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Performance Optimization of Variable Speed Room Air-Conditioner Under Intermediate Speed Working Condition

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ABSTRACT

Through the method of experiment and simulation, some optimized designs have been carried out to improve the performance in intermediate working conditions during the process of performance matching for APF of R32 variable speed room air-conditioner. In this paper the main reason for the poor performance in intermediate conditions of R32 air conditioning system is clarified by the analysis of the heat exchange between the refrigerant side and the air side, and some optimized designs of the indoor unit heat exchanger's pipeline layout and pipe diameter have been carried out. The optimization results show the performance of the intermediate condition has been improved by about 4%, and the performance of APF of the R32 room air conditioning system has been increased by 3.8% compared with the original unit.

1. INTRODUCTION

R32 has the similar thermodynamic characteristics to R410A, and it is a hot alternative refrigerant for R410A due to its low GWP value. Although flammability is one of the disadvantages of R32, some countries have revised R32 refrigerant flammability regulations to improve the possibility of R32 replacing R410A. Legalization of R32 has applied to room air-conditioner (below 3 USRT) by the High Pres-sure Gas Safety Institute of Japan since 2013. R32 is determined as weak flammable refrigerant by ANSI/ASHRAE standard 34/15-2010. R32 is considered "non-flammable" during transportation process in the transportation regulations of DOT173.115 by the U.S. Department of Transportation. And some standards for the installation and maintenance of flammable refrigerant applying in room air-conditioner are in the process of development in China.

Although R32 has excellent thermodynamic characteristics，the poor performance under intermediate conditions of R32 room air-conditioner affects the further improvement of APF performance during the matching process of R32 replacing R410A. This paper analyzes the reason for the poor performance in intermediate conditions of R32 room air-conditioner from the theoretical and experimental point of view. The corresponding optimization designs are put forward to effectively improve the APF performance.

2. THE COMPARISON OF PHYSICAL PROPERTIES BETWEEN R32 AND R410A

Table 1 shows that R32 has the similar physical properties to R410A, which has the better environmental characteristics: the GWP value of R32 is about 1 / 3 of R410A. And the theoretical cycle performance of air conditioner system under rated intermediate cooling working condition is approximate. The discharge pressure of R32 is 2% higher than R410A, the discharge temperature is 6.5 ℃ higher than R410A, and the theoretical cop of R32 system is 0.43% better than R410A under rated intermediate cooling working condition seen from the Table 1.

Table1: Refrigerant property comparison between R410A and R32

Refrigerant type		R410A(baseline)	R32
Chemical formula / mixture components		R32/R125（50/50）	CH_2F_2
Physical properties	Relative molecular mass	72.58	52.02
	Critical pressure /MPa	4.9	5.78
	Critical temperature /℃	71.35	78.1
Rated intermediate cooling performance	Suction pressure/Mpa	1.365	1.395
	Discharge pressure/Mpa	2.419	2.478（↑2%）
	Theoretical discharge temperature /℃	54.2	60.7
	Theoretical COP	11.691	11.741（↑0.43%）
Environmental parameters	ODP	0	0
	GWP(100 a)	2100	675
Security	Security level	A1	A2L

Note: Rated intermediate cooling working condition: Evaporation temperature 18℃, Condensation temperature 40 ℃, suction superheat 5 ℃, sub-cooling degree 8.3 ℃;

3. THE EXPERIMENTAL PERFORMANCE COMPARION BETWEEN R410A AND R32

Figure 1 shows the experimental performance comparison between R410A and R32, which is based on the same variable speed room air-conditioner with a cooling capacity of 3.5KW. And this variable speed room air-conditioner is designed for R410A refrigerant.

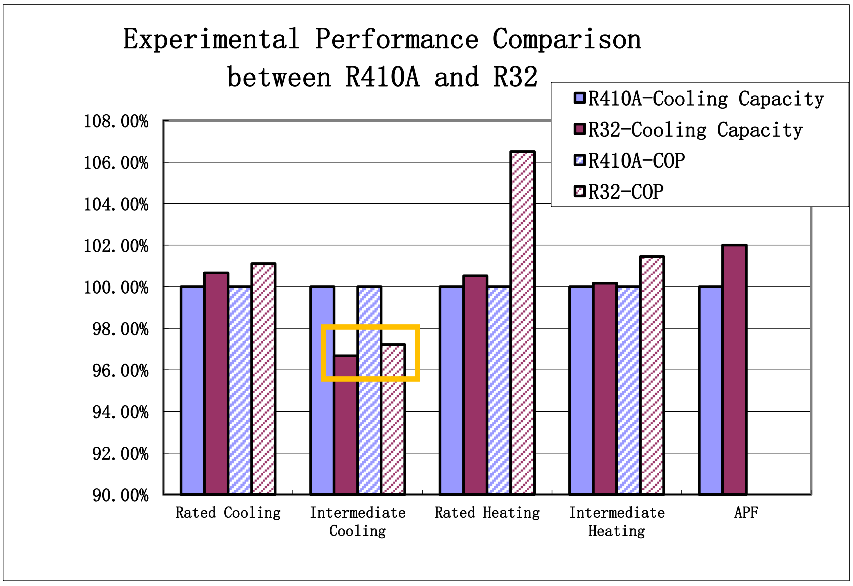


Fig1: Comparison of experimental performance

According to the experimental results shown in Fig 1, the APF of the variable speed air-conditioning system with R32 refrigerant is 2% higher than that of system with R410 refrigerant. The air-conditioning system with R32 refrigerant has the obvious performance advantages under rated cooling conditions and rated heating conditions compared with R410A. But the cooling capacity and COP of R32 air conditioning system are significantly lower than that of R410A system under rated intermediate conditions, especially under rated intermediate cooling conditions seen from the figure1.
When R410A refrigerant is replaced by R32 refrigerant in the same air-conditioner, even though the APF of R32 system has better behavior, the performance of R32 system under intermediate working condition is relatively low. It is expected to improve the performance of R32 system under intermediate conditions through optimization design, so that the APF of R32 system can be further improved.

4. SIMULATION METHOD OF R32 AIR CONDITIONING SYSTEM

In this paper, the analysis method of simulation and test is used to discuss the performance of R32 system under intermediate working conditions.

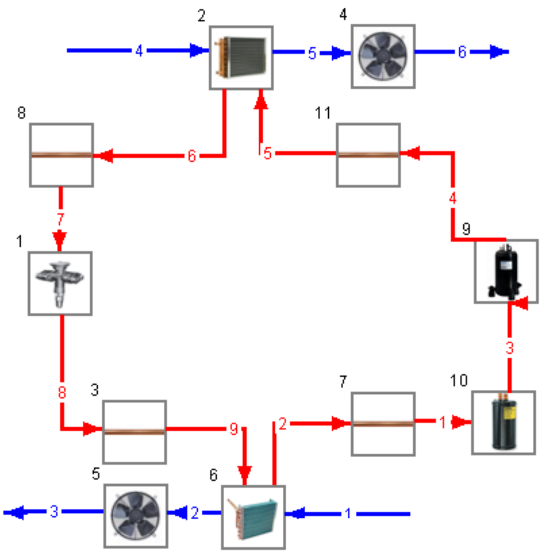


Figure2：Simulation diagram of variable speed air conditioning system

Firstly, the parameterized mathematical model of the variable speed room air-conditioner is built, which mainly includes：

- The structural parameters of the indoor and outdoor heat exchangers of the room air-conditioner are modeled, and the appropriate heat exchange correlations are determined.

  The C.C.Wang model（C.C.Wang et al.1997, 1999, 2001） is used for heat exchange of the air-side. The Shah-Tang model（Shah et al.1979,1982 and Tang et al.1997）is used for the heat exchange of the condensation side. The Kattan-Thome-Favrat mode（Kattan, N., Thome, J.R. and Favrat et al.1998）is used for boiling heat exchange of the evaporation side.
- The mathematical model of compressor is established with 10 coefficient model of AHRI .
- The mathematical model of fan and EEV（electronic expansion valve）is established. The fan adopts efficiency model, and the orifice valve model is used for the EEV, which includes the physical and mathematical models according to the two-phase flow characteristics of refrigerant.

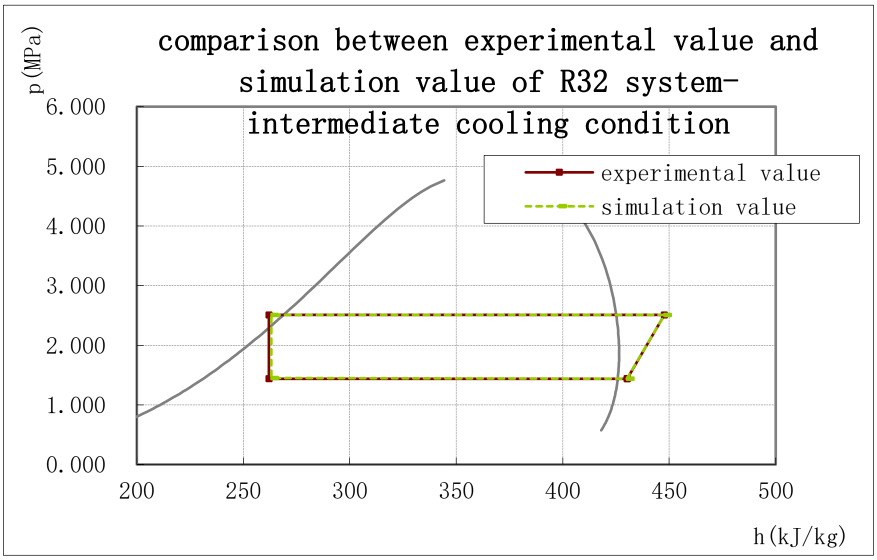


Figure3: Calibration diagram under intermediate cooling condition

Note: Test/Simulation Condition: Outdoor air temperature 35 ℃/24 ℃, Indoor air temperature 27 ℃/19 ℃

After the above components are modeled, a steady-state simulation model of variable speed air conditioning system is formed shown in Figure 2. Calibrate the steady-state simulation model of R32 air conditioning system according to the test data in Figure 1. The calibration result (shown in Figure 3) shows that the deviation between the experimental value and the simulation value is less than 5% under the rated intermediate working condition. Therefore, the following system simulation in this paper is based on the steady-state simulation model of the variable-frequency air conditioning system.

5. PERFORMANCE OPTIMIZATION OF R32 VARIABLE SPEED AIR CONDITIONING SYSTEM UNDER INTERMEDIATE COOLING CONDITION

5.1 Analysis of poor performance of R32 system under intermediate condition

5.1.1 Analysis of refrigerant-side

It has been shown that the refrigerant-side flow and heat transfer characteristics of R32 are different from that of R410A. Gavallini et al.（2001）tested the heat transfer performance of R32 in a horizontal smooth tube with a diameter of 8 mm, the results showed that the heat transfer coefficient of R32 increases with the increase of refrigerant flow, but the heat transfer coefficient of R32 increased much faster at the high flow area. Chen et al.（2013）found through experiments that the heat transfer coefficient of R410A changed smoothly with the increase of flow compared with R22 in a rifled tube.

Experiments prove that the charge of R32 air conditioning system is 70% ~ 80% of R410A system（Zhang et al.2017）during the matching process of R32 replacing R410A, that is to say, the refrigerant flow of R32 system is lower than that of R410A system. Based on the air conditioning system simulation model shown in Figure 2，the heat transfer coefficient with refrigerant flow is compared between R32 and R410A shown in Figure 4 by using the steady-state simulation model of variable air conditioning system shown in Figure 2.

It can be seen from Figure 4 that at the low flow area (intermediate condition), the heat transfer coefficient of R32 system has no advantage compared with R410A system， with the increase of refrigerant flow, the heat transfer coefficient of R32 system is better than that of R410A system under rated condition, which is consistent with the experimental results that the performance of R32 system is lower under intermediate condition and slightly more superior under rated condition during the matching process of R32 replacing R410A shown from Figure 1. The change trend of simulation is in accord with the heat characteristics of R32 described in the above literature.

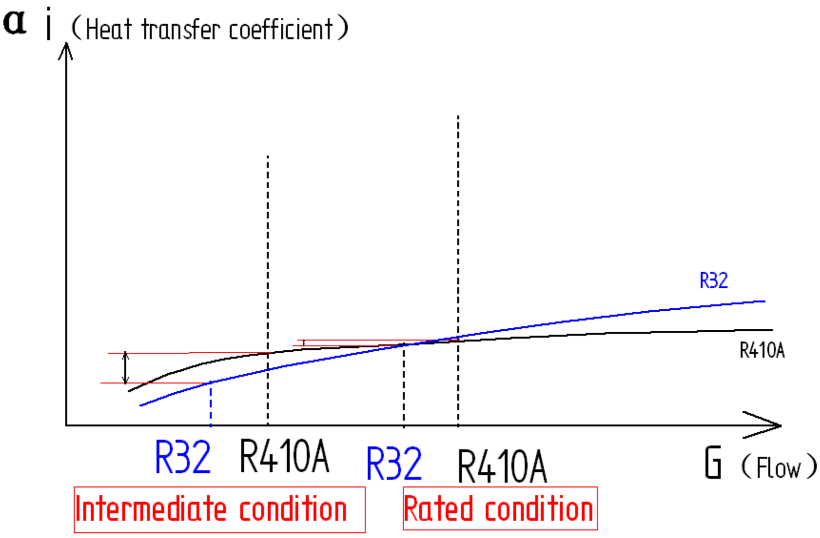


Figure4：Comparison of heat transfer coefficient with refrigerant flow

Note: Test/Simulation Condition: Outdoor air temperature 35 ℃/24 ℃，

Indoor air temperature 27 ℃/19 ℃

From the above analysis, it can be seen that the refrigerant flow has a great impact on the heat transfer performance of R32. The heat transfer effect of R32 at the low flow area (intermediate condition) is significantly lower than that of R410A refrigerant, which results in the poor performance of R32 system under intermediate condition.

5.1.1 Analysis of Air-side

The higher evaporation temperature of R32 room air-conditioner results in small temperature difference of heat exchange between the refrigerant-side and air-side under intermediate cooling condition. Therefore, the quantity of heat exchange of the second row heat exchange tube of the indoor unit decreases, and the available heat transfer area of the second row indoor reduces.

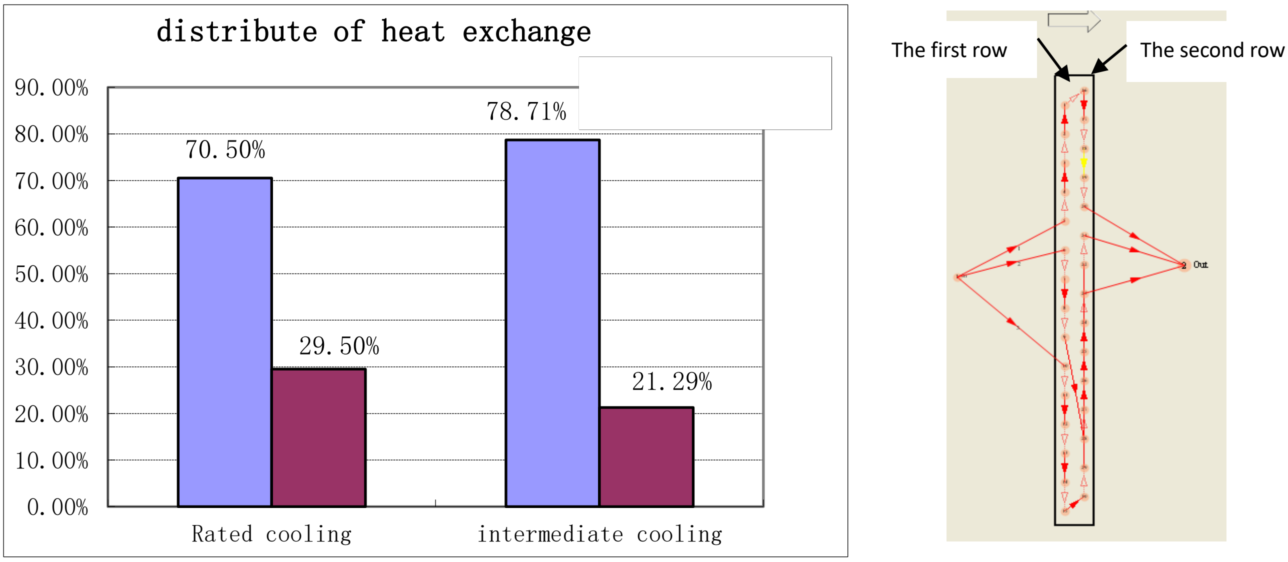
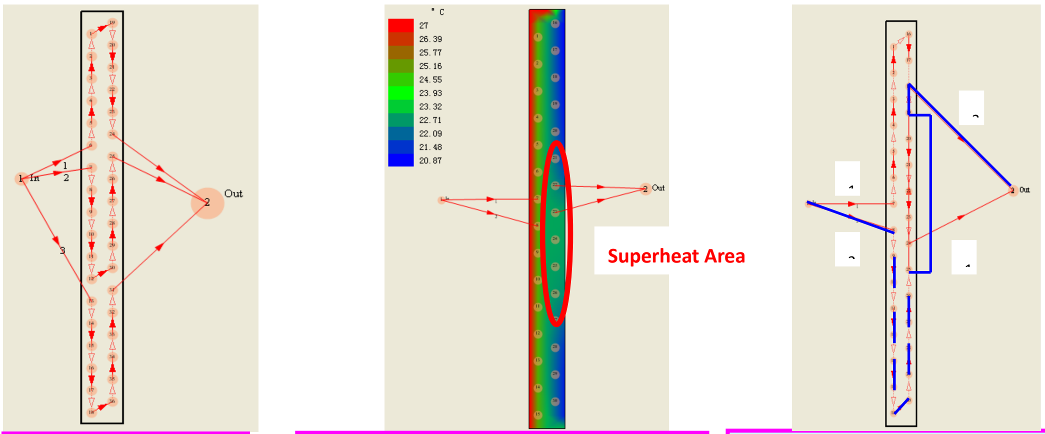


Figure5: distribute of indoor heat exchange

The heat transfer distribution of indoor heat exchanger is calculated by simulation method. Figure 5 shows the ratio of the quantity of heat exchange of the second row to the total under APF condition. The ratio of the quantity of heat exchange of the second row is about 30% under the rated cooling condition, and the ratio is about 20% under intermediate condition because of the high evaporation temperature and the small temperature difference of heat exchange. Therefore, it is necessary to optimize the indoor unit heat exchanger's pipeline layout to improve the effective heat exchange area of the second row, and the ultimate goal is to improve the performance of R32 system under intermediate condition.

5.2 Optimization designs for the performance of R32 system under intermediate condition

5.2.1 Optimization designs for the heat exchanger's pipeline layout of indoor unit



(a) Three pipelines

(b) Three pipelines →two pipelines

(c) Three pipelines →two pipelines (cross arrangement)

Figure6: Optimization designs for the heat exchanger's pipeline layout of indoor unit

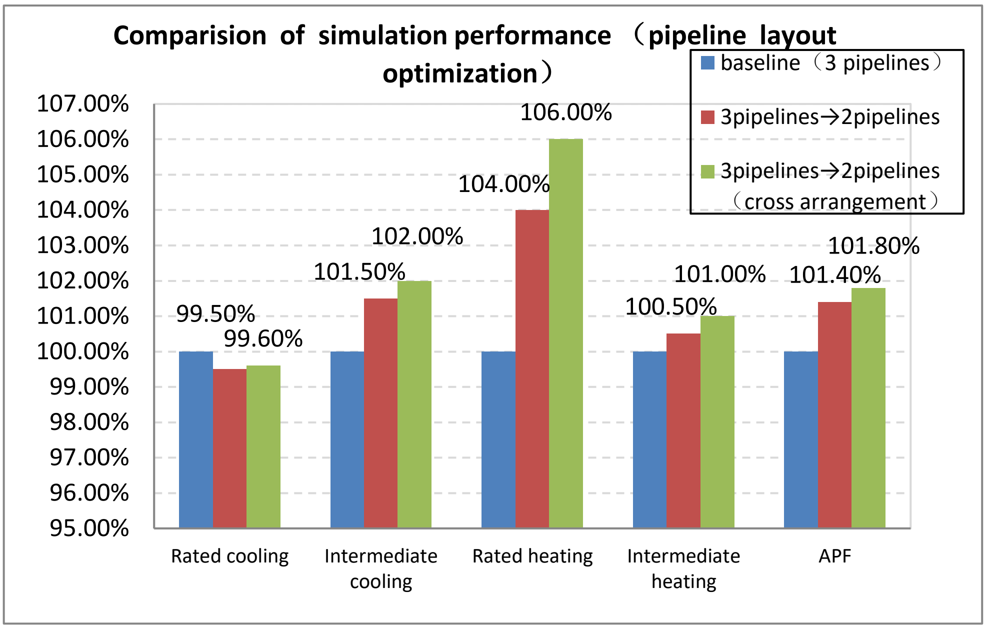


Figure 7：Comparison of simulation performance （pipeline layout optimization ）

According to the above analysis, the number of the pipelines of indoor heat exchanger is reduced to increase the refrigerant flow of R32 system. And the pipeline layout is optimized to the cross arrangement in order to increase the quantity of heat exchange of the second row. These optimized designs are shown in Figure 6. **W**hen adopting the design of across arrangement （shown in Figure 7（c)），the performance of R32 system under intermediate cooling condition are improved by 2%, and the APF of R32 system is increased by 1.8%，according to the simulation results shown in Figure 7.

5.2.2 Optimization designs for the pipe diameter

It can be shown from the simulation results of Figure 8 that the heat transfer coefficient of refrigerant-side with diameter Φ5.25mm of indoor heat exchanger increases to 200.09% compared with diameter Φ7.35mm under intermediate cooling condition and 166.46% under intermediate heating condition.

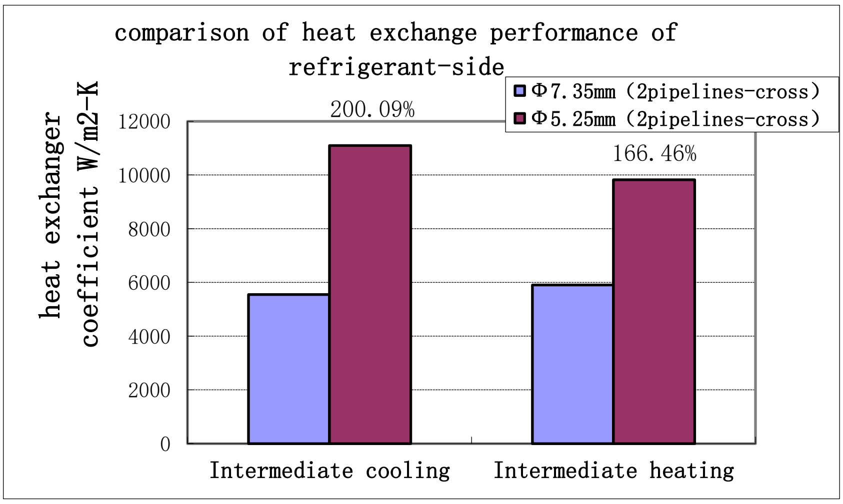


Figure 8：Comparison of heat exchange performance of refrigerant-side

The simulation performance with diameter Φ5.25mm of indoor under intermediate condition is shown in Figure 9 that the intermediate cooling capacity is increased by 4% and the intermediate heating capacity is increased by 2%. The APF of the system is increased by 3.8% and the weight of copper tube of indoor unit is reduced by 18.4%, cost advantage is obvious seen from Figure 10.

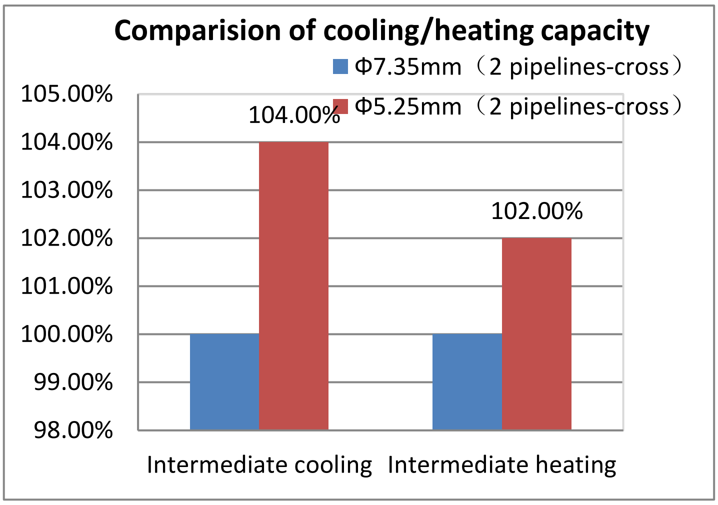


Figure9：Comparison of cooling/heating capacity

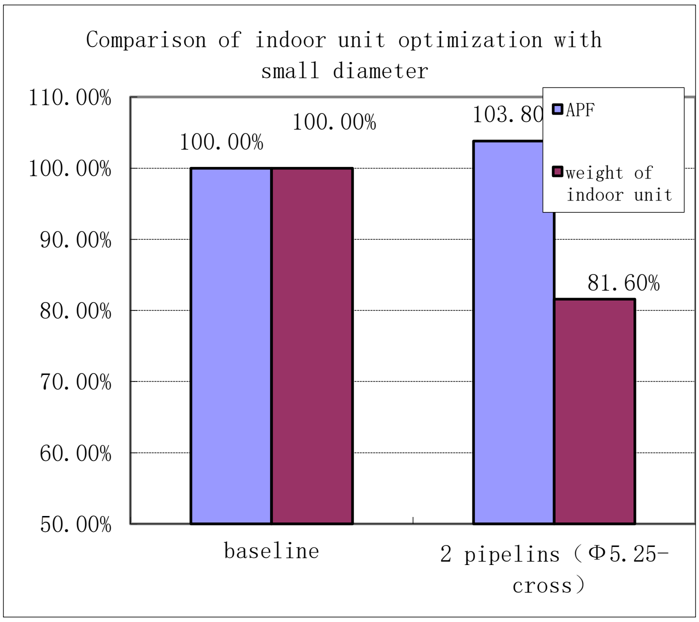


Figure 10：Comparison of indoor unit with small diameter

6. CONCLUSIONS

（1） Although R32 system has more excellent APF performance than R410 system, the performance of R32 room air-conditioner under intermediate conditions during the matching process of R32 replacing R410A is relatively low.

（2） It is clarified that the lower heat transfer coefficient at low flow area and lower heat exchange capacity of the second row of indoor unit of R32 system results in poor performance under intermediate cooling condition compared with R410A system.

（3） The intermediate cooling capacity is improved by 4% and the APF performance is improved by 3.8% through the optimization designs for the heat exchanger's pipeline layout and the reducing pipe diameter of the indoor unit, which has the obvious cost advantage.

NOMENCLATURE

Acronyms and Abbreviations:

APF	Annual Performance Factor
GWP	Global Warming Potential
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration and Air conditioning Engineers
DOT	United Stated Department of Transportation
COP	Coefficient of Performance
ODP	Ozone Depression Potential
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
EEV	Electronic Expansion Valve

Symbols:

P	Pressure, *Pa*
h	Refrigerant enthalpy, *J/kg*
αi	Heat Exchanger Coefficient, $W/m^2\text{-}K$
G	Mass Flow Rate, *kg/s*

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