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Experimental study of lab-scale steam generation heat pump with waste heat recovery

Jin Woo YOO, Dae Hoon KANG, Min Soo KIM*

Dept. of Mechanical Engineering, Seoul National University, Seoul 08826, Korea

Abstract

Steam is one of the most commonly used energy form in industrial factories for manufacturing processes and it takes about 40% of industrial energy use. After process, liquid water remains as a form of waste heat and it needs to be cooled before exhaust because the temperature of hot liquid water is still high ranging from 50 to 90°C. This cooling process requires additional energy. Introduction of a heat pump system which can generate steam using this waste heat has a positive effect on both additional steam generation and waste heat recovery from hot liquid water. Steam is usually generated through boiler, but the heating capacity of boiler never can exceed the capacity of input energy. On the other hand, heat pump can usually generate several times more heat compared with the input energy. Therefore, in this study, a lab-scale steam generation heat pump system is designed and experimental results are shown in order to verify the performance of the system. It is concluded that coefficient of performance and heating capacity increase 1.25 and 1.8 times respectively as heat source temperature rises from 60 to 80°C. Optimal operational point of the system is found and the performance at this condition was also verified.

Keywords: Steam generation; Heat pump; Wast heat recovery; R245fa; Coefficient of performance

* Corresponding author. Tel.: +82-2-880-8362; fax: +82-2-873-2178.
E-mail address: minskim@snu.ac.kr.

1. Introduction

Steam is one of the widely used energy form in manufacturing facilities and it takes about 40% of total energy use in industrial sector in the United States. [1] Such demand of steam is coming from its diverse application at manufacturing factories such as food processing [2-3], pulp and paper [4], textile [5-6], hydrogen reforming [7], chemicals, petroleum refining, primary metals, cement, electrical equipment, glass products, plastic and rubber product, textiles, etc. After steam is used at these processes, high temperature liquid water (or mixture with contaminant) remains as a form of waste heat and it needs to be cooled before exhaust because the temperature of hot liquid water is still high ranging from 50 to 90°C. This cooling process requires additional energy. The amount of waste heat is large and still has high quality so the thermal energy can be recycled to reduce the primary energy use and CO_2 emission. [8] However, a direct use of waste heat energy is difficult because of possibility of contamination. In this case, the introduction of a heat pump system would have positive effect which can both generate steam recover waste heat from hot liquid water. Steam is usually generated through boiler, but the heating capacity of boiler could never exceed the capacity of the input energy. On the other hand, a heat pump can usually generate several times more heat compared with the input work. Actually, using heat pump for waste heat energy utilization is nothing new. Many studies showed that heat pump is one of the common systems for heat recovery but most studies are limited to hot water generation instead of steam. [6, 9, 10] However, when comparing steam with hot water, steam has several advantages for processes. First, heat transfer coefficient of saturated steam is much higher (several to hundreds of times) than hot water. Second, temperature control is easier because steam provides energy as a form of latent heat thereby temperature of steam is kept constant during heat transfer process. Third, steam does not require additional pump because it naturally moves from high to low pressure side even with small pressure difference. With these advantages, a steam generation heat pump system is more applicable for industrial purpose.

In this study, a lab-scale steam generation heat pump system is designed and experiments were conducted. Results are shown in order to verify the performance of the system at different heat source and charge amount. It is concluded that coefficient of performance and heating capacity increase as heat source temperature rises. Optimal operational point of the system is found and the performance at this condition was also verified.

2. Experimental setup

2.1. Selection of refrigerant and oil

For installation of a lab-scale steam generation heat pump, the selection of refrigerant and oil is required. As operating temperature is different from usual heat pump systems, refrigerant should be selected according to its temperature range. Target steam generation temperature is at least 100°C which is boiling temperature of water at atmospheric pressure. In order to operate a heat pump at subcritical region, critical temperature of refrigerant should be much higher than target temperature. Suitable refrigerants are chosen through following process. First, refrigerant with critical temperature between 150 and 200°C were selected from REFPROP 9.1 [11]. Second, the refrigerants that have non-zero ozone depletion potential or high flammability or toxicity were removed. R245fa and R1234ze(Z) are considered to be suitable for this experiment. R245fa was chosen for steam generation heat pump system.

In case of oil, viscosity is important. The temperature at operating range of compressor is much higher than conventional heat pump so the oil needed to be replaced with one that has proper viscosity value at operating condition. Using proper oil also prevent carbonization or degradation of compressor. The ISO 320 class oil has chosen for the system.

2.2. System description

A schematic of the steam heat pump system is shown in Fig. 1. Its configuration is almost the same as conventional water to water heat pump with mechanical compressor except the steam reservoir. The first role of steam reservoir is to separate saturated steam and water after heat exchange process at condenser. Without steam reservoir, direct steam generation at condenser outlet should be done and this is difficult in control aspect because water may not fully evaporate to steam at condenser outlet. Another reason for use of steam reservoir is to keep steam at saturated state not superheated state because the heat transfer coefficient of saturated steam is much higher than that of superheated steam. With steam reservoir, generated steam could always be at saturated state. During the experiment, produced steam naturally leaves the steam reservoir by pressure difference with ambient pressure. At the same time, tap water enters to steam reservoir in order to keep the level of water at steam reservoir constant. For the performance evaluation thermocouples, pressure sensors and mass flow meters are installed. Temperature is measured at inlet and outlet of each component and pressure sensor is located at inlet and outlet of condenser and evaporator. Mass flow meters were used to measure refrigerant, steam, water mass flow at heat exchangers. Data were saved after the system had reached steady-state.

Specifications of each component (compressor, heat exchangers, expansion device, water pump, steam reservoir) and experimental conditions of the system are shown in Table 1 and 2 respectively. A reciprocating compressor with 4 cylinders was used and two heat exchangers were selected to make this system at the capacity range around 10 kW. For expansion device, electronic expansion valve was applied. Centrifugal pumps were installed at each side (condenser and evaporator). The state of water at condenser side pump inlet is almost saturated state (not exactly saturated because added tap water), thereby cavitation tends to occur. However, through the calculation of available and required net positive suction head (NPSH), the pump was properly installed and there is no cavitation during the operation.

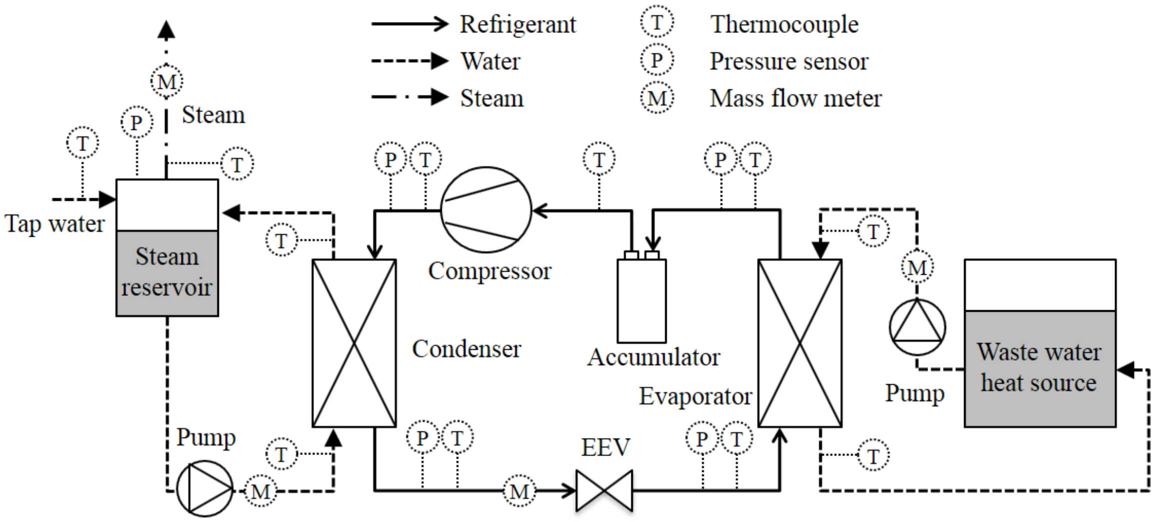

Fig. 1. Schematic of lab-scale steam generation heat pump system

Table 1. Specifications of lab-scale steam generation heat pump

Part	Specifications
Compressor	Reciprocating compressor with 4 cylinders
	Total stroke volume : 308 cm^3
Condenser	Plate heat exchanger with 26 plates
	Width / length / height (mm) : 512 / 124 / 58
Evaporator	Plate heat exchanger with 26 plates
	Width / length / height (mm) : 512 / 124 / 58
Expansion device	Electronic expansion valve
Pump at condenser side	Multistage centrifugal pump
Pump at evaporator side	Centrifugal pump
Steam reservoir	Diameter / height (mm) : 150 / 650

The heat source temperatures of experiment are decided from 60 to 80°C and the sink temperature of the system depends on the operating condition at condenser and steam reservoir. Suction degree of superheat(DSH) of compressor is fixed at 15 K. This value is much larger than conventional heat pump systems which are generally controlled below 10 K. The choice of high suction DSH is related with the characteristic of refrigerant. The temperature-entropy diagram of R245fa is not bell-shape but overhanging type. [12] Many refrigerants have bell-shape temperature-entropy diagram so when suction point of compressor is at superheated state, refrigerant at discharge point automatically becomes superheated state and this is important for safe operation of compressor. On the other hand, overhanging type refrigerant such as R245fa has lower discharge DSH than suction DSH. In order to secure the refrigerant at compressor discharge point at superheated state, high suction DSH is required. Five different charge amount conditions were chosen to find out performance of this system at optimal operational point. Mass flow rate of water at condenser was decided at the condition where cavitation does not occur. During the experiment, refrigerant decomposition does not occur because decomposition temperature of R245fa is around 300°C which is much higher than operating temperature of steam generation heat pump. [13] Maximum temperature in this experiment was 135°C.

Table 2. Experimental conditions

Variable	Value
Type of refrigerant	R245fa
Heat source temperature (°C)	60 / 70 / 80
Suction degree of superheat (K)	15 (All conditions expect one)
	20 (at charge amount = 1.8 kg, T_{source}= 80°C)
Amount of refrigerant (kg)	1.0 / 1.2 / 1.4 / 1.6 / 1.8
Mass flow rate of water at condenser side (g/s)	760
Mass flow rate of water at evaporator side (g/s)	1100
Compressor speed (RPM)	900

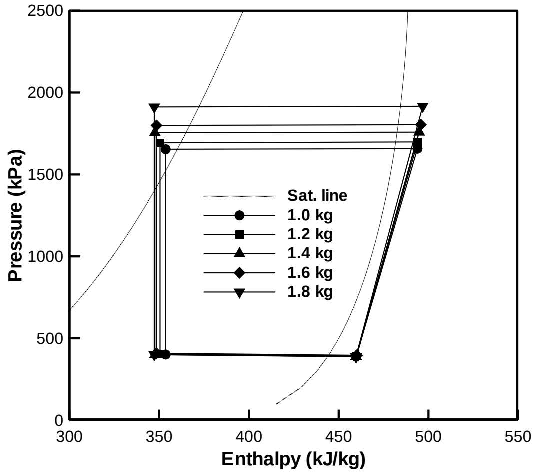

Fig. 2. Pressure-enthalpy various at different charge amount condition (T_{source}=70°C)

3. Results and discussion

System performance with respect to charge amount can be explained with Fig. 2 which represents pressure-enthalpy diagram at the source temperature of 70°C. Evaporating pressure hardly changes mainly because the suction DSH kept constant. On the other hand, condensing pressure increases as charge amount rises. The pressure values at all heat source conditions are shown in Fig. 3. From this graph, it is concluded that at constant suction DSH condition, condensing pressure is highly dependent on charge amount while evaporating pressure show little variation. In Fig. 4, degree of subcool(DSC) becomes higher as charge amount become larger. This is because the pressure increase at condenser makes larger temperature difference between refrigerant and water at condenser which tends to generate higher DSC.

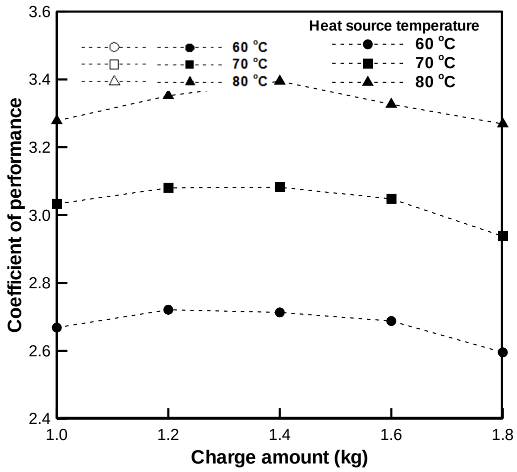

Fig. 3. Charge amount vs. pressure

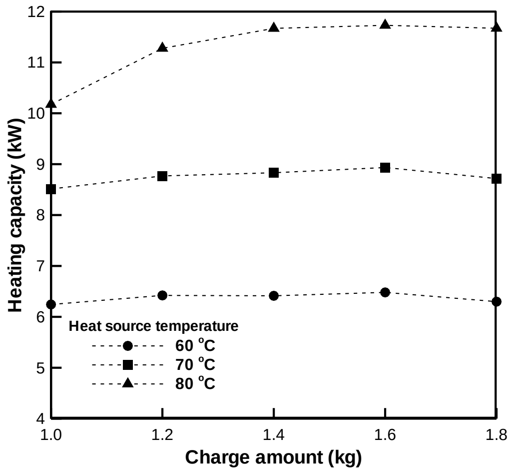

Fig. 4. Charge amount vs. degree of subcool

In order to find optimal operational condition, coefficient of performance(COP) and heating capacity of the system is represented in Figs. 5 and 6 respectively. In the COP aspect, the optimal operational point lies between charge amount of 1.2 and 1.4 kg. This is because maximum COP is shown at 1.2 kg when heat source temperature is 60°C and 1.4 kg at heat source temperature of 70°C and 80°C. Corberan *et al*. (2011) also reported that at lower heat source condition, maximum COP also appears at lower charge amount condition. [14] The maximum COP at 80°C was 3.39 which is about 25% higher than COP at 60°C (2.72). In case of heating capacity, the system has maximum value at charge amount of 1.6 kg at all the heat source temperature in Fig. 6. The heating capacity of the system is the product of mass flow rate of refrigerant and enthalpy difference between inlet and outlet of condenser. In pressure-enthalpy diagram in Fig. 2, the increase of enthalpy difference is clearly checked as charge amount rises. On contrary, mass flow rate of refrigerant in Fig. 7 shows reversible trend which means it tends to drop as charge amount increases. This is due to the pressure variation. A high compression ratio between compressor inlet and outlet usually yields lower mass flow rate. As mentioned earlier, at the same source temperature condition, evaporating pressure does not change while condensing pressure increases when charge amount rises. Therefore, compression ratio increases with charge amount which results mass flow rate decrease. Two different trends make maximum point of heating capacity at 1.6 kg in this experiment. At the condition where charge amount is 1.8 kg and heat source temperature is 80°C, refrigerant mass flow rate is smaller than expected value because suction DSH at this point is 20 K. This happened due to the limit of electronic expansion valve(EEV) opening. When charge amount decreases, the opening value of the EEV tends to increase. Only at this condition, EEV reached the maximum value so the suction DSH is no longer controllable to 15 K. Therefore, the EEV was operated at fully-opened state. Suction DSH was kept at 15 K other than this point. The maximum heating capacity at 80°C was 11.7 kW and it is about 1.8 times larger compare with heating capacity at 60°C which is 6.5 kW.

From these results, the optimal operational point could be decided differently according to the criterion which can be either COP, heating capacity. Additionally, the compressor discharge temperature is another factor that should be into consideration when checking the system characteristics. In Fig. 8, the temperature at compressor discharge points are shown and the trend is similar to condensing pressure that it increases as charge amount rises. Checking compressor discharge point is one of the important task because there is possibility of oil carbonization and refrigerant decomposition. In case of this system, the decomposition of R245fa is 300°C and there are no signs of oil carbonization.

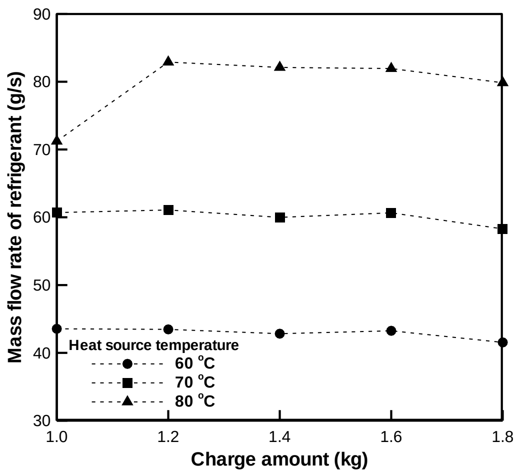

Fig. 7. Charge amount vs. mass flow rate of refrigerant

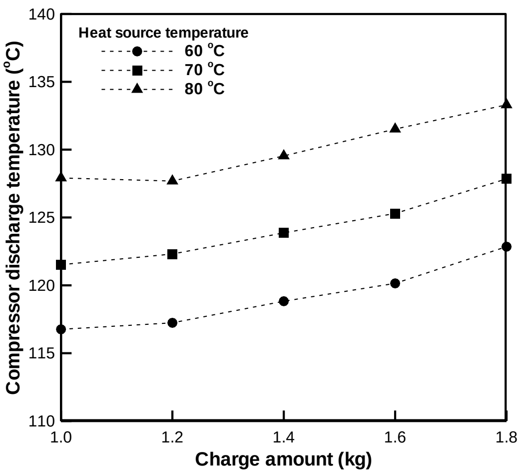

Fig. 8. Charge amount vs. compressor discharge temperature

4. Conclusion

In this study, a lab-scale steam generation heat pump system was designed and experiments were conducted to figure out the characteristics of this heat pump cycle with respect to various charge amount and heat source temperature. The maximum COP was shown at different charge amount with respect to heat source temperature and COP at source temperature of 80°C is 25% higher than that of 60°C. In case of heating capacity, maximum point is shown at the charge amount of 1.6 kg for all the source temperature condition and at 80°C, it is 1.8 times larger compare with the source temperature of 60°C. When deciding the system operational charge amount, COP, heating capacity, and compressor discharge temperature need to take into consideration for both efficient and safe operations.

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References

[1] U.S. Department of Energy. *Improving Steam System Performance: A Sourcebook for Industry*. 2rd ed. Washington: U.S. Department of Energy; 2012.
[2] Nathakaranakule A, Kraiwanichkul W, Soponronnarit S. Comparative study of different combined superheated-steam drying techniques for chicken meat. *Journal of food engineering* 2007;**80**;1023-1030.
[3] Amon R, Maulhardt M, Wong T, *et al*. Waste heat and water recovery opportunities in California tomato paste processing. *Applied Thermal Engineering* 2015;**78**;525-532.
[4] Abrahamsson K, Stenstrom S, Aly G, *et al*. Application of heat pump systems for energy conservation in paper drying. *International Journal of Energy Research* 1997;**21**;631-642.
[5] Filiz Tumen Ozdil N, Tantekin A, Erbay Z. Energy and exergy analyses of a fluidized bed coal combustor steam plant in textile industry. *Fuel* 2016;**183**;441-448.
[6] Wu X, Xing Z, He Z, *et al*. Performance evaluation of a capacity-regulated high temperature heat pump for waste heat recovery in dyeing industry. *Applied Thermal Engineering* 2016;**93**;1193-1201.
[7] Odukoya A, Naterer GF. Upgrading waste heat from a cement plant for thermochemical hydrogen production. *international journal of hydrogen energy* 2014;**39**;20898-20906.
[8] Ammar Y, Joyce S, Norman R, *et al.* Low grade thermal energy sources and uses from the process industry in the UK. *Applied Energy* 2012;**89**;3-20.
[9] Miah JH, Griffiths A, McNeill R, *et al*. Maximising the recovery of low grade heat: An integrated heat integration framework incorporating heat pump intervention for simple and complex factories. *Applied Energy* 2015;**160**;172-184.
[10] Xue B, Meng X, Wei X, *et al*. Dynamic study of steam generation from low-grade waste heat in a zeoliteewater adsorption heat pump. *Applied Thermal Engineering* 2015;**88**;451-458.
[11] Lemmon EW, Huber ML, McLinden MO. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1. National Institute of Standards and Technology, Standard Reference Data Program: Gaithersburg, MD 2013.
[12] Saleh B, Koglbauer G, Wendland M, Fischer J. Working fluids for low-temperature organic Rankine cycles. *Energy* 2007;**32**;1210-1221.
[13] Angelino G, Invernizzi C. Experimental investigation on the thermal stability of some new zero ODP refrigerants. *International journal of refrigernation* 2003;**26**;51-58.
[14] Corberan J, Martinez I, Gonzalvez J. Influence of the source and sink temperatures on the optimal refrigerant charge of a water-to-water heat pump, *International Journal of Refrigeration* 2011; **34**;881-892.