The Study on the Performance Characteristics due to the Degree of Superheat in NH_3 Refrigeration System III

– The Comparison of Heat Exchanger Types –

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ABSTRACT: Because the usage of CFC and HCFC based refrigerants are restricted due to the depletion of ozone layer, the NH_3 gas, in the experiment is evaluated to the performance characteristics for the superheat control to improve the energy efficiency. The experiment is carried out about the condensing pressure of refrigeration system from 1,500 kPa to 1,600 kPa through the degree of superheat from 0 to 10℃ at each condensing pressure. As a result, in the case of shell and tube type of heat exchanger, the COP is more efficient than other cases at the degree of superheat 1℃ at each condensing pressure. In the case of shell and disk type of heat exchanger, the COP is the most efficient at the degree of superheat 0.

Nomenclature

$\dot{m}_r$: refrigerant mass flow [kg/h]

$\dot{m}_{cw}$: cooling water flow [kg/h]

P_e : evaporation pressure [kPa]

Q_e : heat capacity of the evaporator [kW]

1. Introduction

Due to the destruction of ozone layer and global warming that have been globally considered as serious environmental problems, the production of specific freon substances such as R-11, R-12, and R-502 that were used as refrigerants of industrial refrigerating system has been prohibited since January 1st in 1996. Also, as the HCFC refrigerants were classified as harmful substance to the earth environment, its production and usage are restrained. Accordingly, the HFC refrigerants to be substituted with HCFC refrigerants were developed and has been marketed. However, it has low heat conductivity and needs careful choice to refrigerate oils and materials. In particular, as it shows high Global Warming Potential (GWP), its usage is undesirable with aspect to environment. The most appropriate way to resolve the environmental problem is to utilize natural refrigerants such as ammonia, propane, and propylene which are inorganic compounds. They are easy to obtain and purchase cheaply. Furthermore, they are friendly favored to environment.[1-2]

Among them, the ammonia refrigerant has

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been widely used because of its outstanding properties such as good heat conductivity, and high critical temperature and pressure, but it has disadvantages such as toxicity, inflammability and explosiveness.[3-4]

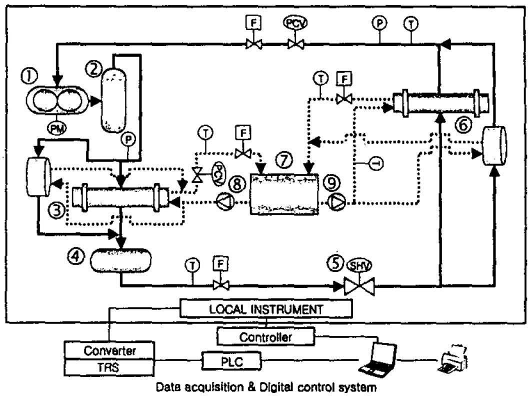
There are many kinds of heat exchangers for refrigerating systems. heat exchangers have been used to the most of the ammonia refrigerating systems including the shell and tube type of condenser and flooded evaporator. However, the shell and tube type of heat exchanger has the following disadvantages: it needs large space because of its heavy weight and large-scale. it needs more refrigerants while a system is working, and refrigerants may be leaked if systems are corroded.

Therefore, this study is conducted for a comparative analysis of condensing pressure and performance characteristics as to superheat between a small-sized and light-weight shell and disk type of optimum heat exchanger[5] which needs the least amount of refrigerant and permits little leakage of refrigerant and a shell and tube type of heat exchanger in ammonia refrigerating systems. Optimal operation conditions are identified with respect to types of heat exchangers and are demonstrated with the superiority of the shell and disk type of high-density heat exchanger that has not been practically used in domestic industrial fields for refrigeration.

2. Experimental equipment and procedure

2.1 Experimental equipment

Figure 1 shows an outline of experimental system for the performance study of the refrigerator based on changing superheating depending on condensing pressure. This study use ammonia as working fluid for this system, which is composed of a compressor, a condenser, a receiver, a thermostat, an expansion valve and other accessories. We give a great



① Compressor ⑤ Expansion valve
② Oil separator ⑥ Evaporator
③ Condenser ⑦ Constant temperature bath
④ Receiver ⑧, ⑨ Circulation pump
P : Pressure sensor PCV : Pressure control valve
PM : Power meter SHV : Superheat controller
F : Mass flow meter T : Temperature sensor
—— : Refrigerant —— : Cooling & chilled water
······ : Electric signal

Fig. 1 The schematic of ammonia refrigeration system.

care on the inside design of the system for the minimization of pressure loss. Its low-pressure part was insulated to meet the KS standard not to be affected by external temperature. To measure the phase change of working fluid within the system, we install a pressure gauge, a thermometer, a mass flow meter, a superheating controller, a pressure adjustment valve, and a power meter in the system. We install a thermo-hydrostat in our laboratory to maintain the range of errors of measurements within the temperature of ±0.1℃, the pressure of ±0.1 bar, the mass flow of ±0.1% and the power of ±0.1%.

We use a screw compressor for an experiment under a constant load and fix a slide to maintain the constant level of load. For a compressor and an evaporator, shell and tube type and shell and disk type of heat exchangers are used. As a fluid for refrigerant phase change, we use water. To maintain fluid temperature for refrigerant phase change at

Table 1 Experimental condition

Condensing pres. (kPa)	1,500 ~ 1,600
Superheat temp. (℃)	0 ~ 10
Bath temp. (℃)	28
Ambient temp. (℃)	24
Chilled water flow rate (kg/h)	6,800
Cooling & Chilled water	Demineralized
Evaporator temp. (Superheat temp. 0)	9℃

constant level, we install a 1 kW heater, a 3-way flow control valve and a thermostat for automatic control of temperature.

For constant water level of chilled water in the evaporator, an inverter circulation pump and a flow control valve are installed.

To control the degree of superheat, we calculate the degree of superheat with regard to suction temperature and pressure of each sensor attached to the outlet of the evaporator and use an electronic expansion valve[6] which automatically controls the opening of the valve through PID control to achieve a set-up valve.

For condensing pressure control, a pressure adjustment valve is used to automatically adjust the cooling water flow of the condenser with respect to the set-up pressures through input value of pressure sensor of the top of the condenser. A flow meter is installed to measure the amount of cooling water flow of the condenser. For the measurement of refrigerant mass flow, a mass flow meter is installed at the outlet of the receiver and the evaporator.

Table 1 shows the experimental conditions.

2.2 Experimental procedure

To maintain the external conditions of the system at constant level in a prior to the test operation of the refrigerating system, this study operates a thermo-hydrostat. To examine that the chilled water flow of the evaporator is being maintained at constant level, we operate a circulation pump for flow check. Before the operation of the system, this study compares the value of each measuring instrument attached to the system with the measured value transmitted on-line to check errors, and then monitors the operation using a monitoring program. When the operation becomes stable, the experiment is conducted at every 0.5 bar in a range from 15.0 bar to 16.0 bar. For the measurement of the degree of superheat, the experiment is conducted every 1℃ in a range from 0 to 10℃. The degree of superheat and condensing pressure are set up through an electronic automatic flow control valve and an electronic expansion valve at the beginning of operation, and then a manual flow control valve and a manual expansion valve are used to maintain the exact set-up value at constant level. To improve the accuracy of experiment data, we conduct repetitive experiments. The experimental data is measured every two seconds through data acquisition system and then the data is analysed using a computer.

3. Results and discussions

This study discusses the comparative results when the degree of superheat is changed step-by-step according to the condensing pressures as the following.

3.1 Refrigerant mass flow and evaporation pressure

Figure 2 shows the result that refrigerant mass flow decreased as the condensing pressure and degree of superheat increased.

If the condensing pressure increases under the same degree of superheat, the compression ratio increases and the actual volume of refrigerant vapors discharged by the compressor per hour decreases. And then the volume efficiency of the compressor decreases because of the decrement of the actual mass of the re-

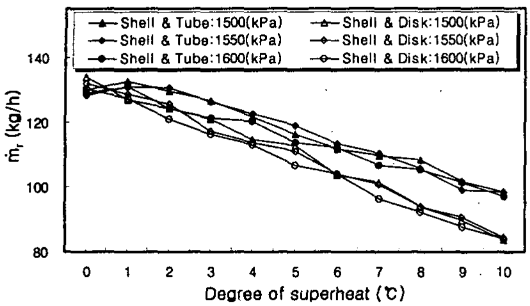


Fig. 2 The relations of refrigerant mass flow rate and degree of superheat at each condensing pressure.

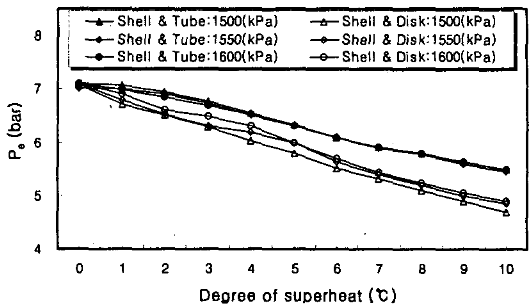


Fig. 3 The relations of suction pressure and degree of superheat at each condensing pressure.

frigerant vapors discharged per hour by the compressor. Therefore, if the condensing pressure increases, the refrigerant mass flow decreases because of the decrement of volume efficiency.

The evaporating pressure shows that the refrigerant mass flow decreased as the condensing pressure and the degree of superheat increased. The more the superheat and condensing pressure increased, the more the refrigerant mass flow decreased. So, the volume efficiency of the compressor decreased and equilibrium temperature of evaporator lowered. Therefore, it is shown that the evaporating pressure is in proportional to the refrigerant mass flow.

Under the same condensing pressure and 0 and 1℃ of superheat, there is the difference in the refrigerant mass flow and the evaporating pressure of the shell and tube type heat exchanger and the shell and disk type heat exchanger.

For the shell and disk type heat exchanger, when the superheat was 0, the refrigerant mass flow and the evaporating pressure were the highest. For the shell and tube type heat exchanger, when superheat was 1℃, the most refrigerant mass flow and the highest evaporating pressure were found.

The reason why the refrigerant mass flow and the evaporating pressure are the highest when the superheat of the shell and tube type is 1℃, is that sub-cooled boiling[7] occured at degree of superheat 1℃.

The reason why the refrigerant mass and the evaporating pressure are the highest when the superheat of the shell and disk type heat exchanger is 0 is presented as follows: as the shell and disk type heat exchanger has a plate-type structure, the interval where heat transfer is made between refrigerant and cooling water is narrow and dense. So, the area of heat transfer per unit volume is maximized and rapid heat exchange is made between two fluids due to the whirlpool by strong turbulence. For the shell and disk type heat exchanger, as turbulence is made at low Reynolds and it shows superior heat exchange to laminar flow required, boiling occurs at equilibrium temperature. Therefore, for the shell and disk type heat exchanger, as the superheat decreased, the refrigerant mass flow and the evaporating pressure increased orderly.

3.2 Cooling capacity of the evaporator

When the superheat increased under the same condensing pressure, both the shell and tube type heat exchanger and the shell and disk type heat exchanger had less cooling capacity of evaporator as shown in Fig. 4. It is because as superheat increased, the refrigerant mass flow decreased and the difference in temper-

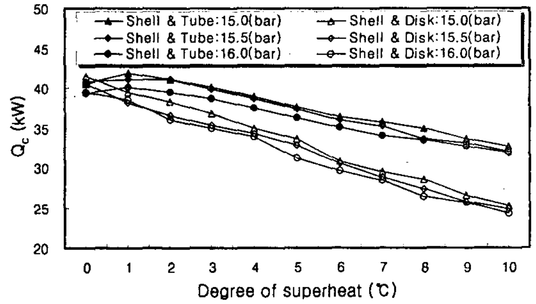


Fig. 4 The relations of evaporator cooling capacity and degree of superheat at each condensing pressure.

atures of the cooling water inlet and outlet of evaporator decreased.

Under the same degree of superheat, as the condensing pressure increased, the refrigerant mass flow of the shell and tube type heat exchanger and the shell and disk type heat exchanger decreased according to increased condensing pressure, but the ratio of change is insignificant. Therefore, the cooling capacity of two type heat exchanger should be almost same but actual data of cooling capacity has insignificant difference.

The reason why the shell and disk type heat exchanger shows insignificant difference is that as the condensing pressure increased, the actual refrigerating effect decreased and cooling capacity of the evaporator decreased accordingly.

As the condensing pressure becomes higher, the refrigerant fluid temperature at the condenser outlet should be higher, but the shell and disk type heat exchanger shows insignificant differences in temperature of the refrigerant fluids at the condenser outlet. The reason is that as the mentioned of 3.1, heat transfer per unit volume of Shell & Disk heat exchanger is superior.

3.3 Power

The power of condenser both for the shell and tube type heat exchanger and the shell and disk type heat exchanger increased as the condensing pressure and the superheat increased.

As the superheat increased with the same condensing pressure, the evaporating refrigerant mass flow decreased. As evaporating pressure decreased, the equilibrium temperature lowered, the specific volume increased, the volume efficiency decreased and the power increased.

When the condensing pressure was changed with the same degree of superheat in Fig. 2, as the condensing pressure increased, the compression rate increased leading to a higher power.

3.4 COP

The COP (coefficient of perform) is a ratio of cooling capacity of evaporator to power. The

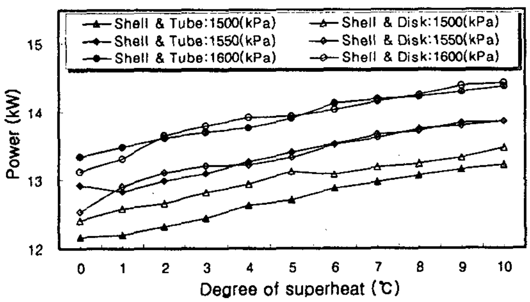


Fig. 5 The relations of power and degree of superheat at each condensing pressure.

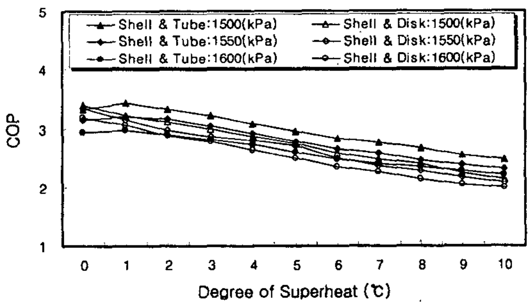


Fig. 6 The relations of COP and degree of superheat at each condensation pressure.

COPs of both the shell and tube type heat exchanger and the shell and disk type heat exchanger decreased with increasing the superheat and condensing pressure, but the difference in COPs according to the types of heat exchanger is insignificant. In particular, the condensing pressure is getting higher, the COPs of the shell and disk type heat exchanger and the shell and tube type heat exchanger are getting similar.

When the superheat increases with the same condensing pressure, the COPs of both the shell and tube type heat exchanger and the shell and disk type heat exchanger decrease. It is because as the superheat increases, the heat cooling of evaporator decreases and the power increases. However, for the shell and tube type heat exchanger, when the superheat is 1℃, the COP is the highest. For the shell and disk type heat exchanger, when the superheat is 0, the COP is the highest. For the shell and tube type heat exchanger, when the superheat is 0, the refrigerant mass flow decreases because of sub-cooled boiling.

As the condensing pressure with the same superheat is increased, the COPs of both the shell and tube type heat exchanger and the shell and disk type heat exchanger decrease. However, for the shell and tube type heat exchanger, the COP is decreased than shell and disk type. It is because for the shell and disk type heat exchanger, the cooling capacity of the evaporator is almost same. On the other hand, Shell and Tube type is getting decreased as the condensing press is higher, but the power is similar for both the two different types of heat exchangers in spite of the change of condensing pressure.

4. Conclusion

This study conducted the comparative analysis of performance of ammonia refrigerating systems using the shell and tube type heat exchanger and the shell and disk type heat exchanger with respect to the change in condensing pressure and superheat and the results are presented as follows:

(1) According to the condensing pressure conditions, the change in cooling capacity of evaporator is insignificant for the shell and disk type heat exchanger. For the shell and tube type heat exchanger, the more decrement of COP is found in comparison with the shell and disk type heat exchanger.

(2) Both for the shell and tube type heat exchanger and the shell and disk type heat exchanger, as the condensing pressure and superheat increases, the refrigerant mass flow input to the evaporator decreases, the compression ratio increases, the power increases, the COP decreases and the more energy was required.

(3) For the shell and tube type heat exchanger, when the superheat is 1℃ according to condensing pressures, the COP is the highest.

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References

1. E. I. du Pont de Nemours & Co. Ltd., Technical Report, 1989, Du Pont Alternative Refrigerants, Applications Testing of HCFC-123 and HFC-134a.
2. Soloman, S. and Wuebbles, D., 1994, ODPs, GWPs, and Future Chlorine/Bromine loading, Scientific Assessment of Ozone Depletion, pp. 131-136.
3. Res. Found. Inform. Bull. Refrig., 1979, Effect of Ammonia, Washington, D.C., p. 4.
4. Mclinden, M., Klein, S., Lemmin, E. and Peskin, A., 1998, NIST Thermodynamic and Transport Properties of Refrigerants and Refrigerant Mixture (REFPROP) ver. 6.0, Nation-

al Institute of Standards and Technology, Boulder, CO, U.S.A.

5. Ha, O. N., Kwon, I. W., Jeon, S. S., Lee, S. J., Jung, S. T., Ha, K. S., Yun, K. S., Lee, J. I. and Hong, K. H., 2005, The study on performance characteristics of NH_3 refrigeration system for various degree of superheat II, Korean Journal of Air-Conditioning and Refrigeration Engineering, Vol. 17, No. 4, pp. 297-302.
6. Higuchi, K., 1986, Electronic expansion valve and control, Refrigeration, Vol. 61, pp. 45-52.
7. Cengel, Y. A. 2002, Heat Transfer, McGraw-Hill, New York, pp. 461-505.