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Study on the Analysis of Cycle Characteristics in Low Temperature Environment of Automatic Cascaded Air Source Heat Pump with Flash Tank and Economizer

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Abstract **Original Research Article**

The COP of the air source heat pump drops as the ambient temperature decreases, and then the exhaust gas temperature of the compressor rises, making it impossible to use it.

In this paper, an automatic cascaded air source heat pump system (ACASHPs) with flash tank and economizer is presented to improve the performance of air source heat pump in cold regions where the ambient temperature is below -10℃.

The refrigerant R600/R143a (0.2/0.8) mixture was used and the characteristics of the ACASHPs with flash tank and economizer were compared with typical heat pump systems when producing hot water at 50 ℃ under ambient temperature of -25~5 ℃.

The analysis shows that the COP can be 6.2~30.3% higher and the discharge temperature can be 48.7~10.5 ℃ lower than the conventional ACASHPs.

It was also shown that the high COP and stable operating conditions can be achieved compared with the single stage compression systems with the economizer.

Keywords: Air source heat pump, Auto-cascade heat pump, COP, cold region, mixed refrigerant.

1. Introduction

Air source heat pumps (ASHP) are widely used for heating and cooling in domestic and commercial buildings due to their inexhaustible heat sources, ease of installation and use.

Especially, ASHP is becoming a promising device because it is energy-saving, environmentally friendly cooling and heating system that do not emit pollutants such as particles, nitrogen oxides, and sulfur oxides [1, 2].

Nomenclature	
ASHP	air source heat pump
ACHP	auto-cascade heat pump
ACASHPS	auto-cascade air source heat pump system
COP_h	coefficient of performance of heating
VI	vapor injection

Symbols	
h	specific enthalpy (kJ/kg)
p	pressure (MPa)
m	mass flow rate (kg/s)
n	power consumption (kW)
q	heating capacity (kW)
x	vapor quality
η	efficiency
Subscript	
1,2,3 et al. the state points of the ACHPS with VI	
e	evaporator
c	condenser
s	isentropic

However, ASHP still has many problems.

Frost formation has a negative effect on the heating by the air source heat pump, such as the reduction of the operation time of the heat pump, the performance degradation, and the additional energy consumption for defrosting [3].

The ozone depletion potential (ODP) and global warming potential (GWP) of refrigerants used in ASHP are also another challenge to heat pumps. [4].

When the winter air temperature is below -10 degrees, compared to the ground source heat pump, the heat capacity and COP of the ASHP are much reduced and the discharge gas temperature of the compressor is higher.

Such characteristics under low ambient temperature limit the popularization of ASHP [5].

Therefore, a great deal of previous research into ASHP system has been carried out to address such problems

Therefore, there have been a great deal of previous research on the performance enhancement in the low temperature environment of ASHP system [6].

Bertsch et al. [7] simulated and tested an air-source two-stage heat pump using R410a as refrigerant to produce hot water of 50℃ at ambient temperature of -30 ℃. The system could run in single-stage and in two stage operation mode, and at the same ambient temperature, the heat capacity in the two-stage operation mode was almost twice that of the single-stage operation mode. Results showed that the COP of 2.1 and second law efficiency of 45% could be achieved and the compressor discharge temperature was stayed below 105℃ at all times.

Jin et al. [8] considered a two-stage heat pump with flash tank. Experimental results showed that the system was capable of providing 50 ℃ heating hot water at ambient temperature of -20℃, with COP of 1.76, and the discharge temperature of the high-stage compressor was below 100 ℃.

Wang et al. [9] proposed a double-stage coupled heat pumps combined with ASHP and water source heat pump to improve the heating performance of the heat pump system under the low temperature environment.

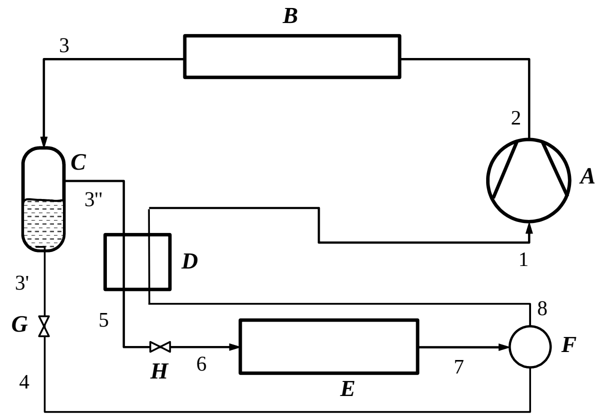
Field test results showed that the compression ratio of the double stage system decreased 50% and the energy efficiency ratio (EER) was improved by 130% compared to the single stage system when the ambient temperature and supply water temperature were −6.3 ℃ and about 43 ℃, respectively.

Wu et al. [10] experimentally investigated a cascaded SAHP water heater with phase change material under various operating conditions and obtained results that the COP ranged from 1.74 to 2.55 when the ambient temperature was -7 ℃.

Ma et al. [11, 12] proposed an improved heat pump system with a scroll-compressor economizer to increase the heating capacity in severe winter temperatures, and compared with a conventional ASHP system, the energy efficiency of the improved system increased 6.0% when the condensing temperature and evaporating temperature were 45 ℃ and −15 ℃, respectively.

There were many studies about auto cascade air source heat pump sytem (ACASHPS).

Li et al. [13] studied a thermodynamic analysis of an auto-cascade heat pump(ACHP) cycle for heating application in cold regions. The performance of the ACASHPS was analyzed for 15 binary mixtures and it was found that the maximum COP of 2.15 was obtained for the R134a/R600 (0.8/0.2) mixture at ambient temperature of -10 °C and heating temperature of 50 °C.



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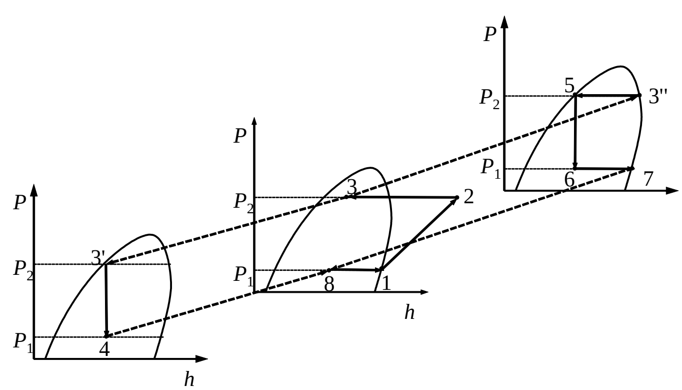

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fig1. Conventional auto-cascade heat pump cycle (a: systematic diagram, b: cycle in the p-h diagram, A-compressor, B-condenser, C-accumulator, D-evaporative condenser, E-evaporator, F-mixer, G, H-expansion valve)

Fan et al. [14] proposed an ejector enhanced internal auto-cascade heat pump cycle (EIHP) with zeotropic mixture R32/R290 for district heating in cold region. The simulation results showed that EIHP had 19-9% higher COP and 7-12% higher volumetric heating capacity than the conventional heat pump in the case of evaporator outlet temperature ranging from -25 to 5 ℃and condenser outlet temperature of 60 ℃.

E.A. et al. [15] proposed two systems using injectors to improve performance in an auto cascade chiller and analyzed the performance of using mixtures R600a and R1150.

Vapor injection (VI) has been a crucial technology to improve the performance of ASHP in low ambient temperature.

Wang et al. [16] established experimental bench to study the small EVI scroll compressor and investigated the influence of the place and size of economizer holes.

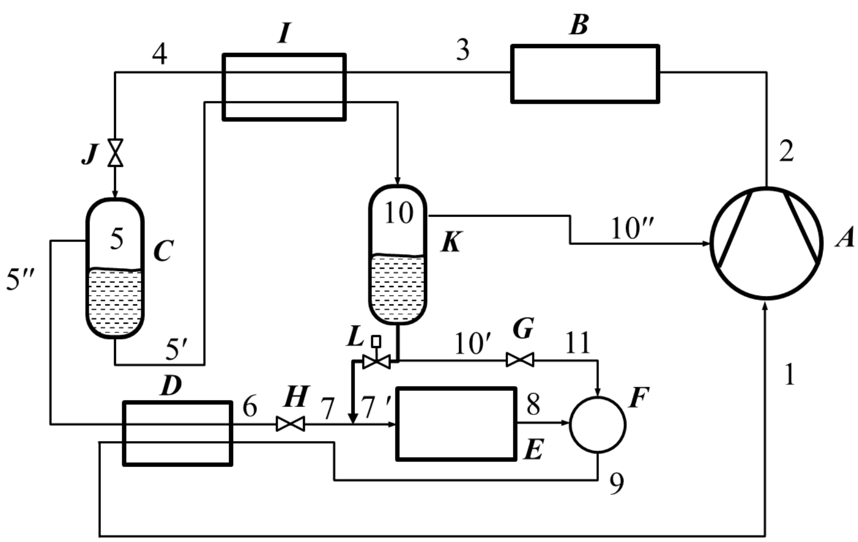
The ASHP system has almost similar COP compared to other types of heat pump systems at low ambient temperature.

However, overall, the COP is still not high compared to other types, and it is only superior to temperatures below -10℃

In this study, a system combining flash tanks is proposed and analyzed to improve the performance of an automatic cascade heat pump system.

2. Cycle description and modeling of Automatic cascaded air source heat pump with flash tank and economizer

2.1. Cycle description



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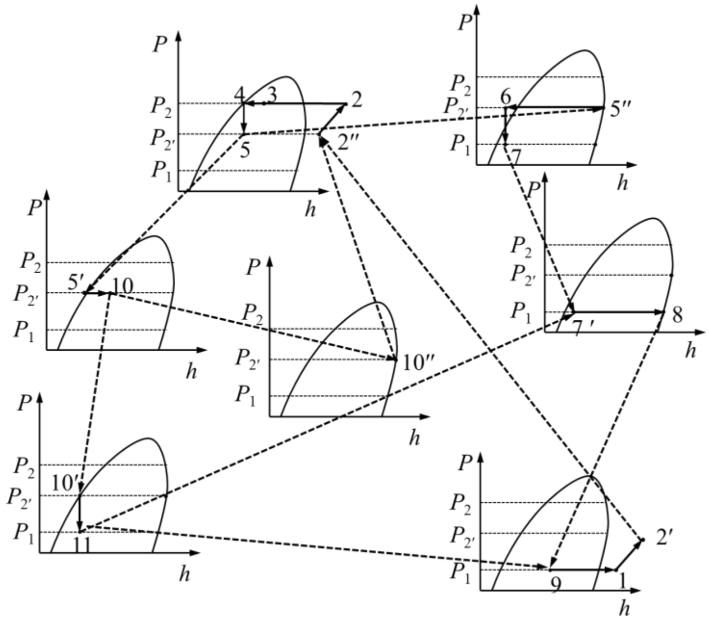

b)

fig2. Auto cascade air source heat pump system with flash tank and economizer (a: systematic diagram, b: cycle in the p-h diagram, A-compressor, B-condenser, C, K -flash tank, D, I-evaporative condenser, E-evaporator, F-mixer, G,H,J-expansion valve, L-electrical expansion valve)

The mixture gas discharged from the compressor (A) is cooled in the condenser (B) and partially condensed, heating the water.

The refrigerant from condenser (B) is cooled in evaporative condenser (I) to become saturated liquid.

This saturated liquid then passes through the expansion valve (J) and enters the flash tank (C).

The pressure of flash tank (C) is the intermediate pressure of evaporation and condensation pressure.

Here, the refrigerant is divided into gas and liquid components, which differ in the concentration of the mixture of the gas phase and the liquid phase.

The liquid component from the flash tank (C) is partially evaporated and enters the flash tank (K) by heat exchange with the refrigerant from the condenser through the evaporative condenser (I).

The gas component from the flash tank (K) is sucked into the compressor.

In the flash tank (C), the gas component condenses in the condenser evaporator (D) and passes through the expansion valve (H) and enters the evaporator (E).

In the evaporator (E), the liquid refrigerant evaporates and is mixed with the refrigerant coming from the flash tank (K) in the mixer (F), passing through the evaporative condenser (D) and sucked into the compressor.

According to the superheat of the refrigerant from the evaporative condenser (D), a part of the liquid refrigerant from the flash tank (K) is mixed with the refrigerant entering the evaporator through the solenoid valve L.

2.2. Modeling

The refrigerant gas sucked in the compressor is compressed to a primary intermediate pressure. The compressional work is

$$N_1 = m_1(h_{2'} - h_1) \quad (1)$$

The output enthalpy at the middle point of the compressor is

$$h_{2'} = h_1 + \frac{(h_{2's} - h_1)}{\eta_s} \quad (2)$$

At the mid-point of the compressor, it is mixed with the refrigerant vapor coming from the flash tank (k). The work during mixing is as follows.

$$N_2 = m_1 v_{2'} \left(p_{2''} - p_{2'} \right) \quad (3)$$

In the secondary compression process, the work of compression is

$$N_3 = m_2(h_2 - h_{2''}) \quad (4)$$

Here

$$m_2 = m_1 + \Delta m \quad (5)$$

$$h_{2''} = \frac{m_1 h_{2'} + \Delta m h_{11}}{(m_1 + \Delta m)} \quad (6)$$

$$h_2 = h_{2''} + \frac{\left(h_{2s} - h_{2''}\right)}{\eta_s} \quad (7)$$

The compressor isentropic efficiency η_s [17] is given as

$$\eta_s = 0.875 - 0.135 \frac{P_2}{P_1} \quad (8)$$

Thus, the power consumption in the compressor is

$$N = N_1 + N_2 + N_3 \quad (9)$$

The heat load of the condenser is :

$$Q_c = m_2 \left(h_2 - h_3 \right) \quad (10)$$

The heat balance equations in the evaporative condenser (I) is:

$$\left(1 - x_1\right) m_2 \left(h_{10} - h_{5'} \right) = m_2 \left(h_3 - h_4 \right) \quad (11)$$

where x_1 is gasification rate in the flash tank (C).

The enthalpy of the refrigerant entering the evaporator is

$$h_{7'} = \frac{m_3 h_{11} + x_1 m_2 h_7}{m_1} \quad (12)$$

Where m_3 is the refrigerant mass flow rate entering the evaporator from the flash tank (K).

The refrigerant quantity entering the mixer from flash tank (K) is

$$m_4 = (1 - x_1)(1 - x_2) m_2 - m_3 \quad (13)$$

Where x_2 is gasification rate in the flash tank (K).

Enthalpy in mixer (F) is as follows:

$$h_9 = \frac{m_4 h_{11} + \left(x_1 m_2 + m_3 \right) h_8}{m_1} \quad (14)$$

The heat balance equation in the evaporative condenser (D) is

$$x_1 m_2 \left(h_{5''} - h_6 \right) = m_1 \left(h_1 - h_9 \right) \quad (15)$$

The cooling load in the evaporator is:

$$Q_e = \left(x_1 m_2 + m_3 \right) \left(h_8 - h_{7'} \right) \quad (16)$$

The refrigerant mass flow rate (m_3) entering from flash tank (K) to evaporator (E) is adjusted according to the superheat of refrigerant at the compressor suction pipe.

3. Simulation results

3.1. Setting of operating condition

The refrigerant in the system is a mixture of R600 and R143a.

From ref [13], the COP was maximized for the mixture of R600 and R143a of 0.2/0.8 in the automatic cascade heat pump system.

In this study, the R600/R143a mixture refrigerant with a mixing ratio of 0.2/0.8 is used based on the previous study.

The heat pump is used in the process of producing hot water, hence the low temperature air in winter is used as heat source, and the evaporator is a plate finned tube heat exchanger.

The condenser is a plate heat exchanger.

The inlet and outlet temperature difference of air in the evaporator is set to 10 °C.

The temperature difference between the inlet water temperature and the outlet refrigerant in the condenser is set to 5 °C.

The opening of the electronic expansion valve (L) is adjusted so that the temperature difference at the condenser evaporator inlet is 5°C.

His refrigerant state entering the compressor from the tank (K) is considered as saturated gas, the refrigerant state at the evaporative condenser (I) outlet is considered as saturated liquid, and the refrigerant state at the evaporator (E) outlet is considered saturated gas.

In the calculations, the refrigerant state values are calculated using the refrigerant property program NIST Refprop9.1 [17].

The working conditions are set as shown in Table 1.

Table 1. Operating conditions of the automatic cascade heat pump system.

item	value	other
Ambient temperature (℃)	-25~5	
Hot water Temperature in condenser inlet (℃)	40	
Hot water temperature in condenser outlet (℃)	50	
Refrigerant	Mixed refrigerant of R600/R143a (0.2/0.8)	mixing ratio in condenser: 0.2/0.8

3.2. Performance with varying ambient temperature

The cycle characteristics of the improved automatic cascaded air source heat pump system under the above conditions were analyzed using Eqs. 1-16.

Table 2 shows the characteristics of the improved heat pump ambient temperature.

Table 2. The characteristics of the improved heat pump with ambient temperature.

Ambient temperature (℃)	-25	-20	-15	-10	-5	0	5
Condensation pressure (MPa)	1.524	1.524	1.524	1.524	1.524	1.524	1.524
condenser outlet refrigerant temperature (℃)	45	45	45	45	45	45	45
evaporation pressure (MPa)	0.149	0.184	0.226	0.275	0.331	0.396	0.470
Refrigerant pressure in flash tank E (MPa)	0.476	0.530	0.587	0.647	0.711	0.777	0.846
Gas-Phase Concentration of R143a in flash Tank E	0.91803	0.92	0.92	0.92	0.92	0.92	0.92
Liquid phase concentration of R143a in flash tank E	0.743	0.74	0.749	0.753	0.743	0.743	0.766
temperature of the refrigerant in flash tank E (℃)	0.6	3.9	7.1	10.3	13.4	16.5	19.4
vaporization rate of refrigerant in flash tank E	0.360	0.337	0.315	0.292	0.269	0.246	0.222
temperature of the refrigerant in flash tank K (℃)	5.3	8.6	11.9	15.2	18.5	21.6	24.7

vaporization rate of refrigerant in flash tank K	0.530	0.523	0.509	0.485	0.477	0.468	0.461
Gas-Phase Concentration of R143a in flash Tank K	0.872	0.873	0.873	0.873	0.873	0.873	0.873
Liquid phase concentration of R143a in flash tank K	0.596	0.609	0.61	0.61	0.61	0.61	0.61
compressor suction enthalpy ($kJ \cdot kg^{-1}$)	432.1	435	438.8	442.45	446.41	451	458.2
compressor discharge enthalpy ($kJ \cdot kg^{-1}$)	505.3	496.7	491.7	487.7	483.4	481.4	480.7
compressor discharge temperature (℃)	83.4	77.54	74.0	71.3	70.9	68.4	66.5
condenser outlet enthalpy ($kJ \cdot kg^{-1}$)	349.7	349.7	349.7	349.7	349.7	349.7	349.7

In the flash tank (C), the R143a concentration in the gas refrigerant was 91.8-92.0%, which did not change significantly with ambient temperature, and was maintained at a fairly high concentration.

However, the gasification rate varied greatly from 36.1 to 22.2%.

In the flash tank (K), the concentration of refrigerant R143a in the gas refrigerant was about 87.3%, which was almost constant regardless of the ambient temperature, and the gasification rate was 59.6-61.0%, which did not change significantly.

The compressor discharge gas temperature was 66.5-83.4 ℃, which was very stable.

3.3. Comparison of conventional ACASHPs

The performance of the improved ACASHP was analyzed in comparison with other systems.

To consider the performance of the improved system, a comparative analysis with the system presented in fig.1 was carried out.

The concentrations of R143a refrigerant entering the evaporator at conventional ACASHP and improved ACASHP under the conditions presented above are shown in Table. 3.

Table 3. Concentration of R143a in the evaporator at conventional ACASHP and improved ACASHP

Type of system	Conventional ACASHP	Improved ACASHP
Concentration of R143a in the refrigerant entering the evaporator (%)	88.0	91.8~92.0

As can be seen, the concentration of R143a in the improved system was about 3.8~4% higher.

Table 4 shows the variation of conventional ratio with ambient temperature.

Table 4. Compression ratio in standard ACASHP and improved ACASHP with ambient temperature.

Ambient temperature (℃)	Conventional ACASHP	ACASHP with flash tank	reduction rate (%)
-25	12.34	10.23	17.09
-20	9.87	8.24	16.51
-15	7.99	6.72	15.84
-10	6.53	5.53	15.20
-5	5.38	10.23	14.61
0	4.47	8.24	14.04
5	3.75	6.72	13.51

As shown in Table. 4, the compression ratio of the improved system is 13.5-17.0% lower.

This is because the concentration of R143a in the mixture entering the evaporator is higher in the improved system.

Table. 5 shows the variation of COP with ambient temperature for conventional ACASHP and improved ACASHP.

Table 5. COP in convention ACASHP and improved ACASHP with the ambient temperature.

Ambient temperature (℃)	-25	-20	-15	-10	-5	0	5
COP of Conventional ACASHP	1.87	2.01	2.16	2.32	2.51	2.73	3.01
COP of improved ACASHP	1.99	2.16	2.40	2.65	2.99	3.35	3.92
rate of increase (%)	6.20	7.38	11.00	13.87	18.91	22.58	30.37

As can be seen, the COP of the improved system is 6.2-30.3% higher.

This is due to the reduction in compression ratio, the increase in heat production by the vapor injection and the decrease in power consumption in the improved system.

Table. 6 shows the variation of discharge temperature for both systems with ambient temperature.

Table 6. Exhaust temperature at conventional ACASHP and improved ACASHP with varying ambient temperature.

Ambient temperature (℃)	-25	-20	-15	-10	-5	0	5
Compressor discharge temperature of Conventional ACASHP (℃)	132.1	119.2	108.2	99.1	90.9	83.7	77.1
Compressor discharge temperature of improved ACASHP (℃)	83.4	77.5	74.0	71.3	70.9	68.4	66.5

deviation, (℃)	48.7	41.8	34.2	27.8	20.0	15.3	10.6

As shown in Table 7, the compressor discharge temperature in the improved system is 48.7-10.5 ℃ lower.

This is due to the lower compression ratio and the cooling effect by vapor injection into the compressor in the improved system

3.3. Comparison of different conventional air source heat pump systems

The performance of the improved ACASHP system was compared with that of conventional ASHP systems.

Heat pump systems using refrigerants R22, R410A and R404A were selected as the heat pump systems for comparison.

Table. 7 shows the characteristic changes in ambient temperature -25~5 ℃ for conventional ASHPs and improved ACASHPs.

Table.7. Variation of characteristics with ambient temperature in air source heat pump system using R22, R410A and R404A and in improved ACASHPs

Item	Ambient Temperature (℃) / Type of system	-25	-20	-15	-10	-5	0	5
Compression ratio	Improved ACASHP	10.23	8.24	6.72	5.53	4.59	3.84	3.24
	ASHP with using R22	20.72	16.52	13.30	10.88	8.88	7.35	6.14
	ASHP with using R410A	19.46	15.60	12.66	10.37	8.56	7.13	5.98
	ASHP with using R404A	19.20	15.40	12.49	10.22	8.44	7.27	6.09
COP	Improved ACASHP	1.99	2.16	2.40	2.65	2.99	3.35	3.92
	ASHP with using R22	1.95	2.14	2.38	2.62	2.85	3.11	3.38
	Increase rate, %	2.10	0.91	0.61	1.16	4.76	7.84	16.11
	ASHP with using R410A	1.90	2.09	2.29	2.49	2.69	2.88	3.14
	Increase rate, %	4.53	3.54	4.73	6.31	11.18	16.69	24.69
	ASHP with using R404	1.82	1.99	2.20	2.38	2.58	2.81	3.05
	Increase rate, %	9.56	8.68	8.77	11.15	15.72	19.41	28.31
Compressor discharge temperature (℃)	Improved ACASHP	83.4	77.5	74.0	71.3	70.9	68.4	66.5
	ASHP with using R22	153.0	132.9	118.8	108.8	100.4	94.8	89.6
	ASHP with using R410A	129.7	115.0	104.8	97.3	91.5	87.1	82.9
	ASHP with using R404A	104.0	95.1	88.5	84.1	81.0	78.8	76.8

As shown in the table.7, the improved system has a compression ratio of about 50% lower than conventional heat pump systems.

Moreover, the discharge gas temperature is much lower than other systems, which can improve the compressor operating conditions significantly.

The COP of improved ACASHP is almost similar in the range of -25 ~ -5℃ of ambient temperature compared to the R22 system, and is higher than 7 ~16% in the range of 0 ~ 5℃.

And then the COP of improved ACASHP is significantly higher than the R410A system, which is 3.5~6.3% higher in the range of -25 ~ -10℃ of ambient temperature and 10~24% higher in the range of -10~5℃

In particular, the improved system shows superior performance compared to the R404A system.

The COP increased by 8.6-28.3% in -25~5℃ of ambient temperature

4. Conclusions

In this paper, an improved automatic cascaded air source heat pump system is presented to improve the performance of air source heat pump in low temperature environment, and a comparative analysis is carried out with the conventional automatic cascaded air source heat pump system and the conventional air source heat pump systems using R22, R410A and R404A.

The following conclusions were obtained from the analysis.

(1) The improved automatic cascaded air source heat pump system has a higher COP compared to the conventional automatic cascaded air source heat pump system, and the discharge gas temperature is very low, which can ensure the stability of working.

The improved system has 13.5~17.0% lower compression ratio, 6.2~30.3% higher COP and 48.7~10.5 ℃ lower discharge temperature compared to conventional ACASHP.

(2) The improved automatic cascaded air source heat pump system has higher operational stability and improved performance at low ambient temperature compared to the conventional air source heat pump systems with the economizer.

Compared with heat pump systems using R22, the COP is almost similar at -25 ~-5℃ and higher than 7 ~ 16% at 0 ~ 5℃.

Moreover, the discharge gas temperature at R22 is very high, 132-153 ℃ at -25~- 20 ℃, which has a detrimental effect on compressor life and operation, but the improved ACASHP system can provide safe operation with discharge gas temperature below 85 ℃ even at -25 ℃.

The COP is significantly higher than the R410A system, with 3.5~6.3% higher in -25 ~-10℃, 10~24% higher in 10~5℃, and the discharge gas temperature is much lower.

The improved ACASHPs shows superior performance compared to the system using R404A.

The COP increased by 8.6~28.3% in -25~5℃.

As a result, the improved ACASHP system was found to be safe to operate and high COP compared to other heat pump systems when the ambient temperature was below -10℃.

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Declaration of conflicting interests

The authors declare that they have no conflicts of interest.

Data availability statement

All data that support the findings of this study are included within this article.

Code availability

The code that supports the findings of this study is available from the corresponding author [Won-Chol Yang], upon a reasonable request.

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