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Experimental Investigation of the Performance of a Single-stage Auto-cascade Refrigerator

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Abstract: Auto-refrigerating cascade (ARC) systems possess many advantages comparing with traditional cascade refrigeration systems. This work proposed a novel ternary mixture, R600a/R23/R14, for ARC systems for 190K applications. The performance of the ternary mixture and the influences of compositional ratio and bypass scheme were assessed in a modified domestic cooler. The results demonstrated the feasibility of the proposed R600a/R23/R14 ternary mixture as an environmental benign alternative for ARC systems. The performance varied little within a certain composition range and a mass ratio of 35/30/35 for R600a/R23/R14 mixture was recommended. It also showed that the two bypass schemes, which can regulate more effectively the refrigerant compositions, were better than the conventional hot-gas bypass approach. The variation of the evaporator temperature suggested the presence of local dryout at high heat loads (i.e., larger than the design value), which should be carefully prevented.

Keywords: Auto-refrigerating cascade system, Ternary refrigerants, R600a/R23/R14, Pressure characteristics, Bypass control

1 Introduction

Although pure fluids and azeotropic refrigerant mixtures have been traditionally used in refrigeration systems, many attempts have been endeavored to develop zeotropic refrigerant mixtures for specific applications. In recent years, there has been a remarkable development in mixed refrigerant systems. Single-stage vapor compression refrigerators using mixed refrigerants can provide 80~230K low temperature environment, which have been widely used in the fields of gas chiller, liquefaction, cryosurgery, cryopreservation, semiconductor fabrication, infrared sensors and water vapor trapping [1, 2]. Two types of systems have been successfully developed that can reach below 230K. One is known as the Linde–Hampson refrigerator (LHR) and the other is the auto-refrigerating cascade (ARC) systems. The ARC systems have many advantages comparing to traditional cascade refrigeration systems, and are widely used to provide refrigeration temperature down to 230K [3, 4].

Significant progress has been made on the development of refrigerant mixtures for ARC systems in recent years, especially for lower temperature applications. Three categories of refrigerants were generally used to form different mixtures, (i) high boiling components such as R600a (isobutene), R134a (CF_3CH_2F), R22 ($CHClF_2$) and R290 (propane); (ii) middle boiling components including R23 (CHF_3), R170 (ethane), R744 (carbon dioxide), and R1150 (ethylene); and (iii) low boiling components, i.e., R14 (CF_4), R50 (methane), N_2 (nitrogen) and Ar (argon). Gong et al. [5] investigated the refrigeration performance of binary zeotropic refrigerant mixtures of R170/R23 (37.2/62.8) and R170/R116 (34/66), and a ternary zeotropic refrigerant mixture of R170/R23/R116 (26/41/33), where the numbers in the parenthesis refer to the mass ratio of the refrigerants. These mixtures showed good potentials for refrigeration in the 190K temperature range. Many other refrigerant pairs were proposed by Venkatarathnam et al. for ~200K applications.

Examples included R23/R142b (7.9/92.1), R22/R142b (57.7/42.3), R23/R134a (18/82) and R23/R125/R134a (15/25/60) [6, 7]. Wang et al. [8, 9] experimentally studied the influence of compositions of various refrigerant mixtures, i.e.,R600a/R290/R1150/R50/R728, R23/R134a and R170/R290, and suggested to use mixtures with appropriate combination of inflammable refrigerants and natural refrigerants to achieve good performance. Kim et al. [10] investigated an ARC system using zeotropic refrigerant mixtures of R744/R134a and R744/R290. Comparing with pure carbon dioxide based vapor compression systems, better refrigeration performance could be achieved by using appropriate composition of the refrigerant mixtures. The search of the best refrigerant mixtures for different ARC system applications is always on the going. R14 is one of the low temperature refrigerants with stable performance, R23 is currently one of the best refrigerants at 200K for cascade refrigeration systems, and R600a is the most commonly-used refrigerant in household refrigerators. Surprisingly there is still no proposition to use refrigerant mixtures of R14, R23 and R600a for ARC systems. The potential use of R600a/R23/R14 mixture would provide an environmental benign (i.e. no chlorine atoms and zero Ozone Depletion Potential) and chemically stable refrigerants if its refrigeration performance can be accepted.

To reduce the energy consumption, a lot of investigations have been performed recently to understand the behavior of refrigeration systems under different bypass conditions. In cold conditions, defrosting is always required to remove accumulated frosts from heat exchangers by using a hot gas bypass scheme. Comparing with the on–off cycling under frosting/defrosting conditions, defrosting by a hot-gas bypass showed a higher refrigeration capacity with less temperature fluctuations, but at the cost of more compressor power [11]. Tso et al. [12] compared the performance of the hot gas bypass control and the suction modulation control in a refrigerated shipping container, and showed that the suction modulation control strategy was more energy efficient. Yaqub et al. [13] compared three different bypass schemes (i.e., hot gas injection into the suction line, liquid and hot gas injection into the suction line, and hot gas injection into the evaporator) for a R134a refrigeration system, and found that the coefficient of performance (COP) was the highest for the hot gas injection directly into the evaporator. However it is of note that the influence of bypass schemes on the performance of an ARC system has not been reported. It is expected that the variation of the bypass mode could affect the flow regime, pressure and temperature distribution of the refrigerator, which would influence subsequently the thermodynamic performance of the ARC system.

This work aimed to conduct a feasibility study to assess if the ternary system R600a/R23/R14 could be used on an ARC system for 190K applications. Experiments were performed in a modified domestic cooler, and the influences of the component and composition of a ternary mixture ternary composition and different bypass schemes on the ARC performance were experimentally investigated.

2 Refrigerants and experimental system

2.1 Selection of the refrigerant mixture

The choice of the refrigerants, which has a direct impact on the reliability of the system, is the first question to answer for a single-stage compression mixed-refrigerant throttling refrigerator. The variation of refrigerant pairs and their compositional ratios could produce different evaporating temperature. Per the design requirement, the application temperature of the cooler is ~190K and the refrigerants shall be condensed by water at ambient temperature. The saturation pressure and temperature curves of ten commonly-used refrigerants in ARC systems are calculated through software NIST Refprop8.0, shown in Fig.1. The evaporation temperature of R50 is the lowest and

R600a is the highest under the same evaporation pressure, and the evaporation pressure of R600a is the lowest and R50 is the highest for a given evaporation temperature.

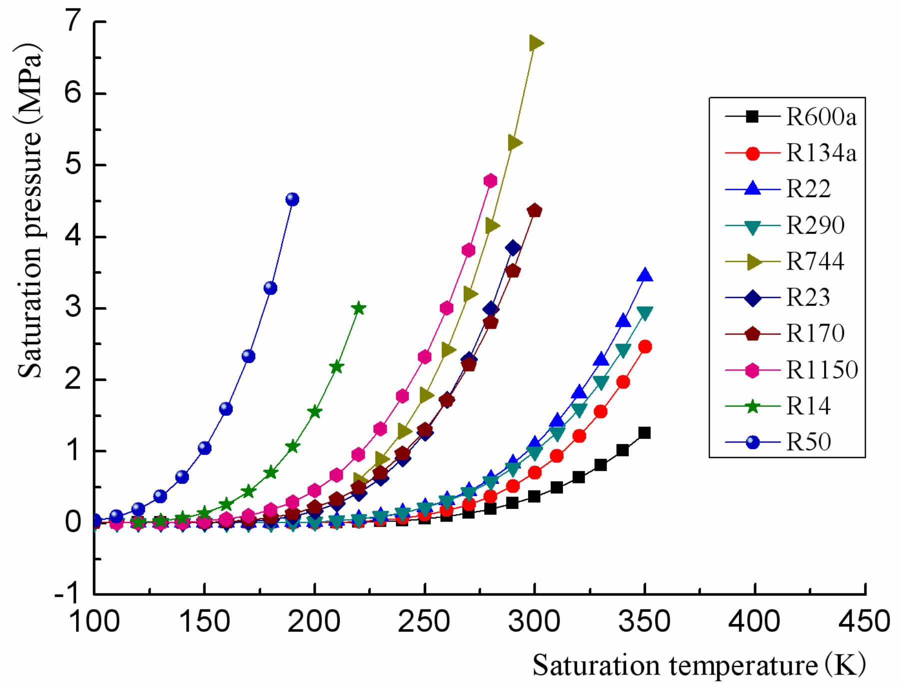


Fig.1 The relationship of saturation pressure and saturation temperature of various refrigerants

Three different categories of refrigerant mixtures were assessed with different amounts of high, low and middle boiling components to identify an appropriate refrigerant mixture. The classification is temperature range generally based on the normal boiling temperature of a refrigerant, T_b, at one atmospheric pressure. For refrigerant with $T_b > 230K$, it is defined as high boiling refrigerant; for Tb< 150K, it is the low boiling refrigerant; and for 150 K<T_b<230 K, it is the medium boiling refrigerant. The properties of the candidate refrigerants are shown in Table 1. Among all the high boiling refrigerants compared, i.e. R600a, R22, R290 and R134a, R600a has the Ozone Depletion Potential (ODP) of zero and the Global Warming Potential (GWP) of 20, which are much smaller than R134a and R22. As the flammability and explosion risk of R290 is significantly greater than R600a, R600a was chosen as the high boiling refrigerant. Comparing all the middle boiling refrigerants, i.e. R744, R23, R170 and R1150, the freezing point of R744 is 216.59K, which is too high for an evaporating temperature of 190K. As the flammability and explosion risk of R170 and

R1150 are significantly greater than R23, R23 was selected as the middle boiling component. Similarly R14 was proposed as the low boiling component as R50 is more dangerous from the flammability concern. It should be noted that although the ODP values of R23 and R14 are both zero, the GWP values are relatively high. Both are not the perfect refrigerant components. However it is very difficult to find other better refrigerants in the required temperature range. So considering the temperature range, the applicability, and the zeotropic feature, the mixture of R600a, R23 and R14 was proposed as a novel refrigerant mixture for ARC systems. Comparing to other refrigerant mixtures, there is no chlorine atom in all three selected refrigerants, and R23 and R14 have good chemical stability.

Another criterion needs to be considered is the boiling temperature difference between refrigerants, which decides the effect of phase separation. If the boiling temperature difference is too small, the separation of phases becomes difficult and the purity of the low temperature fraction obtained after phase separation is not good. Generally speaking, good separation can be achieved if the boiling temperature difference is in the range of 40~80K. From Table 1, the standard boiling point differences between R600a and R23, and R23 and R14, are 70.27K and 46.03K respectively, which are in the ideal temperature range. Appropriately mixed, the ternary system would provide good refrigeration performance yet with less environmental and hazardous influence.

Table 1 Thermal properties of commonly-used refrigerants in ARC systems

Refrigerant	Chemical formula	Mol. Mass /(kg·kmol^{-1})	ODP	GWP	Normal boiling temperature at one Atm.(K)	Freezing point(K)	Critical temperature(K)	Critical pressure (KPa)	Isentropic index

R600a	C_4H_{10}	58.12	0	20	261.42	113.7	407.81	3.629	1.09
R134a	CH_2FCF_3	102.03	0	1300	247.08	169.85	374.21	4.059	1.11
R22	$CHFCL_2$	86.47	0.055	1700	232.34	115.73	369.3	4.99	1.12
R290	$CH_3CH_2CH_3$	44.1	0	~20	231.04	85.53	369.89	4.251	1.13
R744	CO_2	44.01	0	1	216.59	194.75	304.13	7.377	1.22
R23	CHF_3	70.01	0	12100	191.13	118.02	299.29	4.832	1.19
R170	CH_3CH_3	30.07	0	20	184.57	90.368	305.32	4.8722	1.20
R1150	$CH_2{=}CH_2$	28.05	0	20	169.38	103.99	282.35	5.042	1.14
R14	CF_4	88.01	0	5700	145.1	120	227.51	3.75	1.22
R50	CH_4	16.04	0	24.5	111.67	90.69	190.56	4.6	1.31

2.2 Experimental system

The experimental system was modified from an existing small-scale chiller by changing the components and charging different refrigerants. To achieve the best configuration, thermodynamic analysis on the performance of the refrigeration cycle were conducted and the pipe dimensions were decided. The optimization was based on the following assumptions:

(i) The discharge pressure of 2.0MPa and suction pressure of 0.2MPa of the compressor was fixed and there was no pressure variation in the system except the compressor and capillaries. No heat loss occurs in the capillaries. The reasons for this pressure selection were based on the consideration of the compression ratio of the compressor (i.e, in the range of 8-10) and the requirement of leakage free to the refrigeration system (i.e. the suction pressure need to be higher than the ambient pressure)

(ii) The mixed refrigerant in the compressor was approximated as an isentropic compression process, which was based on the experimental conditions that there was no active cooling facilities provided for the compressor and the natural convection effect was estimated very small. It was also assumed that the lubricating oil was completely separated in the oil filter and returns to the compressor; so there was no oil component in the mixed refrigerant except the compressor. There were only mixed refrigerants of R600a, R23 and R14 in the pipeline without other impurities and non-condensable gases

(iii) The phase separator was an adiabatic apparatus and had the function of storing excess refrigerant and automatically regulating the refrigerant vapor compositional ratio. The liquid and vapor refrigerants were separated completely in the phase separator without phase transformation.

(iv) There were cold loss with the external surroundings for two heat exchanges and the evaporator. The mixing was an adiabatic process.

A schematic view of the experimental system is shown in Fig.2, and the corresponding pressure-enthalpy diagram (p-h) is included in Fig. 3. The refrigerant mixtures were compressed by the compressor and flow into the condenser, where the temperature was reduced by the condensing process. In the condenser, the high boiling point refrigerant (R600a) was condensed into liquid, while the middle and low boiling point refrigerants were remained in the vapor state. The vapor and liquid mixture from the condenser flowed into the phase separator 1 after a drier-filter, where vapor and liquid were separated by the gravitational force. The high boiling point liquid phase was sent to the capillary 1 throttle via the phase separator 1, and returned to the compressor suction pipeline after evaporation in the heat exchange 1. The middle and low boiling point mixed vapor refrigerants achieved partial condensation in the heat exchange 1, where the majority of the middle boiling point refrigerant (R23) was condensed into liquid, and most of the low boiling point refrigerant (R14) still remained in the vapor status. Second liquid/vapor separation was achieved in the phase separator 2, and the middle boiling point liquid (R23) joined the return flow of the evaporator after throttling in capillary 2. The low boiling point vapor (R14) from the upper phase separator 2 was condensed and subcooled in the heat exchange 2. The condensed liquid flowed into the evaporator absorbing heat after the throttling through the capillary 3. After the evaporative refrigeration, the low boiling point refrigerant (R14) merged with the middle and high boiling point refrigerants in the compressor suction pipeline, and completed the entire refrigeration cycle.

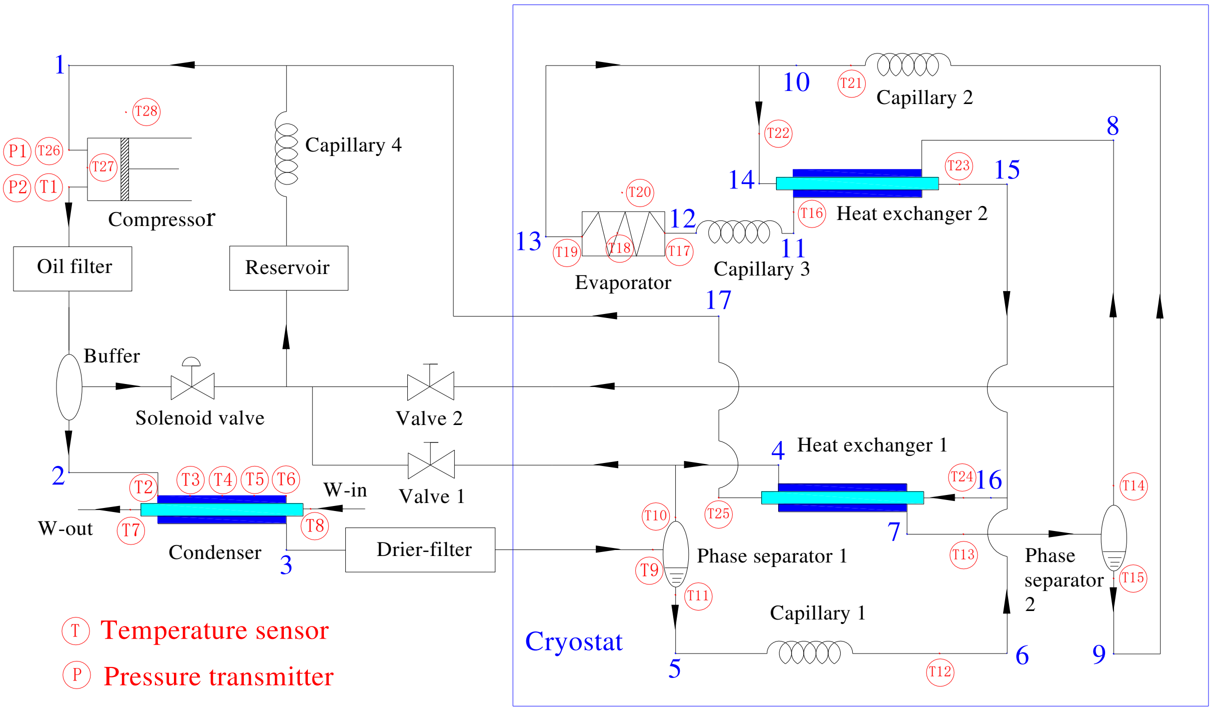


Fig.2 Schematic diagram of the ARC system

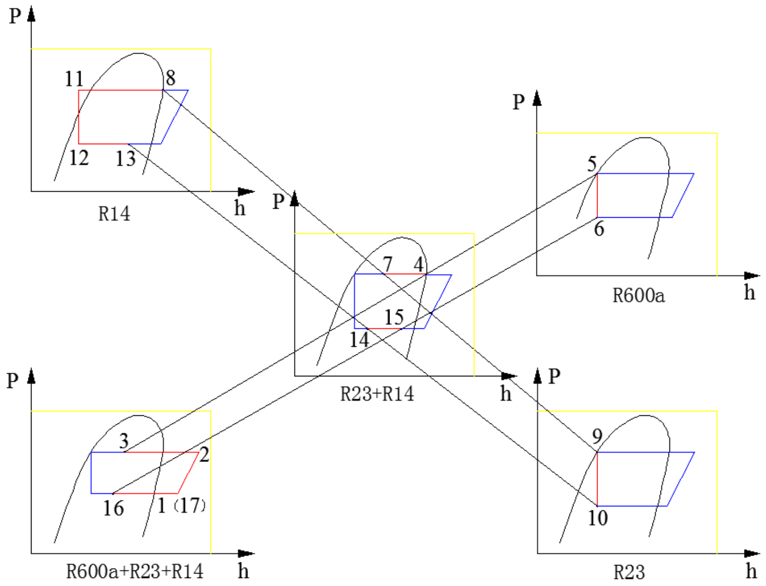


Fig. 3: p-h diagram of the tertiary system

A number of thermocouples were arranged in different characteristic locations of the ARC system as shown in Fig.2. Temperature and pressure data were recorded by a data acquisition system (34970A acquisition instrument of Agilent of American) and stored in a PC. 0.3mm copper-constantan T-type thermocouple wire (Chino, Japan) was used to measure the temperature. They were made with DC welding method, and the accuracy was calibrated in a thermostatic bath against two standard thermometers, and was determined as ± 0.5K. The pressure sensors were NS-I1 type (Shanghai Tianmu Automation Instrument Co., Ltd), and have two measurement ranges, i.e., 0~4.0MPa and 0~2.0MPa of absolute pressure, due to the high system equilibrium pressure. The power supply voltage was 24VDC, and the output signal was 4~20mA, having a precision grade of 0.25 level. The main equipment and model of experimental system is summarized in Table 2.

Table 2 Main equipment and model of experimental system

No.	Equipment name	Number	Model	Note
1	Compressor	1	SC21CL	Danfoss
2	Oil filter	1	OUB 1	3/8 inch

3	Buffer	1	35*1.2 mm, 100 mm	Own manufacture
4	Reservoir	1	3.8 liters	Bingwu
5	Drier-filter	1	DML032S	Danfoss
6	High pressure controller	1	KP5	Automatic reset
7	Condenser	1	Φ6.35*0.8 mm; Φ12.7*0.8 mm	3.25 m
8	Heat exchanger	2	Φ6.35*0.8 mm; Φ12.7*0.8 mm	1.40 m
			Φ9.52*0.8 mm; Φ15.88*1.0 mm	1.55 m
9	Capillary	4	0.8 mm	5.78 m
			0.8 mm	7.55 m
			0.8 mm	1.59 m
			1.0 mm	0.67 m
10	Phase separator	2	Φ35*1.2 mm, 100 mm	Own manufacture
11	Evaporator	1	Φ6.35*0.8 mm	4.0 m
12	Valve	2	Danfoss GBC	1/4 inch
13	Solenoid valve	1	EVR3	1/4 inch

The evaporator's cooling capacity was initially identified as 50W, and the evaporator temperature was designed at 190K. The compressor was SC21CL of Danfoss, and had the suction and discharge pipe size of Φ9.52*0.8mm and Φ6.35*0.8mm respectively. The compressor speed was calculated according to a nominal speed of 2800r/min. The nominal power of the compressor was 590W under the isentropic case, and the mass flow rate of refrigerant was 6.23g/s under standard conditions. The designed evaporating pressure and condensing pressure were 0.2MPa and 2.0MPa respectively. The mixed refrigerant temperature of the suction line was designed at 270K because it required a certain degree of superheat. As the discharge temperature increased with the increase of the suction temperature, proper control of the suction temperature was assured to control the discharge temperature below 390K. Higher discharge temperature would produce issues of the decomposition and carbonization of the lubricating oil. The discharge temperature was designed as 380K in the experiment. The inlet and outlet temperature of cooling water were fixed at 295K and 300K. The refrigerants were charged into the system by the weight filling method. The total filling quantity was in the range of 340~380g, with 360g being the optimal refrigerant charge. The ambient temperature was in the region of 285~290K, corresponding to the spring season in Shanghai. Fig.4 shows a snapshot of the experiment system.

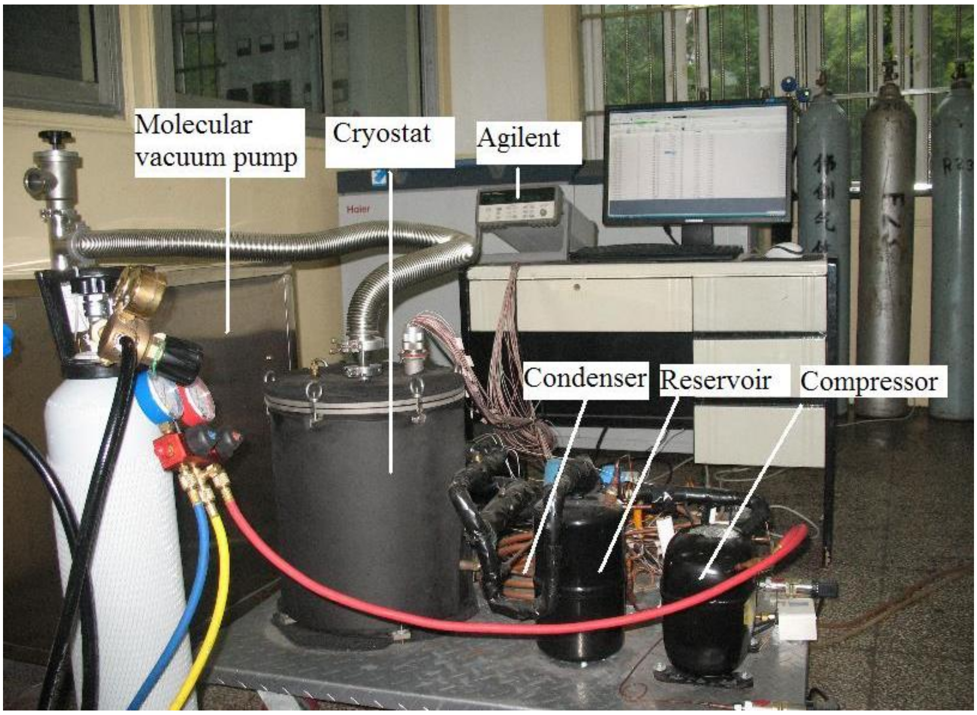


Fig.4 Photo of the experiment system

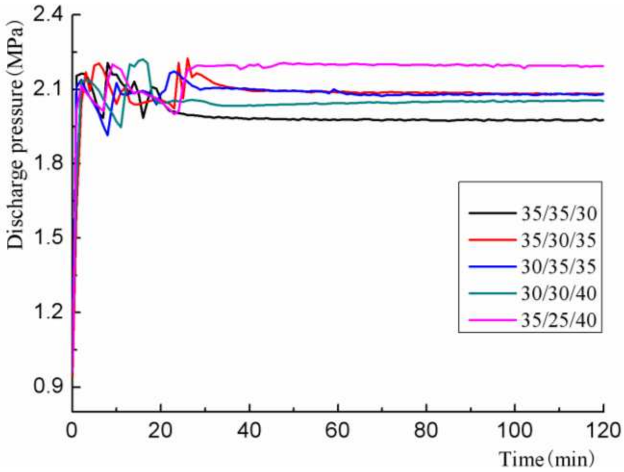
As shown in Fig.2, most of the components including the evaporator and the heat exchangers were located inside a cryostat, which was a stainless steel barrel with diameter of 350mm and height of 750mm. The inner cryostat was polished precisely in order to reduce the radiation heat transfer. Two pumps were used to control the vacuum level inside. A mechanical pump was used first to reach ~10Pa, and then a FF110 molecular vacuum pump (Shanghai FanFeng Vacuum Machinery Co., Ltd) was employed to reach a working pressure of $4.2x10^{-3}$Pa. The molecular vacuum had a pumping speed 400L/S and a limiting vacuum degree of $5.0x10^{-5}$Pa.

After the system installation was completed, three steps were performed, i.e., leakage testing, vacuum pumping and charging refrigerant, before the system diagnosis. The leakage check was performed under both high pressure and low vacuum conditions. For high pressure check, dry nitrogen gas was filled in the system at 2.2MPa for 48h, and for vacuum checking, the system pressure was controlled at 0.002MPa for 24 hour. This was achieved by monitoring constantly the suction and discharge pressure, the suction and discharge temperature, frosting condition on the copper pipeline and the power of compressor. After the leakage check, the amount of mixed refrigerant R600a, R23 and R14 was increased gradually according to the running situation,, and the charging process was finished when the discharge pressure reached ~2.2MPa. Experiments are then performed, with results described below.

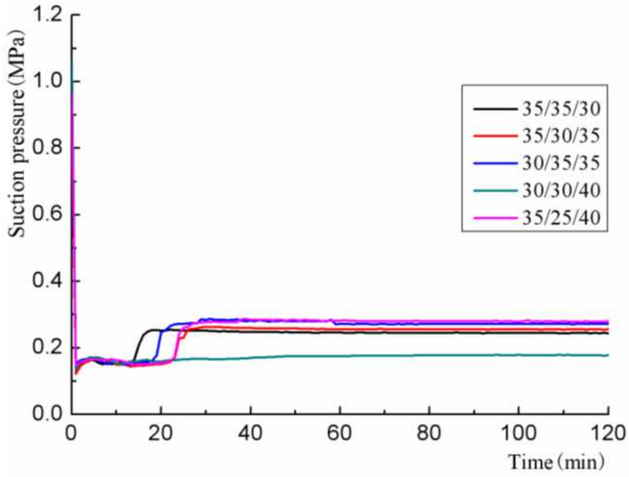
3. Result analysis

3.1 Influence of the refrigeration composition ratios

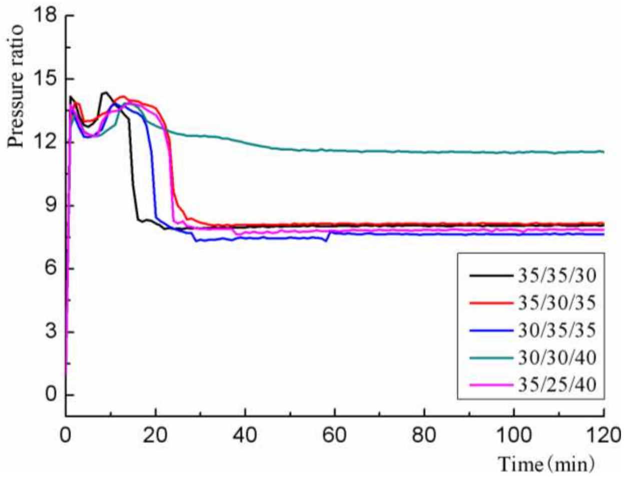
As the performance of the refrigeration system is heavily influenced by the compositions of the ternary system, it is essential to identify an appropriate composition ratio. In the experiment, five different ratios of the refrigerant were examined, namely 35/35/30, 35/30/35, 30/35/35, 30/30/40 and 35/25/40 for mixed refrigerant R600a, R23 and R14, where the numbers refer to the mass ratio of each refrigerant. Six system parameters: the discharge pressure, the discharge temperature, the suction pressure, the suction temperature, the pressure ratio and the evaporation temperature, were used to assess the performance of the refrigerants, shown in Fig.4. In the comparison, the discharge pressure of different compositions was controlled to be roughly the same by adjusting the bypass valve.



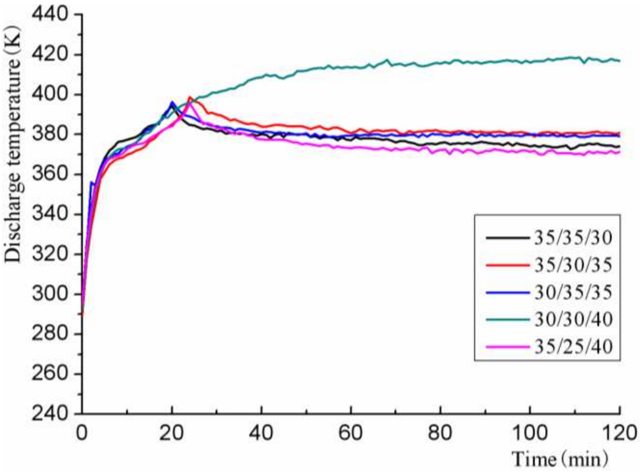
(a) Discharge pressure with different compositions



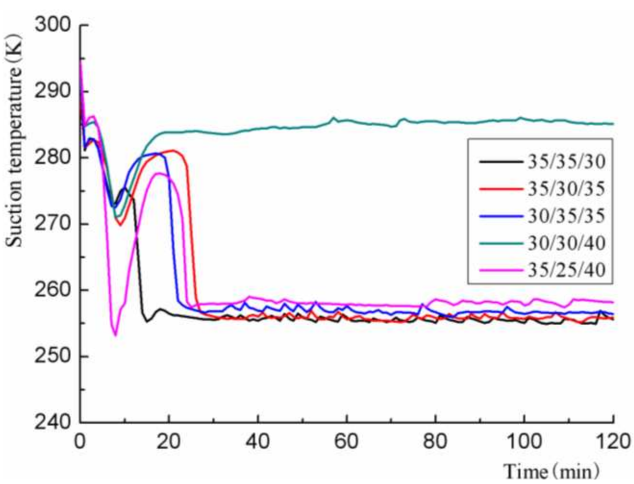
(b) Suction pressure with different compositions



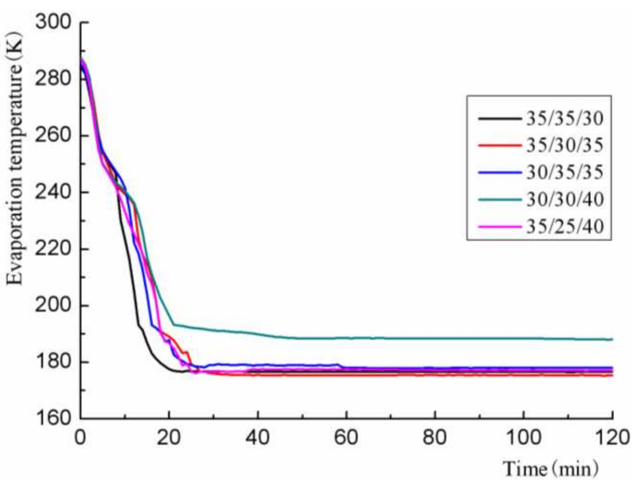
(c) Pressure ratio with different compositions



(d) Discharge temperature with different compositions



(e) Suction temperature with different compositions



(f) Evaporation temperature with different compositions

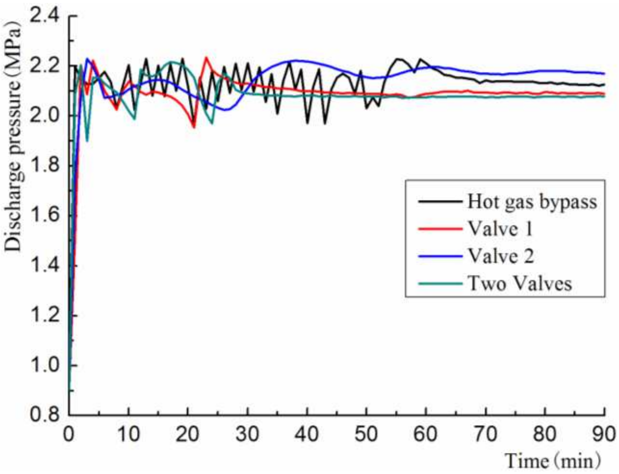
Fig.5 Performance parameters with different refrigerant compositions (the notations in the graph refer to the mass concentration of R600a/R23/R14 in the mixture refrigerant.)

Fig.5 shows that the refrigeration system reached steady state in ~30 minutes time. The discharge pressure, Fig.4a, was in the range of 2.0MPa~2.2MPa regardless of the composition ratios. Apart from one compositional ratio, 30/30/40, all other ratios showed similar behavior in terms of the pressure ratio, the discharge temperature, the suction temperature and the evaporation temperature. A higher discharge temperature was observed for the mixture with the ratio of 30/30/40, which was due to the relative large proportions of the lower boiling point refrigerants, i.e., R14 and R23. Correspondently, it had higher discharge temperature, higher pressure ratio and higher suction temperature. The increased portion of R14 had a similar effect of the non-condensable gas in the refrigeration system that deteriorated the heat transfer rate, resulting in a higher evaporation temperature. Other observation supported such a view, i.e., the cooling process was not very good and the flow of capillary 3 was larger at the ratio of 30/30/40. A lot of uncondensed vapor crowded at the inlet of capillary 3, resulting in a lower suction pressure.

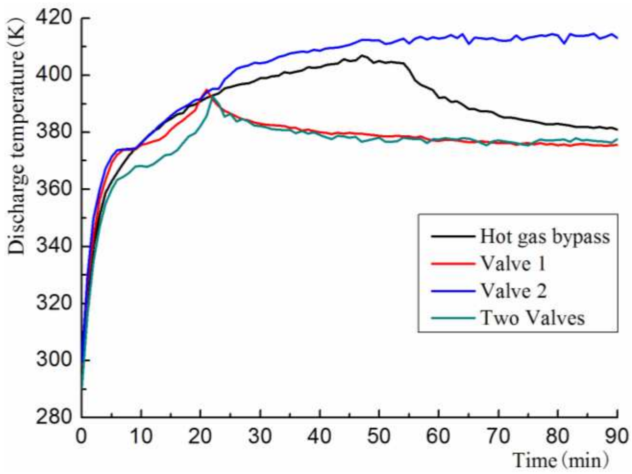
The mixed refrigerant was charged into the system according to the theoretical composition values, there shall have certain deviations. The maximum change of the compositions was up to 6% for different operating periods. The unevenness of the mixture compositions at different positions of the system may be up to 12% or even more [14, 15]. The variation of the environmental temperature and climate change also influenced the optimum composition value. As the ARC system has a self-adjustment function, a small change of the refrigerant components has little effect on the system performance. The mixture ratio of 35/30/35 shows an overall good performance, and is selected in the following comparative experiments.

3.2 Influence of the bypass on the compressor

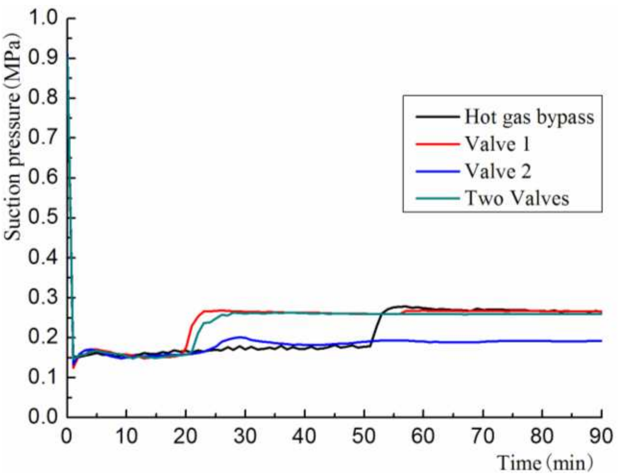
In this work, two groups/four kinds of pressure and composition control schemes were examined for the ARC system: (1) the hot gas bypass; (2) the vapor bypass of the upper phase separator. The vapor bypass of the upper phase separate was further classified as three types: valve 1 control, valve 2 control, and two-valve control, shown in Fig.2. The four type's regulation method can be adjusted either independently or dependently. Fig.6 shows the performance parameters of different pressure and composition control methods.



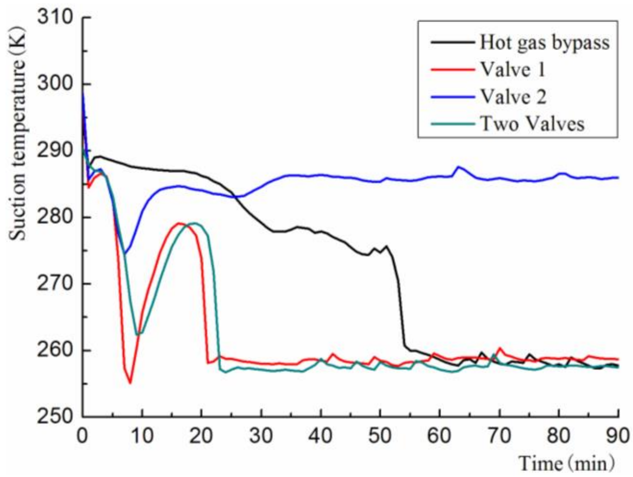
(a) Discharge pressure with different control method



(b) Discharge temperature with different control method



(c) Suction pressure with different control method



(d) Suction temperature with different control method

Fig.6 Performance parameters with different control methods

The traditional ARC systems protect the refrigerant applications by the hot gas bypass method. In the experiment, the hot gas bypass loop is between the compressor and the condenser, followed by a solenoid valve, a reservoir and the capillary 4. If the pressure of the mixed refrigerants discharged from the compressor was higher than the threshold pressure of 2.2MPa, which is the set value for the solenoid valve, the partially mixed refrigerant was bypassed to the reservoir, which could eliminate the potential damage associated with high temperature and high pressure gas. The solenoid valve would close automatically when the pressure was dropped to about 2.0MPa, and the hot gas bypass could make the pressure stabilized at around 2.1MPa. Fig.6 shows that the discharge and suction temperature were higher for the valve 2 control, resulting in low cooling effect. This was because that the bypass valve 2 adjusted the cryogenic refrigerant from the phase separator 2, and caused cooling loss seriously. Clearly valve 2 control should be excluded for controlling the compressor performance. For the valve 1 and two- valve control, similar trends were obtained in terms of the discharge temperature, the suction temperature, and the discharge and suction pressure. The discharge temperature had a turning point at about 20 min for the valve 1 and two-valve control, and finally stabilized at about 370K. The reason was that the system evaporation temperature was reduced to the lowest point, resulting in a decrease in the discharge temperature. The large increase in the suction pressure in ~20min was due to the beginning of the liquid seal at the inlet of capillary 3.

3.3 Influence of the bypass on the evaporation process

The evaporation temperature mainly depends on three factors: the evaporation pressure, the purity of low boiling point refrigerant, and the degree of the subcooling of the low boiling point refrigerant before the throttling. Fig.7 shows the evaporator temperature change by using different control methods. The cooling speed of the evaporator was slower for the hot gas bypass method, and it took 52minutes to reduce the evaporator temperature to the lowest point. This was because that the system pressure was higher and the uncondensed vapor influenced the condensation and cooling process greatly. For the valve 2 control, the cooling speed was relatively fast and the intermediate adjustment time was relatively short, however, the final stable evaporation temperature was higher by ~5K. Because the cold leakage was seriously for the valve 2 individually, cold was discharged to the reservoir together with the low temperature bypass refrigerant. The results from the valve 1 and two- valve control were similar, where the evaporator temperature was reduced to 175K and became stabilized after 20minutes.

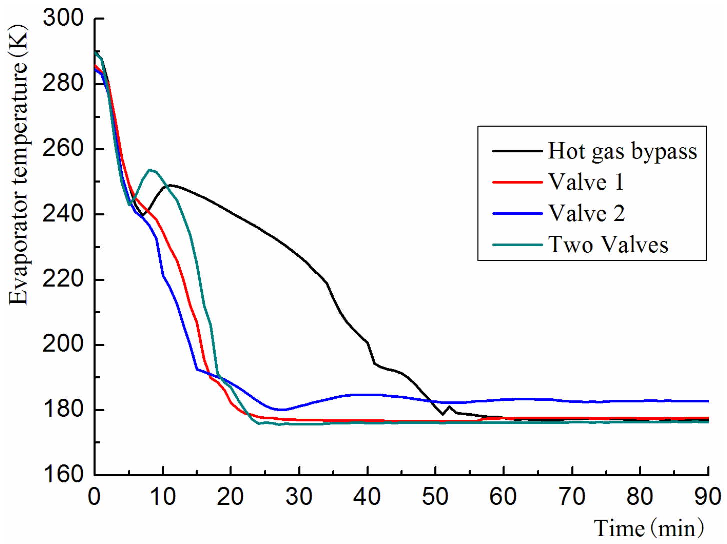


Fig.7 Evaporator temperature with different control method

Inside the evaporator, the evaporation pressure of the mixed refrigerant can be considered as constant, but the evaporation temperature and vapor-liquid phase component will be changed along the streamline of the refrigerants. In the experiments, five temperatures were measured equally along the evaporator length, as shown in Fig.8. At the load of 60W, the evaporation temperature was nearly constant in the first quarter of the evaporator, but it increased relatively quickly afterwards, reaching ~240K at the end of the evaporator. Considering the normal boiling temperature 156K of pure R14 at 0.20MPa, Table 1, local dryouts may be produced towards the end of the evaporator, which deteriorated the heat transfer rate. Such a trend is similar to the pure refrigerant under high heat loads.

At low loads (i.e.<60W), the evaporator temperature curves were similar, i.e., the temperature of mixed refrigerant first decreased and then increased slightly, but maintaining a temperature non-uniformity around 10K across the evaporator. The initial decrease of the evaporator temperature was believed to be associated with the throttling process and the onset of the nucleate boiling of R14. It is also noted in the experiments that the pressure decreased slightly when the mixed refrigerants came into the evaporator from capillary 3, which may also contribute to the initial decrease of the evaporator temperature. Further down the streamline, the temperature of mixed refrigerants increased slightly by absorbing the external heat. Such relative uniform surface temperature suggested that nucleate boiling was occurring inside the evaporator. It shall note that the designed maximum heat load was 50W, and such results actually proved the validity of the experimental system. The cooler shall be operated within the designed range, i.e., <=50 W, for the current systems. Detailed study of the boiling process is currently ongoing.

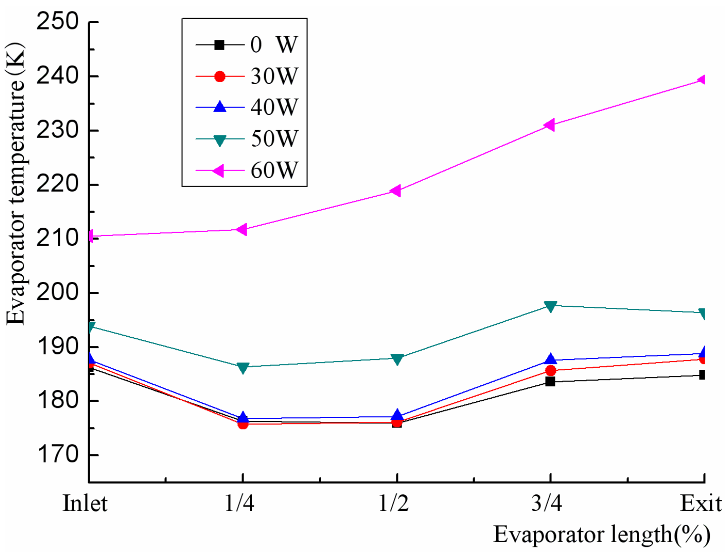


Fig. 8 Evaporator temperature with different refrigeration loads

4. Conclusions

This work proposed a novel ternary mixture, R14, R23 and R600a, for an auto-refrigerating cascade (ARC) system for 190K applications, and assessed their performance in a modified domestic cooler. The influence of composition ratios and the bypass schemes on the ARC performance was studied. In a short summary:

- Both thermodynamic and environmental analysis showed that the proposed R600a/R23/R14 ternary mixture is an environmental benign alternative for ARC systems.
- The assessment of the composition effect showed that the slight composition variation has little effect on the system performance due to the self-adjustment feature of the ARC system. An appropriate mass ratio of the ternary mixture R600a/R23/R14 was determined as 35/30/35.
- The bypass control experiments illustrated that the valve 1 control and two-valve control could effectively regulate the refrigerant compositions, and were better regulation methods than the conventional hot gas bypass approach.
- The large increase of the evaporator temperature at high heat loads (i.e. 60W) suggested that local dryouts occurred inside the evaporator, which shall be avoided for any engineering practice.

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