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# Cycle Performance and Heat Transfer Characteristics of a Heat Pump using R22/R142b Refrigerant Mixtures

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## ABSTRACT

A heat pump system is constructed to evaluate the performance and the heat transfer characteristics. Mixtures of R22/R142b are adopted in the heat pump. The heat transfer in the evaporator and the overall perfomance are measured and analyzed in terms of the compositions and the flow rate of the refrigerants. The evaporator consists of counter-current concentric circular tubes.

Possibility of capacity modulation by changing composition is observed without degradation of heat transfer coefficients and coefficent of performance for a fixed volumetric displacement rate. The evaporating capacity may be varied continuously within 200 percent based on minimum capacity. For similar cooling load, COP is improved by mixing two refrigerants and shows maximum value at 60% mass fraction of R22.

Local heat transfer coefficients are obtained in the evaporator and shown as a function of composition for given values of mass flow rate. Heat transfer coefficients of mixtures are decreased in comparison with pure refrigerants at similar mass flow rate and average evaporating temperature.

## 1. INTRODUCTION

Research and development have been progressed to enhance the system performance and to estimate properties of refrigerant mixtures in a heat pump. In addition, efforts have been made to utilize main advantages of non-azeotropic refrigerant mixtures which may be classified into performance enhancement and capacity modulation.

There are numerous researches on performance of heat pumps in the literature[1,2,3,4]. It is well known that thermal capacity can be controlled and performance is enhanced in some cases by using mixed refrigerants.

When binary non-azeotropic refrigerant mixture is used as working fluid of a heat pump, the mean temperature difference between refrigerant and heat source(or sink) may be reduced because of the temperature gliding effect. Although the coefficient of perfomance can be increased by using refrigerant mixtures, area of heat exchanger has to be changed to compensate the differences of heat transfer coefficients between pure refrigerants and mixtures

Understanding of heat transfer characteristics is important for performance analysis of a heat pump. A number of convective phase change heat transfer researches has been performed in pure refrigerants and mixtures[5,6,7,8]. The results shows that in general heat transfer coefficients are degraded by mixing pure components.

This paper deals with the experiments of a heat pump system using R22/R142b non-azeotropic refrigerant mixtures. Variation of heat tranfer coefficient and overall performance are measured and analyzed in this work. A horizontal annulus type evaporator is fitted to an experimental heat pump system. Analyses are performed in two viewpoints, namely, possibility of capacity modulation and COP enhancement.

## 2. EXPERIMENTS

### 2.1 Experimental Apparatus

The schematic diagram of the experimental apparatus is shown in Fig.1. The system is basically a heat pump composed of a compressor, an evaporator, a condenser, an expansion valve and a subcooler. Miscellaneous equipments are added including an oil separator, a receiver, an accumulator, a strainer, a sight glass, a flow meter and an inverter which modulates the rotational speed of compressor. Temperature and pressure measuring instruments are attached to the system. Also equipped are heat source and heat sink units to supply secondary fluids at nearly uniform temperature.

A schematic view of the test evaporator is shown in Fig.2. The evaporator is constructed of count-current concentric tubes in order to fully utilize the temperature gliding effect of refrigerant mixtures. Refrigerant flows through the inner tube and water flows in the outer annulus side. Both the inner and outer tubes are made of copper. The inner and outer diameters of inner tube are 10.7 and 12.7 mm respectively

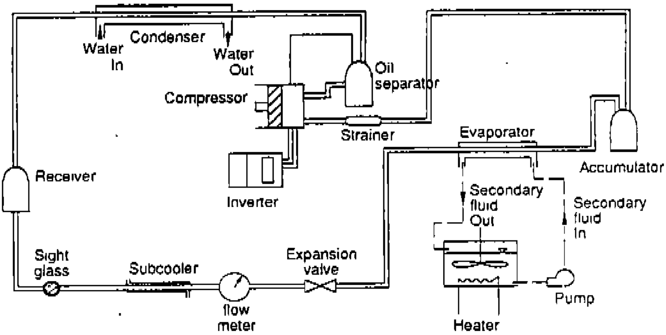


Fig. 1 Schematic diagram of a heat pump system in this study

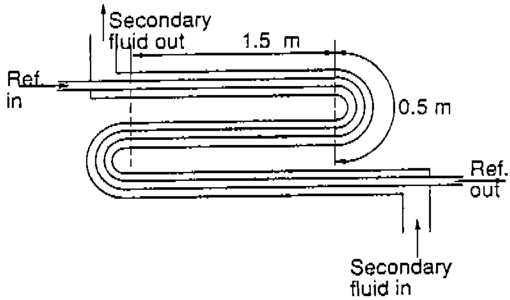


Fig. 2 Overall view of evaporator

and the inner diameter of outer tube is 23.4 mm. Total length of evaporator is 6 m and effective heat transfer area based on inner side of inner tube is 0.20 $m^2$. Outer tube is sufficiently insulated. Water is used as secondary fluid and an electric heater is equipped in a storage unit to make up for heat transferred to refrigerant and to retain constant inlet temperature of water at evaporator.

Temperature and pressure are measured at various locations shown in Fig.3. Thirteen points are chosen to measure refrigerant temperature through the evaporator. Temperature of water is also recorded at the same axial location of refrigerant temperature measurement point and temperature of outer wall of inner tube is measured at mid-position. The wall temperature of each section is measured at three points (top, bottom and side) around the circumference. Temperature of refrigerant is also measured at inlet and outlet of compressor, inlet of expansion valve and through the condenser. T type(copper-constantan) thermocouples are used for all temperature measurements.

Pressure taps are located at 7 positions through evaporator. A strain gauge type pressure transducer is adopted to measure pressure at inlet of evaporator and a diaphram type pressure transducer is used to measure pressure differences between inlet and other measuring positions through evaporator. Bourdon tube type pressure gauges are attached to inlets and outlets of compressor and expansion valve.

The flow rate of refrigerant is measured at exit of subcooler by a positive displacement type (rotary piston type) micro flow meter and degree of subcooling is maintained sufficiently large to avoid existence of vapor through the flow meter.

A semi-hermatic 1HP compressor designed for R22 is used. An inverter which can modulate frequency from 0 to 120 Hz is linked to the compressor to change the rotational speed of compressor. An ammeter, a voltage meter and a watt meter is connected to compressor to measure input power. The refrigerant mixtures are prepared by weighing each component. Condenser and subcooler are also made of concentric tubes and water is used as the secondary fluid.

Working substances are chosen to be two pure refrigerants of R22 and R142b and four different mixtures of R22/R142b with mass fraction of R22 : 20%, 40%, 60% and 80%. For a given substance, experiments are performed by varying compressor speed and flow rates of refrigerant and secondary fluid.

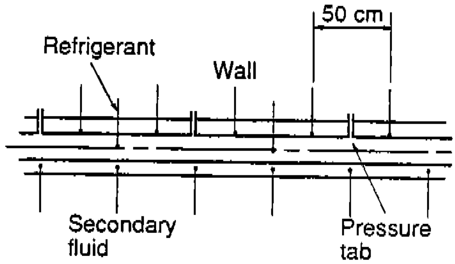


Fig. 3 Measurement points of temperature and pressure of evaporator

2.2 Measurement and data reduction

When system becomes steady state, pressure, power input and frequency output of inverter are recorded. Temperatures of refrigerant, water and wall at designated positions are recorded by personal computer through a data acquisition system. Mass flow rate of water is determined by weighing the amount collected for enough time and that of refrigerant is calculated by multiplying density of subcooled liquid to measured volumetric flow rate at the exit of subcooler.

The total heat transfer rate, $\dot{Q}$, at evaporator can be calculated by

$$\dot{Q} = (\dot{m}C_p\Delta T)_{water} \tag{1}$$

where $\Delta T$ means temperature difference between inlet and outlet of water at evaporator. $\dot{Q}$ also equals to the rate of heat obtained by refrigerant if there is no heat loss to ambient.

Figure 4 shows a subsection adopted to calculate the local heat transfer coefficient. A subsection is a region between two measuring positions of water temperature. Temperature of refrigerant is needed to calculate local heat transfer coefficient and can be determined by two methods. One is to use the directly measured temperature and the other is to use the temperature calculated by energy balance and pressure-temperature relation. The measured temperature slightly deviates from caculated one in this study. Moreover, measured temperature shows higher value than calculated temperature at high quality region due to the superheat of vapor[9]. Therefore the calculated temperature is more suitable to the analysis of the evaporation process because energy balance is basically required. Similar analyses can be found in the literature[6,7]. In this paper we also adopt temperature obtained from pressure measurment and directly measured temperature is used as a reference.

The heat balance for a subsection is represented by either eq. (2) or (3) and they must be identical without heat loss.

$$\dot{Q}_{sub} = \dot{m}_r(i_o - i_i) \tag{2}$$

$$\dot{Q}_{sub} = (\dot{m}C_p\Delta T_{sub})_{water} \tag{3}$$

Here $\dot{Q}_{sub}$ is heat transfer rate through a subsection and subscripts $o$ and $i$ denote outlet and inlet of a

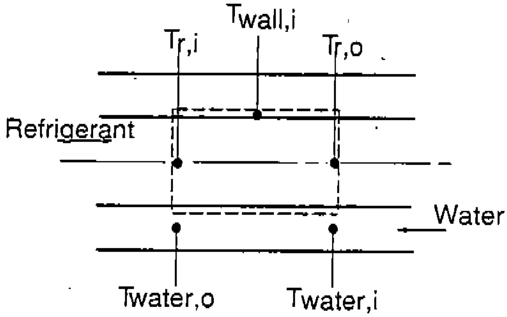


Fig. 4 Control volume for evaluating local heat transfer coefficient

subsection. For a subsection, inlet enthalpy, $i_i$, can be calculated with the known value of outlet enthalpy, $i_o$. Enthaly is denoted by $i$ instead of $h$ in order not to be confused with heat transfer coefficient. Some degree of superheat is always maintained at the exit of evaporator and exit enthalpy can be found as a function of temperature and pressure. Therefore, once $\dot{Q}_{sub}$ is calculated by eq.(3), inlet enthalpies of all subsections can be calculated by eq.(2) in reverse direction from last to first subsection. Temperatures of inlet and outlet of a subsection can be obtained by known enthalpies, pressures and total composition with the help of thermodynamic relations. Pressure can be determined by linear interpolation between measured values of two neighboring positions.

Refrigerant temperature at mid point of a subsection is needed to calculate the local heat transfer coefficient. The mid point temperature is determined through the similar procedure as that mentioned above except replacing $\dot{Q}_{sub}$ by 1/2 $\dot{Q}_{sub}$.

Calculation of thermodynamic properties and determination of vapor liquid equilibrium of pure and mixed refrigerant are all carried out by the well known Peng-Robinson equation of state[10,11].

The local heat transfer coefficient, $h$, is defined by

$$h = \frac{\dot{Q}_{sub}}{A_{sub}(T_{wi,mean} - T_b)} \tag{4}$$

$A_{sub}$ is the heat transfer area of a subsection and calculated by

$$A_{sub} = \pi d_i \Delta z \tag{5}$$

where $\Delta z$ indicates the effective length of a subsection and estimated to be 495 mm. $T_b$ and $T_{wi,mean}$ represent the temperature of refrigerant and the mean temperature of inner surface of inner tube respectively. The inner wall temperature, $T_{wi}$ of each point around circumference can be obtained by well known conduction relation from measured outer wall temperature of $T_{wo}$.

After the inner wall temperatures are obtained for top, bottom and side surfaces, mean inner surface temperature, $T_{wi,mean}$, can be determined by

$$T_{wi,mean} = \frac{T_{wi,top} + 2T_{wi,side} + T_{wi,bottom}}{4} \tag{6}$$

where weight factor 2 accounts for the symmetry of geometry. The average heat transfer coefficient may be defined and represented by eq.(7) since subsections before dryout are equally devided by $N$.

$$\bar{h} = \frac{\int h dA}{\int dA} = \frac{\sum_{i=1}^{N} h}{N} \tag{7}$$

## 3. RESULTS AND DISCUSSIONS

### 3.1 Scope of experiments

The flow rate of refrigerant is varied mainly by changing compressor speed which is controlled by an inverter. Inlet and exit temperature of water at evaporator is maintained at about 10 and 27°C respectively. Average condensing temperature of refrigerant is controlled between 27 and 30°C by varying flow rate of cooling water.

Significant differences appear in vapor density among properties in the mixtures of R22 and R142b. At a given temperature, the vapor specific volume of R142b is larger than that of R22 by a factor of more than two. Therefore one can conjecture that as the fraction of R142b becomes larger, the vomulemetric displacement rate must be increased in order to maintain equal heat transfer rate. The volumetric displacement rate is proportional to the compressor speed in this study.

Variation of compressor speed is conducted by modulating the input frequency of voltage to compressor. The standard frequency of 60 Hz is changed by the inverter. Experiments are performed in frequency range from 30 to 80 Hz with 10 Hz increment. Higher frequency means higher compressor speed.

Total heat tranfer rate, $\dot{Q}$, at evaporator is shown in Fig. 5 for various compositions. $\dot{Q}$ is almost linearly proportional to mass flow rate, $\dot{m}_r$. Since the latent heats of R22 and R142b are similar in magnitude, heat transfer rates of all compositions are nearly equal at given $\dot{m}_r$. This is the reason why all data can be loaded on a line

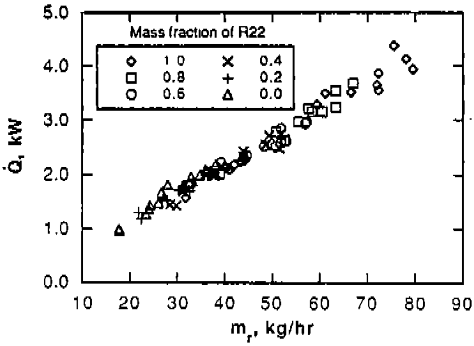


Fig. 5 Variation of total heat transfer rate through evaporator with mass flow rate of refrigerant

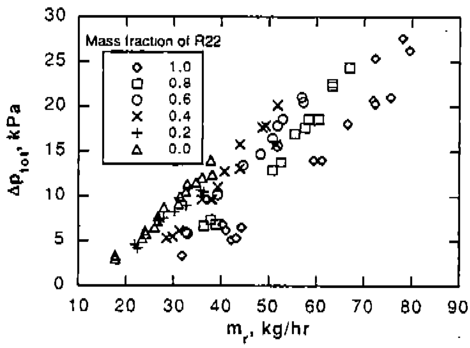


Fig. 6 Variation of total pressure drop through evaporator with mass flow rate of refrigerant

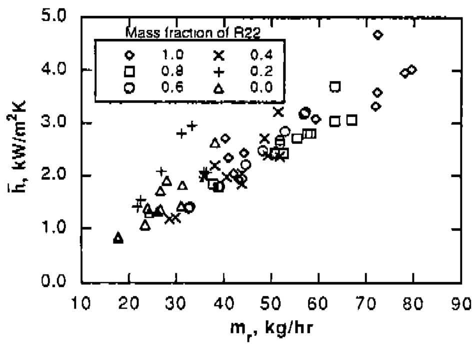


Fig. 7 Variation of average heat transfer coefficient through evaporator with mass flow rate of refrigerant

Total pressure drop through evaporator is shown in Fig.6. $\Delta P_{tot}$ also increases with increasing $\dot{m}_r$. This is trivial because friction and inertia pressure drop increase as flow rate increases. As fraction of R142b becomes larger, pressure drop increases. This can be explained by the relatively large value of vapor specific volume of R142b, which leads to large mean velocity of fluid.

Average heat transfer coefficient, $\bar{h}$, is presented in Fig.7. Mass flow rate is the main parameter which governs the heat transfer coefficient of convective flow. For all compositions, $\bar{h}$ also increases with increasing $\dot{m}_r$ as is the cases for $\dot{Q}$ and $\Delta P_{tot}$.

3.2 Capacity modulation

Experimental results are rearranged to see the variation of evaporating capacity. Figure 8 through 10 show variations of total heat transfer rate, coefficient of performance and average heat transfer coefficient in terms of compressor speed and refrigerant composition.

Coefficient of performance is defined by eq.(8) as usual where $\dot{W}$ is the input power to compressor.

$$\mathrm{COP} = \frac{\dot{Q}}{\dot{W}} \tag{8}$$

At fixed frequency or constant speed of compressor, heat transfer rate $\dot{Q}$ increases as the fraction of R22 becomes large. On the contrary, COP increases as the fraction of R142b becomes large. Especially, $\dot{Q}$ of R22 is about twice as large as that of R142b at same speed of compressor although latent heat of both refrigerants are similar in magnitude. These results relatively agree well with results of a cycle simulation[11] based on similar conditions.

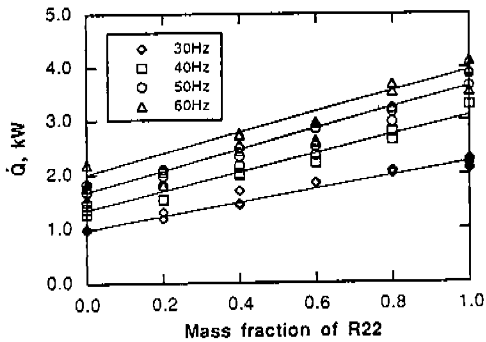


Fig. 8 Variation of total heat transfer rate with mass fraction of R22 (indicated No. is frequency of input voltage)

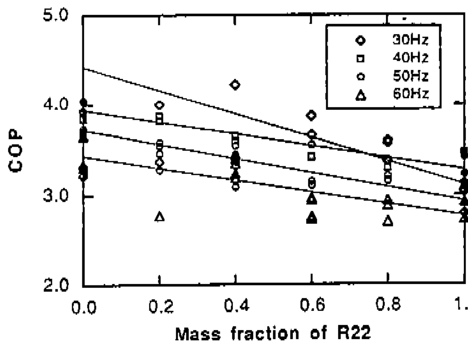


Fig. 9 Variation of COP with mass fraction of R22 (indicated No. is frequency of input voltage)

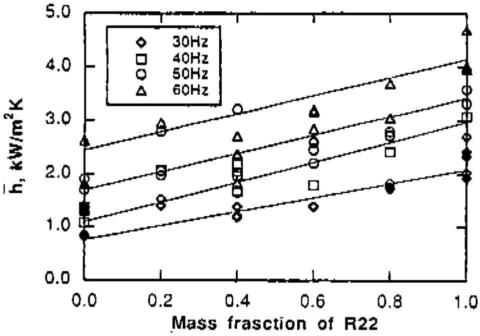


Fig. 10 Variation of average heat transfer coefficient with mass fraction of R22 (indicated No. is frequency of input voltage)

The decrease of COP at 30 Hz for mixtures whose mass fraction of R142b exceed 60% represents the low efficiency of compressor. Under severe conditions where compressor speed is sufficiently low and specific volume at inlet of compressor is large, the compressor efficiency goes down and COP decreases sharply.

Average heat transfer coefficient, $\bar{h}$, increases as fraction of R22 becomes large. It is closely related to large value of $\dot{m}_r$ for R22 at same frequency in contrast to R142b. Generally, heat transfer coefficient varies almost linearly with increasing mass flow rate if other properties are constant. Since both R22 and R142b have similar values of thermal conductivity, specific heat and viscosity of liquid phase which affect the heat transfer coefficient greatly, $\bar{h}$ of R22 is about twice as large as that of R142b due to larger $\dot{m}_r$.

For all compositions, all of $\dot{Q}$, COP and $\bar{h}$ increase as compressor speed becomes faster as a result of increased mass flow rate.

Above results can be understood as follows. Conventionally, compressor is slowed down to meet the reduced thermal load when pure refrigerant is used. On the contrary, the same effect of load management can be carried out by varying composition when mixtures are used. This possibility is also found in R22/R142b mixtures in this study. $\dot{Q}$ of R142b at 50~60 Hz is analogous to that of R22 at 30 Hz. Moreover, COP and $\bar{h}$ of both cases are similar. Therefore there can be no degradation of cycle efficiency and heat transfer coefficient when R22/R142b mixtures are used for capacity modulation.

It can be concluded that capacity control is possible by changing the mixing ratio of R22 and R142b without decrease of COP and heat transfer coefficient.

3.3 Constant heat transfer rate

Analyses are performed and results are given here for a fixed heat transfer rate of evaporator. The evaporating capacity is chosen to be 2 kW among the capacity of 1 to 4 kW because the experimental apparatus can not cover whole range of parameters to be considered.

The heat transfer rate, $\dot{Q}$, and mass flow rate of refrigerant, $\dot{m}_r$, are shown in Fig.11. $\dot{Q}$'s of all mixtures are nearly maintained at 2kW. It is noticeable that mass flow rates of all mixtures have nearly same value. This phenomena result from similar values of latent heat of R22 and R142b. It is interesting to know that the condition of constant heat transfer rate coincides with the condition of constant mass flow rate.

Figure 12 represents the variation of COP with respect to mixing ratio. COP$_{exp}$ is caculated by eq.(8) and COP$_{cal}$ is based on the work calculated by the difference of enthalpies at inlet and exit of compressor. The ratio of COP$_{exp}$ to COP$_{cal}$ may be considered to be the efficiency of compressor.

The input frequency to compressor is maintained at standard or slightly higher values(60~70 Hz) for R142b and lowered as fraction of R22 increases to meet evaporating capacity(2kW). As a result, the compressor efficiency decreases slightly as the fraction of R22 increases.

COP's of mixtures are higher than those of pure refrigerants. Moreover, there exists maximum value of COP which corresponds to mixture with R22 fraction of 60% and about 20% enhancement is possible compared with pure refrigerants. The composition of maximum COP almost corresponds to the composition which have largest temperature difference between bubble and dew points. These results are also similar with those of cycle simulation[11].

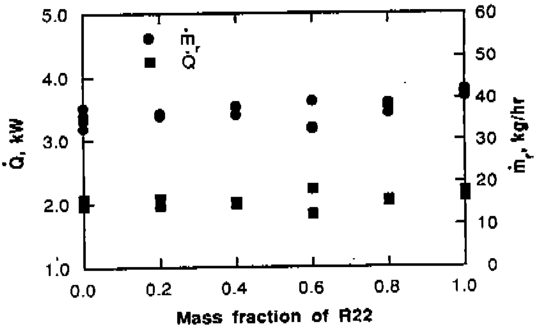


Fig. 11 Ranges of experimental values

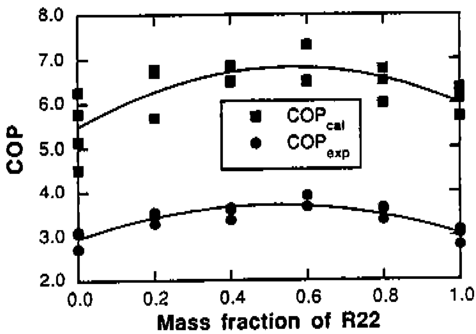


Fig. 12 Variation of COP with mass fraction of R22 ($\dot{Q}$=2kW)

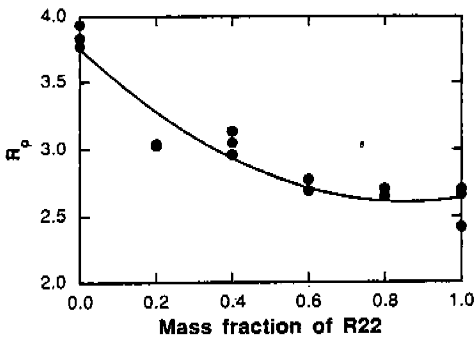


Fig. 13 Variation of pressure ratio with mass fraction of R22 ($\dot{Q}$=2kW)

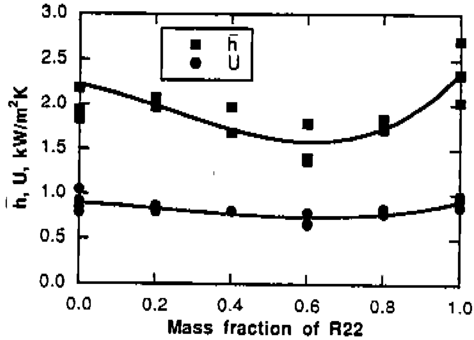


Fig. 14 Variation of average heat transfer coefficient and overall heat transfer coefficient with mass fraction of R22 ($\dot{Q}$=2kW)

Shown in Fig.13 is the pressure ratio $R_p$ which is defined by the ratio of exit to inlet pressure of compressor. $R_p$ increases as fraction of R142b increases. $R_p$ of R142b is about 40% higher than that of R22. The rate of change in $R_p$ with composition is steep at R142b rich mixtures. This higher pressure ratio does not seem to affect the efficiency of compressor largely in our system. But in real system where pressure ratio is larger than that of this study, compressor efficiency may be somewhat lowered for R142b.

The average heat transfer coefficient of refrigerant, $\bar{h}$, is shown in Fig.14. Mixing of refrigerants leads to decrease of heat transfer coefficient. This effect can also be found in other refrigerant mixtures[6,7] such as R22/R114 mixture. The average heat transfer coefficient of refrigerant, $\bar{h}$, has its minimum value at about 60% of R22 fraction which nearly corresponds to the point of maximum COP.

In real system overall heat transfer coefficient is more important than heat transfer coefficient. The overall heat transfer coefficient, $U$, at evaporator is also shown in Fig.14 and shows decrease by mixing as in the case of $\bar{h}$. But degree of decrease is not so large as that of $\bar{h}$. Although $\bar{h}$ decreases by mixing, heat transfer coefficient of water side nearly does not change and is small compared with $\bar{h}$. As a result, the variation of $U$ with respect to composition is relatively small. It can be deduced that no major modification is needed for heat exchanger when R22/R142b mixtures are adopted in a heat pump system designed for R22.

## 4. CONCLUSIONS

Experiments are performed to investigate the capacity modulation and the enhancement of COP in a heat pump system. Mixtures of R22 and R142b are chosen as the working fluids.

It is shown that the evaporating capacity can be varied up to 200% based on minimum load without loss of performance for a fixed rotational speed of compressor.

The results also show that the COP of the heat pump has a maximum value at a certain mixing ratio if the evaporating capacity is fixed. At the condition of fixed evaporating capacity, it is known that the overall heat transfer coefficient is not altered practically although there is some degradation in heat transfer coefficient itself in the mixture side.

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