)p_w L_{TP}\right]_{cond} &= \left[\alpha_{air} L1(T_w-T_{amb})A_{s(P+F)}\right]_{amb} \\
\left[\alpha_{cond_g} L2\left(\bar{T}_{cond_g}-T_w\right)p_w L_g\right]_{cond} &= \left[\alpha_{air} L2(T_w-T_{amb})A_{s(P+F)}\right]_{amb}
\end{aligned} \tag{3}$$

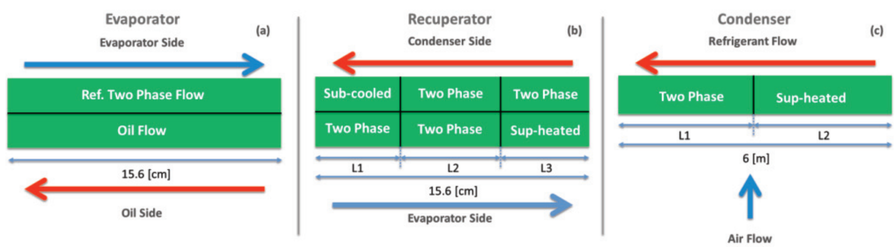

Fig. 3. (a) Evaporator has single flow zone (b) Recuperator has three flow zones (c) Condenser has two flow zones

4.2. Linear Compressor and Expansion Valve

The compressor used in this work is a piston-cylinder compressor type that uses a linear motor to drive the piston. The governing equation to model the compressor is, $\dot{m} = \rho V \omega \eta_v$, where ρ is the density, V is the compression volume, ω is the frequency, and η_v is the volumetric efficiency and it is a function of the pressure ratio across the compressor.

Mass flow rate across the expansion valve is governed as, $\dot{m} = C_v\sqrt{\rho_{in}(P_{in} - P_{out})}$, where C_v is the flow coefficient of the valve and it is a function of the cross sectional area, the fluid specifications and pressure ratio.

5. Model Validation

All component models have been validated with experimental data to predict inlet and outlet conditions of each component. Figure 4-a shows the effect of changing the compressor capacity on recuperator different zones lengths. With increasing compressor capacity heat transfer in the recuperator also increases due to higher temperature difference across recuperator plates. Therefore, refrigerants will gain more super-heated and sub-cooled degrees at higher compressor capacity which affects recuperator two-phase zone. Figure 4-b shows the P-h diagram for the VCC test-bed at different compressor capacities. The graph shows a very good agreement between the experimental data and the component model predictions.

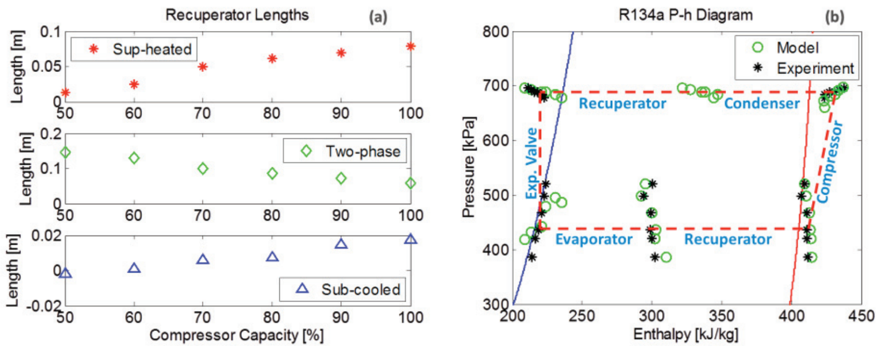

Fig. 4. (a) Effect of changing compressor capacity on the recuperator different zones length (b) VCC component model validation

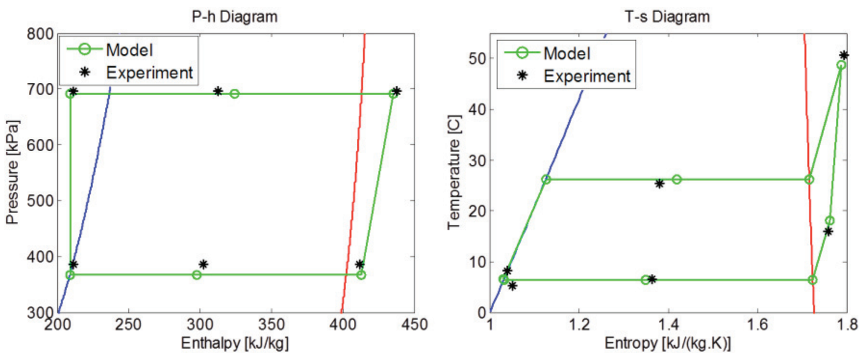

Fig. 5. Pressure-enthalpy and temperature-entropy diagrams of recuperator-integrated refrigeration cycle at the 100% compressor capacity

The cycle model should be self-contained and the model predictions just depend on external boundary conditions: Air velocity and ambient temperature at the condenser side and oil mass flow rate and inlet temperature at the evaporator side. Component models developed in the previous sections are combined and integrated together into the cycle model. Figure 5 shows the cycle model validation at evaporating temperature 25 ^{o}C and compressor capacity 100%. Additionally, the cycle model was able to predict the refrigerant mass flow rate with a maximum prediction error of 9.27%. All graphs and results show very good agreements between cycle model and experimental data.

6. Conclusion and Future Work

A compact VCC prototype was designed and developed. Extensive parametric studies were carried out to better understand system characteristics under changing operating conditions. Moving boundary models were developed for recuperator and condenser to find out the effect of changing the operating condition on these heat exchangers performance. The recuperator model was able to predict the amount of super-heat and sub-cooled degrees added to the refrigerant at different operating points. The model is coupled with other component models and integrated into a cycle-level model. The proposed components and cycle models were validated with experimental results. Components and cycle model predictions agree well with experimental data.

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Biography
Dr. TieJun Zhang has been an assistant professor of Mechanical and Materials Engineering at the Masdar Institute of Science and Technology in UAE since 2011. He was a visiting faculty member at the Massachusetts Institute of Technology. He was also a postdoctoral research associate at the Rensselaer Polytechnic Institute, USA.