Effects of Economizer Type and Size to the Gas Injected System Using Zeotropic R32/R1234ze(E) Mixture

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ABSTRACT

Refrigerant mixtures are commonly regarded as a potential substitute because rare pure refrigerants can totally meet all the requirements of environmental, thermodynamic and safe properties. Meanwhile, as an effective means, gas injected system is widely applied to improve the system performance of heat pump under large compression ratio. However, the effects of the economizer's configuration on the performance is still unclear. In this paper, a water-water heat pump with scroll compressor and charged with R32/R1234ze(E) mixtures is chosen to carry out the investigation. A thermodynamic model which could handle different suction and injected concentration refrigerant is derived. On the basis of this model, a detailed comparison of different gas injected configurations and the size of intermediate heat exchanger (IHX) in IHX system are investigated. The results indicate that both flash tank (FT) gas injection and IHX injection can largely enhance the heating capacity and COP of water-water heat pump using R32/R1234ze(E) mixture. However, IHX system with large IHX is a better choice for zeotropic R32/R1234ze(E) gas injected system.

Keywords: Heat pump; Zeotropic refrigerant; Compressor; Gas injection; Air conditioner

1. INTRODUCTION

With increasing environmental protection requirements, chlorofluorocarbons (CFCs) have been phased out and hydrochlorofluorocarbons (HCFCs) are being alternated for their high ozone depletion potential (ODP). At the same time, some hydrofluorocarbons (HFCs) also will be phased down due to their high global warming potential (GWP). Accordingly, finding the appropriate alternative refrigerant becomes an essential work for refrigeration and heat pump fields. While according to the recent research (Mclinden *et al.*, 2017; Radermacher and Hwang, 2005; Mohanraj *et al.*, 2011), rare pure refrigerants can meet the environmental, thermodynamic, and safety requirements to new-generation refrigerants completely, and the mixed refrigerants provide more flexibility in searching for new alternatives. Among recent studies (Pham and Rajendram, 2012; In *et al.*, 2014) HFC/HFO mixtures have attracted great attention due to their excellent properties, such as R32/R1234ze(E) in the field of air conditioning and heat pumps. Kojima *et al.* (2015) tested a water heat pump with R410A and R32/R1234ze(E) mixtures. They found that when the heat sink water is heated by 10 K, the COP of the mixtures was significantly lower than R410A, but when the heat sink water temperature is increased by 25 K, the COP of the mixtures was generally comparable to that of R410A. Taira *et al.* (2016) evaluated the performance of an air source heat pump with R32/R1234ze(E) (70/30) mixtures, R410A and R32. It was concluded that the COP of R32 was the highest among three refrigerants for the same capacity. Cheng *et al.* (2017) investigated an air source heat pump with R32/R1234ze(E) mixtures. It is concluded that the heating capacity decreased by 67.2% while the COP continuously increased by 70.3% when the mass fraction of R1234ze(E) changes from 0% to 100%. To sum up, this pair of refrigerant mixtures has been widely recognized with its great environmental and thermodynamic potential, and has a good application prospects in refrigeration and heat pump fields.

At the same time, the key issue in the heat pump system that has been focused but still needs to be addressed is the poor performance under large pressure ratio. Gas injected system is widely considered as a good solution to this problem (Xu *et al.*, 2011) because it can increase the heating capacity and improve the system energy efficiency at low ambient temperature. Ma *et al.* (2003) analysed the performance of improved gas injected system under different condensing and evaporating temperature, it was founded that this technology can help system run smoothly for a long time when the ambient temperature was near -15°C and exhibited larger COP than conventional heat pump system. Heo *et al.* (2009) studied the effects of flash tank vapor injection on the heating performance of a two-stage heat pump at ambient temperatures of -15, -5, and 5 ℃. The COP and heating capacity of the injection cycle were enhanced by 10% and 25%, respectively, at the ambient temperature of -15 ℃. The total mass flow rate of the injection cycle was 30–38% higher than that of the non-injection cycle.

There are some researchers concentrated in the gas injected system with refrigerant mixtures. Högberg and Berntsson (1994) investigated the difference between zeotropic mixtures and pure fluids in two-stage cycles by simulation. R22/R152a, R22/R142b, R22/R114 were selected as the zeotropic mixtures. The main results showed that the COP and capacity of zeotropic mixtures were more largely increased than those of pure fluids by adopting the two-stage cycle with intermediate heat exchanger. Jung *et al.* (1999) simulated the multi-stage heat pumps with a heat exchanger economizer. R22/R142b/R134a, R32/R134a, R125/R134a mixtures were studied. The results indicated that the three-stage heat pump is up to 27.3% more energy efficient than the conventional single-stage with pure refrigerant. D'Angelo *et al.* (2016) presented a performance evaluation of a gas injected refrigeration system using R290/R600a. The results showed that the maximum COP was obtained for a mixture containing 40 wt% R290 and the COP was enhanced 16–32% by gas injection.

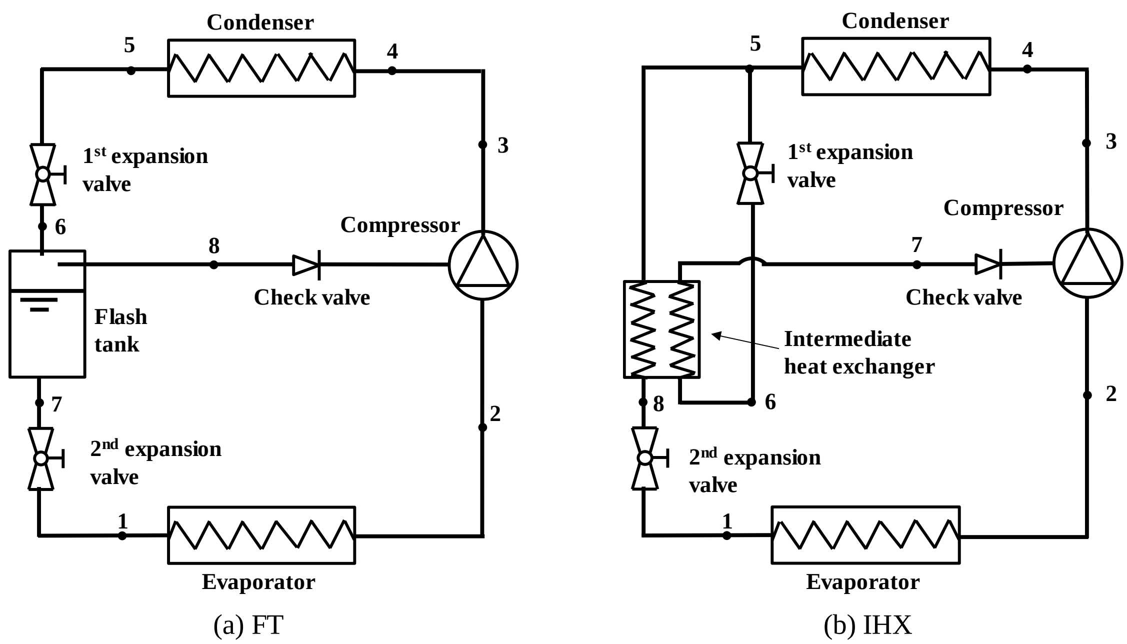

Fig. 1 Schematic diagram gas injection systems

Though there are many researches focusing on the gas injected system, few of these researches uses refrigerant mixtures, which may limit the further application of gas injected system, especially under the context of refrigerant substitution in which refrigerant mixtures may form the trend. Gas injected system with refrigerant mixtures is different from that with pure refrigerant, especially for FT system, the suction fraction varies from the injection fraction, which brings the difference between FT system and IHX system. The schematic diagram this two systems is presented in Fig .1, to illustrate the composition change in FT system, the temperature-fraction diagram of FT system is shown in Fig .2. It can be observed that all of suction fraction, injected fraction and charged fraction are different, while in IHX system, the circulation fraction keeps as the charged fraction all the time. Though as a commonly used and energy efficient method, gas injected system with refrigerant mixtures has not been researched adequately, it is not clear whether some conclusions for pure refrigerant can be used in gas injected system for refrigerant mixtures and which configuration is better.

From above research, it is found that most of the research are carried out without mention about the different fraction at suction port and injected port, or conduct the numerical analysis based on compressor efficiency model which only consider the compression ratio regardless of the thermodynamic properties of refrigerant. At the same time, which configuration is better and the improved potential are both not clear. In this paper, R32/R1234ze(E) mixtures are chosen as refrigerant pairs for its great potential in the future, the scroll compressor is chosen to carry out the research for its wide application in gas injected system. The thermodynamic model with different suction and injected concentration is derived, on the basis of this model, a detailed system performance comparison of different gas injected configurations with R32/R1234ze(E) mixtures is investigated.

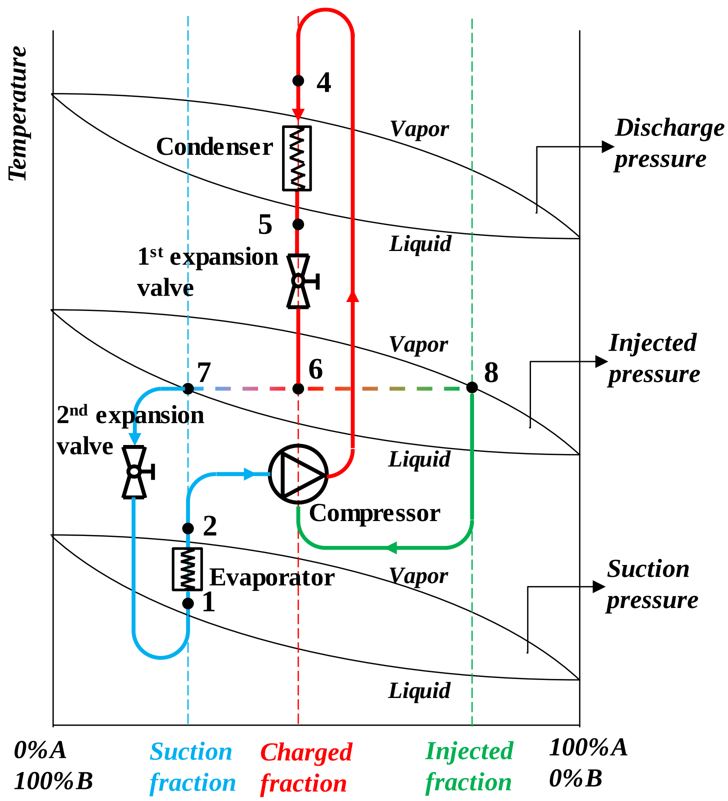

Fig. 2 Temperature-fraction diagram of FT gas injection system with zeotropic refrigerant

2. MODELING

A gas injected heat pump with FT and a gas injected heat pump with IHX are modelled in this section. To make the results comparable, the parameters of the condenser, evaporator and compressor are kept same. In order to simplify the simulation properly, the following assumptions are made:

- The superheated degree of at the evaporator outlet is fixed as 5K and the subcooling degree at the condenser outlet is fixed as 3K;
- In the IHX system, the vapor at the injection outlet is saturated.
- The difference between the charged composition and circulation composition in condenser is neglected.

2.1 Compressor

On the one hand, the traditional lumped parameter model could not be used in the simulation of the injected scroll compressor, because there is continuous refrigerant inflow or outflow in the compression process and the refrigerant states in the compression pocket keeps changing during the whole injection process. On the other hand, for gas injected scroll compressor with FT as illustrated in abovementioned analysis, the fraction from suction port and injection port is different, which varies from traditional injected scroll compressor model. The distributed parameter scroll compressor model is applied and thermodynamic model is derived for refrigerant mixtures in this section.

In the previous study, Wang *et al.* (2005) has proposed a general geometrical model including scroll wraps, working chambers volume and leakage areas, which can be applied in simulation of scroll compressor with discretional initial angle and present the variation of the volume in a general subsection function style and provided the precise mathematical formula for four kinds' leakage areas. This geometrical model is applied in this research.

Configuration parameters of the system used in simulations are shown in Table 1.

Table 1: Scroll compressor configuration parameters

Base circle radius (mm)	Thickness of scroll (mm)	Height of scroll (mm)	Cycles of scroll	Injection port position (°)	Injection port area (mm^2)
2.70	3.68	35.7	2.79	0	50.27

In every chamber of scroll compressor, the temperature, pressure, and mass change is with respect to orbiting angle θ. The thermodynamic process of refrigerant in every chamber can be regarded as an unstable flow process in an open control volume. The governing differential equations have been derived and widely applied in the compressor simulation (Chen *et al.*, 2002; Wang *et al.*, 2008). These equations are both derived based on pure refrigerant system, in which any state parameters are determined by another two parameters, but for gas injected system with flash tank, the refrigerant from injection port and the leakage refrigerant from adjacent chamber will both change the refrigerant fraction in governing chamber. So for this system, any state parameters are determined by *C*+1 parameters for a single phase system, where *C* is the total number of components. In this section based on the first law of thermodynamics, material balance and equation of state, the compressor thermodynamic model of refrigerant mixtures is derived. Considering the change of orbiting angle θ, the relation between temperature and orbiting angle can be written as follows:

$$\frac{dT}{d\theta} = \frac{\sum \frac{dN_{in}}{d\theta} h_{in} - \sum \frac{dN_{out}}{d\theta} h - T\left(\frac{\partial p}{\partial T}\right)_{V,N_i} \frac{dV}{d\theta} - T\sum_i \left(\frac{\partial S}{\partial N_i}\right)_{T,V,N_{j,j\neq i}} \frac{dN_i}{d\theta} - \sum_i \mu_i \frac{dN_i}{d\theta} + \frac{dQ}{d\theta}}{Nc_v} \tag{1}$$

The material balance for the control volume yields:

$$\frac{dN_i}{d\theta} = \sum \frac{dN_{in}}{d\theta} w_{i,in} - \sum \frac{dN_{out}}{d\theta} w_{i,out} \tag{2}$$

The relation between fraction and orbiting angle yields:

$$\frac{dw_i}{d\theta} = \frac{Nw_i + \sum dN_{in} w_{i,in} - \sum dN_{out} w_{i,out}}{\left(N + \sum dN_{in} - \sum dN_{out}\right) d\theta} \tag{3}$$

2.2 Heat Exchanger

Table 2 shows the specific configuration and size parameters of the evaporator and condenser.

Table 2: Configuration parameters of heat exchangers

	Length (mm)	Inside diameter of outer tube (mm)	Inside diameter of outer tube (mm)	Inside diameter of inner tube (mm)	Number of inner tube
Evaporator	6200	37	9.52	8.72	6
Condenser	6200	42.6	9.52	8.72	10

A distributed parameter model of double pipe heat exchanger is adopted to simulate the heat exchange process accurately. In the calculation for each section, local properties such as the compositions, specific heat, coefficient of viscosity, and heat conductivity coefficient are adopted, and on basis of these properties the overall heat transfer coefficient in every section is calculated. Moreover, the heating transfer fluids in evaporator side is a mixture of 40: 60% (by volume) of ethylene glycol and water.

In order to put emphasis on the effects of economizer's configuration, plate type heat exchanger is utilized as the IHX. Moreover, the size of IHX is also our concentrate, the heat exchange area of 0.2 m^2 and 0.54 m^2 are both involved in this study. In the later analysis, 0.2 m^2 and 0.54 m^2 represent smaller and larger IHX respectively.

3. RESULTS AND DISCUSSION

Heating capacity and COP is the evaluation index in this research. The heating capacity is the product of the refrigerant mass flow rate and enthalpy difference at the condenser, and is defined as follows:

$$Q_{con} = m_r \cdot \left(h_{con,out} - h_{con,in}\right) \tag{4}$$

The COP of the heat pump can be defined as follows, where W_{com} is the power consumption of compressor:

$$COP = \frac{Q_{con}}{W_{com}} \tag{5}$$

The different system performance at different composition of refrigerant mixtures is taken into account at one fixed working condition in this section. The working conditions are presented in Table 3. System performances are simulated when the molar fraction of R32 changes from 0 to 100% in increments of 10%.

Table 3: Working conditions

Inlet temperature of water in condenser (℃)	Inlet temperature of HTF in evaporator (℃)	Inlet mass flowrate of water in condenser (kg/s)	Inlet mass flowrate of HTF in evaporator (kg/s)
40	0	0.55	0.7

3.1 Heating Capacity

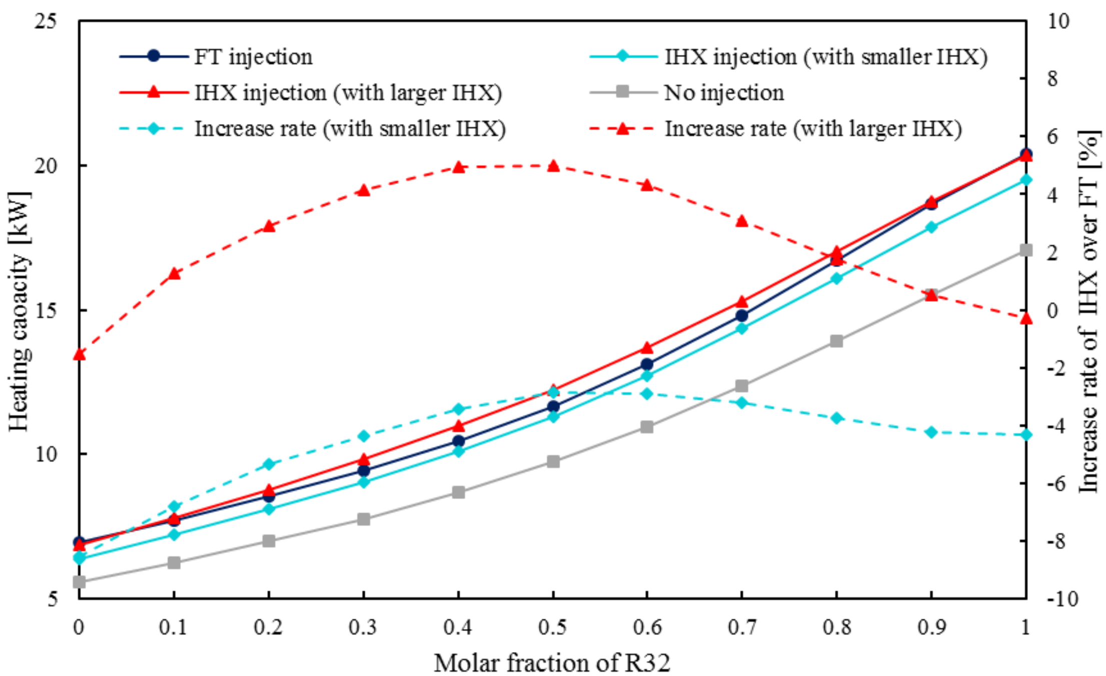

Figure 3: Heating capacity of FT system, IHX system and no injection systems

The heating capacities of FT system, IHX system and no injection system at various refrigerant fractions are shown in Fig. 3. As the Fig. 3 shows, the heating capacities of all systems increase with enriching R32 in the refrigerant. It also can be found that compared with no injection system, the heating capacities of both of the FT system and IHX system are larger. Compared with FT gas injection system, the heating capacity of IHX injection system with small IHX is smaller, as presented with dotted line, the increase rate is -8.5%~-2.4%. Meanwhile, it can be seen that the change in increase rate is not very regular, which is caused by different matching relationship of refrigerant temperature glide and water temperature shift at different molar fraction of R32.

Oppositely, when using a larger IHX, the heating capacity of IHX injection system is almost same as the one of FT injection system especially when the system operates with pure R32 or R1234ze(E), which is expectable for IHX system with quite large IHX and saturated injection gas. On the other side, with mixtures charged to the system, the heating capacity of IHX is higher than that of FT system. The IHX injection appears 5.0% increase in heating capacity over the FT system when the molar fraction of R32 is close to 0.4. That can be attributed to the richer low pressure component in the evaporator due to component shifting in the flash tank.

Compared with no injection system, the FT injection and IHX injection with larger IHX enhance the heating capacity 19.5~25.4% and 19.2~26.6%, respectively.

3.2 Power Consumption

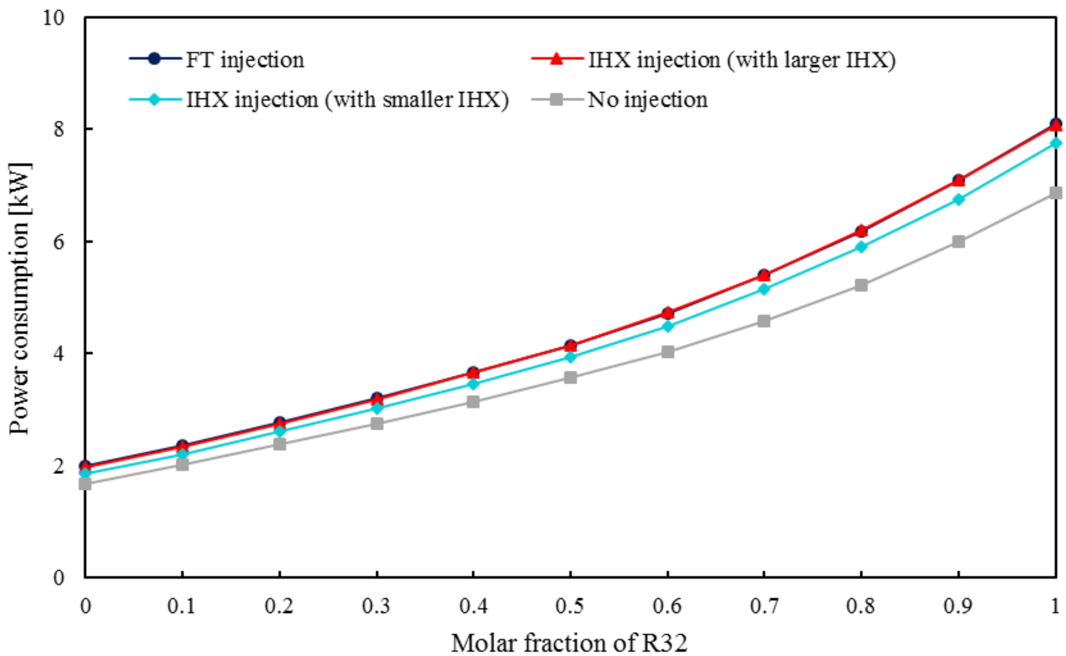

Figure 4: Power consumption of FT system, IHX system and no injection systems

The power consumption of FT system, IHX system and no injection system are presented in Fig. 4. Comparing the power consumption of FT system with IHX system with smaller IHX, it can be found that at each charged concentration, the power consumption of FT system is higher, which is caused by the higher injected pressure and the lower evaporating pressure of FT gas injection system. Nevertheless, when utilizing larger IHX, the injected pressure will increase, which causes more refrigerants injected to IHX system. So, in the IHX system with larger IHX, the power consumption will increase, the power consumption of FT injection system keeps close as IHX system in this situation.

3.3 COP

As shown in Fig. 5, the COP of all systems decrease with enriching R32 in the refrigerant. Under the combined effects of heating capacity and power consumption, the COP of IHX system is higher than FT system. Actually, the bad performance of the FT system can be attributed to the component shift in the flash tank. Meanwhile, it can be clearly seen that when utilizing larger IHX, the COP of IHX system is higher, which is caused by more obvious improvement of heating capacity with larger IHX. For larger IHX system, IHX system performs 5.3% higher COP than that of FT system as the most obvious improvement when charged with R32/R1234ze(E) (40/60) mixtures. For smaller IHX system, IHX system performs 2.5% higher COP than that of FT system as the most obvious improvement when charged with R32/R1234ze(E) (50/50) mixtures. Compared with no injection system, the FT injection and IHX injection with larger IHX enhance the COP 1.2~5.3% and 1.4~9.3%, respectively.

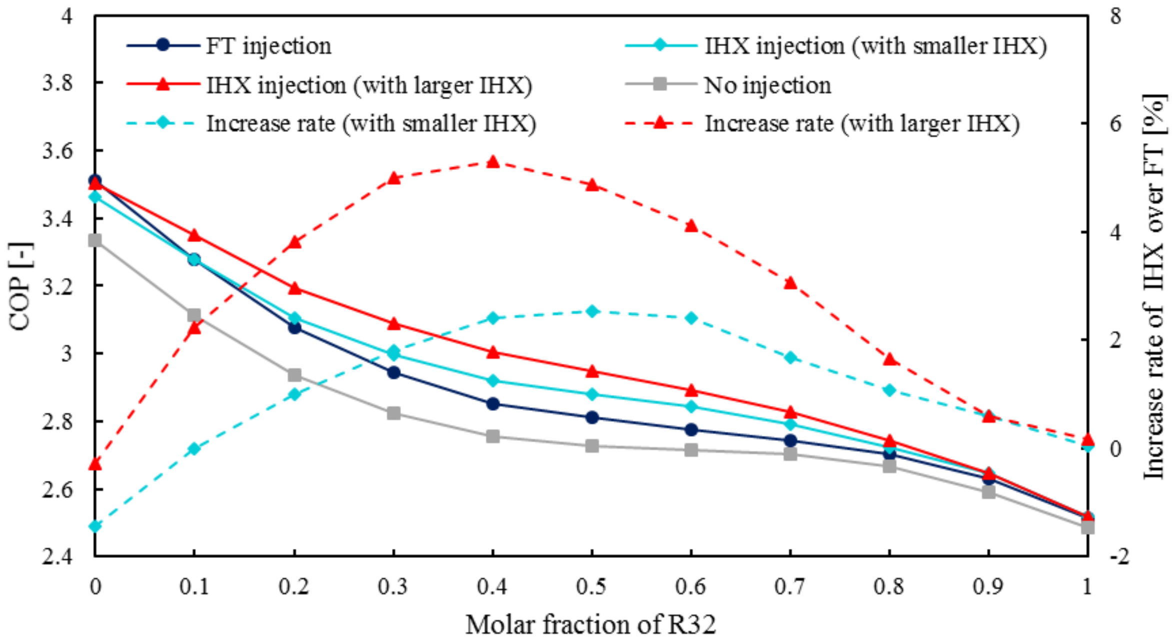

Figure 4: Power consumption of FT system, IHX system and no injection systems

4. CONCLUSIONS

Based on a comprehensive model of gas-injected water-water heat pump with R32/R1234ze(E), the performance of the gas injected heat pump with FT economizer and IHX economizer are comparatively researched. Meanwhile the effect of the IHX size is also involved. The main research conclusions are as follows:

- Both FT gas injection and IHX injection can largely enhance the heating capacity and COP of water-water heat pump using R32/R1234ze(E) mixture, the FT injection enhances the heating capacity by 19.5~25.4% and COP by 1.2~5.3% over the without injection system. The IHX injection with larger IHX increases the heating capacity by 19.2~26.6% and COP by 1.4~9.3% over the without injection system;
- The heating capacity of IHX system with larger IHX is higher than FT system. For COP, the IHX system is higher than FT system.
- When using gas injected system with zeotropic refrigerants, IHX gas injection configuration is a better choice.

NOMENCLATURE

COP	coefficient of performance	(–)
CFCs	chlorofluorocarbons	(–)
FT	flash tank	(–)
GWP	global warming potential	(–)

HCFCs	hydrochlorofluorocarbons	(–)
HFCs	hydrofluorocarbons	(–)
IHX	intermediate heat exchanger	(–)
C	total number of components	(–)
h	specific enthalpy	(kJ mol^{-1})
N	the number of moles	(mol)
p	pressure	kPa)
Q	heating capacity	(kW)
S	entropy	(kJ K^{-1})
T	temperature	(K or ℃)
V	volume	(m^3)
w	molar fraction	(–)
W	power consumption	(kW)
μ	chemical potential	(kJ mol^{-1})
θ	orbiting angle	(–)

Subscript

con	condenser
in	inlet
i,j	serial number of components, , j=1, 2, 3…C
out	outlet
r	refrigerant

REFERENCES

Chen, Y., Halm, N.P., Groll, E.A., Braun, J.E.(2002). Mathematical modeling of scroll compressors - Part I: Compression process modeling. *Int. J. Refrig*. 25, 731–750.

Cheng, Z., Wang, B., Shi, W., Li, X. (2017). Numerical Research on R32/R1234ze(E) Air Source Heat Pump under Variable Mass Concentration. *Int. J. Refrig*. 82, 1–10.

D'Angelo, J.V.H., Aute, V., Radermacher, R. (2016). Performance evaluation of a vapor injection refrigeration system using mixture refrigerant R290/R600a. *Int. J. Refrig*. 65, 194–208.

Heo, J., Jeong, M.W., Kim, Y. (2010). Effects of flash tank vapor injection on the heating performance of an inverter-driven heat pump for cold regions. *Int. J. Refrig*. 33, 848–855.

Höbberg, M., Berntsson, T. (1994). Non-azeotropic mixtures as working fluids in two-stage economizer-type heat pumps. *Int. J. Refrig*. 17, 417–429.

In, S., Cho, K., Lim, B., Kim, H., Youn, B. (2014). Performance test of residential heat pump after partial optimization using low GWP refrigerants. *Appl. Therm. Eng*. 72(2), 315–322.

Jung, D., Kim, H.J., Kim, O. (1999). Study on the performance of multi-stage heat pumps using mixtures. *Int. J. Refrig*. 22, 402–413.

Kojima, H., Fukuda, S., Kondou, C., Takata, N., Koyama, S.(2015). Comparative assessment of heat pump cycle operated with R32/R1234ze(E) and R32/R1234yf mixtures. In: *Proceedings of the 24th International Congress of Refrigeration, 16–22 August, Yokohama, Japan*.

Ma, G., Chai, Q., Jiang, Y. (2003). Experimental investigation of air-source heat pump for cold regions. *Int. J. Refrig*. 26, 12–18.

McLinden, M.O., Steven Brown, J., Brignoli, R., Kazakov, A.F., Domanski, P.A. (2017). Limited options for low-global-warming-potential refrigerants. *Nat. Commun*. 8, 14476.

Mohanraj, M., Muraleedharan, C., Jayaraj, S. (2011). A review on recent developments in new refrigerant mixtures for vapour compression-based refrigeration, air conditioning and heat pump units. *Int. J. Energy Res*. 35, 647–669.

Pham, H., Rajendran, R. (2012). R32 and HFOs as Low-GWP Refrigerants for Air Conditioning. In: *13rd International Refrigeration and Air Conditioning Conference, Purdue University, West Lafayette, Indiana, USA*.

Radermacher, R., Hwang, Y. (2005). Vapor compression heat pumps with refrigerant mixtures. CRC Press.

Taira, S., Haikawa, T., Minamida, T. (2016). Evaluation of performance of heat pump using R32 and HFO-mixed refrigerant by refrigeration cycle simulation and loss analysis. In: *15th International Refrigeration and Air Conditioning Conference, Purdue University, West Lafayette, Indiana, USA*.

Wang, B., Li, X., Shi, W. (2005). A general geometrical model of scroll compressors based on discretional initial angles of involute. *Int. J. Refrig*. 28, 958–966.

Wang, B., Shi, W., Li, X., Yan, Q. (2008). Numerical research on the scroll compressor with refrigeration injection. *Appl. Therm. Eng*. 28, 440–449.

Xu, X., Hwang, Y., Radermacher, R. (2011). Refrigerant injection for heat pumping/air conditioning systems: Literature review and challenges discussions. *Int. J. Refrig*. 34(2): 402-415.

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