

Development of steam generation heat pump through refrigerant replacement approach

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Abstract

In this study, design of high temperature heat pump to produce low pressure steam for industrial processes was conducted. For efficiency and reliability refrigerant of high temperature heat pump should have critical point above steam generation condition. Among commercial refrigerant satisfying this, R245fa was selected. For the generation of low pressure steam, pressure reduction method was applied. R245fa and pressure reduction methods made the high temperature heat pump cycle similar to that of hot water of R134a. Therefore refrigerant change approach from R134a to R245fa was selected for the preliminary research. Under same suction superheat R245fa tends to have smaller discharge superheat than R134a. To secure superheat without increasing temperature difference in evaporator, internal heat exchanger was applied to R245fa heat pump cycle. Sensitivity analysis to suction superheat was performed for R245fa and R134a. The results showed that R245fa has better performance than R134a. This means R245fa has more advantages in applying internal heat exchanger compared to R134a.

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1. Introduction

Emission of greenhouse gases is largely related to energy consumption and energy efficiency is expected to play an important role next to renewable energy to deal with global warming issues.^[1] Among residential, commercial, industrial and transport sectors, the industrial sector generally contributes the largest portion in energy consumption. For South Korea the industrial sector consumed about 61.7% of final energy in 2012.[2] Energy consumption by functions in manufacturing showed that feedstock accounted for the largest portion of 57.2% followed by heat energy of 29.1%.[3] (Table 1) Therefore it is an important research target to find methods of reducing heat energy in industrial sector. Conventional heating methods like combustion and electric heating is considered to reach limit in efficiency. Alternative heating methods are being sought to improve thermal efficiency. Among them heat pumps are considered as one of the most prominent candidates. Heat pump systems satisfy heat demand by transferring energy from low temperature source to high temperature sink. Energy is used in changing pressure and temperature of working fluid – refrigerant in heat pumps. The total transferred heat is the sum of energy input and extracted energy from low temperature source. Therefore heat pump can deliver more heat than input energy.

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Table 1. Energy consumption in manufacturing industry by functions in 2010, South Korea

Function	PJ (10 ¹⁵ J)	Portion (%)
Heat	1,312	29.1
Power	398	8.8
Feedstock	2,574	57.2
Miscellaneous	216	4.8
Total	4,501	100

Heat pump can be applied to use low temperature heat source to generate hot temperature energy such as space heating and hot water. As a low temperature heat source, heat pumps can use air, river, lake/sea water or even ground (underground) energies.^[4] This versatility of energy source has made heat pump to be considered as a renewable energy system. In EU, heat pump satisfying minimum SPF (Seasonal Performance Factor) is regarded as renewable energy system.^[5] Some countries have implemented subsidy programs to expand installation of heat pump since it can emit less greenhouse gases than conventional boilers using fossil fuels.

Currently main market of heat pump is residential, commercial and district heating/cooling market. The penetration status of heat pump in the industrial sector is very low. In one milk powder factory in Denmark, a compression/absorption heat pump was installed to preheat process air up to 80°C by extracting heat from coolant of 40°C.^[6] In England one chocolate factory installed a heat pump system to simultaneously heat the process water and cool glycol mixture.^[7] Most of heat pump application in industrial sector are in production of hot water for processes. For a higher temperature application, Costa et al. proposed the application of absorption heat pump type 2 to reduce middle pressure steam in pulp factory and showed high potential in energy saving.^[8]

Table 2 is the temperature distribution of heat demand in German in C. Lauterbach's report.^[9] According to the report, heat demand of 100°C or less occupies 9% and that of 100-200°C was 8% of similar contribution. Temperature distribution of heat demand however showed different pattern among industrial sectors. For example, paper and rubber product sector need almost twice amount of heat in 100-200°C as that in below 100°C zone. Fig. 1 is thermal energy demand distribution in French industry by D. Bobelin et al.^[10] The analysis focused on heat demand below 140°C. The heat demand below 100°C was reported to be small compared to heat demand above that temperature.

Table 2 Breakdown of the industrial heat demand with detailed temperature distribution between 100 and 500°C^[9]

Industrial sector	Hot water	Space heating	Process heat [TWh]					sum
			<100°C	<200°C	<300°C	<500°C	>500°C	
Chemicals and chemical products	0.2	6.7	13.5	9.5	5.9	5.5	55.7	96.9
Food products and beverages	0.3	8.3	11.8	13.7	0.9	0	0	35
Motor vehicles and trailers	1	7.3	2.7	1.1	0	0.8	3.7	16.8
Paper and paper products	0.1	2.4	2.7	5.6	0.2	4.1	0	15.1
Fabricated metal products	0.9	6.3	2.3	1	0	0.8	3.4	14.8
Machinery and equipment	0.6	4.5	1.6	0.7	0	0.5	2.3	10.3
Basic metals	0.2	4.4	0.9	1.5	0.1	1.1	154.9	163.1
Non-metallic mineral products	0.1	3.5	1.2	1	0	0.7	82.7	89.3
Rubber and plastic products	0.1	1.6	0.9	2	0.1	1.4	0	6.1
Electrical equipment	0.3	2.4	0.9	0.6	0	0.5	1.1	5.8
Textiles	0.1	1.2	2	0	0	0	0	3.3
Printing and recorded media	0	0.4	0.2	1.5	0.1	1.1	0	3.3
Wood and wood products	0	0.3	1.5	0.2	0	0.2	0	2.1
Furniture and other goods	0	0.7	0.4	0.6	0	0.4	0.2	2.4
Computer, electronic, optical products	0.1	0.9	0.3	0.1	0	0.1	0.4	2
Other transport equipment	0.1	0.9	0.3	0.1	0	0.1	0.4	2
Leather and related products	0	0.1	0.2	0	0	0	0	0.3
Wearing apparel	0	0.1	0.2	0	0	0	0	0.3
Tobacco products	0	0	0	0	0	0	0	0.1

Sum	4.2	52	43.6	39.3	7.5	17.4	304.9	468.9
Share	1%	11%	9%	8%	2%	4%	65%	100%

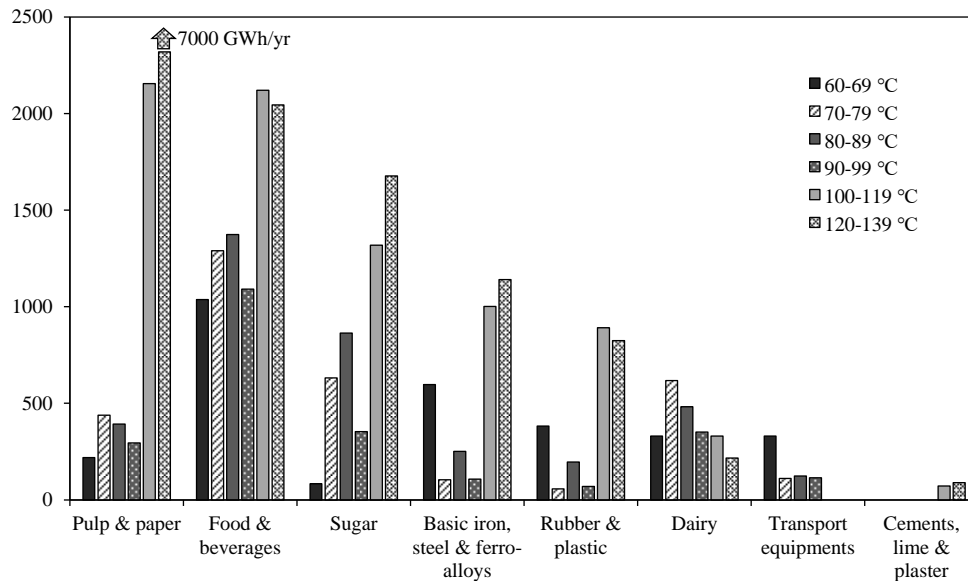


Figure 1 Distribution of the heat demand per industrial branch^[10]

Most of the thermal energy in industrial processes is supplied in the form of steam in order to maintain a stable temperature. Typically, steam below 140°C is termed as low pressure steam. Since there is high demand of heat above hot water temperature, many researches have been conducted to increase the temperature of heat pump cycle. In this paper, high temperature heat pump cycle to generate low pressure steam was developed. The design temperature of heat sources are 60-70°C waste heat and the product target was 120°C low pressure steam. In addition to basic cycle design, application of internal heat exchanger was considered. Test rig of small scale steam heat pump was constructed and performance evaluations were conducted.

2. Design of high temperature heat pump cycle for steam generation

As a base cycle for high temperature heat pump, vapor-compression cycle was applied. In this cycle, compressor increases the pressure and temperature of refrigerant by transforming mechanical energy. The discharged refrigerant releases heat energy to high temperature heat sink in condenser. The temperature difference between heat sink and refrigerant in condenser is generally in order of 10°C. The lower the temperature difference is, the lower the input energy that compressor requires. This eventually increases the efficiency of the system. The optimization of heat pump system is directed to the minimization of temperature difference in condenser.

Fig. 2 is heat and mass transfer diagram of steam generation heat pump cycle. In closed loop, the generated steam will be circulated to boiler again, while in open loop additional water should be supplied as the steam is exhausted to low temperature process or in the air. Fig. 2 is open loop configuration. Steam can be produced by either direct heating or by pressure reduction in circulating water (flashing method). Since the density difference between saturated liquid and saturated vapor of water is so large, the condenser of heat pump may become too bulky under direct heating method. Therefore in this study, pressure reduction method was used for steam production. In this method, only small portion of circulating water is transformed to low pressure saturated vapor. (less than 10%) Large amount of saturated liquid is recirculated to condenser and make-up water is mixed along the circulation. After circulation pump the pressure of process water is increased. Since the portion of saturated liquid is very high, the circulating water undergoes single phase heat transfer in condenser.

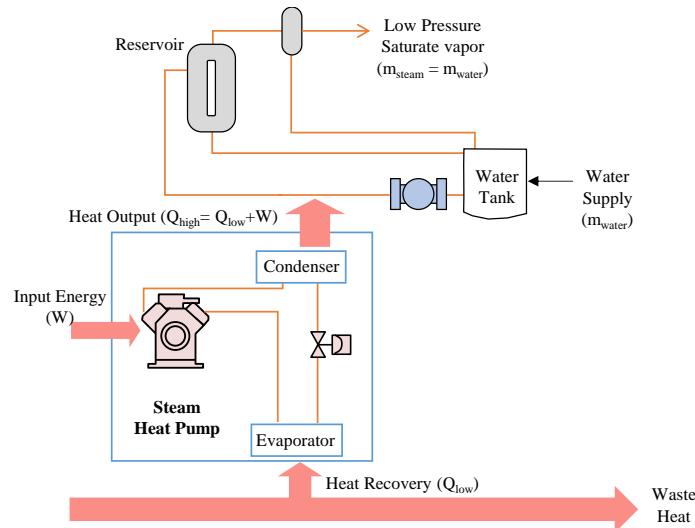


Figure 2 Cycle diagram of steam generation heat pump

To produce low pressure steam with heat pump, the condensation temperature should rise up to 130°C. Table 3 gives characteristics of refrigerant candidates for high temperature application.^[11] R134a is one of the most commonly used refrigerant in current market. It has critical temperature around 100°C. This means that heat pump cycle of R134a would undergo supercritical process to generate steam. If refrigerants have critical temperature above 130°C, they would have advantage in efficiency. R1234ze(Z), R245fa and R365mc satisfy this requirement for critical temperature. Among them R245fa was chosen since it is not flammable and is easily purchased.

Table 3 Basic characteristics of refrigerants for high temperature heat pump applications

Refrigerant	GWP	Flammability	T _c [°C]	P _c [MPa]
R134a	1430	-	101.1	4.06
R744	1	-	31.0	7.38
R1234yf	4	Low	94.7	3.38
R1234ze(E)	6	Low	109.4	3.64
R1234ze(Z)	<10	Low	153.7	3.97
R245fa	1030	-	154.0	3.65
R365mfc	794	Low	186.9	3.27

Heat pump system of R134a makes hot water with low temperature heat source (such as geothermal energy, sea water, ambient air). The target heat pump of R245fa makes low pressure steam with waste heat source of 60-70°C. Considering the difference in the operation temperature range R245fa and R134a have similar operation pressure. Thermo-physical properties related to heat transfer are listed in Table 4 at different saturation pressures. Chen suggested boiling heat transfer as combination of convective heat transfer and nucleate heat transfer.^[12] Column 10 and 11 of Table 4 are the relative convective heat transfer coefficient when values of R134a are assumed as 1. The velocity was assumed to be the same. Dittus-Boelter correlation was used to calculate heat transfer coefficient. From the relative values, R245fa may show similar heat transfer performance in vapor phase but have poor performance in liquid phase. This is because R245fa has disadvantage in thermal conductivity and viscosity in liquid phase. For the enhancement factor of convection heat transfer many researchers suggested Martinelli parameter as key parameter.^[12] Since R245fa has higher density in saturated vapor phase than R134a, enhancement factor of R245fa will decrease in evaporation heat transfer compared to R134a. In condensation, R245fa may show similar heat transfer performance as R134a.

Although the final product of steam heat pump cycle is vapor, the heat transfer in condenser is single phase if pressure reduction method is applied. Considering the working temperature difference, the condensation heat transfer will not change much compared to R134a. All of these will imply that conventional cycle design approach of R134a could be applied to the design of R245fa based heat pump design with only minor modifications. Therefore refrigerant change approach from R134a to R245fa was applied in the construction of test apparatus for the preliminary research.

3. System Reliability and Efficiency

The most common causes of compressor failure are refrigerant flood back, flooded starts, liquid slugging, overheating, lack of lubrication, etc.^[13] Refrigerant flood back occurs when refrigerant is not heated up to superheated phase in the evaporator. This is caused by low evaporation load or failure of evaporator. Under compression with refrigerant flood back, foaming phenomenon in lubricant of compressor generally occurs. Flooded start is when compressor starts its initial operation with a large amount of liquid refrigerant within the compressor shell. This happens when the system is over-charged with refrigerant or the system does not operate for a long period. Under flooded start, massive and instant exit of lubricant occurs. This causes temporal lack of lubrication of compressor. Overheating operation is caused by high compression ratio, low suction pressure, high superheat, etc. High compressor temperature causes negative effect to lubrication.

Adequate superheat is needed for the protection of refrigerant flood back. It is also important to secure lubrication performance in compressor. Compressor lubricant performs roles such as sealing, anti-friction, cooling, etc. Under normal operation lubricant mixes with refrigerant in compressor. Daniel chart shows trends of lubricant/refrigerant mixture with isobaric curves.^[14] It shows refrigerant concentration and viscosity at different temperature and pressure of lubricant.^[15] Since the pressure of lubricant is the same as that of refrigerant the difference between lubricant temperature and refrigerant saturation temperature can be used as the index of reliability. This definition is similar to the definition of superheat. In practical application, superheat would be used as an alternative index to lubricant superheat. Under constant pressure, concentration ratio of refrigerant stiffly increases as temperature approaches saturation temperature of refrigerant. Since the viscosity of HFC refrigerants is much lower than that of pure lubricant, lubrication performance is impaired as superheat becomes small.^[16] Whether using suction superheat or discharge superheat as the criteria of reliability depends on the position of oil reservoir in the compressor. In low pressure shell, oil reservoir is placed in suction part. Low suction superheat may tremendously increase the concentration of refrigerant damaging the lubrication performance. However in high pressure shell compressor, the oil reservoir is placed in discharge part. Therefore the negative effect of low suction superheat can be compensated in high pressure shell type compressor.^[17] In summary, adequate superheat where the oil reservoir is placed should be maintained to secure lubrication performance.

Table 4 Properties of R134a and R245fa at saturation conditions

P (MPa)	R134a				R245fa						
	T (K)	ρ_{liq} (kg/m ³)	ρ_{vap} (kg/m ³)	h_{gf} (kJ/kg)	T(K)	ρ_{liq} (kg/m ³)	ρ_{vap} (kg/m ³)	h_{gf} (kJ/kg)	$\alpha_{liq,relative}$	$\alpha_{vap,relative}$	$X_{trelative}$
0.3	274	1293	15	198	319	1281	17	178	0.87	1.05	1.09
0.7	300	1200	34	176	348	1187	39	156	0.88	1.05	1.09
1.1	316	1134	54	160	367	1118	62	140	0.89	1.05	1.09
1.5	328	1077	77	145	381	1059	88	126	0.90	1.06	1.09
1.9	338	1024	101	132	392	1003	117	113	0.91	1.08	1.09
2.3	347	972	129	118	402	947	151	99	0.92	1.11	1.10

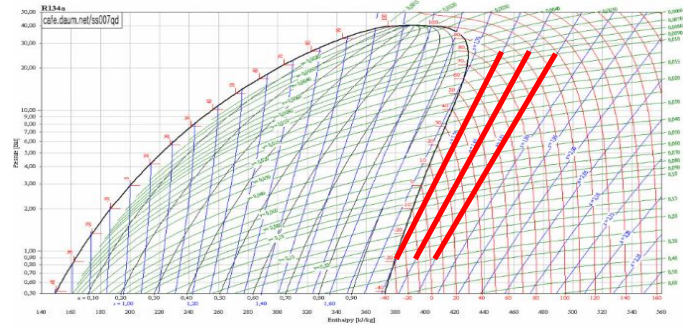
“relative” means values of R245fa compared to R134a under same pressure

$$\text{Dittus-Boelter correlation: } \alpha_{liq} = 0.023 \text{Re}_{liq}^{0.8} \text{Pr}_{liq}^{0.4} \left(\frac{k_{liq}}{d_i} \right)$$

$$\text{Martinelli parameter: } X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_{vap}}{\rho_{liq}} \right)^{0.5} \left(\frac{\mu_{liq}}{\mu_{vap}} \right)^{0.1}$$

For commonly used R134a the isentropic line moves away from saturated vapor line (bubble line) as the pressure increases. However R245fa shows different trend, moving toward the bubble line (Fig. 3) Discharge superheat affects lubrication of high pressure part in compressor. Therefore certain amount of discharge superheat should be maintained for reliability.

[R134a P-h diagram]



[R245fa P-h diagram]

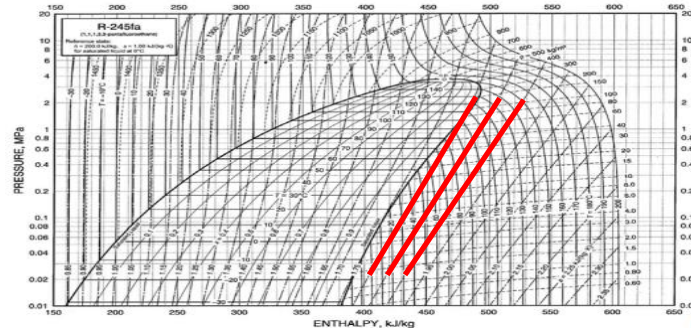


Figure 3 Isentropic lines in P-h diagram

Fig. 4 shows discharge superheat trends with different suction superheat and operation pressures. The suction pressure was fixed as 400 kPa and the isentropic efficiency of compressor was assumed as 0.8. If compressor has to have adequate superheat for the entire operation range, more suction superheating is required for R245fa. This means that the design point of suction superheat of R245fa should be higher than R134a. High suction superheat requires large temperature difference between refrigerant and secondary fluid in evaporator. This causes low evaporation pressure and low efficiency.

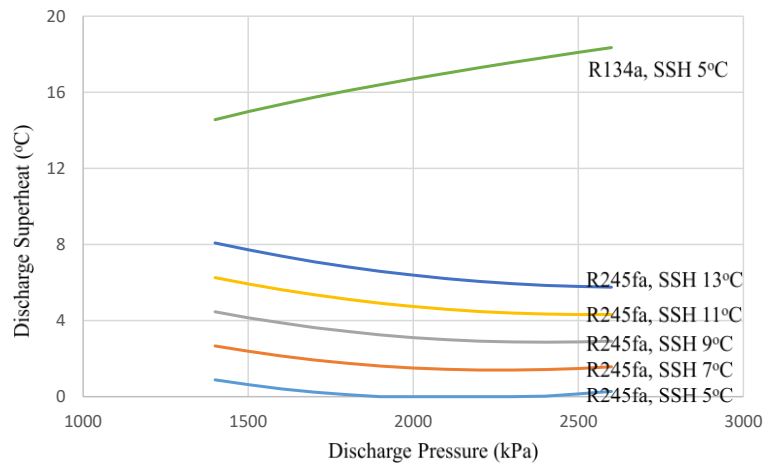


Figure 4 discharge superheating trend of R245fa and R134a

4. Application of Internal Heat Exchanger

In the previous chapter it was shown that the requirement of superheat of R245fa may cause performance decrease in heat pump cycle. With evaporator only, the temperature gradient of refrigerant will increase to meet the requirement. The temperature difference between heat source and evaporator will also increase. To meet this,

evaporation pressure of heat pump system will drop resulting in high compression work and low capacity. High temperature heat pump system for steam generation was designed to examine this trend. Fig. 5 shows the experimental apparatus. Fig. 6 is the system performance changes under different suction superheat. If the minimum value of discharge superheat is set as 10°C, about 7% of COP was decreased due to high suction superheat.



Figure 5 Experimental apparatus of steam generation heat pump development

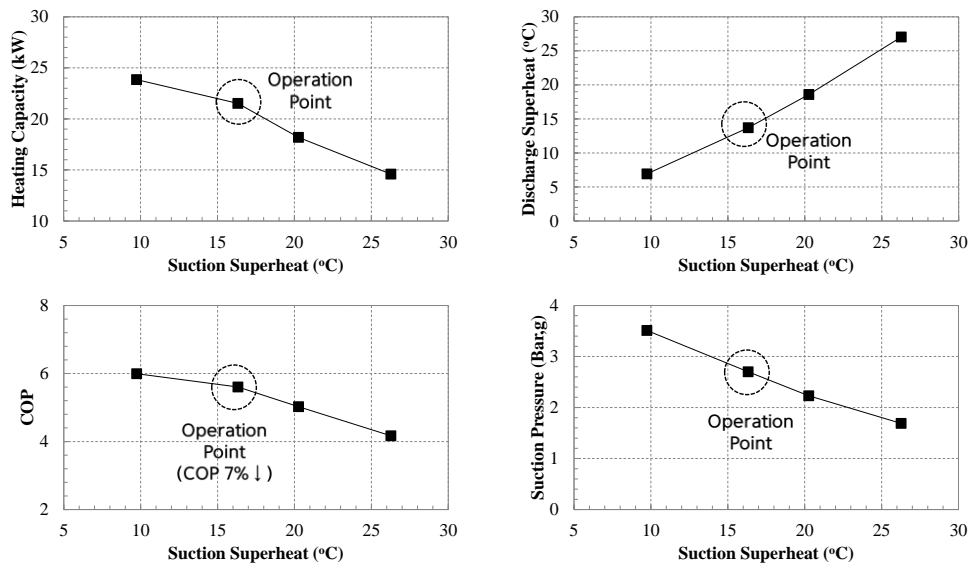


Figure 6 Performance characteristics of R245fa heat pump cycle with various suction superheat

Internal heat exchanger (IHEX) transfers heat between condenser outlet fluid and evaporator outlet fluid. With this, system has additional suction superheat and heating capacity. As IHEX provides superheat for compressor, unnecessarily low pressure in evaporator can be eliminated. Table 5 shows performance comparisons with and without IHEX. The operation condition is the hot water generation case under low load. In this condition IHEX shows the increase of the evaporation pressure by 0.4 bar.

Table 5 Performance comparisons between without and with internal heat exchange

Parameters		No IHX	With IHX	
Evaporation P	bar,g	2.88	3.64	0.76
Condensation P	bar,g	11.09	11.5	0.41
Subcooling	°C	2.06	13.67	11.61
Discharge superheat	°C	15.21	25.82	10.61
Evaporator superheat	°C	17.29	11.82	-5.47
Suction superheat	°C	16.21	30.21	14.00
Heating capacity	kW	21.52	24.55	3.03
Heating COP		5.60	6.22	0.61

For the steam generation flashing process was applied. Since the circulated water is pressured the outlet status of water of the condenser can remain subcooled even when the temperature is above 100°C. By flashing the pressured water to the two-phase region, small portion of water is changed to steam. Fig. 7 shows the system diagram of the steam generation heat pump. Water inlet and outlet temperatures and the temperature of the flash tank are displayed in the diagram when the steam temperature of produced steam is 104.5°C. Fig. 8 shows the trend of the heating capacity and the COP at different compressor frequencies. As the frequency increases from 30 Hz to 60 Hz, the heating capacity also increases by 90% while the COP decreases by 10%.

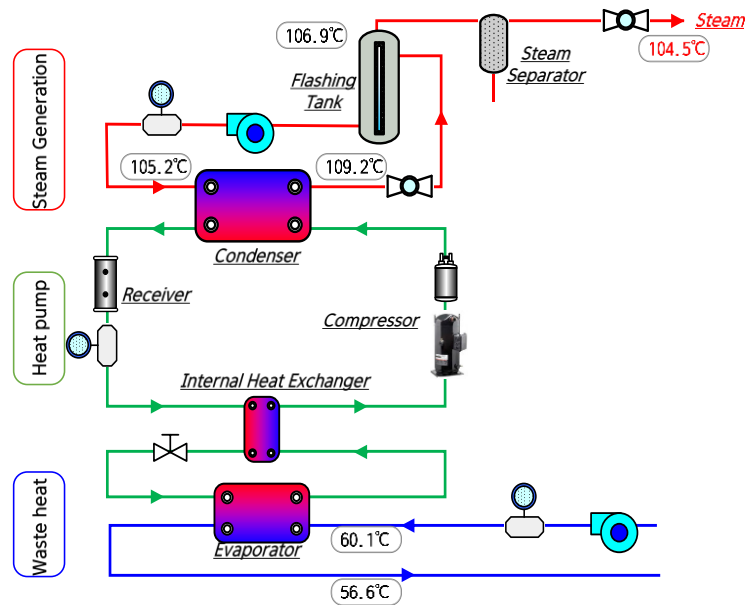


Figure 7 Cycle diagram of steam heat pump with internal heat exchanger

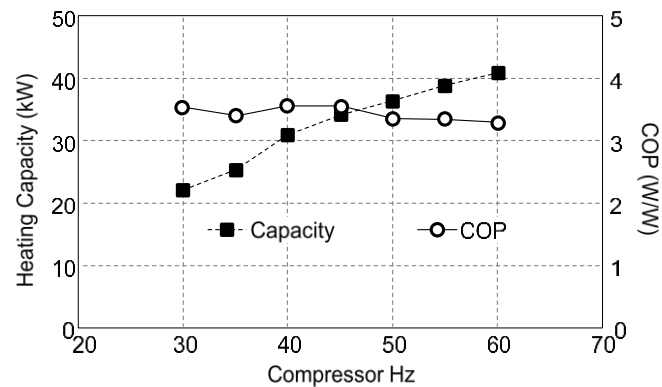


Figure 8 Trend of heating capacity and COP at different compressor frequencies

Table 6 shows the comparison of the system parameters at the different steam temperatures. Compared to the large increase of condenser pressure increases, the evaporator pressure shows a minor change.

Table 6. System parameters at the different steam temperatures

System Parameters	Steam Temperature: 104°C	Steam Temperature: 123°C
Water temperature (°C, condenser outlet)	109.2	128.9
Flash tank temperature(°C)	106.9	125.2
Condenser pressure (bar,g)	16.4	24.7
Evaporator pressure (bar,g)	2.7	2.4
Water temperature (°C, evaporator inlet)	60	60

Table 7. Effects of suction superheat on discharge superheat and COP

Suction superheat	R134a		R245fa	
	Discharge superheat	Relative COP	Discharge superheat	Relative COP
11	15.63	1.013	0.02	1.037
13	17.36	1.018	1.42	1.050
15	19.10	1.022	2.85	1.062
17	20.86	1.027	4.33	1.074
19	22.64	1.031	5.83	1.086
21	24.42	1.036	7.37	1.097
23	26.22	1.040	8.93	1.109
25	28.03	1.045	10.52	1.120

To further analyze the effect of IHEx simple parametric analysis was performed for R134a and R245fa. Since the performance of IHEx affects the degree of superheat at the compressor suction, the analysis was conducted at different suction superheats. The operation temperatures of R134a and R245fa are different since R134a is for the hot water generation system and R245fa is for the steam generation system. Instead of temperatures condensation pressure and evaporation pressure were set as 2200 kPa and 400 kPa for the both refrigerants. Isentropic compression, constant volumetric efficiency, no subcooling were assume. Under these conditions, performance with 5°C suction superheat was selected as reference point. The relative performances with different suction superheat were listed in Table 7. According to the table, increasing superheating also increases the compression work. However it also increases the heating capacity of the system. Therefore the heating COP have the tendency to increase as well. More improvement is seen in R245fa cases. This is due to higher specific heat at the vapor phase and the relatively low latent heat. R245fa has lower sensitivity in discharge superheat and higher sensitivity in efficiency than R134a. Therefore R245fa has more advantage in applying IHEx than R134a.

5. Conclusions

In this study cycle characteristics of high temperature heat pump cycle were analyzed. As the demand for energy saving has increased in the industrial sector, interest for heat pump which recycles waste heat to produce low pressure steam has increased. R245fa was considered as suitable refrigerant for a heat pump cycle for steam generation condition since the critical temperature of R245fa is above the operation temperature of steam generation condition. Similar design concepts of R134a hot water heat pump could be applied to design steam heat pump cycle when steam is produced through flashing process.

Superheat is an important index for reliability and efficiency. In compressor, lubricant mixes with refrigerant. Small temperature difference to saturation temperature implies stiff concentration increase of refrigerant - high solubility of refrigerant to lubricant. Therefore certain amount of superheat should be maintained in oil reservoir part of compressor. R245fa tends to decrease superheat under compression process which is contrary to R134a. In designing R245fa cycle, design point of suction superheat must be higher because of this characteristic.

With internal heat exchanger, discharge superheat can be secured without increasing temperature difference in the evaporator. Simple parametric analysis was performed for R134a and R245fa to see effect of suction superheat. The results showed that R245fa has more favorable thermo-physical characteristics in applying internal heat exchanger compared to R134a. Therefore internal heat exchanger can be suggested as an essential component in R245fa heat pump cycle.

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References

- [1] Energy Technology Perspectives 2012 – Pathways to a Clean Energy System, IEA, France
- [2] Energy Info. Korea, Korea Energy Economics Institute, December 2013.
- [3] Annual Report 2011, Korea Energy Management Corporation
- [4] J.T. Park and G.C. Jang, An investigation on quantity of unused energy using temperature difference energy as heat source and its availability, *Energy Eng. J.*, 11, 2002, 106-113.
- [5] IEA-ETSAP and IRENA© Technology Brief E12, Heat Pumps – Technology Brief, January 2013.
- [6] Annual Report 2012, Danish Technological Institute
- [7] Application of Industrial Heat Pumps, Task 4: Case Studies, IEA Industrial Energy-related Systems and Technologies Annex13/ IEA Heat Pump Programme Annex 35, Final Report, 2014.
- [8] A. Costa, B. Bakhtiari, S. Schuster, J. Paris, Integration of absorption heat pumps in a Kraft pulp process for enhanced energy efficiency, *Energy*, vol 34, 2009, pp. 254-260.
- [9] C. Lauterbach, Potential, system analysis and preliminary design of low-temperature solar process heat systems, Kassel university press GmbH, Kassel, 2014
- [10] D. Bobelin, A. Bourig, J. Peureux, Experimental results of a newly developed very high temperature industrial heat pump (140°C) equipped with scroll compressors and working with a new blend refrigerant, *Proceedings of International Refrigeration and Air Conditioning Conference at Purdue*, July 2012.
- [11] Application of Industrial Heat Pumps, Task 3: R&D Projects, IEA Industrial Energy-related Systems and Technologies Annex13/ IEA Heat Pump Programme Annex 35, Final Report, 2014.
- [12] J.R. Thome, *Engineering Data Book III*, Wolverine Tube, Inc.
- [13] Why compressors fail part 1-7, Danfoss, <http://danfoss.com>.
- [14] D. R. Henderson, 1994, Solubility, viscosity and density of refrigerant/lubricant mixtures – Final technical report, Spauschus Associates, Inc., 1994, DOE/CE/23810-34.
- [15] T. Matsumoto, M. Kaneko, Y. Kawaguchi, Development of PVE Refrigeration Lubricants for R32, *Proceedings of International Refrigeration and Air Conditioning Conference at Purdue*, July 2014.
- [16] R. Dossat, *Principles of Refrigeration*, fifth edition, Prentice Hall, Columbus, Ohio.
- [17] W. Tanawittayakorn, P. Phrajunpanich, S. Siwapornphaisarn, Heat Pump Efficiency Improvement by Discharge Superheated Control, *Proceedings of International Refrigeration and Air Conditioning Conference at Purdue*, July 2012.