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공학석사 학위논문

Study on the Performance of Steam Generation Heat Pump using Waste Heat Source with Suction-Line/Liquid-Line Heat Exchanger

내부열교환기가 적용된 폐열원 이용 스팀생산 히트펌프의 성능에 관한 연구

2018 년 2 월

서울대학교 대학원

기계항공공학부

이 재 호

Abstract

Study on the performance of steam generation heat pump using waste heat source with suction-line/liquid-line heat exchanger

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Steam generation heat pump, which is a heat pump cycle using waste heat source to produce industrial steam whose temperature is about 120℃, has different characteristics with other heat pump cycle like air source heat pump. It has high compression ratio because the maximum temperature of refrigerant in the cycle is more than 125℃. Besides, the temperature gradient in the heat exchanger is also different because the secondary fluid at condenser is steam which is two-phase state. Thus, the experiment of SGHP basic cycle is conducted first in order to investigate the distinctive features of SGHP and

possibility to improve performance and to replace boiler. By conducting basic SGHP experiments of 8 kW lab-scale experimental equipment, heating capacity, coefficient of performance(COP) are calculated with variables of refrigerant charge, waste heat source temperature and degree of superheat. Also, other results such as compressor work, cooling capacity, refrigerant mass flow rate and so on are calculated with estimated temperature, pressure and mass flow rate at important points in order to analyze the cycle in detail.

Additional experiments are also conducted after applying suction-line/liquid-line heat exchanger which is internal heat exchanger in order to solve the performance degradation problem of basic SGHP cycle due to high degree of superheat and compression ratio. Performance improvement is confirmed compared to basic cycle : COP increase of 0.4 and heating capacity increase of 38~62%. And the cause of improvement is analyzed by pressure-enthalpy diagram comparison with basic cycle: evaporator pressure and degree of sub-cooling increase. Also, guideline for using SLHX to SGHP cycle efficiently are suggested.

Keywords: Steam generation heat pump, Waste heat recovery, Internal heat exchanger, Suction-line/liquid-line heat exchanger.

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Nomenclatures

A	area (m^2)
C_p	specific heat (kJ/kg·K)
COP	coefficient of Performance
DSC	degree of sub-cooling (K)
DSH	degree of superheat (K)
EEV	electronic expansion valve
h	specific enthalpy (kJ/kg)
I	current (A)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (kPa)
PD	piston displacement per second (m/s)
q	specific capacity (kJ/kg)
Q_h	heating capacity (kW)
Q_c	cooling capacity (kW)
SGHP	Steam Generation Heat Pump
SLHX	Suction-Line/Liquid-Line Heat Exchanger
T	temperature (K)
U	overall heat transfer coefficient ($kW/m^2 \cdot K$)

V	voltage (V)
v	specific volume (m^3/kg)
W	Compressor work (kW)
x	vapor quality
ρ	density (kg/m^3)
η_v	volumetric efficiency

Subscript

Cond	condenser
comp	compressor
Eva	evaporator
in	inlet
liq	liquid
out	outlet
ref	refrigerant side
steam	steam side
sat	saturation
source	waste heat source side
vap	vapor
water	tap water side

Chapter 1. Introduction

1.1 Background of the study

As the consumption of fossil fuel is increasing, environmental issues are emerging, and it becomes necessary to use energy efficiently in order to prevent waste of energy while reducing the use of fossil fuels. One of the way is to recovery the unused waste heat source. Rattner and Garimella (2011) researched that the energy of high temperature exhaust from power plant is same to 27% of residential air conditioning in the USA. They also investigated the potential to cut primary energy demand in USA by 12% and CO2 emissions by 13% through waste heat recovery. In the report by USA Department of Energy (2008), about 60% of waste heat are under 230℃ and many facilities only cool the exhaust gases to about 150℃. Fig. 1.1 shows there is the most waste heat and work potential at low temperature under 230℃. Therefore, it is necessary to utilize the waste heat source at low temperatures as much as possible in order to save energy and additional costs for cooling the waste heat.

In the report by USA Department of Energy (2003), Steam generation heat pump is a kind of industrial heat pump system that utilize the waste heat source to produce higher temperature products whose temperature is more than 100℃. Steam is very useful because latent heat is used in heat exchange so has good heat transfer performance and constant temperature in heat exchange process (Cengel, Y., 2003). By these advantages, steam takes a third of total industrial energy in USA and it is widely being used in various field such as papermaking,

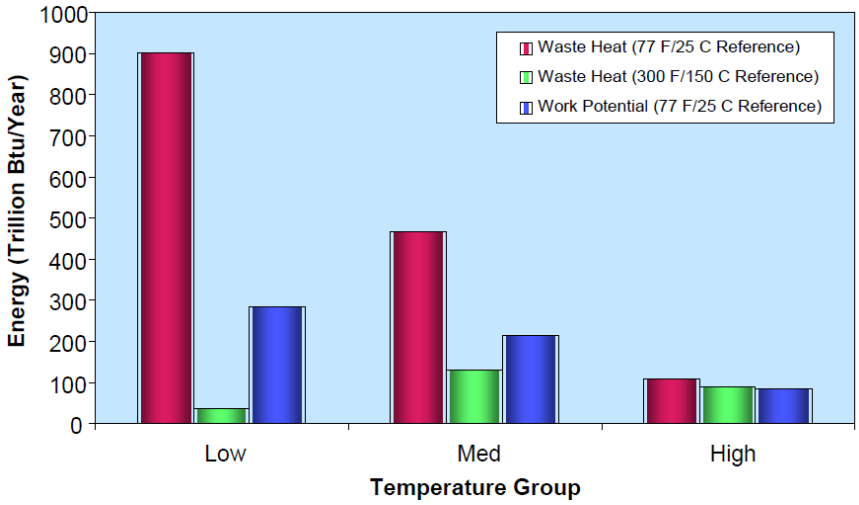

Fig. 1.1 Unrecovered waste heat in different temperature groups

(U.S. Department of Energy, 2008)

drying, sterilization and so on (US Energy Information Administration, 2012).

Existing way to generate steam is boiler, which is easy to make a lot of steam with fossil fuel in Fig. 1.2. However, its efficiency is less than 1 and a lot of waste heat come out. Also there is a lot of heat loss at pipe and pipe cost because boiler tends to be very far from the process for thermal management and exhaust gas treatment. These drawbacks of boiler can be solved if heat pump cycle is used to generate steam. Fig. 1.3 is SGHP cycle installed at the process. SGHP generates more heat than compressor work. Also contaminated water, which is most of waste heat source form, could be used, which helps energy saving. According to the report from IEA, heat pump could cut global CO2 emissions up to 8%. In addition, SGHP can be installed in each process and supply steam with the specific requirement for each industrial process, and waste heat from each process can be managed well.

SGHP operates in high temperature conditions, whose secondary fluids are 120℃ at condenser and 50~70℃ at evaporator. So it has a high compression ratio because the maximum temperature of the refrigerant is more than 125 degree Celsius. Also some refrigerants usually used at air source heat pump could not be used in SGHP cycle because refrigerant must not be go through the property change or decomposition at high temperature. So at SGHP cycle, different refrigerant, R245fa in this study, operates in different high temperature at different pressure. Besides, the temperature gradient in the heat exchanger is also different because the state of secondary fluid at some part of condenser is two-phase state which is steam. These various factors cause distinctive tendency of SGHP which is not possible to know by simulation or analytic

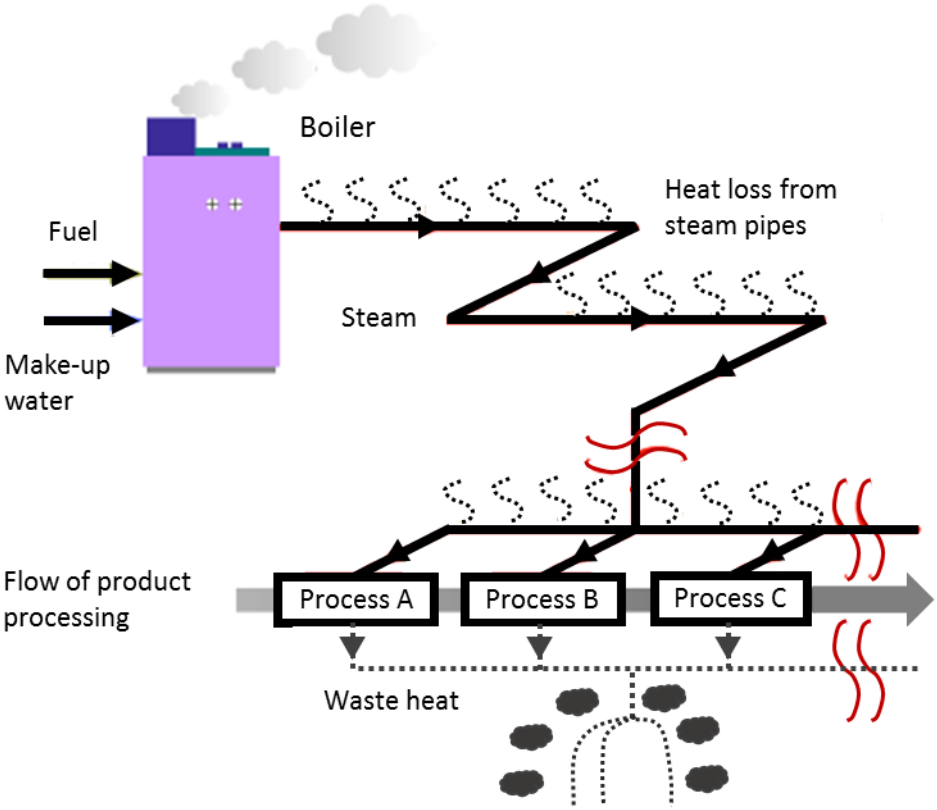

Fig 1.2 Supplying steam with boiler

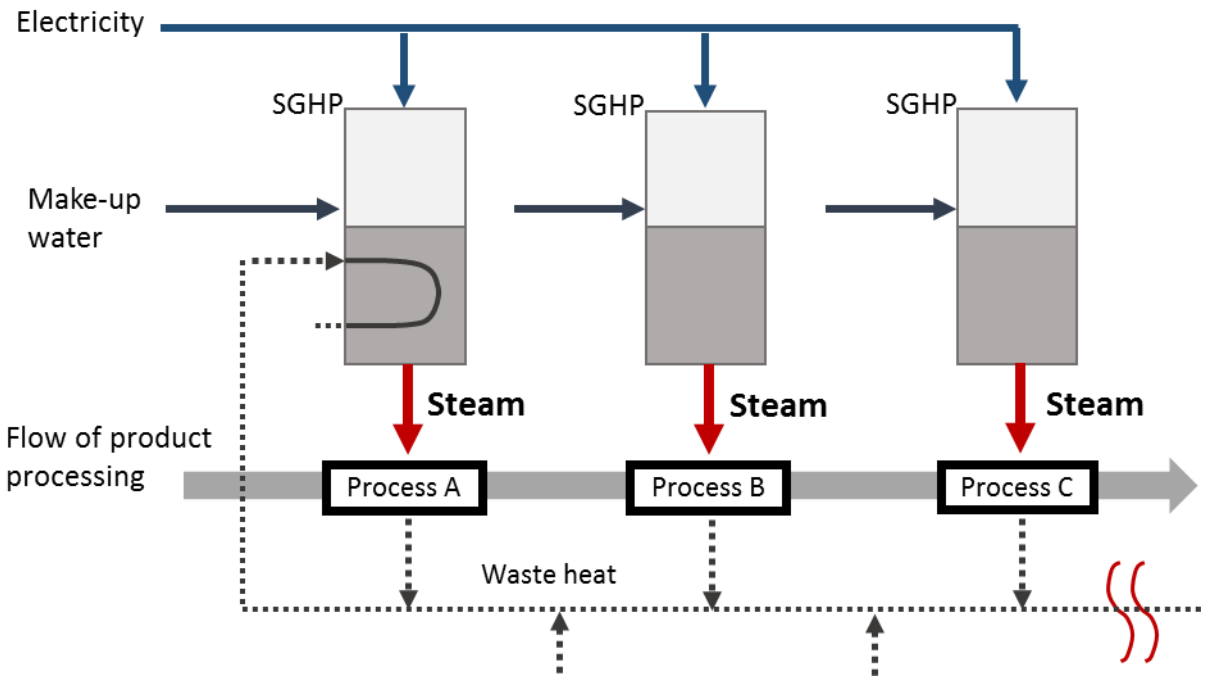

Fig 1.3 Supplying steam with SGHP system

access. So it is necessary to investigate about the distinctive features of SGHP for improving efficiency and performance. Also the performance of SGHP system should be compared to boiler in order to check if SGHP is possible to replace the boiler. Therefore, the experiments of SGHP basic cycle is conducted first.

Another important thing is to find the way to improve the performance of SGHP system. Because SGHP is different to other heat pump cycle, the most efficient way to improve the performance is different as well. High temperature operating condition causes some performance degrading issues of SGHP basic cycle, which mainly result from decrease of evaporator pressure. Because there is big temperature difference between refrigerants at high and low pressure, there is possibility to improve efficiency and solve the basic cycle problems. Fig. 1.4 is the heat pump cycle with a suction-line/liquid-line heat exchanger. By adding the heat exchanger, the refrigerant at evaporator outlet can be heated again by the refrigerant at condenser outlet. So it is necessary to investigate the characteristics and tendency change when SLHX is applied to SGHP basic cycle. In order to use the SLHX to SGHP cycle well, the guideline for using SLHX is also needed. And checking how much the performance of SGHP get improved by SLHX is important to know the effect of SLHX. Therefore, the experiments of SGHP cycle applied by SLHX is conducted and the results are compared to the basic cycle.

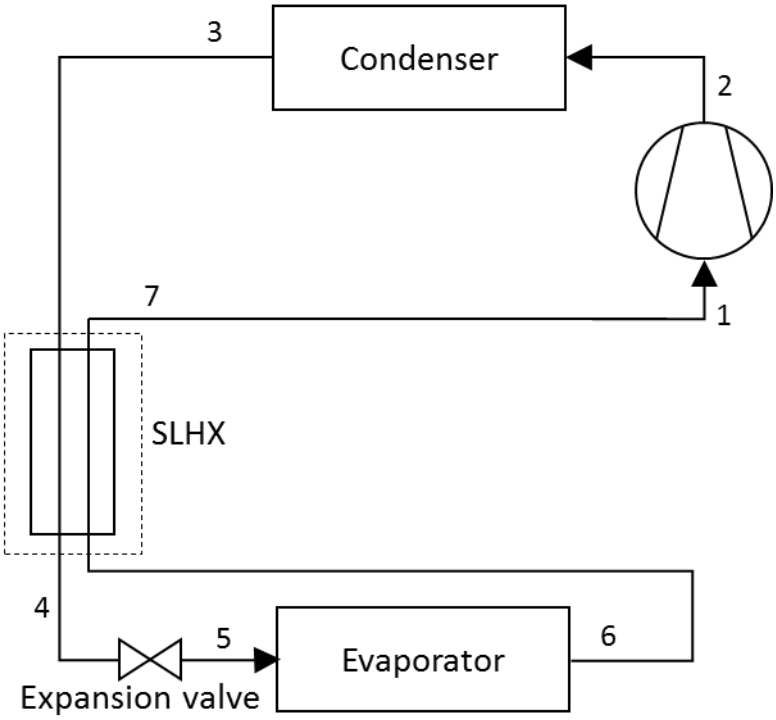

Fig 1.4 Schematic of heat pump cycle with a suction-line/liquid-line heat exchanger

1.2 Literature survey

1.2.1 Industrial heat pump system

Many researchers studied on heat pump system to rise the temperature of fluid to more than 100℃ by using waste heat. There were many attempts to make heat pump for increasing the temperature of something to more than 100C using waste heat. Chamoun et al. (2014) investigated a new high temperature heat pump with water as a refrigerant in the condensing temperature range of 130-140℃. By experimental and numerical simulation, the start-up phase, heat enhancement and transient trends were analyzed. Boblin and Bourin (2012) presented the feasibility and low CO2 emissions of lab-scale high temperature industrial heat pump whose maximum temperature is 140℃ based on experimental data with ECO3™, R245fa, R1234ze, R345mfc, R236fa and R717. They demonstrated that it was possible to generate steam efficiently until 125℃, but there was degradation of COP between 125 and 140℃.

Kaida et al. (2015) evaluated the technical possibilities for better performance of lab-scale SGHP system generating 165℃ steam, which motivate the ready-made SGHP system whose steam temperature is 120 °C (SGH120) and 165°C (SGH165) by Kobe Steel, Ltd. They investigated steam generation rate, energy efficiency and performance of the SGH165 with various operating conditions. Many researchers also studied on high temperature industrial heat pump system under maximum temperature of 120℃(Lee et al, 2017; Cao et al, 2014; Assaf et al, 2010; Kim et al, 2005). In order to generate

steam of 120℃ using various waste heat source temperature more efficiently, investigation of the characteristics and trends in SGHP cycle is necessary.

1.2.2 Application of SLHX

There are some studies on SLHX applied at the refrigeration cycle, which are usually air source heat pump. Apera et al. (1999) calculated the advantages of applying SLHX to the ideal refrigerant cycle. COP difference of the SGHP cycle with and without SLHX are calculated with own simple equation and compared through various (CHC)s and (HCFC)s types of. However, this paper are analyzed at only one condition of outdoor temperature. Domanski et al. (1994) screened twenty-nine fluids for their sensitivity to SLHX cycle which has constant pressure on a theoretical basis. The fluids which has different properties such as heat capacity, latent heat, and coefficient of thermal expansion are compared with COP and volumetric capacity. Klein et al. (2000) take into account the mass flow rate degradation with DSH and pressure drop for calculating the effect of SLHX analytically. They conclude some refrigerants such as R22 and R32 have degradation of performance with SLHX, and SLHX which has a minimal pressure are useful for other refrigerant systems such as R134a, R410A and so on. Hermes (2013) calculated the improvement by SLHX more similarly to real cycle at the conditions of constant pressure, constant cooling capacity. However, previous studies saw the effect of SLHX to vapor compression refrigeration system on a theoretical basis. Because ideal case is different to real case, it is necessary to consider some

variables such as pressure change of evaporator and condenser, non-isentropic compression and so on. There is also DSH and DSC at basic cycle in order to prevent the compressor, so it is necessary to compare both cycle at the same DSH.

In order to see the performance improvement in more real case, in this paper, additional experiments are conducted after applying suction/liquid line heat exchanger which is internal heat exchanger. And the cause of improvement is analyzed by pressure-enthalpy diagram comparison with SGHP basic cycle. Also, guideline for using SLHX to SGHP cycle efficiently are suggested.

1.3 Objectives and scopes

There is different features compared to air source heat pump system, and the tendency of SGHP cycle are also different whether SLHX is applied or not. A goal of this study is to investigate the characteristics of SGHP cycle in detail in order to find the way for improving the performance. And another goal is to compare to SGHP cycle with and without SLHX. For these objectives, two different experiments are conducted.

In chapter 2, many cases of experiments of SGHP basic cycle which is mainly composed of compressor, condenser, expansion valve and evaporator are conducted. Many variables such as heating capacity, heat transfer rate at each components, compressor work and so on are calculated by some equations. For this variables, properties of refrigerant and secondary fluid at some key

points of the cycle are estimated such as temperature, pressure and flow rate.

In chapter 3, another type of experiments are conducted after applying SLHX. The different features of the cycle compared to basic cycle are checked. Also the way for applying SLHX to SGHP cycle efficiently is analyzed and suggested. And the efficiency and capacity is calculated for judging if it`s possible to replace the boiler.

Chapter 2. Study on Performance of SGHP basic cycle

2.1 Introduction

In this chapter, the characteristics and trends of the basic SGHP cycle are identified through lab scale experiments. The efficiency and capacity are also calculated in order to judge if SGHP is possible to replace the boiler. Experiments were conducted with respect to refrigerant charge, degree of superheat and waste heat source temperature. When SGHP is applied at the industrial process, determining refrigerant charge is necessary. So by varying the refrigerant charge, change of performance and the pressures is investigated. The performance are also compared according to the waste heat source temperature because the efficiency and capacity of the cycle depend on the temperature. Some variables such as refrigerant flow rate, compressor work, cooling and heating capacity are also analyzed in order to improve the performance. And experiments are conducted with variable of DSH because SGHP needs high DSH and optimal DSH for efficiency should be determined.

2.2 Experimental method

2.2.1 Experimental equipment

Fig. 2.1 and Fig 2.2 is a schematic and photo of experimental equipment. Specification of the equipment is at Table 2.1. Basic SGHP cycle is mainly composed of compressor, condenser, expansion valve and evaporator. In order to improve heat transfer performance and to supply steam stably, there is a flash tank at which steam is generated. The 120℃ water moves from the tank to condenser, and water from thermostat which imitate the waste water heat source moves to evaporator. Because the flow rate of tap water is exactly same to the rate of steam, water level at flash tank is unchanged. So COP and capacities can be calculated by secondary fluid's energy balance equation. Heating capacity and cooling capacity is calculated from Eq. (2.1) and Eq. (2.2)

$$Q_{\boldsymbol{h}} = \dot{m}_{steam} h_{out} - \dot{m}_{water} C_{water} T_{in} \tag{2.1}$$

$$Q_{\boldsymbol{c}} = \dot{m}_{source} C_{source} (T_{in} - T_{out}) \tag{2.2}$$

However, because it is necessary to compare the results in detail according to main variables, the capacity are calculated with refrigerant properties from Eq. (2.3) and Eq. (2.4). Compressor work is estimated by digital power meter and calculated by multiplying voltage and current from Eq. (2.5). COP is calculated by dividing heating capacity by compressor work from Eq. (2.6). By estimating

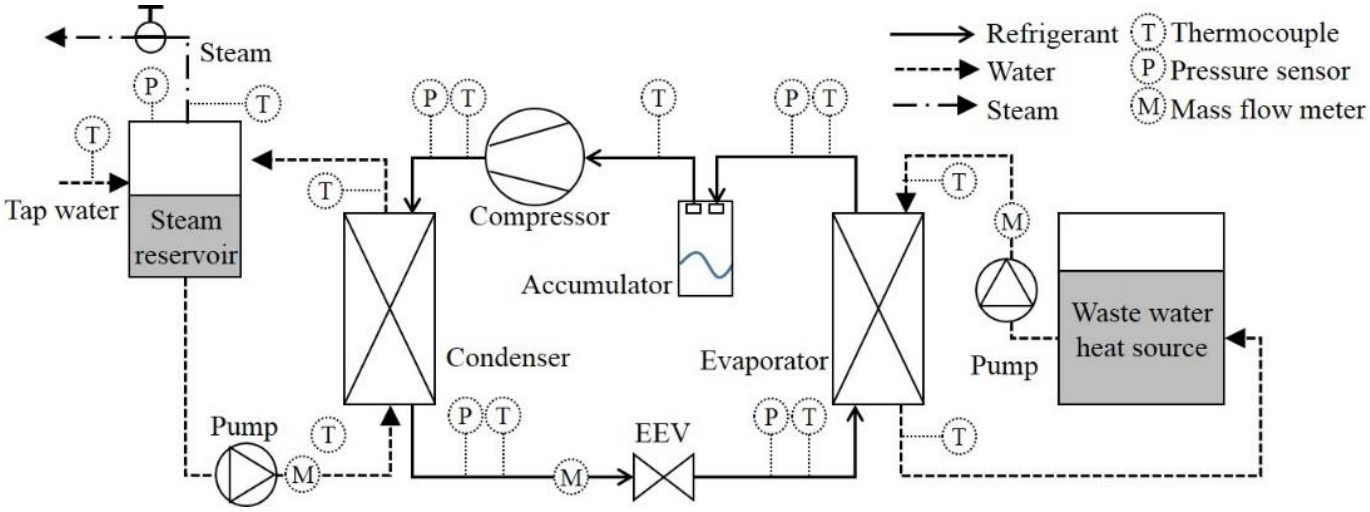

Fig 2.1 Schematic of experimental equipment

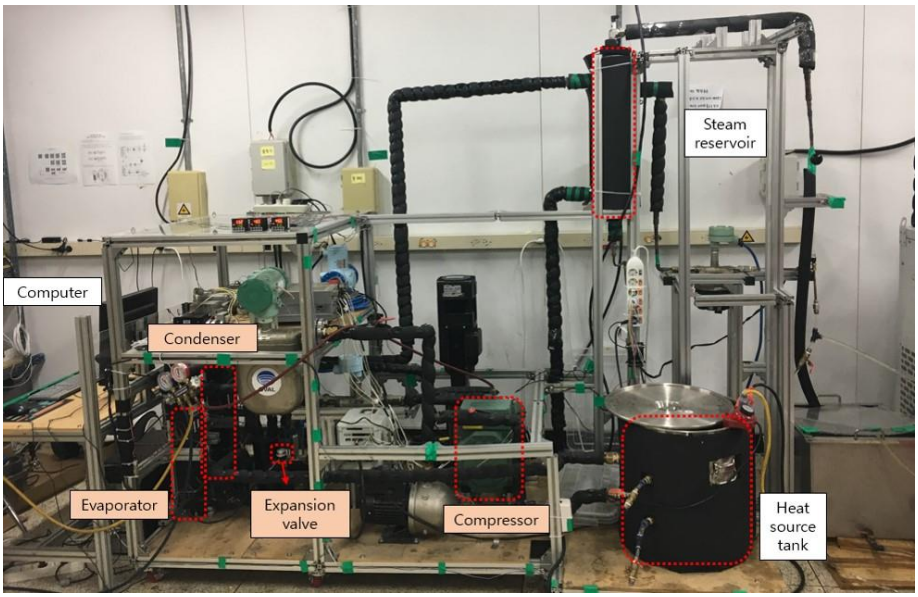

Fig 2.2 Photo of experimental equipment

Table 2.1 Specification of experimental equipment

Component	Model (manufacturer)	Contents
Compressor	4DES-7-20D (Bitzer)	Reciprocating compressor Frequency is controlled by inverter
Compressor oil	CPI-4214-320	ISO 320 standard Oil for high temperature
Heat exchanger	B25TH X 26 (SWEP)	Brazed plate heat exchanger Plate number of 26
Expansion valve	DPF(O)-3.2 (SANHUA)	2000 step controllable EEV
Water pump for condenser	MGE80B (Grunfos),	Pump for high temperature Frequency is controllable
Water pump for evaporator	HBI (Daeyoung)	Frequency is fixed
Flash tank	Self-design	Width x depth x height = 150x150x550mm

temperature, pressure, and mass flow rate at important point, various variables are calculated such as enthalpy, entropy, DSH and so on.

$$Q_{\boldsymbol{h}} = \dot{m}_{ref}(h_{Cond,in} - h_{Cond,out}) \tag{2.3}$$

$$Q_{\boldsymbol{c}} = \dot{m}_{ref}(h_{Eva,out} - h_{Eva,in}) \tag{2.4}$$

$$W_{comp} = \mathrm{V} * \mathrm{I} \tag{2.5}$$

$$\mathrm{COP} = Q_{\boldsymbol{h}}/W_{comp} \tag{2.6}$$

2.2.2 Experimental conditions

Main variables are refrigerant charge, waste heat source temperature and suction DSH. Experiment are conducted by fixing the value of all variables but three main variables. EEV opening percent, valve opening for supplying tap water, heat source temperature are controlled through LABVIEW program in Fig. 2.3 and Fig. 2.4. All the values are obtained at steady state.

Steam supply rate are controlled through needle valve after flash tank in order to maintain the pressure of flash tank to 2 bar. Compressor frequency, steam temperature, flow rate of secondary fluid at evaporator and condenser and so on are fixed of the value at Table 2.2.

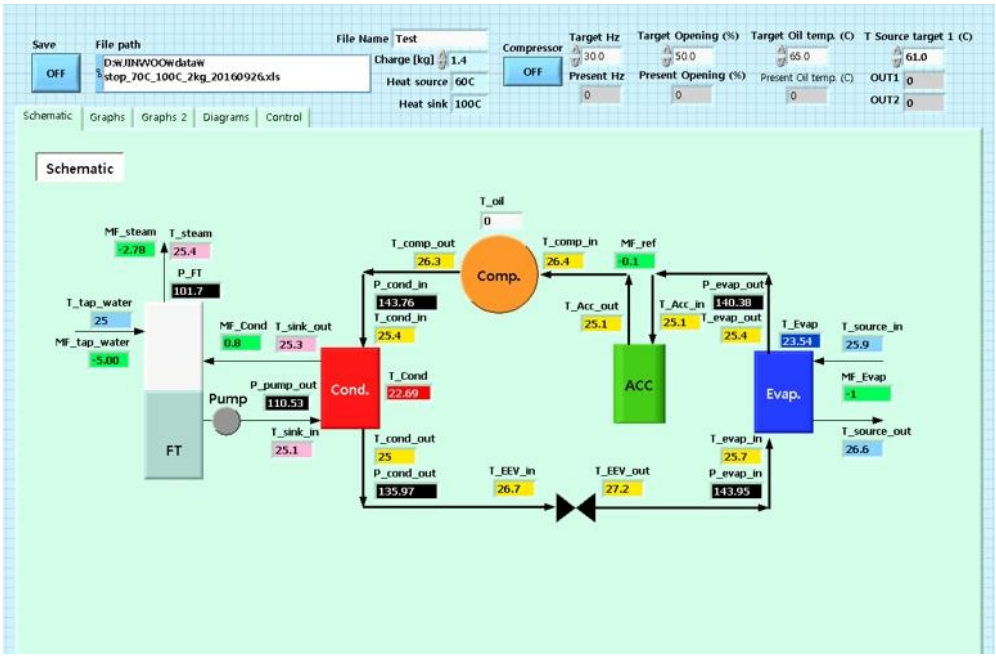

Fig 2.3 Real time estimation of variables in LABVIEW program

Fig 2.4 Control panel of LABVIEW program

Table 2.2 Experimental conditions of SGHP basic cycle

Variable	Value
Compressor frequency (Hz)	40
Steam temperature (℃)	120
Flash tank pressure (bar)	2
Flow rate of 2nd fluid at evaporator (g/s)	1100
Flow rate of 2nd fluid at condenser (g/s)	760
Refrigerant charge (kg)	1.0 ~ 1.6
Waste heat source temperature (℃)	50, 60 ,70
Suction DSH (℃)	12, 15, 18

2.3 Experimental result

2.3.1 Characteristics of SGHP with waste heat source temperature

Fig. 2.3 and Fig. 2.4 show COP and heating capacity of SGHP basic cycle with various refrigerant charge and heat source temperature at suction DSH of 15℃. The results represent that performance increases as waste heat source temperature increases: COP is 1.73, 2.07, 2.48 and heating capacity is 4.34, 6.65, 9.69 kW at each optimal charge. These values represent that SGHP basic cycle has nice performance to replace the boiler for generating steam.

Some features of SGHP cycle are confirmed through pressure-enthalpy diagram of Fig. 2.5. First, Compression ratio, which is defined from Eq. (2.7), is very high compared to other heat pump cycle because temperature difference of secondary fluids of condenser and evaporator is 50~70℃; compression ratio is about 5 as source temperature is 70℃ and even about 10 as the temperature is 50℃. So the way to decrease the compression ratio is necessary.

$$Compression\ ratio = \frac{condenser\ pressure}{evaporator\ pressure} \tag{2.7}$$

Secondly, evaporator pressure increase of 207, 285, 388 kPa as the source temperature increase. It cause the compression ratio and compressor work to decrease. Plus, heat transfer performance and waste heat recovery rate also high at high source temperature because the refrigerant quality at evaporator inlet after expansion process,

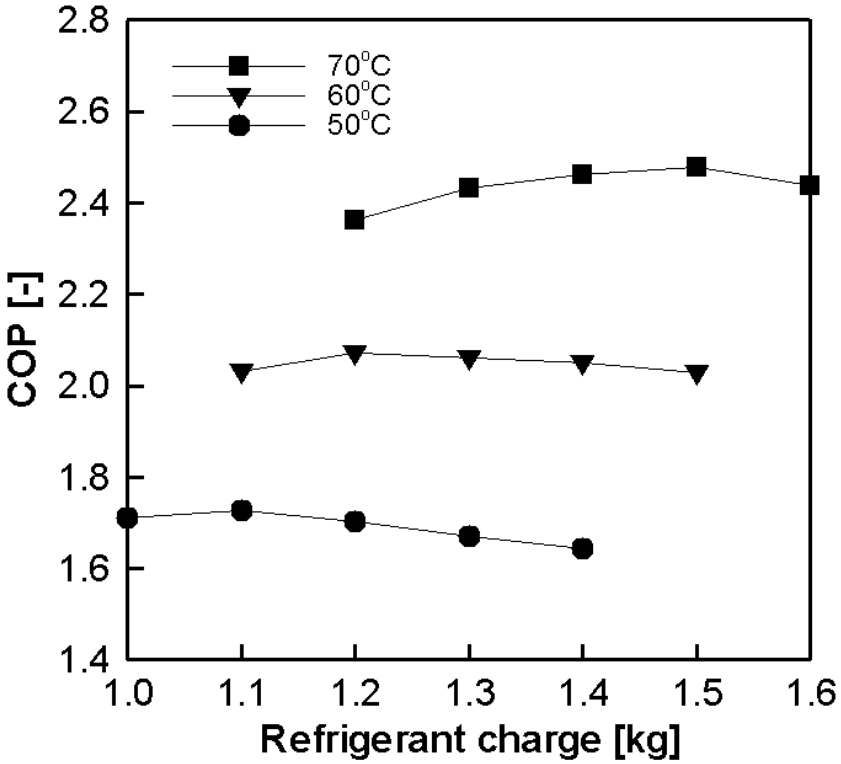

Fig. 2.5 COP with respect to refrigerant charge for various waste heat source temperature at DSH 15℃ and constant mass flow rate of waste heat source

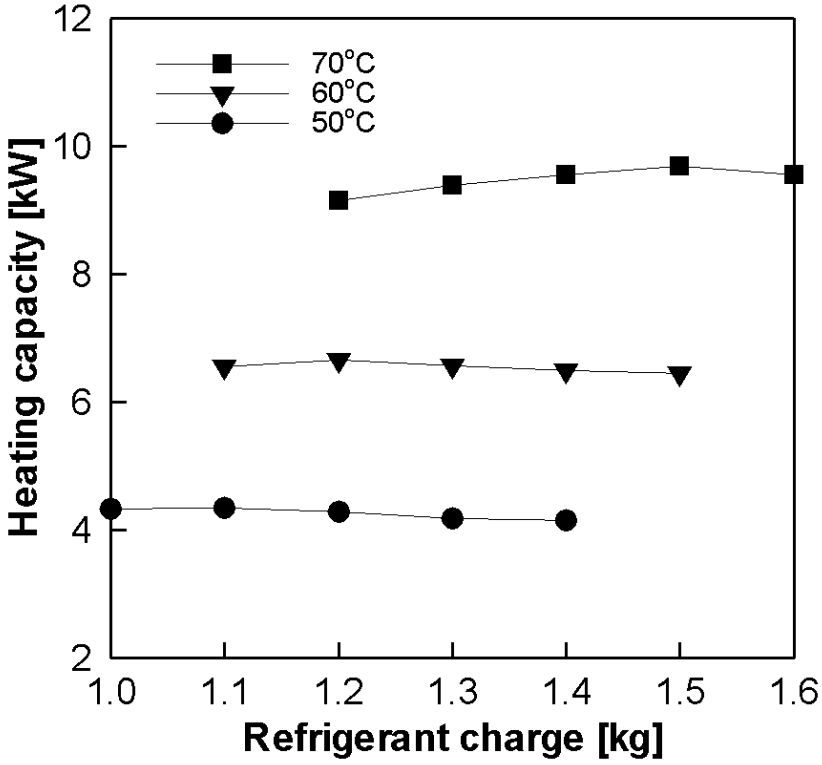

Fig. 2.5 Heating capacity with respect to refrigerant charge for various waste heat source temperature at DSH 15℃ and constant mass flow rate of waste heat source

which is calculated from Eq. (2.8), get lower as the evaporator pressure increase.

$$x = \frac{h_{Eva,in} - h_{sat,liq}}{h_{sat,vap} - h_{sat,liq}} \tag{2.8}$$

Thirdly, SGHP needs high DSH for safe operation by preventing liquid state refrigerant from entering to compressor. SGHP needs more DSH than air source heat pump which needs about 5℃ because the saturation vapor line of the refrigerants operating in high temperature condition changes sharply at low pressure.

There is another variable which is not possible to see by P-h diagram. Fig. 2.6 shows that mass flow rate also increases as the source temperature increase. Because mass flow rate is related to the specific volume at compressor inlet by eq. (2.9), high pressure refrigerant at higher source temperature has bigger mass flow rate. Here, η_v is volumetric efficiency of compressor and PD is piston displacement per second. Increasing of refrigerant flow rate cause heating and cooling capacity to increase. Also, steam generation rate is calculated by eq. (2.10). Because the amount of tap water supplying is exactly same to the steam mass flow rate, steam generation rate is calculated.

$$\dot{m}_{ref} = \eta_v \cdot \frac{PD}{v_{comp,in}} \tag{2.9}$$

$$\dot{m}_{steam} = \dot{m}_{ref}(h_{in} - h_{out}) \ / \ (h_{steam,out} - C_{p,water}T_{in}) \tag{2.10}$$

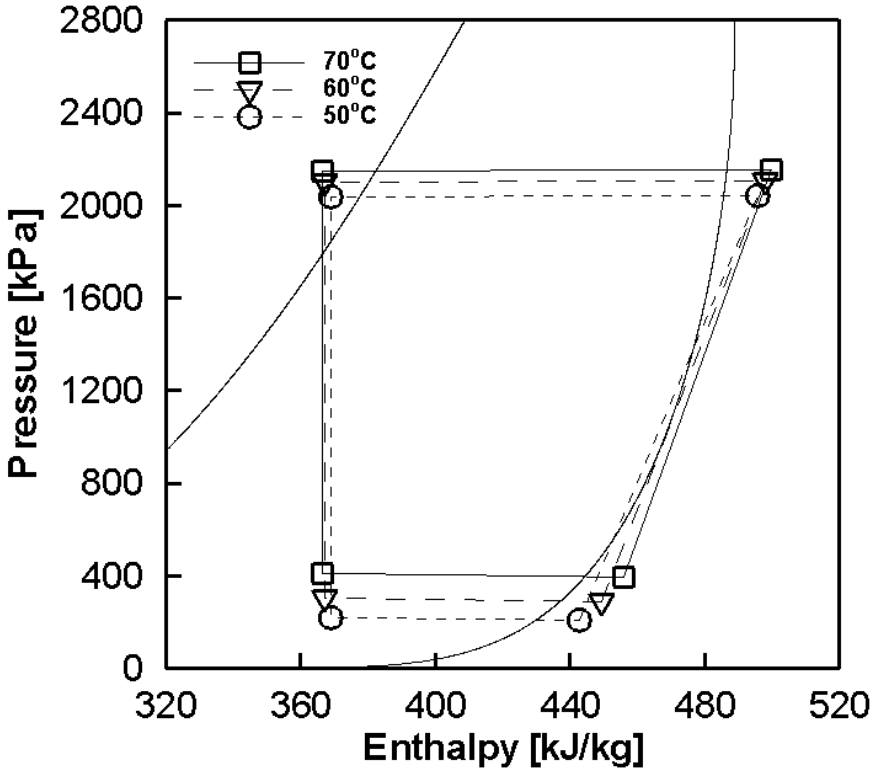

Fig 2.7 Pressure-enthalpy diagram of SGHP basic cycle for various waste heat source temperature at refrigerant charge of 1.3 and DSH of 15℃

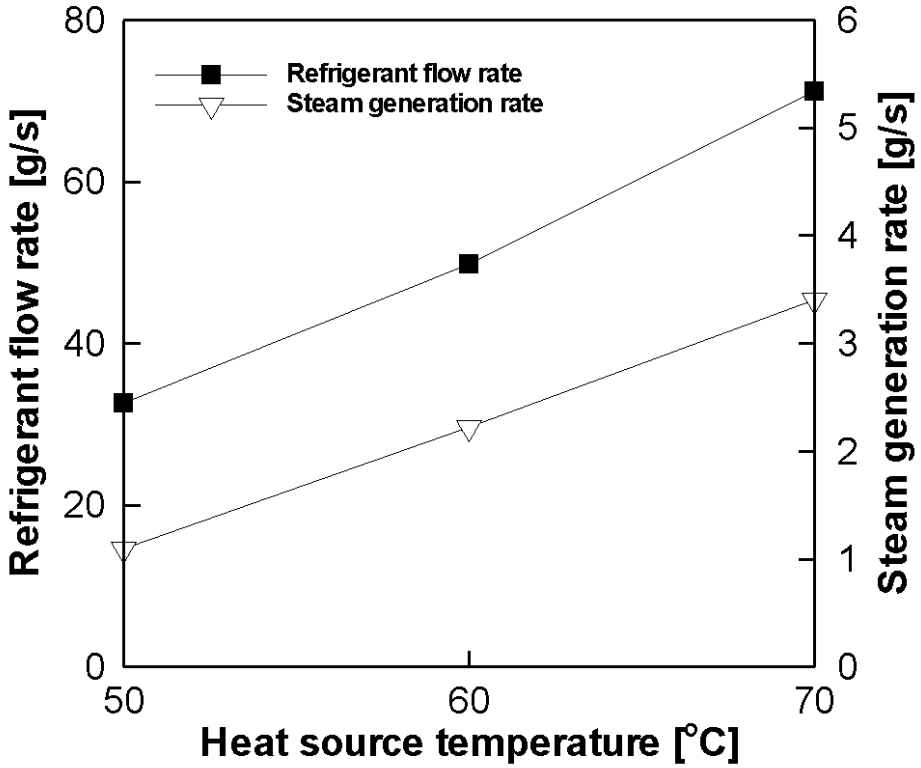

Fig. 2.8 Refrigerant mass flow rate and steam generation rate with respect to waste heat source temperature at DSH 15℃

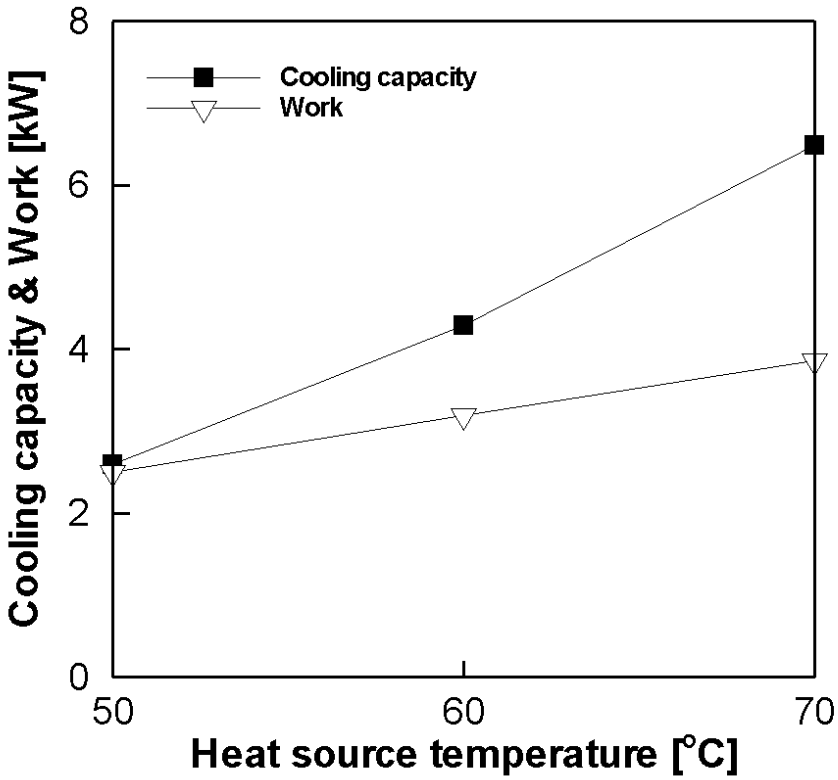

Fig. 2.9 Cooling capacity and compressor work with respect to waste heat source temperature at DSH 15℃

Increase of refrigerant flow rate cause cooling capacity and compressor work to increase as well in Fig. 2.7. Cooling capacity which is waste heat recovery rate increase as the source temperature increase. By the result, COP get increase because compressor work increase less than steam generation rate and cooling capacity.

2.3.2 Performance with refrigerant charge

There is optimal charge at each source temperature. When the refrigerant are charged less or much more than optimal charge, COP degradation happens. The results are related to condenser pressure. As Fig. 2.8, condenser pressure of the cycle increase when the refrigerant charge increase. Fig. 2.9 shows degree of sub-cooling increases as the charge increase because saturation temperature of refrigerant at condenser increases, which contribute to performance of heat transfer. Heat transfer rate and DSC is calculated from Eq. (2.11) and Eq. (2.12).

$$Q_{\boldsymbol{h}} = UA(T_{ref} - T_{steam}) \tag{2.11}$$

$$DSC = T_{sat,liq} - T_{Eva.out} \tag{2.12}$$

However, when refrigerant is overcharged, compressor work get too high. That's the reason that optimal charge exists. And the optimal charge varies as the source temperature change because refrigerant density at some points changes due to difference of evaporating pressure. Optimal charge is 1.1, 1.2, 1.5kg each.as the source temperature is 50, 60, 70℃.

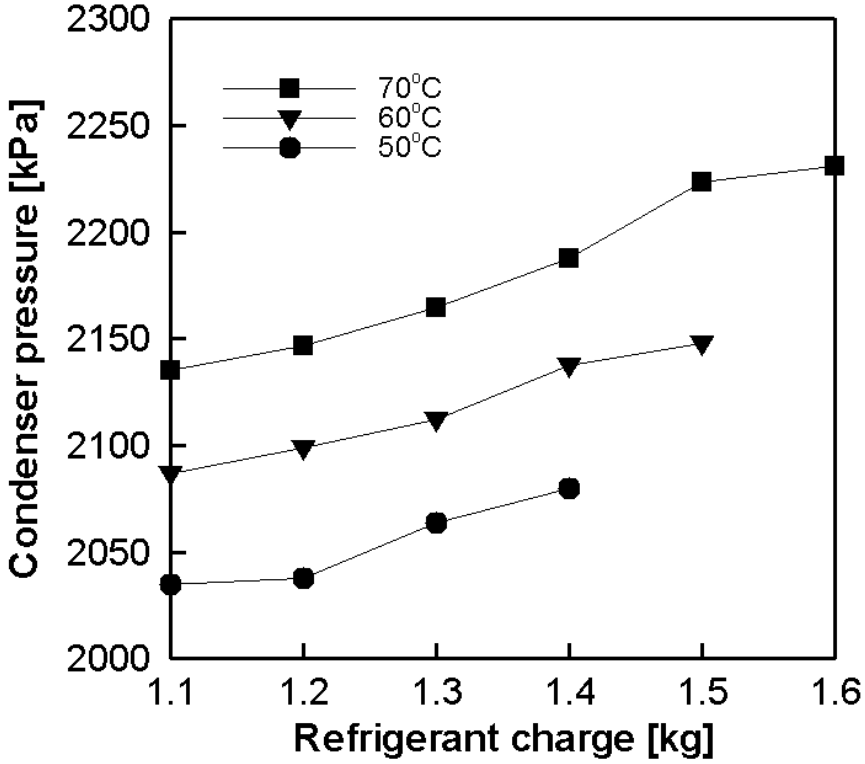

Fig 2.10 Condenser pressure with respect to refrigerant charge for various heat source temperature at DSH of 15℃

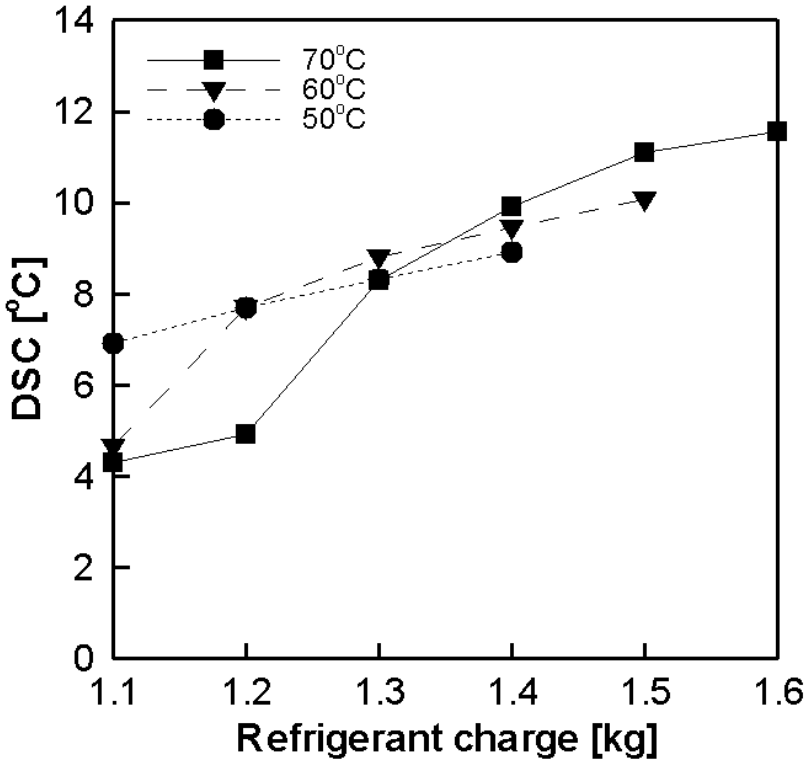

Fig 2.11 Degree of sub-cooling with respect to refrigerant charge for various heat source temperature at DSH of 15℃

2.3.3 Performance with degree of superheat

Fig. 2.10 shows the variation of COP and heating capacity with respect to degree of superheat which is defined from Eq. (2.13). Although high DSH is necessary at SGHP cycle, COP decreases as DSH increase. It is because as DSH increases, evaporating pressure decreases in Fig. 2.11. Refrigerant temperature at outlet must be less than heat source inlet temperature, causing saturation temperature to decrease by Eq. (2.14).

$$DSH = T_{comp,in} - T_{sat} \tag{2.13}$$

$$Q = UA(T_{waste} - T_{ref}) \tag{2.14}$$

So compressor work increase and the quality of refrigerant at evaporator inlet increase. Additionally Fig. 2.12 shows refrigerant mass flow rate also decrease. Therefore, it is good for performance to have the lowest DSH without liquid refrigerant inflow into the compressor. That DSH is calculated to 18, 15, 12℃ at the source temperature of 50, 60, 70℃ by various experiment cases.

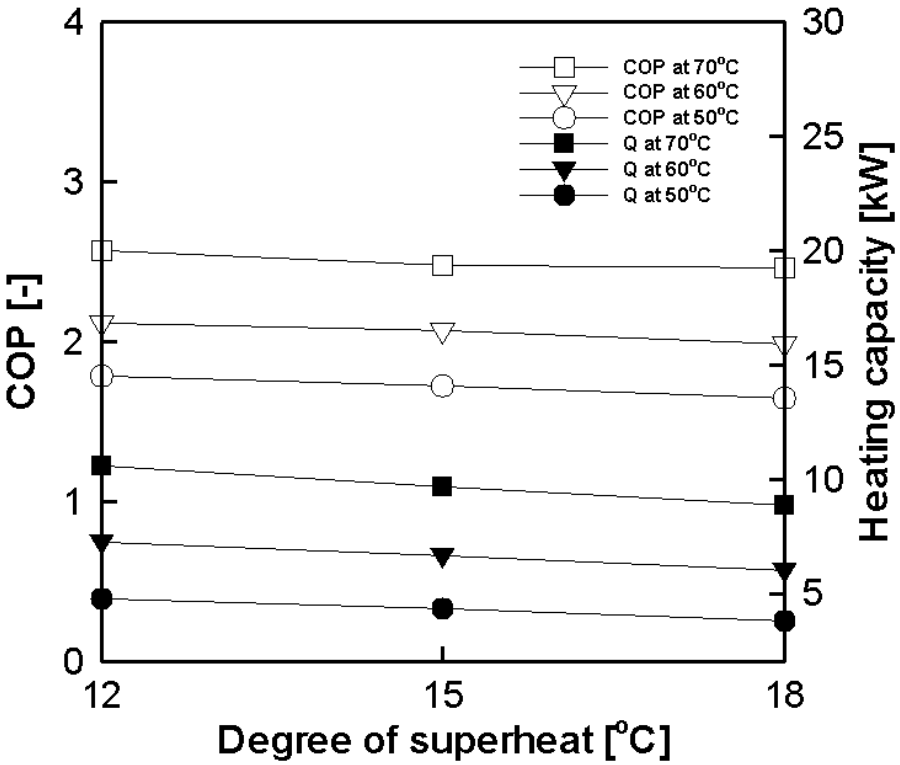

Fig. 2.12 COP and heating capacity with respect to DSH for various waste heat source temperature

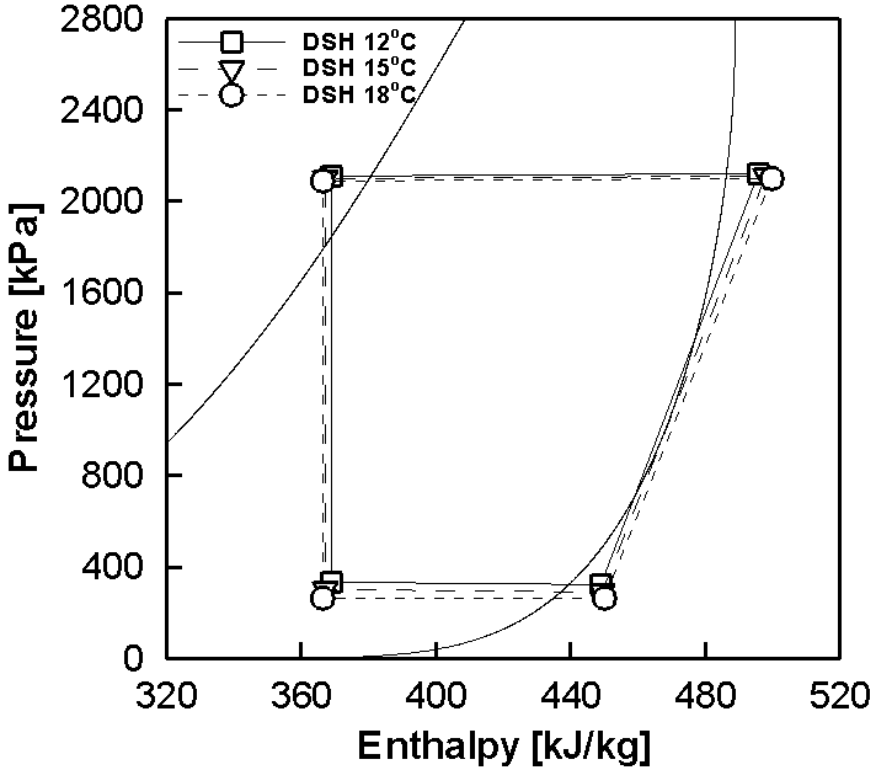

Fig. 2.13 Pressure-Enthalpy diagram for various DSH at waste heat source temperature 60℃

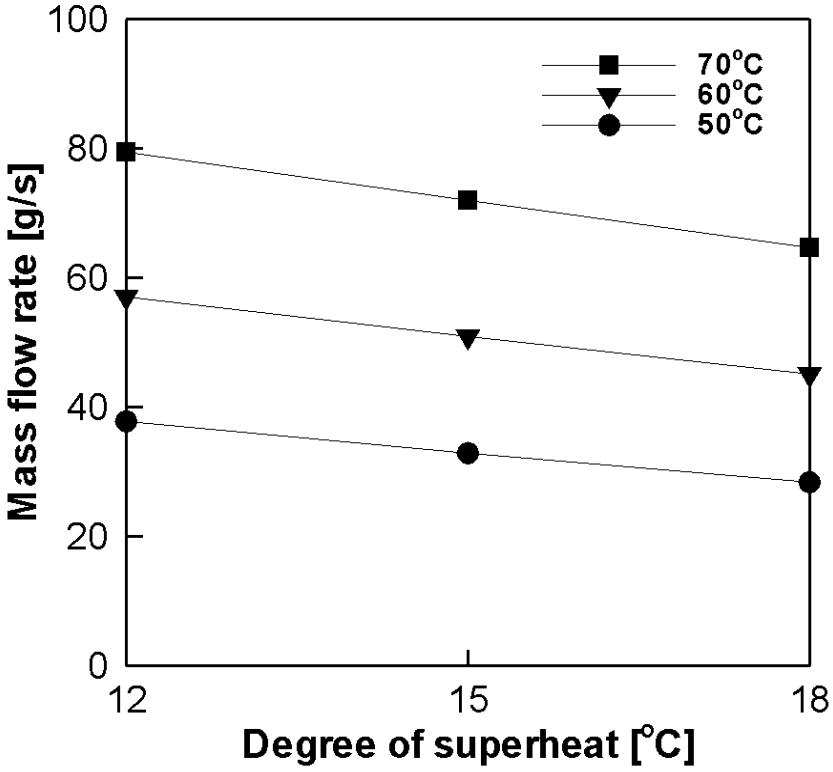

Fig. 2.14 Refrigerant mass flow with respect to DSH for various waste heat source temperature

2.4 Summary

Several features of SGHP cycle was investigated through the experiments of SGHP basic cycle. The performance of cycle changes according to DSH, waste heat source temperature and refrigerant charge were evaluated. There is optimal charge at each SGHP cycle because heating capacity depends on condenser pressure. Performance improvement was also confirmed as the source temperature increase. And there are proper DSH at each source temperature, which has better efficiency without liquid refrigerant inflow to compressor. The results show that performance of cycle is quite nice to replace the boiler for generation steam.

Chapter 3. Research on Performance of SGHP with Suction/Liquid Line Heat Exchanger

3.1 Introduction

The characteristics of SGHP basic cycle were analyzed at chapter 2. However, some disadvantages causing performance degradation were discovered. The compression ratio of SGHP basic cycle is too high because evaporator pressure decrease in the process of obtaining high DSH as Fig 3.1. In case of R245fa, evaporator pressure decrease of about 170 kPa when 15 ℃ DSH is obtained compared to 0 ℃ DSH. Also, sharp change of saturation vapor line of R245fa at low pressure cause the quality of refrigerant increase. So heat transfer performance decrease because the region of latent heat decrease and refrigerant temperature increase at superheated region.

These problems can be solved by adding SLHX. Because heat of high temperature refrigerant at condenser outlet are transferred to low temperature refrigerant at evaporator outlet, DSH are obtained without degradation of evaporator pressure and DSC are additionally obtained in Fig. 3.2.

In this chapter, the experiments of SGHP cycle applied by SLHX are conducted in order to investigate the performance improvement and the reason of improvement are analyzed by pressure-enthalpy diagram and some variables. Also, proper guideline for applying SLHX to SGHP cycle is suggested by analyzing performance change according to refrigerant state at evaporator outlet and DSH at compressor inlet.

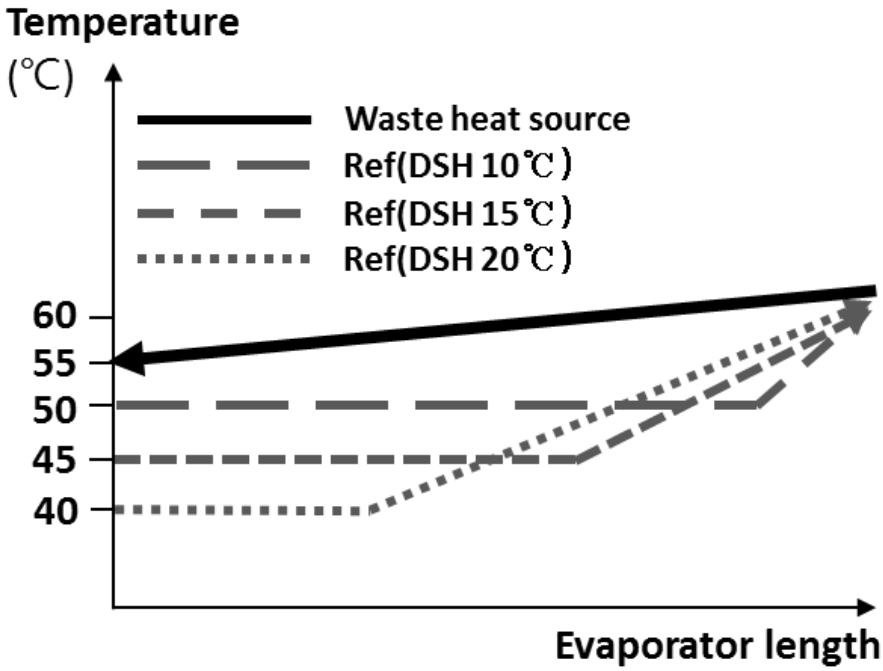

Fig. 3.1 Saturation temperature and pressure decrease at evaporator as degree of superheat increase

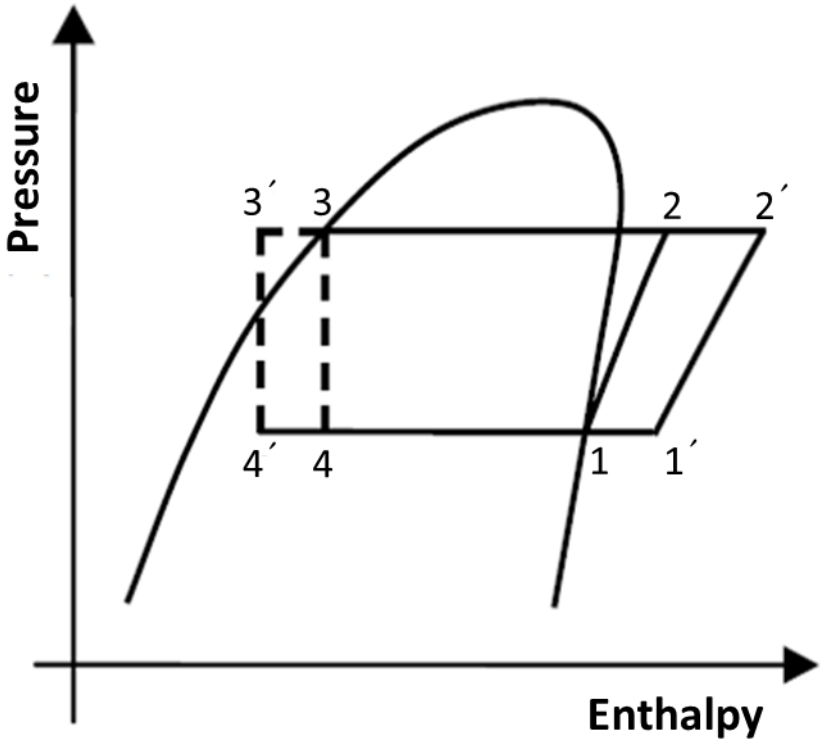

Fig. 3.2 The effect of suction-line/liquid-line heat exchanger in pressure-enthalpy diagram

3.2 Experimental method

Fig. 3.3 shows SLHX is newly added to the experimental equipment of SGHP basic cycle. Two valves are also added after evaporator outlet in order to control the mass flow rate to the SLHX. All variables are same but main variables: Refrigerant charge is 1.8 to 3.6 kg, waste heat source temperature is 50, 60, 70℃, and especially DSH is 12, 15, 18, 21, 24 in order to see the state of evaporator outlet and compressor inlet in detail. By increasing the DSH, it is possible to know when the state of evaporator outlet would be superheated vapor state. Revised experimental conditions are at Table. 3.1.

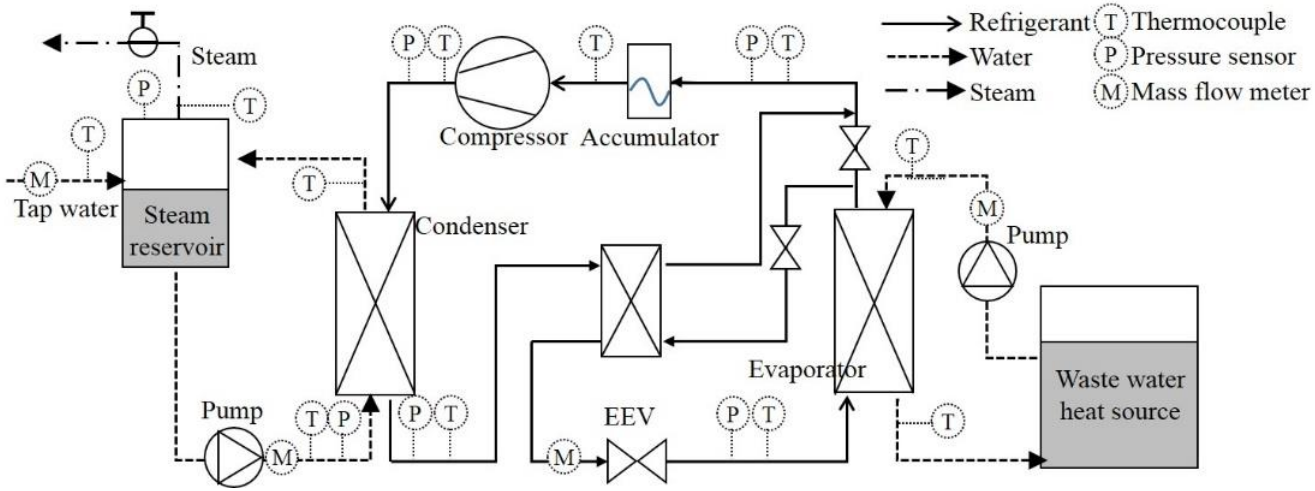

Fig. 3.3 Schematic of SGHP cycle with suction-line/liquid-line heat exchanger

Table 3.1 Experimental conditions of SGHP cycle with SLHX

Variable	Value
Compressor frequency (Hz)	40
Steam temperature (℃)	120
Flash tank pressure (bar)	2
Flow rate of 2nd fluid at evaporator (g/s)	1100
Flow rate of 2nd fluid at condenser (g/s)	760
Refrigerant charge (kg)	1.8 ~ 3.6
Waste heat source temperature (℃)	50, 60 ,70
Suction DSH (℃)	12, 15, 18, 21, 24

3.3 Experimental result

3.3.1 Performance with waste heat source temperature

Fig. 3.4 and Fig. 3.5 show COP and heating capacity of SGHP cycle with and without SLHX for various refrigerant charge at each source temperature. Table. 3.1 is the result of COP and heating capacity. COP increase about 0.4 compared to basic cycle. Especially heating capacity has considerable increase. For example, heating capacity at heat source temperature of 60 is bigger than the capacity of basic cycle at heat source temperature of 50℃. Compared to basic cycle, heating capacity increase 62% at waste heat source temperature of 70℃ and 38% at 50℃.

The performance improvement results from several reasons. First, evaporator pressure increases. Fig. 3.6 and Fig 3.7 is pressure-enthalpy diagram at 50℃ and 70℃. As evaporator pressure increase from 207 to 295 kPa, compression ratio decrease. Secondly, quality of the refrigerant decrease by additional cooling before expansion, causing enthalpy difference at evaporator and heat transfer coefficient increase. Thirdly, refrigerant mass flow rate also increases as Fig 3.8., causing heating capacity increase. In Fig 3.9, cooling capacity and COP increase as well, causing increase of COP.

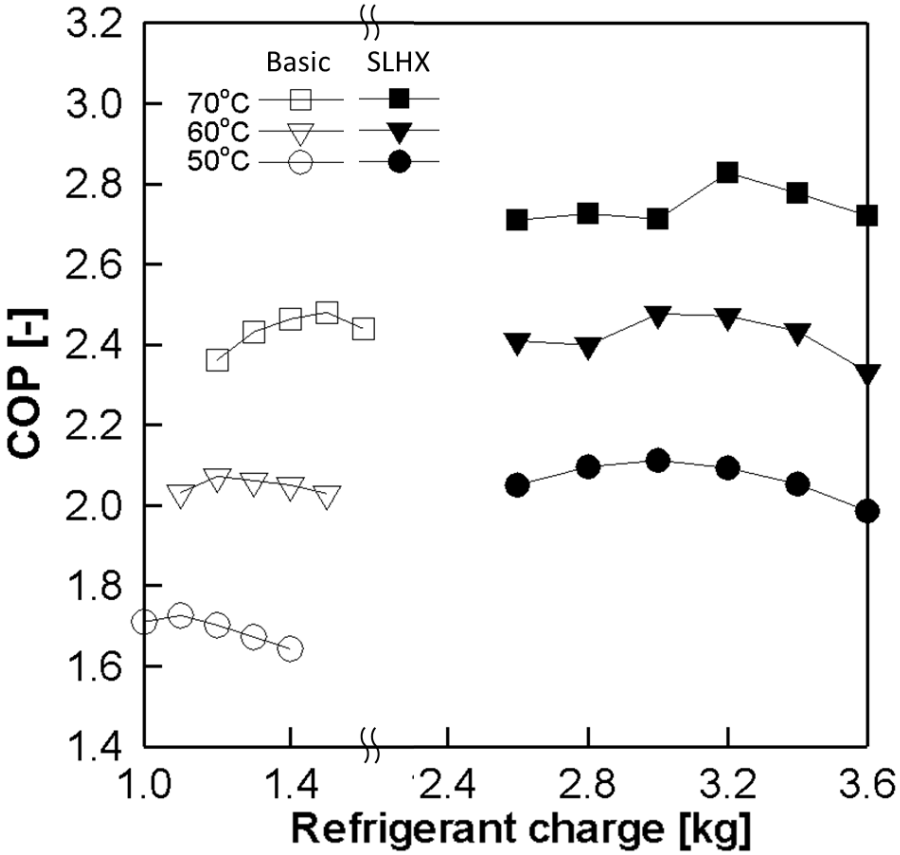

Fig. 3.4 COP with respect to refrigerant charge for various waste heat source temperature at DSH of 18℃ and constant mass flow rate of waste heat source

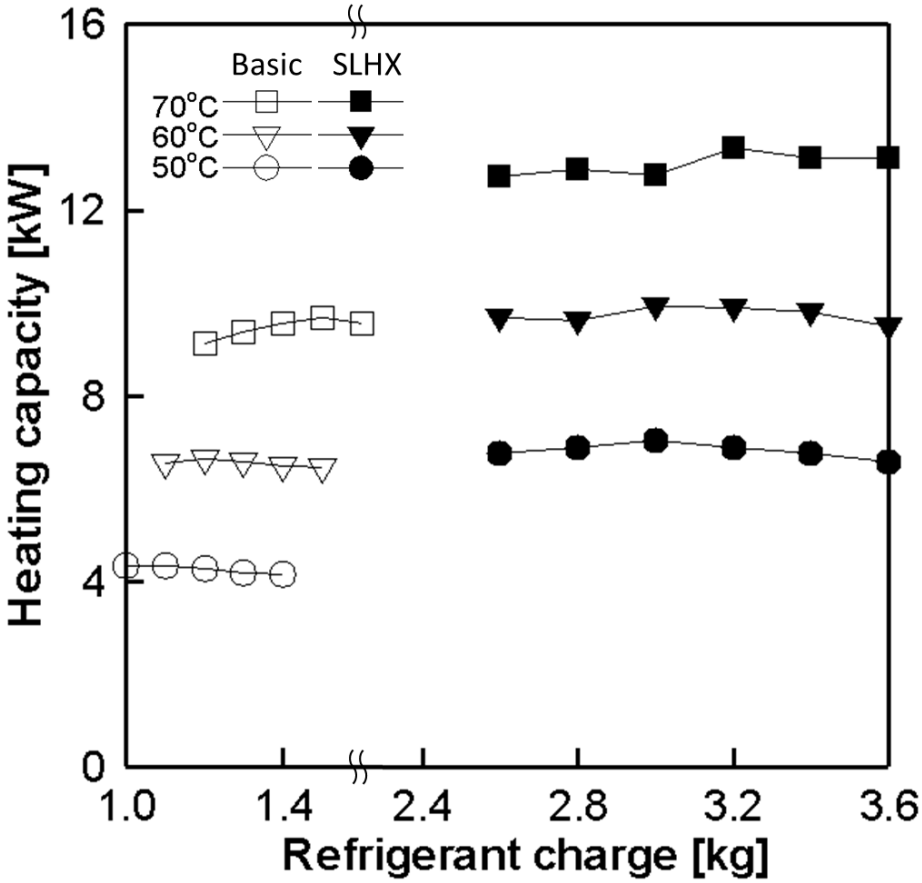

Fig. 3.5 Heating capacity with respect to refrigerant charge for various waste heat source temperature at DSH of 18℃ and constant mass flow rate of waste heat source

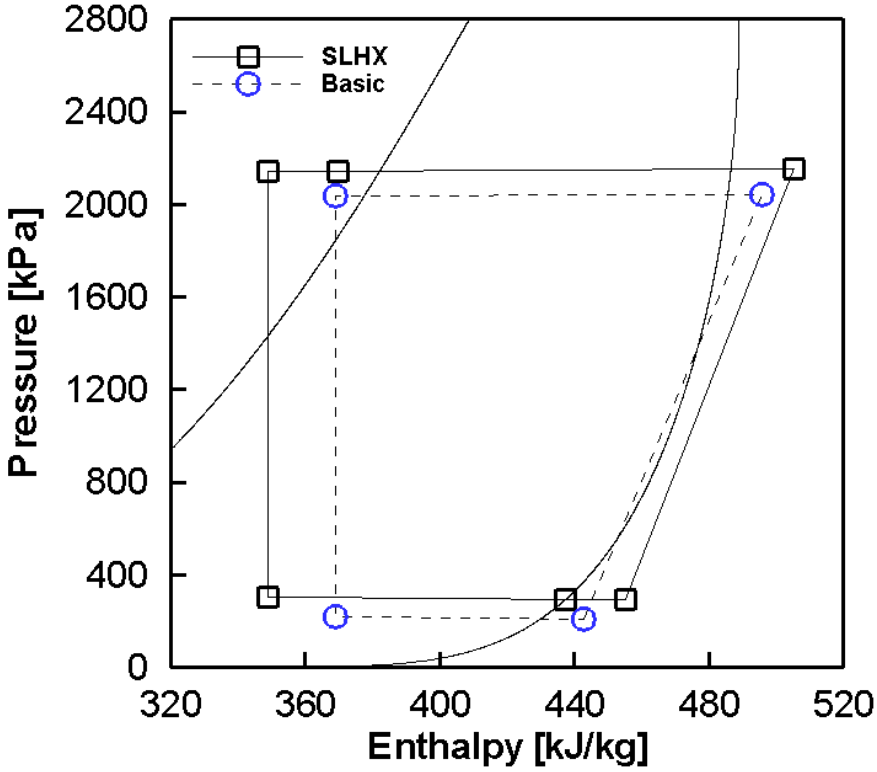

Fig. 3.6 Pressure-enthalpy diagram comparison of SGHP cycle with and without SLHX at waste heat source of 50℃

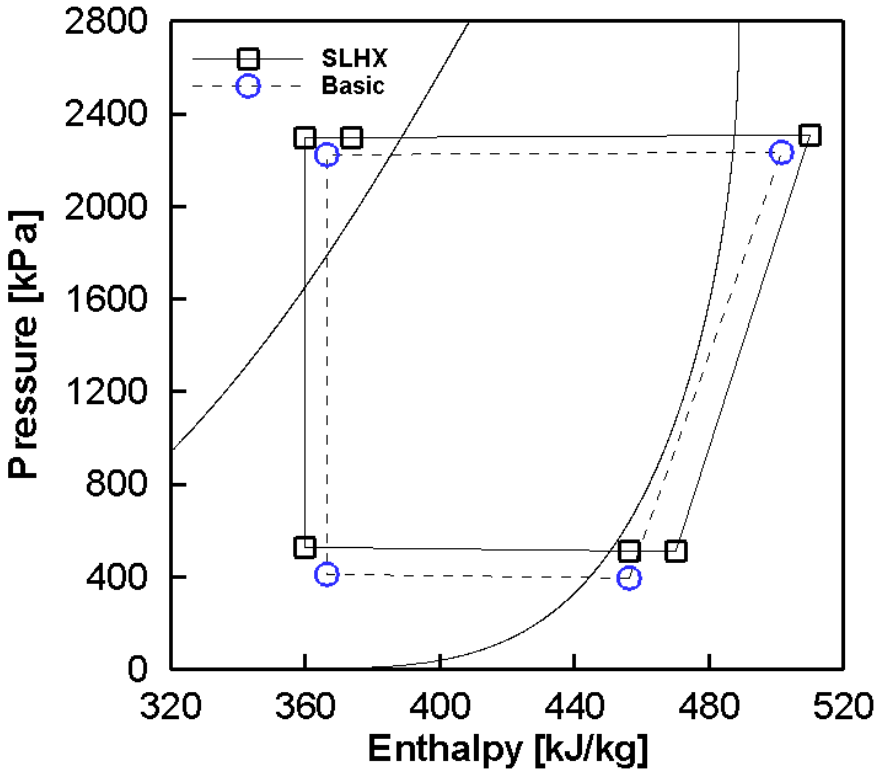

Fig. 3.7 Pressure-enthalpy diagram comparison of SGHP cycle with and without SLHX at waste heat source of 70℃

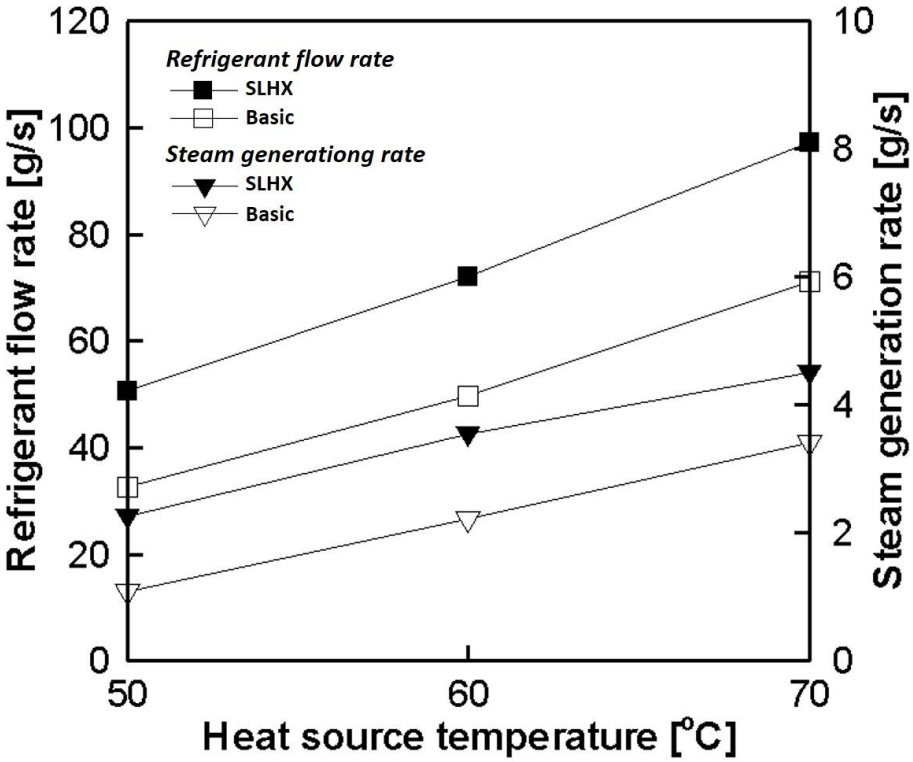

Fig. 3.8 Refrigerant mass flow rate and steam generation rate of SGHP cycle with respect to waste heat source temperature

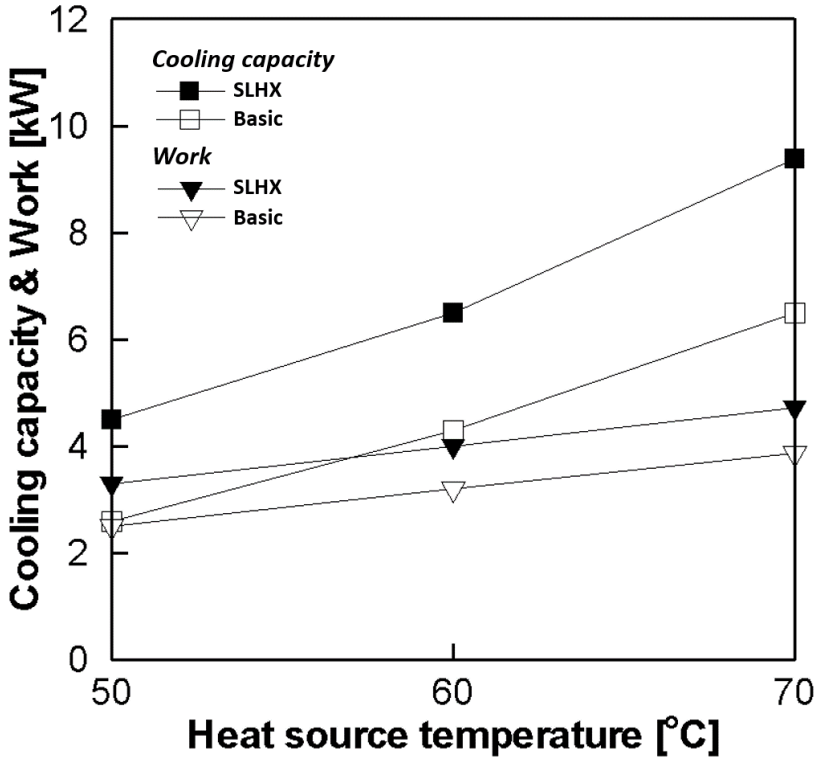

Fig. 3.9 Cooling capacity and compressor work of SGHP cycle with respect to waste heat source temperature

3.3.2 Performance with refrigerant charge

There is also optimal charge at SGHP cycle with SLHX. As Fig. 3.10, condenser pressure increases as the charge increases. And it goes through sudden pressure increases when refrigerant is charged more than optimal charge. Consequently, heating capacity increases due to the amount of heat transfer and DSC. However, COP decreases because compressor work get too high. On the contrary when refrigerant charge is too small, DSC is not obtained enough, resulting in decrease of heating capacity. Because specific heating capacity and heat transfer coefficient is low, COP decreases even if specific compressor work is less than optimal charge. That`s why the optimal charge exists.

3.3.3 Performance with degree of superheat

Fig. 3.11 is the COP with various DSH at source temperature of 60℃. COP of basic cycle decrease as DSH increase. However, SGHP cycle with SLHX has optimal DSH at which COP is maximum. This characteristics is related to the state of refrigerant at evaporator outlet. According to Fig. 3.12, as DSH at compressor inlet increase, quality at evaporator outlet also increase. The refrigerant is two phase state until DSH is 18℃, and superheated vapor state at DSH of 21 and 24℃. So there is no evaporator pressure decrease until DSH is 18℃. However, when DSH is lower than optimal DSH, enthalpy difference and heat transfer performance decrease at evaporator, causing specific cooling

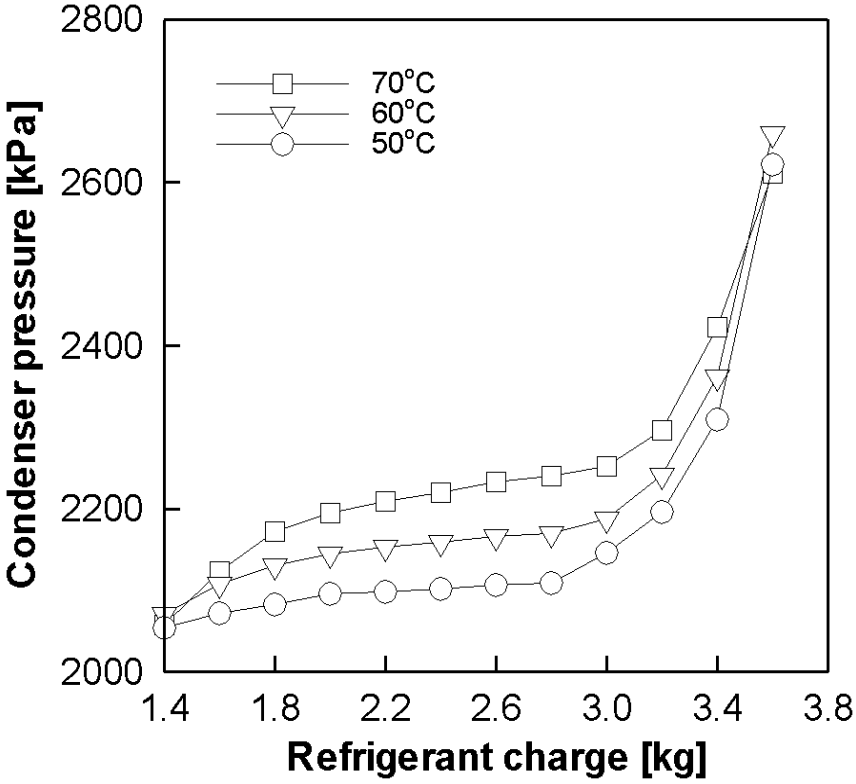

Fig. 3.10 Condenser pressure with respect to refrigerant charge for various waste heat source temperature at DSH of 18℃

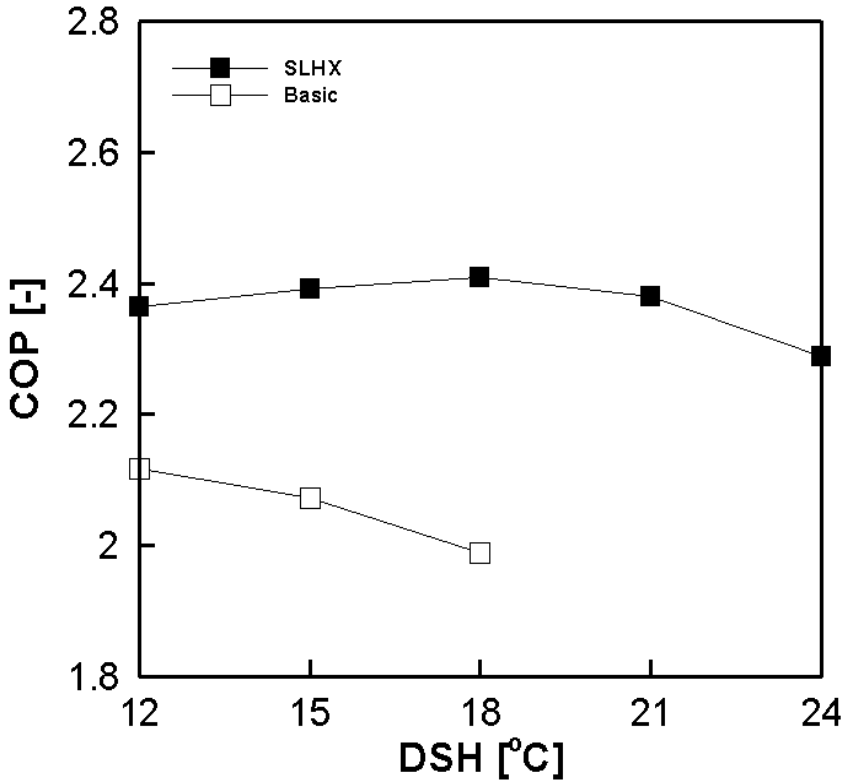

Fig. 3.11 COP comparison of both cycles with respect to DSH at waste heat source temperature of 60℃

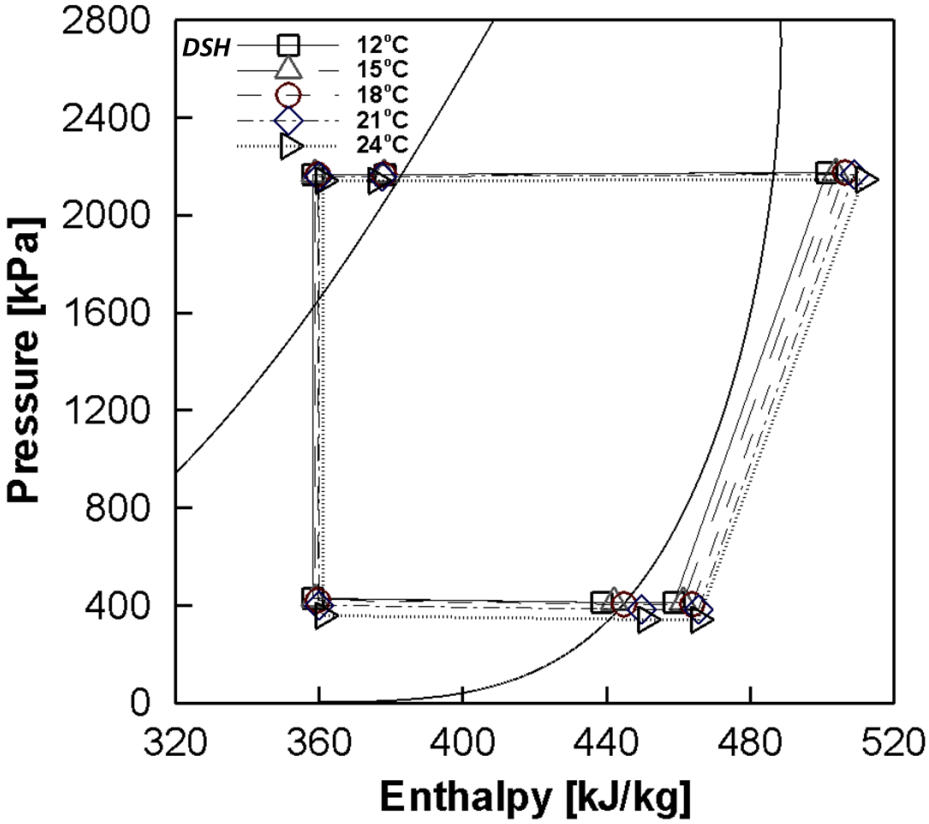

Fig. 3.12 Pressure-enthalpy diagram of SGHP cycle with SLHX at waste heat source temperature of 60℃

capacity decrease.

On the contrary, when DSH is higher than optimal DSH, there is same problem of basic cycle, decrease of evaporator pressure. Therefore, optimal DSH contributed by the refrigerant state at evaporator outlet is near saturation vapor state.

There is also degradation of mass flow rate as DSH increase. Therefore, by combination of several factor such as refrigerant state at evaporator outlet, mass flow rate, and compressor work, optimal DSH is determined.

3.3.4 Suggestion for applying SLHX

When SLHX is applied at SGHP cycle, various capacity of SLHX can be used. If the size of SLHX is too small compared to the cycle capacity, SLHX is not able to cover whole needed DSH, causing decrease of low pressure and less effect by SLHX. On the contrary, too big size SLHX cause increase of refrigerant temperature at high and low pressure, which is bad effect. Also the refrigerant quality at evaporator outlet is very low even if DSH at compressor inlet is obtained enough. That is, enthalpy difference of refrigerant at evaporator is very low compared to the enthalpy difference at compression process, resulting in COP decrease. Because heat is absorbed enough at low pressure side of SLHX, the capacity of condenser and evaporator can be less than the capacity of SLHX. Consequently, steam generation rate and heat recovery rate from waste heat source could decrease. Therefore, proper size of SLHX is necessary to be applied.

The important point is evaporator outlet. It is efficient to set the state of evaporator outlet around saturation vapor state. For example, if the capacity of SLHX is higher than this experiment case, DSH should be also higher in order to make the refrigerant at evaporator outlet be around saturation gas state. And if there is some problems due to high DSH such as decrease of mechanical efficiency or limitation of temperature, SLHX should be replaced to the smaller one.

3.4 Summary

Applying SLHX was considered as a method of improving performance of SGHP cycle. After applying SLHX to the experimental equipment of SGHP basic cycle, experiments were conducted at the same experimental conditions of basic cycle. SLHX covering the whole DSH which is necessary in order to prevent compressor, evaporator pressure increases about 100 kPa, causing decrease of compression ratio and specific compressor work, and increase of heating capacity and cooling capacity due to mass flow rate increase. Also, because DSC before expansion gets larger due to additional heat transfer at SLHX, refrigerant enthalpy at evaporator inlet decreases, causing heat recovery rate and whole performance increase. COP increases of about 0.4.

Performance according to refrigerant charge is also investigated. As the charge increase, condenser pressure also increase, causing more heat transfer at condenser, and increase of specific heating capacity which is enthalpy

difference. However, the pressure increase sharply as the refrigerant is over charged, the increase of compressor work beat the good effect of pressure increase, causing COP decrease. That`s the reason of existence of optimal charge. Guideline for applying SLHX to SGHP is also suggested. There is the optimal DSH at which specific cooling capacity is large enough without decrease of evaporator pressure. The DSH is related to the refrigerant state at evaporator outlet, saturation vapor state.

Chapter 4. Conclusion

In this study, another method of generating steam was suggested, steam generation heat pump. Because of high efficiency and many advantages, SGHP can replace the boiler in economical view. With COP of 2.83 at the source temperature of 70℃, the lab scale SGHP cycle could save the energy for generating steam. If higher source temperature is used and compressor frequency increases, COP and heating capacity would be improved much more.

Because of high temperature condition, basic characteristics are different to other heat pump cycle. Therefore, those distinctive features were confirmed and analyzed by conducting the SGHP cycle, according to variables of refrigerant charge, waste heat source temperature and DSH. SGHP has much better performance at high source temperature. Because the saturation vapor line is sharply changed at low pressure, latent heat region get shorter and high DSH is needed for preventing compressor. So high temperature waste heat source is better to be used at the process. If the source temperature is low, it would be better to heat the heat source before being used at SGHP or to generate the less temperature steam. It is also important to get optimal charge which has enough heating capacity and proper compressor work.

By adding SLHX to SGHP basic cycle, performance get improved a lot. At each temperature, COP increase of about 0.4 and heating capacity of 38~62%. If SLHX is applied to SGHP, the tendency according to DSH get changed. Evaporator pressure increasing, minimum DSH for preventing compressor get lower, so having optimal DSH for highest COP is important.

The DSH is related to the refrigerant state at evaporator outlet, which is saturation vapor state. At the optimal DSH, SGHP has enough cooling capacity without decrease of low pressure.

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국 문 초 록

스팀생산히트펌프는 폐열원을 이용해 약 120도의 산업용 스팀을 만드는 히트펌프로, 공기열원히트펌프를 비롯한 기타 히트펌프 사이클과는 다른 특징을 갖는다. 냉매의 최고온도가 125℃가 넘는 굉장히 높은 온도 조건에서 작동하기 때문에 높은 압축비를 가지며, 응축기의 2차유체가 2상상태인 스팀이므로 열교환기에서의 온도구배 또한 다르다. 이러한 이유에 의해 생기는 기타 히트펌프 사이클과 구별되는 특징을 알아보고자 기본사이클 실험을 수행하고, 기존의 스팀 생산 방식인 보일러를 대체할 수 있을만한 효율을 갖는지 알아본다. 8kW급 스팀생산히트펌프 기본사이클 실험장치를 구성해 다른 조건들은 고정시킨 채 주요 변수인 냉매 충전량, 폐열원의 온도, 과열도를 변화해가며 실험을 수행하였다. 주요 지점에서 온도와 압력, 유량을 측정해 COP와 난방용량, 스팀생산량 등의 결과값을 구했으며, 그 밖의 냉매유량과 압축일, 열 회수량 등의 변수와 압력-엔탈피 선도를 통해 사이클의 성능을 평가하였다.

스팀생산히트펌프 기본사이클의 높은 과열도와 압축비로 인한 성능저하 문제를 해결하기 위해 내부열교환기를 적용 후 기본사이클과 같은 조건에서 실험을 수행해 기본사이클 대비 성능향상을 확인하고 그 이유를 분석하였다. 기본사이클의 문제점을 보완함으로써 0.4의 COP 향상과 37~62%가량의 난방용량 증가를 확인하였다. 또한 압력-엔탈피 선도를 통한 비교를 통해 증발기 압력과 과냉도의 증가 등의 성능 개선 원인을 분석하였다. 또한 내부열교환기를 스팀

생산히트펌프 사이클에 효율적으로 적용하기 위한 방법을 제시한다.

주요어: 스팀생산히트펌프, 폐열 회수, 내부 열교환기

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