

Experimental Analysis of the Performances of Unit Refrigeration Systems Based on Parallel Compressors with Consideration of the Volumetric and Isentropic Efficiency

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Abstract: The performances of a refrigeration unit relying on compressors working in parallel have been investigated considering the influence of the compressor volumetric efficiency and isentropic efficiency on the compression ratio. Moreover, the following influential factors have been taken into account: evaporation temperature, condensation temperature and compressor suction-exhaust pressure ratio for different opening conditions of the compressor. The following quantities have been selected as the unit performance measurement indicators: refrigeration capacity, energy efficiency ratio (COP), compressor power consumption, and refrigerant flow rate. The experimental results indicate that the system refrigeration capacity and COP decrease with a decrease in evaporation temperature, increase of condensation temperature, and increase in pressure ratio. The refrigerant flow rate increases with the increase in evaporation temperature, decrease in condensing temperature and increase in pressure ratio. The compressor power consumption increases with the increase in condensing temperature and increase in pressure ratio, but is not significantly affected by the evaporation temperature.

Keywords: Parallel compressor unit; evaporation temperature; condensation temperature; pressure ratio; refrigeration capacity; energy efficiency ratio (COP)

1 Introduction

Considering the increase in load requirements of air-conditioning and refrigeration systems in application conditions and limitations in the serviceable range of system equipment, equipment combinations such as cascade refrigeration systems, multi-stage compression refrigeration systems, and parallel compressor refrigeration systems have been widely employed. Therefore, analyzing the impact of each application condition on the system performance [1–5] is vital for selection the optimal system for system equipment combination.

Several scholars from China and other countries have conducted extensive numerical simulations and experimental studies on the compressor performance and refrigerant selection [3–12]. Niu et al. [3] proposed a liquid bypass refrigeration system which regulates the refrigeration capacity by adjusting the



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flow of refrigerant entering the evaporator. The system effectively reduces the compressor discharge temperature without the risk of liquid hammer. Ning et al. [4] compared the effects of regenerators on the performance of three cascade refrigeration systems, R290/CO₂, NH₃/CO₂, and R404A/CO₂ in several proportions. The group concluded that at a certain condensation temperature, evaporation temperature, and heat transfer temperature difference of condensation evaporator, the regenerative cycle can effectively improve the performance of the R290/CO₂ and R404A/CO₂ cascade refrigeration systems, and reduce the charge of high temperature circulating refrigerant in the cascade refrigeration systems; thereby, improving the safety of the cascade refrigeration systems. Yang et al. [5] compared different cascade refrigeration cycles and found that R290/CO₂ cascade refrigeration cycle has higher energy efficiency ratio (COP) than R404A/CO₂ cycle and there is an optimal low temperature cycle condensation temperature in both refrigeration cycles; hence the maximum system COP value is obtained. It is advisable to enhance the stability and efficiency of systems via increase in evaporation temperature, reduction in condensation temperature, and reduction in the heat transfer temperature difference of the condensation evaporator. Wang et al. [6] established a thermodynamic model of CO₂ transcritical refrigeration cycle for two-stage compression belt expander. Based on an irreversible loss in the actual cycle, the performance coefficient, thermodynamic perfection degree, and unit refrigerating capacity exergy loss of the system were analyzed and calculated.

Dokandari et al. [7] derived three novel correlations for ejector-expansion cascade cycle. The maximum COP increased by approximately 7 percent in the novel configuration. The exergy destruction rate decreased with ejector utilization and the second law efficiency increased by roughly 5 percent in the novel layout.

Purohit et al. [8] compared the performance of five CO₂ booster refrigeration systems for supermarket application. The investigated configurations included a standard booster system, and the maximum annual energy savings were found to be 22.16% for BC5 in New Delhi. Economic analysis revealed a recovery time of less than four years for the additional investment made in BC5. The slope of recovery time was found to be steeper at lower tariff.

Carlos et al. [9] presented the experimental evaluation of a R134a/CO₂ cascade refrigeration plant designed for low evaporation temperature in commercial refrigeration applications. The test bench incorporated two single-stage vapor compression cycles driven by semi hermetic compressors that were thermally coupled through two brazed plate cascade heat exchangers working in parallel and were controlled by the electronic expansion valves. The experimental evaluation (45 steady-states) covered evaporation temperatures from -40°C to -30°C and condensation temperature from 30°C to 50°C. In each steady-state, a sweep of the condensation temperature was performed for the low temperature cycle with speed variation in high temperature compressor.

Torrella et al. [11] described a general methodology suitable for analyzing any intermediate configuration considered in staged vapor compression refrigeration cycles. This general methodology only depends on two basic parameters related to sub-cooling and de-superheating obtained in the inter-stage system. A COP expression based on the two basic parameters was obtained from the general configuration. Based on the particularization of methodology to seven common configurations, an energy comparison of various configurations was performed for two fluids appropriate for low temperature domain such as ammonia and R-404A. Using the average values of system volume and operating pressure, Heysari et al. [12] calculated the settle-out pressure.

Arora et al. [13] established a theoretical model of two-stage compression refrigeration system based on three laws of energy conservation, momentum conservation, and mass conservation. The effects of evaporation temperature, condensation temperature, suction gas superheat, pre-valve sub-cooling, compressor entropic efficiency, etc., on the optimal intermediate saturation temperature were calculated and analyzed. Du et al. [14] took the mid-temperature gravity heat pipe exchanger as the research object,

simulated the fluid flow field, temperature field and the working state of heat pipe in the heat exchanger by Fluent software. According to the principle of superposition, a calculation model of unfrozen and frozen soils were established. Informed by a laboratory experiment, the latent heat of the adjacent zone was calculated for the freezing stage based on different water contents in the temperature section. Both the latent and specific heat of water, ice, and particles were calculated via superposition of the weight percentage content. A calculation model of the specific heat of the freezing stage was built, which provided both guidance and theoretical basis for the calculation of the specific heat of frozen soil [15]. Zhuang et al. [16] studied the variation of optimal intermediate temperature, optimal high/low pressure gas transmission volume ratio, and maximum COP value of ammonia piston type two-stage compression refrigeration system with the condensation temperature and evaporation temperature. Microcomputer was adopted to determine the relationship between high/low pressure gas transmission volume ratio and intermediate temperature at optimal intermediate state and any intermediate state under any operating conditions. Further comparisons of system comprehensive performance of two-stage machines with fixed volume ratios of 1/2 and 1/3 were conducted under various operating conditions.

In summary, most of the research works analyzed the performance parameters of a single compressor. To enrich the study on parallel compressor systems, herein, based on the water-cooled test bench of parallel compressor unit, the effect of operating conditions such as evaporation temperature, condensation temperature, compressor suction-exhaust pressure ratio, etc. on the system performance was analyzed under different opening conditions of the compressor. These results can provide an experimental basis and theoretical support for the development of parallel compressor unit systems.

2 Description of Experimental System

The test bench had a structure similar to a small water-cooled parallel compressor unit water-cooled chiller and primarily consisted of four parts: refrigeration system, cooling water system, chilled water system, and data acquisition system. The specific principle structure is shown in Fig. 1.

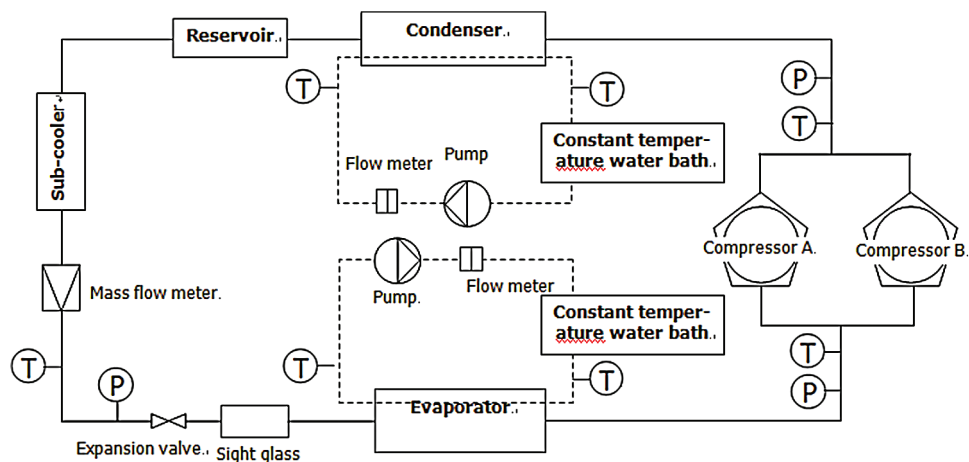


Figure 1: Schematic of the experimental system

The refrigeration system mainly included parallel compressor unit, condenser, reservoir, sub-cooler, mass flow meter, expansion valve, sight glass, evaporator, etc. The compressor was Danfoss scroll compressor. As the compressor unit is composed of invariable frequency compressor, the system operation was mainly adjusted by adjusting the power-on state of the compressor; thereby, achieving the adjustment of the system load.

The cooling water system and the chilled water system were primarily composed of components such as constant temperature water bath, water pump, and electromagnetic flow meter, which provided experimental heat exchange environment for the condenser and the evaporator. The condensation temperature and evaporation temperature were controlled by adjusting the temperature of the constant temperature water bath.

The refrigerant state parameters (including flow rate, temperature, and pressure) and water cycle state parameters (including flow rate and temperature) were measured for the system. The system parameters were collected by using Siemens 300PLC, and the experimental operation state was monitored in real time using a three-dimensional force control program to ensure safe and stable operation of the experiment. The instrument and apparatus performance parameters of the system are shown in [Tabs. 1 and 2](#).

Table 1: The instrument and apparatus performance parameters of the refrigeration system

Parameter	Instrument model	Range	Accuracy
Evaporation temperature	Matsuno PG1300	0~5 MPa	Level 0.1
Discharge pressure	Matsuno PG1300	0~5 MPa	Level 0.1
Compressor suction temperature	JUMO PT-1000	-50°C~300°C	±0.15°C
Compressor discharge temperature	JUMO PT-1000	-50°C~300°C	±0.15°C
The front of expansion valve temperature	JUMO PT-1000	-50°C~300°C	±0.15°C
Mass Flow-meter	DS-CMFI Coriolis Mass Flow-meter	0~700 kg/h	Level 0.2

Table 2: Instrument and apparatus performance parameters of water cycle

Parameter	Instrument model	Range	Accuracy
Chilled water-flow	Electromagnetic flow-meter	0.87~43.42 m ³ /h	±0.15%
Cooling water-flow	Electromagnetic flow-meter	0.87~43.42 m ³ /h	±0.15%
Chilled water inlet and outlet temperature	JUMO PT-1000	-50°C~300°C	±0.15°C
Cooling water inlet and outlet temperature	JUMO PT-1000	-50°C~300°C	±0.15°C

3 Processing of Experimental Data

The system parameters available according to the instrument and apparatus included: evaporation pressure P_{evp} , condensation pressure P_{con} , compressor suction temperature T_{suc} , compressor discharge temperature T_{dis} , the front of expansion valve refrigerant temperature T_{inEXV} , refrigerant mass flow rate m_r , chilled water inlet and outlet temperature $T_{evw,win}$ and $T_{evw,out}$, respectively, chilled water circulation flow m_{evw} , cooling water inlet and outlet temperature $T_{cow,in}$ and $T_{cow,out}$, respectively, and chilled water circulation flow m_{cow} . Primarily, the effects of condensation temperature, evaporation temperature, and system pressure ratio on the system performance were explored. Pre-expansion sub-cooling degree was set to 8°C and the compressor suction gas superheat was set to 5°C. The parameters can be calculated according to [Eqs. \(1\) and \(2\)](#):

$$T_{sc} = T_{suc} - T_{evp} \quad (1)$$

where: T_{evp} is the system evaporation temperature that is calculated from the measured evaporation pressure P_{evp} , °C

$$T_{sh} = T_{dis} - T_{con} \quad (2)$$

where: T_{con} is the condensing temperature that is calculated from the measured condensation pressure P_{con} , °C.

The average value of the heat exchange amount on the chilled water side and the heat exchange amount on the refrigerant side were used as the calculation standard to obtain system refrigeration capacity, i.e.,

Cooling water side heat exchange amount:

$$Q_w = m_{evw} \cdot (T_{evw,out} - T_{evw,in}) \quad (3)$$

where: C_p is the constant pressure specific heat of cooling water, J/(kg·°C).

Refrigerant side heat exchange amount:

$$Q_r = m_r \cdot (h_{suc} - h_{inEXV}) \quad (4)$$

where: h_{suc} is compressor suction specific enthalpy, kJ/kg; h_{inEXV} is refrigerant specific enthalpy of expansion valve; kJ/kg. Both were calculated from the measured temperature, pressure value.

System refrigeration capacity:

$$Q_{evp} = (Q_w + Q_r)/2 \quad (5)$$

COP was used to evaluate the system performance. In the calculations, only the compressor power consumption was calculated.

Compressor power consumption:

$$P = U \cdot I \quad (6)$$

where: U is the compressor input voltage, V; I is the compressor input current, A.

System performance COP:

$$COP = Q_{evp}/P \quad (7)$$

4 Analysis of Experimental Results

The evaporation temperature and condensation temperature were mainly controlled by adjusting the temperature of chilled water and cooling water. When analyzing the effect of evaporation temperature on system performance, the condensation temperature, pre-expansion valve sub-cooling, and compressor suction gas superheat were kept constant. The condensation temperature was set to 40°C and the evaporation temperature was designed at intervals of 2°C, namely, -4°C, -2°C, 0°C, 2°C, 4°C, and 6°C. When analyzing the effect of condensation temperature on the system performance, the evaporation temperature, pre-expansion valve sub-cooling, and the compressor suction gas superheat were maintained constant. The evaporation temperature was set to 4°C and condensation temperature was designed at intervals of 2°C, namely 36°C, 38°C, 40°C, 42°C, 44°C, and 46°C. The expansion valve opening during operation was automatically adjusted by using the PID meter according to the set pre-expansion valve sub-cooling and suction gas superheat.

4.1 Effect of Compressor On-State on Refrigeration Performance

Driven by the compressor, the refrigerant in the system achieves a transition from low-pressure, low-temperature state to high-temperature, high-pressure state. The expansion valve opening could be automatically adjusted according to the system pressure ratio change; thereby, ensuring the constant compressor suction gas superheat and pre-expansion valve sub-cooling.

Under same operating conditions, for the cases when the compressor A ran alone, the compressor B ran alone, and when the compressors A and B ran simultaneously, the refrigerant circulation flow rate and the system refrigerating capacity, respectively, are shown in Figs. 2 and 3. When the compressors A and B ran simultaneously, the system refrigerating capacity was about 2.09~2.15 times and 1.62~1.68 times the capacity as compared to when compressor A ran alone and compressor B ran alone, respectively. While the refrigerant circulation flow when compressors A and B ran together was about 2.15~2.23 times and 1.71~1.75 times, respectively, the capacity when compressor A ran alone and compressor B ran alone. Under same operating conditions, the refrigerant per unit mass has equal refrigeration capacity. Therefore, different refrigerant circulation flows in the system under different compressor opening states result in different load states of the unit.

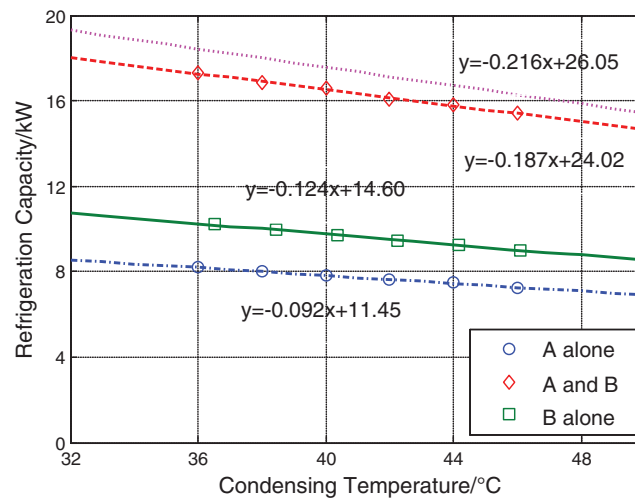


Figure 2: Comparison of system refrigeration capacity under different states of compressor

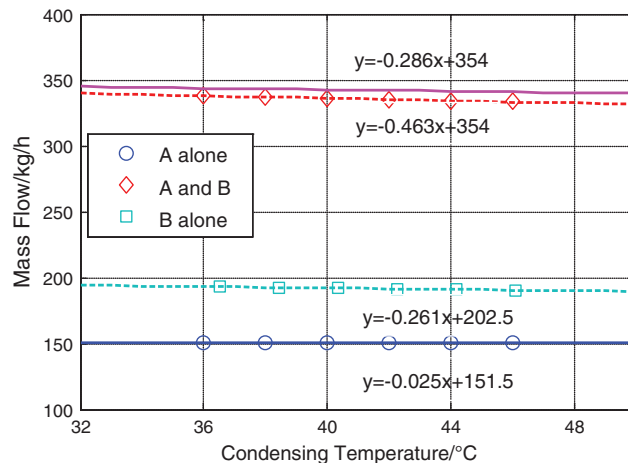


Figure 3: Comparison of refrigerant circulation flow under different compressor states

4.2 Effect of Evaporation Temperature on Refrigeration Performance

The control of the system evaporation temperature was achieved by adjusting the temperature of the chilled water. The variation of the system refrigeration capacity and COP with the evaporation temperature when the compressor A ran alone is presented in Fig. 4. Both the system refrigeration

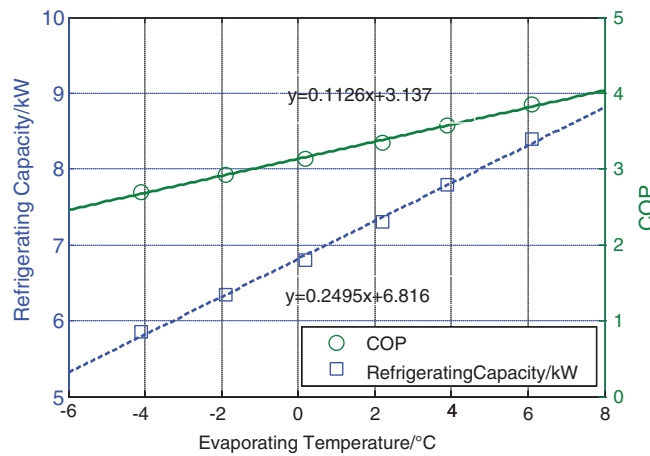


Figure 4: Effect of evaporation temperature on refrigeration capacity and COP

capacity and COP gradually decrease with the decrease in evaporation temperature. At this time, condensation temperature and pre-expansion valve sub-cooling were maintained constant, suggesting that the refrigerant enthalpy at the evaporator inlet remains unchanged, while the decreased evaporation temperature and unchanged compressor suction gas superheat mark the reduction in refrigeration capacity per unit mass of the refrigerant.

Although a decreased evaporation temperature causes the refrigerant to decrease in specific volume, i.e., the compressor can act on more refrigerant in a single stroke, and the volumetric efficiency of the compressor decreases as the system pressure ratio is increased [16]. For a constant condensation temperature, the system pressure ratio decreases as the evaporation temperature increases. Therefore, the refrigerant circulation flow rate increases as the evaporation temperature increases. Although the refrigerant flow rate increases as the evaporation temperature increases, the unit mass compressor work of the compressor on the refrigerant decreases as the evaporation temperature increases.

In addition, the isentropic efficiency of the compressor decreases with the increase in system pressure ratio [17,18]. The three effects complemented each other; hence, the compressor power consumption is not affected by the evaporation temperature (see Fig. 5). Therefore, evaporation temperature has similar effect on COP and refrigeration capacity. The results of the experiments are similar to the analysis conducted by Guo et al. [19].

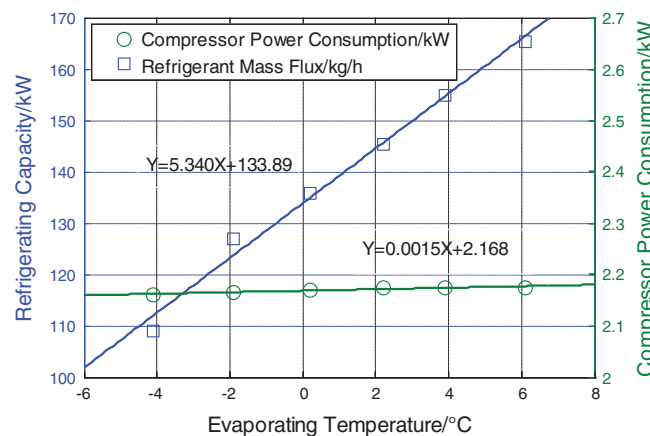


Figure 5: Effect of evaporation temperature on compressor power consumption and refrigerant flow rate

4.3 Effect of Condensation Temperature on Refrigeration Performance

The regulation of system condensation temperature was achieved by adjusting the cooling water temperature. The effect of condensation temperature on the system refrigeration capacity and COP when the compressor A ran alone is shown in Fig. 6. Both the system refrigeration capacity and COP decrease as the condensation temperature increases. The unchanged evaporation temperature and compressor suction gas superheat mark kept refrigerant enthalpy of the evaporator outlet unchanged, but the increase in condensation temperature and unchanged pre-expansion valve subcooling mark decrease the refrigerant enthalpy of the evaporator inlet. In other words, per unit mass refrigeration capacity of the refrigerant decreases with the increase in the condensation temperature. In addition, the refrigerant circulation flow decreases as the condensation temperature increases, both of which verify that the system refrigeration capacity decreases as the condensation temperature increases.

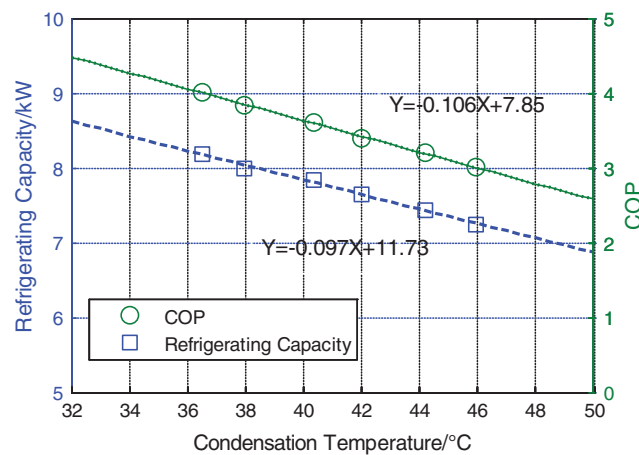


Figure 6: Effect of condensation temperature on cooling capacity and COP

The specific volume of compressor inlet refrigerant was maintained constant, while the compressor volumetric efficiency decreases with the increase in pressure ratio; hence, the refrigerant flow rate decreases as the condensation temperature increases. In contrary to the effect of evaporation temperature on the compressor power consumption, although condensation temperature contribute little towards the effect on refrigerant flow rate, unit mass compression work of the compressor on the refrigerant increases as the condensation temperature increases, thereby, causing the compressor power consumption to increase with the increase in condensation temperature (see Fig. 7). With the increase in condensation temperature, the system refrigeration capacity decreases while the compressor power consumption increases; hence, COP decreases as the condensation temperature increases (see Fig. 6).

4.4 Effect of System Pressure Ratio on Refrigeration Performance

With reference to study by Yu et al. [20] condensation temperature was controlled by adjusting the cooling water temperature, while the evaporation temperature was controlled by adjusting the chilled water temperature. Both evaporation temperature and condensation temperature affect the system load by influencing the unit mass refrigeration capacity. The condensation temperature affects the enthalpy of refrigerant at evaporator inlet, while the evaporation temperature affects the enthalpy of refrigerant at evaporator outlet.

Different from using single variable method to control the effect of evaporation temperature and condensation temperature on the refrigeration capacity and COP, evaporation temperature and condensation temperature accordingly change with the modification in system pressure ratio. In other

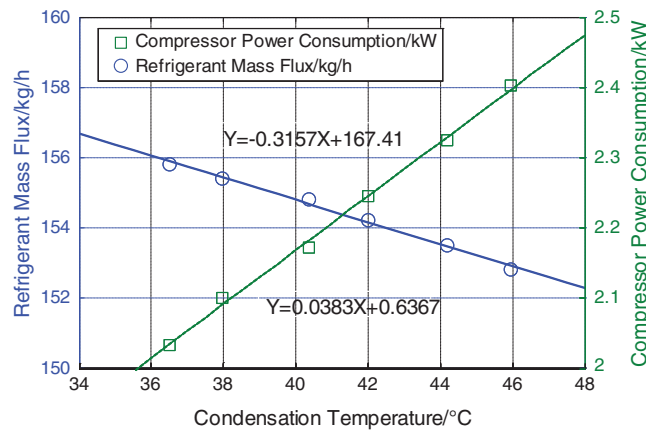


Figure 7: Effect of condensation temperature on compressor power consumption and refrigerant flow rate

words, the enthalpy of refrigerant at evaporator inlet and outlet change accordingly. The effect of pressure ratio on the system refrigeration capacity and COP is shown in Fig. 8. The refrigeration capacity and COP increase as the pressure ratio decreases.

Although the compressor volumetric efficiency decreases as the system pressure ratio increases, when the compressor operating frequency remains unchanged, the refrigerant circulation flow rate decreases with the increase in pressure ratio; thereby, reducing the compressor power consumption. However, an increase in system pressure ratio lowers the compressor isentropic efficiency and increases the compression work per unit mass of the refrigerant. As shown in Fig. 9, the combination effect of both increase the compressor power consumption. Eventually, the effect of system pressure ratio on the compressor power consumption and refrigeration capacity us decrease in COP with an increase in system pressure ratio (see Fig. 8).

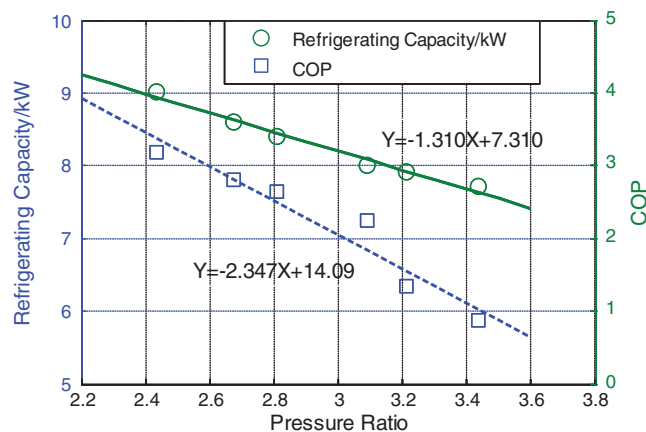


Figure 8: Effect of system pressure ratio on refrigeration capacity and COP

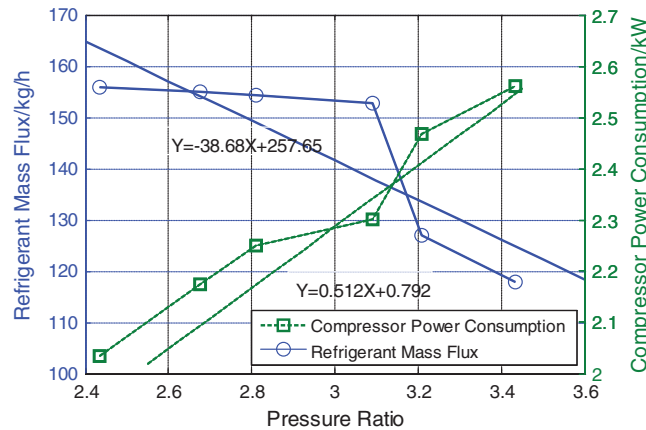


Figure 9: Effect of system pressure ratio on compressor power consumption and refrigerant flow rate

5 Conclusion

Based on the water-cooled parallel compressor water-cooled chiller, combining the effect of compression ratio on compressor volumetric efficiency and isentropic efficiency, the effects of evaporation temperature, condensation temperature, and system pressure ratio on the refrigeration capacity, COP, and compressor power consumption were analyzed. The experiment results show that:

1. For parallel compressors, different refrigerant circulation flow rates in the system lead to different load states.
2. With the decrease in evaporation temperature, system refrigeration capacity and COP gradually reduce, and the refrigerant flow rate increases. However, the compressor power consumption is not much affected by the evaporation temperature.
3. The system refrigeration capacity and COP decrease with the increase in condensation temperature. Also, with the increase in condensation temperature, the refrigerant flow rate decreases and compressor power consumption increases.
4. With the decrease in pressure ratio, the system refrigeration capacity and COP increase. With the increase in pressure ratio, the refrigerant flow rate decreases, while the compressor power consumption increases.

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Conflicts of Interest: The authors declare that they have no conflicts of interest to report regarding the present study.

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Appendix

C_p : constant pressure specific heat of cooling water, J/(kg•°C)

h_{suc} : compressor suction specific enthalpy, kJ/kg

h_{inEXV} : refrigerant specific enthalpy of expansion valve; kJ/kg

I : the compressor input current, A

m_{cow} : chilled water circulation flow, kg/s

m_{evw} : chilled water circulation flow, kg/s

P_{con} : measured condensation pressure Pa

P_{evp} : measured evaporation pressure, Pa

P_{evp} : evaporation pressure, Pa

Q_{evp} : system refrigeration capacity, kJ

Q_{w} : cooling water side heat exchange amount, kJ

Q_{r} : refrigerant side heat exchange amount, kJ

$T_{\text{evw,out}}$: chilled water outlet temperature, °C

$T_{\text{cow,in}}$: cooling water inlet temperature, °C

$T_{\text{cow,out}}$: cooling water outlet temperature, °C

T_{evp} : system evaporation temperature, °C

T_{con} : condensation temperature, °C

U : compressor input voltage, V