

The analysis and experimental investigation of heat pump system of R404A and R410A using thermobank and a two-phase ejector

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Abstract

This paper shows the results of CFD simulation and experimental investigation in which the heat pump system performance of R404A and R410A was compared between an ejector cycle and standard cycle. The special features of this study are to use the thermobank and an ejector in heat pump system. The thermobank stored heat from superheat vapor of refrigeration cycle and its energy is used for heating room and defrosting process. The ejector is an expansion device capable of work recovery. Therefore, the thermobank and ejector can help improving coefficient of performance (COP). When compared to standard cycle, the heat pump system using thermobank and ejector got maximum COP improvements of 38.5% with R404A and 42.7% with R410A. This heat pump system can be used to keep preservation of agricultural products in cold storage warehouse together with floor panel heating for room in winter in Korea and other country.

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Keywords: Ejector, Thermobank, Heat pump system, COP, Energy saving

1. Introduction

Refrigeration system has been used to remove heat from a source and discard it to other places. The refrigeration system applications include air conditioning, heat removal from chemical processes in petrochemical plant, medical equipment, special applications in the manufacturing and construction industries and processing and preservation of food. The processing and preservation of food uses refrigeration for their cold storage. The small size cold storage warehouse under 20 M/T that can preserve and trade agricultural, sea products in countryside and coastal areas in Korea. All of them are located inside or outside of house where is built closely from oil boiler equipment for heated rooms in individual house. A refrigeration unit in cold storage warehouse operates with electric power and heating room is heated with using oil boiler. Therefore spaces of those occupy much large in small house and energy is wasted doubly in spite of saving them in rural. Furthermore, it can reduce originated CO₂ that the cause increase global warming potential (GWP). To overcome these problems, it need to develop heat pump system that can be used to keep preservation of agricultural products in cold storage warehouse together with floor panel heating for room in rural house. There are various numerical models with thermobank have been developed base on physical equations, empirical correlations using curve fitting method or combination of both. This model uses physical equations combined with the curve fitting empirical correlations to develop a model with simple and minimum input requirements. Based on the system operating

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conditions, refrigeration unit models can be divided into steady state and transient models [2]. The steady state model is used because in the refrigeration systems of small size cold storage warehouse, the equipment is mainly intended to keep the refrigerated room in steady state conditions within the set temperature range. Preservation of agricultural products from producing district to the market and consumer forms an important parts of the product's cold chain which is mainly distributed in market which is located at small, medium and large cities. The performance and efficiency of such refrigeration unit is essential to maintain good quality of the agricultural products and fishery foods, saving energy and minimize environmental impacts. This study develops heat pump system based on investigation and CFD simulation the thermal behavior of a thermobank and COP for heat pump and cold storage. The special feature of the system is the heat storage equipment and ejector installed in the refrigeration system. Water is used as the heat storage material. During the refrigeration cycle operation, the heat storage equipment acts as a pre-condenser and stores a portion of refrigerant heat rejection in the thermobank. During the defrost cycle, the system uses superheating refrigerant gas discharged from the compressor, to defrost the evaporator coils. Then the heat stored in the thermobank is used to completely re-evaporate the refrigerant from the defrosting process in the evaporator, before it enters the compressor and continues its cycle. The ejector is used in system which the purpose increases in coefficient of performance (COP) and decrease in compressor displacement relative to a standard vapor compressor cycle.

2. Analysis

2.1. Heat transfer analysis in thermobank, condenser and evaporator

A steady state energy balance equation over the entire system is described in Eq. (1).

$$q_A + q_e = q_c + q_{th} \quad (1)$$

The compressor is modeled based on published manufacturers' specification data and empirical correlations from previous study [4]. The direct expansion evaporator is modeled based on the single zone heat transfer process with the two – phase refrigerant heat transfer process, between the refrigerant inside the evaporator coils and the air flow.

It is derived from the e-NTU relation for two-phase heat exchanger [5]. A quasi-steady state approach is applied to the heat transfer process in the heat storage area. For a determined period of the time, the heat storage temperature increases by Eq. (2).[3]

$$\Delta T_w = (T_2 - T_1) \frac{c_p \cdot \dot{m}}{M \cdot c_v} \left(1 - \exp \left[-\frac{UA}{c_p \cdot \dot{m}} \right] \right) \cdot \Delta t \quad (2)$$

The air-cooled condenser used in this study is modeled based on two-zone analysis, namely super cooling and condensation zones. The refrigerant leaving the condenser is assumed to be in a saturated liquid phase. The heat transfer within the super cooling is obtained from following Eq. (3).

$$q_{c,sp} = \dot{m}_r c_r (T_3 - T_{b,in}) \varepsilon_c \quad (3)$$

Where ε_c is the heat transfer effectiveness.

$$\varepsilon_c = 1 - \exp \left(-\frac{c_a \cdot \dot{m}_a}{c_r \cdot \dot{m}_r} \left(1 - \exp \left(\frac{U_{c,sp} \cdot A_{c,sp}}{c_a \cdot \dot{m}_a} \right) \right) \right) \quad (4)$$

The two-phase heat transfer for the condensation zone is given by Eq. (5).

$$q_{c,tp} = \dot{m}_b c_b (T_c - T_{b2}) \cdot \left[1 - \exp \left(-\frac{U_{c,tp} \cdot A_{c,tp}}{c_b \cdot \dot{m}_b} \right) \right] \quad (5)$$

The two-phase heat transfer process between the refrigerant inside the evaporator coils and the air flow is given by Eq. (6).

$$q_e = \dot{m}_a c_a (T_a - T_r) \cdot \left[1 - \exp \left(-\frac{U_{Ae}}{c_a \cdot \dot{m}_a} \right) \right] \quad (6)$$

The temperature in thermobank is calculated by Eq. (7).

$$T_{w(t)} = T_{in} - (T_{in} - T_{wo}) \exp \left[-\frac{\varepsilon \cdot t}{M \cdot c_v / c_p \cdot \dot{m}} \right] \quad (7)$$

Where, ε is

$$\varepsilon = 1 - \exp \left[-\frac{UA}{c_p \cdot \dot{m}} \right] \quad (8)$$

Coefficient of performance (COP) is calculated by Eq.(9).

$$\text{COP} = \frac{Q_e}{Q_A} = \frac{q_e \cdot \dot{m}_r}{q_A \cdot \dot{m}_{suc}} \quad (9)$$

2.2. Ejector analysis

Numerical model presented here deals with mathematical description of processes occurring in two-phase ejector. Following assumptions were made: equilibrium phase change; mixing process occur in mixing chamber; shock wave occur after mixing of streams; ejector walls are adiabatic; mass, momentum and energy exchange between vapour and liquid in suction chamber are neglected. It is assumed in this model, that the pressure p_i is equal to pressure at the motive nozzle outlet. Depending on the thermodynamic conditions from motive nozzle can emanate liquid stream or two-phase stream if partial evaporation during expansion occur. Since the driving parameters at nozzle inlet are known, the enthalpy h_{li} and entropy s_{li} of liquid can be found. Thermodynamically, the motive stream can have the subcooled liquid state or wet vapour state. It is assumed that at the nozzle outlet is two-phase state. Such conditions are practically in every application of two-phase ejector in refrigeration technology.[14]

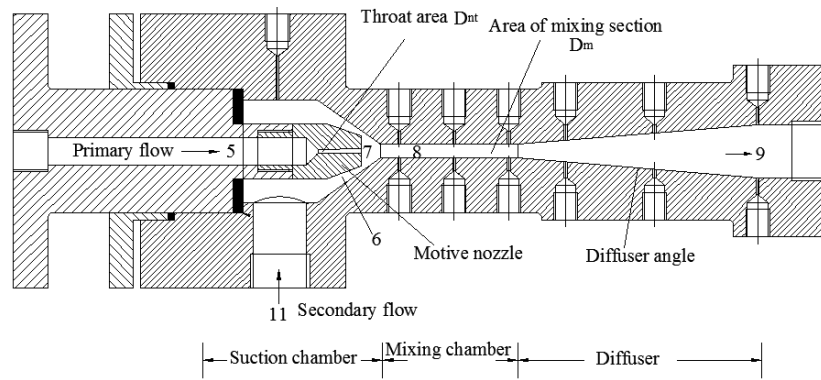


Fig.1. Ejector geometry.

In that case the isentropic quality of the vapour can be found:

$$x_{ns} = \frac{s_{li} - s'(p_i)}{s_{fg}(p_i)} \quad (10)$$

The specific enthalpy of the motive liquid after isentropic expansion is:

$$h_{ns} = h'(p_i) + x_{ns} h_{fg}(p_i) \quad (11)$$

Similarly, it is assumed that the velocity coefficient for motive nozzle ϕ_n is known. The experimental values of the velocity coefficient ϕ_n will be showed later. Specific enthalpy of motive fluid after irreversible expansion is:

$$h_n = h_{li} - \phi_n^2 (h_{li} - h_{ns}) \quad (12)$$

And the quality is:

$$x_n = \frac{h_n - h'(p_i)}{h_{fg}(p_i)} \quad (13)$$

Neglecting the kinetic energy of the motive fluid at the nozzle inlet, the velocity at the nozzle outlet is given by

$$w_n = \sqrt{2(h_{li} - h_n)} \quad (14)$$

And the density is given by

$$\rho_n = [v'(p_i) + x_n v_{gf} v'(p_i)]^{-1} \quad (15)$$

Pressure in suction chamber is assumed equals evaporating pressure. Since the parameters at suction chamber inlet are known, the enthalpy h_{vi} and entropy s_{vi} of vapour can be found. The vapour in suction chamber can be superheated or wet. In case of wet vapour the isentropic quality of the vapour can be found:

$$x_{vs} = \frac{s_{vi} - s'(p_i)}{s_{fg}(p_i)} \quad (16)$$

Where p_i is seek pressure at the mixing chamber inlet, and isentropic enthalpy of vapour is given by:

$$h_{vs} = h'(p_i) + x_{vs} h_{fg}(p_i) \quad (17)$$

The specific enthalpy in a real process is given by:

$$h_v = h_{vi} - \phi_s^2 (h_{vi} - h_{vs}) \quad (18)$$

Where, velocity coefficient for suction chamber ϕ_s is assumed as known. The experimental values of the velocity coefficient ϕ_s will be showed later. Quality of the vapour in real the process is:

$$x_v = \frac{h_v - h'(p_i)}{h_{fg}(p_i)} \quad (19)$$

Density of the vapour is given by:

$$\rho_v = [v'(p_i) + x_v v_{fg}(p_i)]^{-1} \quad (20)$$

If the vapour is superheated, the temperature and enthalpy of vapour in isentropic condition can be found from equation of state:

$$t_{vs} = t(p_i, s_{vi}) \quad (21)$$

$$h_{vs} = h(p_i, s_{vi}) \quad (22)$$

Equation (18) for specific enthalpy is valid also for superheated vapour, therefore the density of superheated vapour:

$$\rho_v = \rho(p_i, h_v) \quad (23)$$

Farther calculation are independent of the state of vapour. Velocity of vapour at the mixing chamber inlet is equal to:

$$w_v = \phi_s \sqrt{2(h_{vi} - h_{vs})} \quad (24)$$

From continuity equation:

$$w_v = \frac{\dot{m}_{vi}}{A_{vi} \rho_v} \quad (25)$$

Where A_{vi} is cross section area, given by:

$$A_{vi} = \frac{\pi}{4} (D_k^2 - D_{no}^2) \quad (26)$$

In mixing chamber the momentum transfer between both streams occurs. After mixing process it can be assume that the flow is homogeneous. Mass velocity in mixing chamber is:

$$G = \frac{\dot{m}_{li} (1 + U)}{A_m} \quad (27)$$

Specific enthalpy at stagnation conditions at the mixing chamber inlet is given by:

$$h_{iz} = \frac{h_{li} + U h_{vi}}{1 + U} \quad (28)$$

Once again, it is assumed that the velocity coefficient for mixing chamber ϕ_m is known. In presented model friction between stream and ejector wall is treated separately. Therefore, it require to calculate average values for two-phase flow at the inlet to the mixing chamber.

Static enthalpy at the mixing chamber inlet is:

$$h_{it} = \frac{h_n + U h_v}{1 + U} \quad (29)$$

Average quality in mixing chamber is given by:

$$x_{it} = \frac{h_{it} - h'(p_i)}{h_{fg}(p_i)} \quad (30)$$

An average value of density at mixing chamber inlet:

$$\rho_i = \left[v'(p_i) + x_{it} v_{fg}(p_i) \right]^{-1} \quad (31)$$

The specific enthalpy after mixing:

$$h_m(p_m, \rho_m) = h'(p_m) + x_m(p_m, \rho_m) \cdot h_{fg}(p_m) \quad (32)$$

Momentum conservation equation for mixing process has a form:

$$A_m(p_i - p_m - \Delta p_f) = \dot{m}_{li}(1+U)w_m - \dot{m}_{li}w_n - \dot{m}_{li}Uw_v \quad (33)$$

It is assumed in this model, that in diffuser velocity of two-phase homogenous flow decreases to $w_d = 0$. Velocity coefficient for diffuser ϕ_d is assumed known. Specific enthalpy of fluid is also known and equal stagnation enthalpy:

$$h_d = h_{iz}$$

Momentum conservation equation for diffuser has a form:

$$\frac{1}{2}(p_d - p_t)(A_d - A_m) = \dot{m}_{li}(1+U)w_t\phi_d \quad (34)$$

Where, pressure distribution in diffuser was assumed as linear. Finally the ejector outlet pressure can be calculated from equation:

$$p_d = p_t + 2Gw_t\phi_d \left(\frac{A_d}{A_m} - 1 \right)^{-1} \quad (35)$$

Overall compression ratio is defined by equation:

$$\Pi = \frac{p_d - p_{vi}}{p_{li} - p_{vi}} \quad (36)$$

3. Experimental apparatus

The Figure.2 shows the schematic and operating principle of heat pump system with using ejector and thermobank. The refrigeration system, like most of other refrigeration systems, utilize similar components used in all vapor compression cycles: a compressor, condenser, an expansion device, an evaporator. But in this system has also thermobank and ejector. A vapor compression refrigeration cycle, working principle operates in the following sequences as depicted in Figure.2 (dot-line): in the compressor, low temperature and low pressure refrigerant (1) is pressurized into high temperature and high pressure refrigerant (2). The refrigerant then enters a thermobank where heat from the refrigerant is removed and water in thermobank is warmed up. Hence, the refrigerant is desuperheating before going to condenser (3). The desuperheating refrigerant goes into condenser and it is condensed into liquid (4). The liquid refrigerant is continuously cooled into subcooling state (5) before entering ejector. The subcooling liquid refrigerant (5) combines with the vapor refrigerant after evaporator (12) in ejector to liquid and vapor refrigerant at (9) state. In ejector where takes place heat transfer, mixing, reduced pressure and separate processes (6,7,8,9). The liquid and vapor refrigerant (9) enter separator; the vapor goes into compressor (1); the liquid refrigerant (10) is then throttled through an thermo expansion valve (11) where a small liquid refrigerant portion is evaporated to vapor and bigger liquid refrigerant portion is continuously reduced pressure to evaporating pressure before entering the evaporator. In the evaporator the liquid refrigerant at low pressure and temperature, passes through the evaporator and absorbs heat from the cold storage.

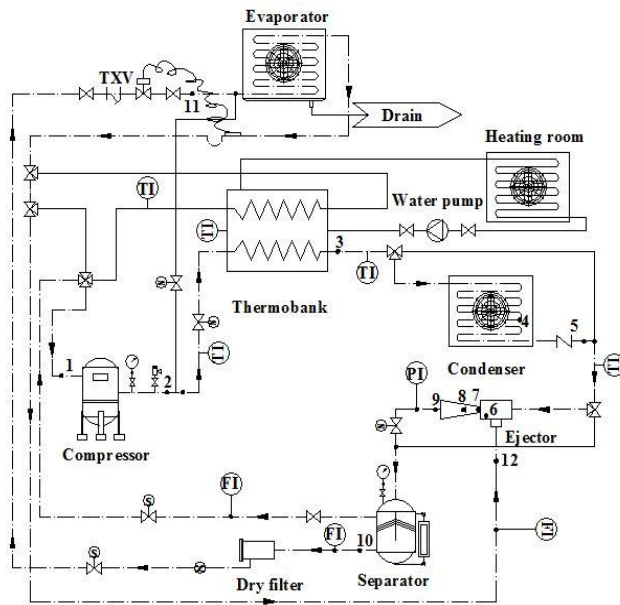


Fig.2. Apparatus of heat pump system with using ejector and thermobank

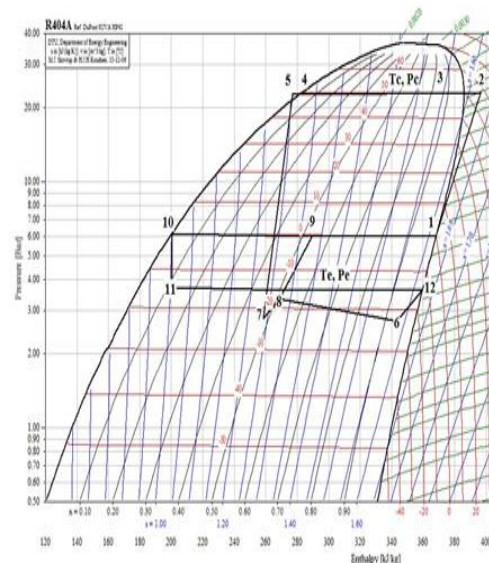


Fig.3. P-h diagrams of the refrigeration cycle with a two-phase ejector. (Points: 1-2-3-4-5-7-8-9-1 and 9-10-11-12-6-8-9-1)

And liquid refrigerant becomes vapor refrigerant (12); before going to ejector and go back to suction line of the compressor (1) for continuing its cycle again. Water is used in the thermobank which stores and transfers the heat. Hot water in thermobank is cyclically pumped to heating panel. Heating panel is used for warming up the house. The heat storage is also used for defrosting process. During the defrosting process, the hot refrigerant from compressor goes through the evaporator for evaporator coils defrosting process. The refrigerant ejects its heat and condenses into liquid. It flows back into the heat storage equipment (thermobank). The liquid refrigerant is re-evaporated by the heat storage of thermobank before entering the compressor and circulates again. So that, the system do not need an accumulator. The Fig.3 indicates more clearly the principle of operation of the system by P-h diagrams.

System specification and test conditions:

- Compressor: Seltec TM-15HD, 4 cylinder, Nominal 2000 rpm, Displacement 147 cm³
- Refrigerant: R404A, R410A.
- Evaporator: Aluminum finned, 6 rows, 9.53mm diameter aluminum tube, total area 8.52 m², frontal area 0.172 m², Fan 2×250 mm diameter, 80w, 12/24V, 1400m³/hr.
- Condenser: Aluminum finned, 4 rows, 9.53mm diameter cooper tube, total area 13.635 m², frontal area 0.2 m², Fan 2×250 mm diameter, 80w, 12/24V, 1400m³/hr.
- Expansion valve: FSE-1/2-C.
- Heat storage: 2 rows × 4 circuit 9.53mm diameter cooper tube immersed in 80 litter of water.
- Test condition: Heat leak and cooling capacity for -10, -5 and 0°C box set temperature based on ARI 1110 at ambient temperature of 20°C.

Boundary condition and assumptions for CFD simulation:

Thermobank model:

- Dimension: 600×300×400 mm
- Model: 2D
- The fluid in thermobank(water): $V_{in} = 0.28$ m/s, T_{in} (water): 20°C.
- Velocity of fluid (gas, refrigerant): 1.72 m/s, T_{in} (gas): 80°C.
- Solid material: cooper
- Mesh quality: Minimum orthogonal quality: 7.228 e-01, Maximum aspect ratio: 7.384e+00, node: 1711

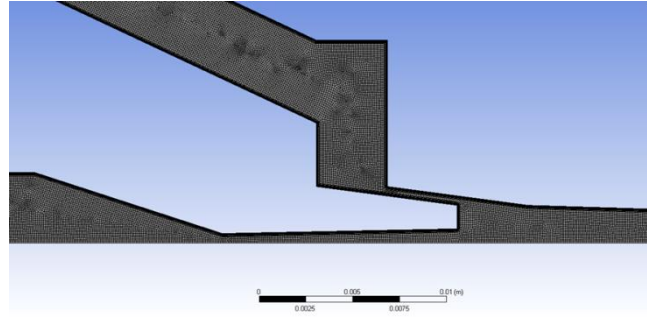


Fig.4. Sample of 2D ejector grid model (a half)

Ejector model:

- Assumptions:
 - The flow is steady one dimensional flow
 - The gravitational force effect on the flow is neglected
 - The isentropic efficiency of the nozzle is 0.98
 - The heat transfer between the fluid and nozzle wall is neglected
- Dimension: Nozzle $\phi 1\text{mm}$, ejector total length 180mm
- Model: 2D, STT k- ω turbulence model.
- The primary fluid (liquid refrigerant): $\dot{m} = 0.07\text{ kg/s}$.
- The secondary fluid (vapor refrigerant): $\dot{m} = 0.031\text{ kg/s}$.
- Solid material: cooper
- Mesh quality: Minimum orthogonal quality: 5.018 e-01, Maximum aspect ratio: 1.650e+01, node: 98056

The quality simulation depends on meshing process. The Fig.4 illuminates a symmetrical haft of meshing ejector.

4. Results and Discussion

The inputs for the computer simulation include condenser model and evaporator, air flow velocity and ambient temperature is constant assumption and cold storage temperature is set point temperature. The results of CFD simulation of thermal behavior in the thermobank are shown in Fig.5. The water temperature in thermobank increases by time. The water temperature in front of the tube (at 0° degree) is lower than the water temperature at 90° degree because the water flow goes directly in front of the tube and after of the tube is dead point (90° degree). The simulation and experimental investigation of heat storage temperature in thermobank is shown in Fig.6. Water is used for heat bearing agent, both simulation and experimental results are in a good agreement. The experimental results become lower than the simulation values at higher heat storage temperature in thermobank . The deviation may be caused by heat losses in the thermobank .

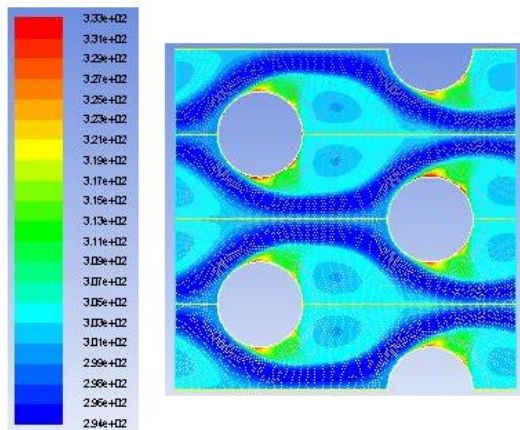
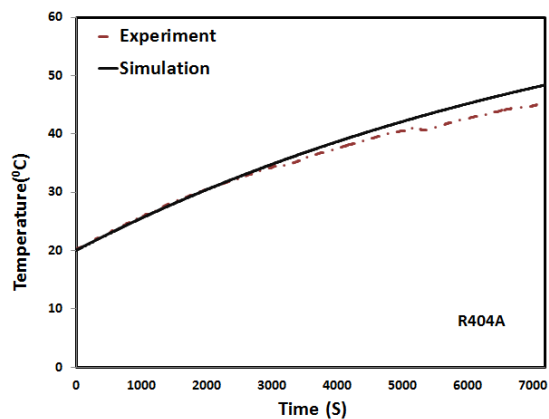
Fig.5. The contours of static temperature magnitude of liquid fluid in thermobank ($^\circ\text{K}$)Fig.6. Water temperature in thermobank by the time ($^\circ\text{C}$)

Fig.7 shows the water temperature in thermobank and heating panel. When pump is turn on, hot water in thermobank is pumped cyclically to heating panel. Heating panel is used for warming up the house. Therefore, water temperature decreases when comes back to thermobank.

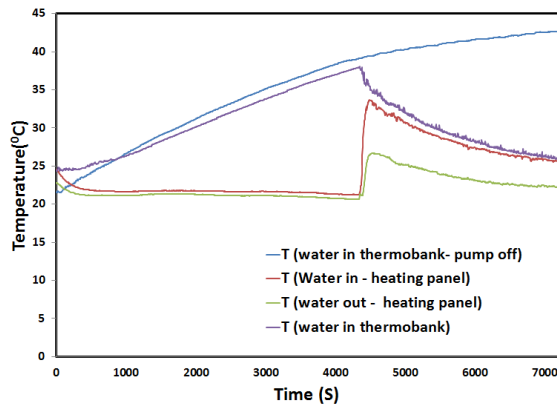


Fig.7. Water temperature in thermobank and heating panel ($^{\circ}\text{C}$)

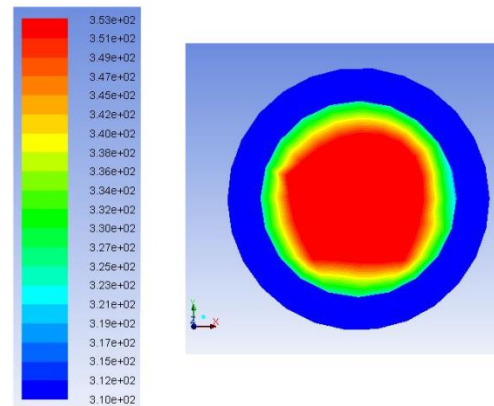


Fig.8. The contours of static temperature of refrigerant fluid go into thermobank ($^{\circ}\text{K}$)

The refrigerant fluid temperature before going to thermobank is illuminated in Fig.8. The outermost layer is tube wall. We can see that in the fluid center, the refrigerant temperature is the highest and smaller at the tube wall. Fig.9 shows the temperature of refrigerant go into thermobank along the tube length. The refrigerant fluid temperature after going out of the thermobank is displayed in Fig.10. The refrigerant fluid temperature decreases because heat rejects by cold water in thermobank. Therefore, the refrigerant is precooled before going to condenser. And Thermobank can help increasing cooling capacity and COP. The Fig.11 shows the velocity profile inside ejector nozzle. The velocity is maximum at the inlet of motive nozzle and decreasing before going to mixing chamber. The maximum velocity at the inlet of motive nozzle is 112 m/s.

The effect of mass entrainment ration to COP at zero degree Celsius set point in cold storage is shown in Fig.12. COP increases when mass entrainment ration increases. When using R410A refrigerant, COP is higher than 4.2% versus R404A. The effect of mass entrainment ration to compression ratio is displayed in Fig.13. When mass entrainment ration increases, compression ratio decreases.

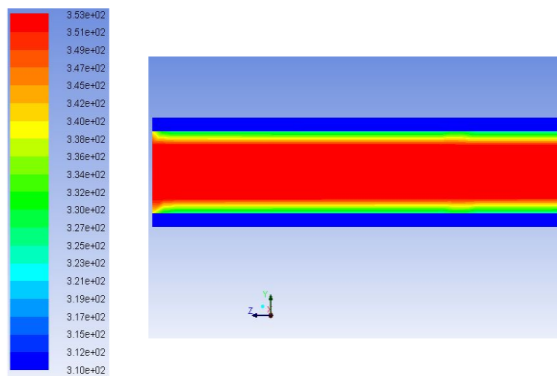


Fig.9. The contours of temperature of refrigerant fluid go into thermobank along the length ($^{\circ}\text{K}$)

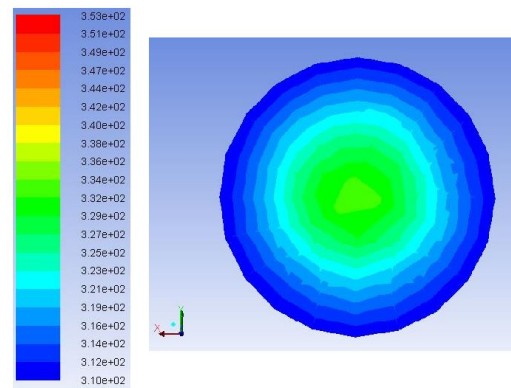


Fig.10. The contours of static temperature magnitude of refrigerant fluid go out of thermobank ($^{\circ}\text{K}$)

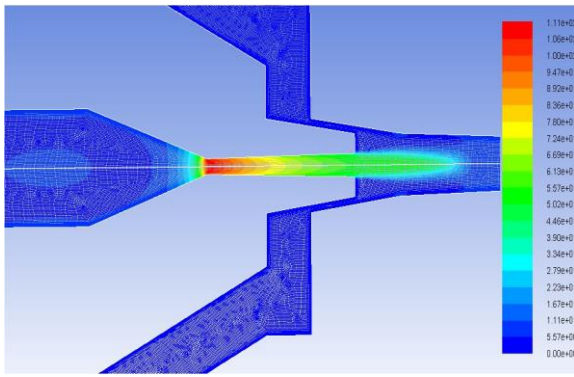


Fig.11. The contours of velocity magnitude of fluid in ejector nozzle (m/s)

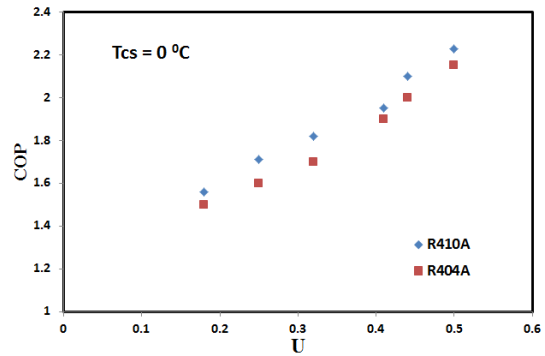


Fig.12. The effect of entrainment ratio to COP

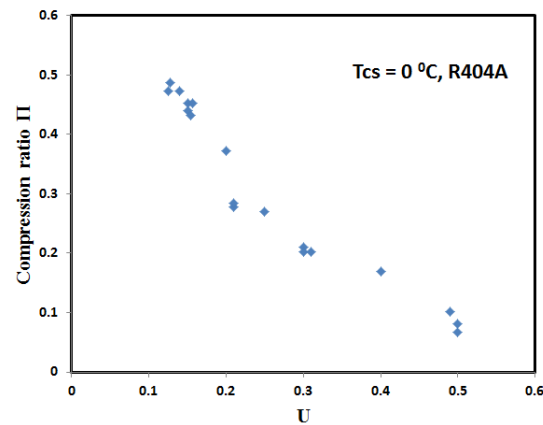


Fig.13. The effect of entrainment ratio to compression ratio

The power input and COP of system is depicted in Fig.14 with R404A refrigerant and Fig.15 with R410A. The measured power input includes compressor's motor, evaporator and condenser fans and control operations and other losses. The compressor power is more required at higher refrigerated set point temperature to achieve the balance point of the system, hence the system power input increases with the increase of the temperature in cold storage.

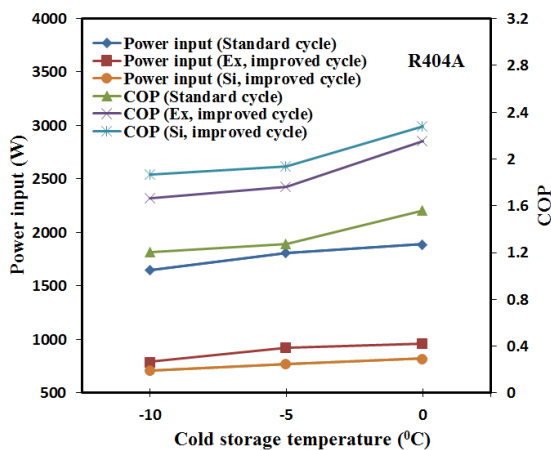


Fig.14. Experiment, simulation results of power input and COP with cold storage temperature (R404A)

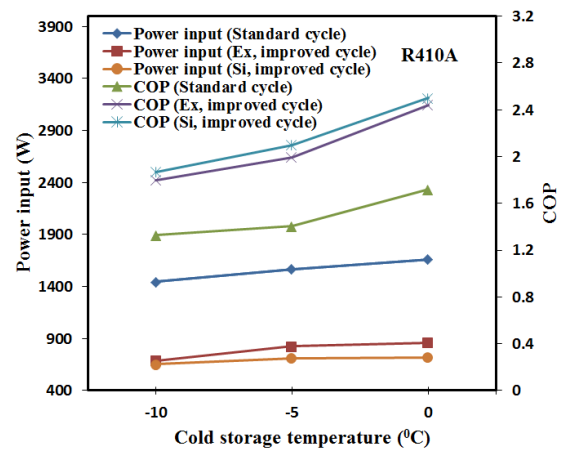


Fig.15. Experiment, simulation results of power input and COP with cold storage temperature (R410A)

Given the COP is directly proportional to cooling capacity, the COP increases as the refrigerated box temperature increases. When compared to standard cycle, the heat pump system using thermobank and ejector got maximum COP improvements of 38.5% with R404A and 42.7% with R410A.

5. Conclusions

The comparison simulation and experimental results using thermobank for defrosting process and cold storage of preservation process in heat pump system show good agreement. The ejector is used in system which the purpose increases in coefficient of performance (COP) and decrease in compressor displacement relative to a standard vapor compressor cycle. The heat storage of thermobank can be used for heating room in individual house in rural of Korea and cold storage for food preservation. The power consumption of heat pump system decreases because the thermobank save heating energy for floor panel heating and increasing cooling capacity. When compared to standard cycle, the heat pump system using thermobank and ejector got maximum COP improvements of 38.5% with R404A and 42.7% with R410A.

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