Comparison of Performance of a Residential Air-Conditioning System Using Microchannel and Fin-and-Tube Heat Exchanger

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ABSTRACT

In this study the seasonal performance of a residential air conditioning system having either a fin-and-tube condenser or a microchannel condenser is experimentally investigated. Microchannel heat exchangers offer a higher volumetric heat exchange capacity and a reduced refrigerant charge amount. However, the operating characteristics and the seasonal energy efficiency ratio (SEER) of the residential air conditioning system using a microchannel condenser have not been well known.

For this investigation, a commercially available 7 kW capacity residential air conditioning system having a fin-and-tube condenser served as the base system. After testing the base unit with the fin-and-tube condenser, the condenser was replaced by a microchannel heat exchanger with the same face area under identical test conditions. The test results show that the system with a microchannel heat exchanger has a reduced refrigerant charge amount of 10%, the coefficient of performance increased by 6% to 10%, and the SEER increased by 7% as compared with those of the base system. Moreover, the condensing pressure of the system is decreased by 100 kPa and the pressure drop across the condenser is decreased by 84%. The microchannel heat exchanger enhances the SEER of the residential air conditioning system by providing better heat transfers at reduced pressure drops.

1. INTRODUCTION

To improve the system performance of air conditioners and to develop environmentally safe air-conditioning systems, research for each component of air conditioning systems has been extensively conducted. For heat exchangers, compactness and higher volumetric heat transfer capacity are required. The microchannel heat exchanger has a great potential for condensers or evaporators in these respects. Due to a higher air-side heat transfer performance of microchannel heat exchangers, the volume of the heat exchanger can be significantly reduced at the same cooling or heating capacity. This high air-side heat transfer performance is attributed to the small hydraulic channel diameter, the characteristics of air flow over flat channel-and-fin geometry, and the reduced contact resistance between fin and tube. With the decrease of the heat exchanger volume, the refrigerant charge amount can be decreased as well. Furthermore, it has a merit for recycling process. For typical fin-and-tube heat exchangers, tubes and fins have different materials. However, fins and tubes of microchannel heat exchangers are both made of the same material, aluminum. Therefore, it has a much higher recycling potential.

Since the microchannel heat exchanger technology has been developed recently, its research is limited. Kim and Groll (2003) compared the performance of a heat pump having a fin-and-tube condenser with that having a microchannel condenser. They replaced the fin-and-tube condenser by a microchannel heat exchanger without changing other components of system including the expansion device. The microchannel condenser has about 23% smaller face area and 32% smaller internal volume than those of the fin-and-tube condenser. Among their test results, a vertically oriented microchannel condenser with 20 fpi fin density shows 2.7% lower cooling capacity and 3.2% lower compressor power. However, the same microchannel heat exchanger slanted 15^{o} from the vertical has 4.3%

higher cooling capacity and 1% lower compressor power consumption. Their COPs are 1.0 and 4.8% higher than the baseline, respectively. Kim and Bullard (2002) investigated the performance of a window room air conditioner with a microchannel condenser. They replaced the fin-and-tube heat exchanger with a microchannel heat exchanger having 50% smaller volume than the original fin-and-tube condenser, and the charge amount was 35% smaller than that of the baseline fin-and-tube condenser system. For the microchannel condenser system, the degrees of subcooling and superheating were controlled to the same values of the baseline fin-and-tube condenser system. They obtained almost the same COP, but the compressor power of the microchannel condenser system is 2% less than that of the baseline system. Cho et al. (1999) evaluated the system performance of the package air-conditioner having a microchannel condenser. They reported that the same cooling capacity was obtained with the smaller microchannel condenser having 82% face area of the fin-and-tube condenser. In addition, the charge amount of the refrigerant is decreased by 35% and 45% when the fin pitch decreased from 3.0 mm to 2.5 and 2.0 mm, respectively. Jeong et al. (2004) investigated the performance of three microchannel condensers having different air-side heat transfer areas but the same face area, which is 78% of the fin-and-tube condenser, by adjusting tube and fin pitches. With increase of the heat transfer area by 73.9%, 84.2%, and 88.5% from that of fin-and-tube heat exchanger, the cooling capacities and COPs approached to that of the fin-and-tube condenser system. Bea and Han (1996) experimentally studied the potential application of the microchannel condenser to the residential air conditioner. They suggested that the microchannel heat exchanger can reduce the condenser volume by 40% at the same condenser heat transfer rate compared to that of the fin-and-tube condenser, and the charge amount can be reduced to by 22%. They also investigated the effect of the different number of pass of the microchannel heat exchanger on the system performance. When the pass of microchannel condenser was changed from 4 to 6, the cooling capacity was increased by 4%, and the compressor power was decreased by 0.9%.

As summarized, most of previous studies focused on reducing the condenser size at a lower or similar COP as compared to that of fin-and-tube condenser system rather than enhancing system performances, such as the COP and SEER. As achieving higher SEERs became an important issue, the current study investigated the performance enhancement while using the same face area microchannel condenser with that of fin-and-tube condenser. Moreover, its steady state and cyclic operating characteristics was investigated.

2. EXPERIMENTAL SETUP

2.1 Test Set-up and Test Unit

All tests were conducted in an outdoor chamber and an indoor loop as shown in Fig. 1. Outdoor test unit of the air conditioner was installed inside the outdoor chamber, and the indoor unit was installed in the indoor loop. Details of the indoor loop are illustrated in Fig. 2. Air flow rate was measured by using the 0.127 m diameter nozzle. The test unit used in this study is a residential air conditioner having a rated cooling capacity of 6.25 kW and R22 as its working fluid. The unit is composed of the basic cycle components, a rotary compressor, a condenser, a short tube orifice, and an evaporator as shown in Fig. 1. Table 1 shows the specification of the fin-and-tube and microchannel heat exchanger. They have the same number of rows, face area, and finned length. The total number of tubes of the fin-and-tube heat exchanger is 18, and 71 for the microchannel heat exchanger. The fin-and-tube heat exchanger has a single refrigerant circuit. The microchannel heat exchanger is divided in two parallel tube groups, 48 tubes and 23 tubes.

Table 1 Specification of Test Heat Exchangers

	Fin-and-tube heat exchanger	Microchannel heat exchanger
Fin shape	Plate Fin	Louvered fin
Number of rows	1	1
fpi	18	17
Tubes per row	28	71
Face area	0.56 m²	0.56 m²
Finned length	850.9 mm	850.9 mm

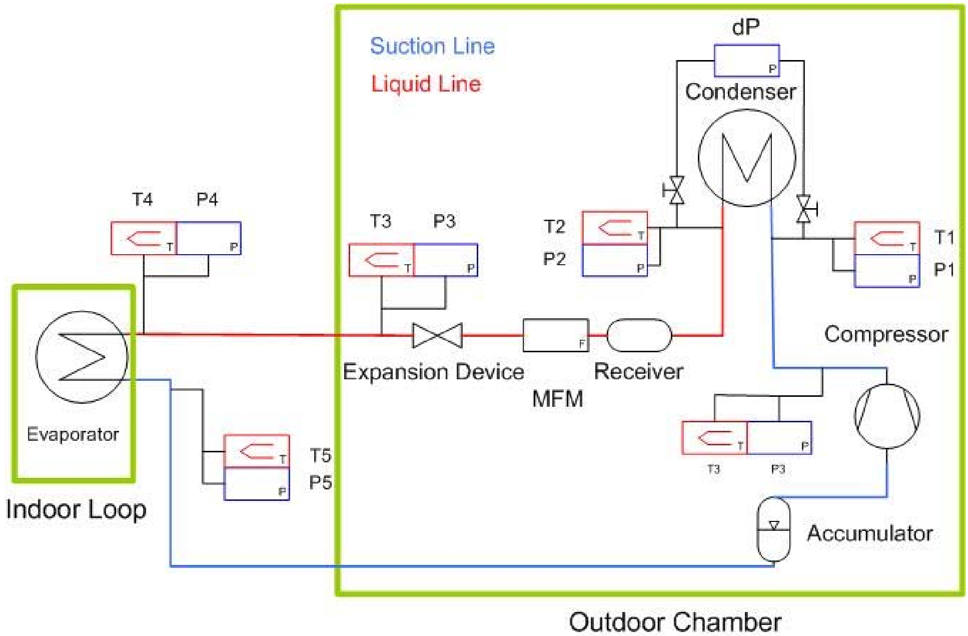


Figure 1 Schematic of Test Set-Up

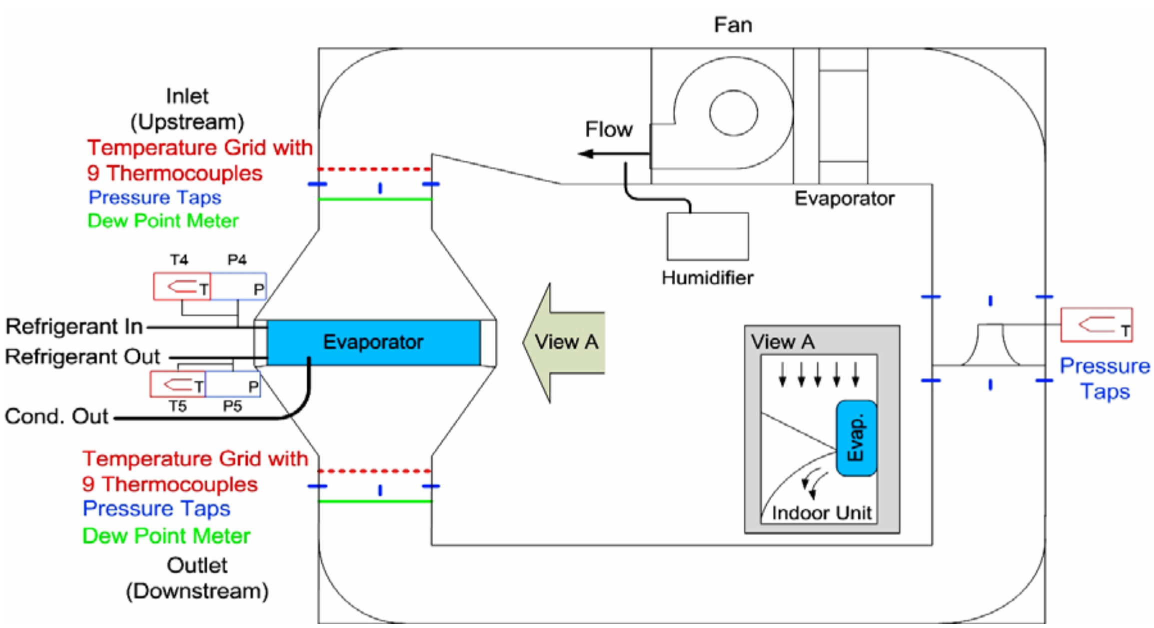


Figure 2 Details of Indoor Loop

2.2 Instrumentations and Test Procedures

Fig. 1 indicates the measurement points of the experiments. The refrigerant pressures were measured by using pressure transducers in 3.5 MPa and 1.8 MPa full scales. They have an accuracy of ± 0.11% of full scale. Five in-stream T-type thermocouples with an accuracy of ± 0.2^{o}C were installed at the same places with the pressure transducers. To measure a differential pressure drop of refrigerant across the condenser a differential pressure transducer was installed. It has a measuring range of 0 to 1.0 MPa and an accuracy of ± 0.2% of full scale. Thermocouples were attached on the U-bend surfaces of the fin-and-tube condenser and on the microchannel tubes close to the headers of the microchannel condenser to observe the refrigerant phase transition in condensers. To measure the average air-side temperatures, nine thermocouples were evenly placed on a grid at the inlet and outlet of the evaporator as shown in Fig. 2. The inlet and outlet humidity ratio of the evaporator were measured by using two chilled mirror dew-point sensors with an accuracy of ± 0.2°C. Two watt transducers were used to measure the power consumptions of the compressor and the outdoor unit fan. The accuracy of watt transducers is ± 0.5% of full scale. Indoor unit fan power was independently measured before system tests. All data were logged and written to a file by the PC. Scanning time for each data point was 5 seconds.

The baseline tests were first conducted by using the original product provided from the manufacturer without any system modification. After the baseline test, the condenser of the original product was replaced by the microchannel heat exchanger without changing other components of the system. These tests were conducted at ARI test A, B, C, and D conditions as shown in Table 2 (ARI, 1989). Except test D, other tests are steady-state tests. Test D is a cyclic test, in which the unit shall cycle with the compressor on for 6 minutes and off for 24 minutes. Baseline system tests and microchannel system tests were conducted with a constant indoor air volumetric flow rate of 0.25 m^3/s.

Table 2 Test Conditions

Cooling Test Condition	Indoor unit		Outdoor unit	
	Air entering		Air entering	
	DB (°F/°C)	WB (°F/°C)	DB (°F/°C)	WB (°F/°C)
Test A in steady state	80/26.7	67/19.4	95/35	75/23.9
Test B in steady state	80/26.7	67/19.4	82/27.8	65/18.3
Test C in steady state (dry coil)	80/26.7	57/13.9	82/27.8	65/18.3
Test D in cyclic (dry coil)	80/26.7	57/13.9	82/27.8	65/18.3

2.3 Data reduction

Air-side cooling capacity was calculated by eq. (1). In eq. (1), the heat loss from the air in duct to ambient was considered. Energy balance between refrigerant-side and air-side was checked at every test by using eq. (2). The cooling capacity of refrigerant side was calculated by eq. (3). The refrigerant inlet enthalpy of the evaporator was assumed to be the same as that of condenser outlet. Energy balance of all tests was within 6%.

$$\dot{Q}_a = \dot{Q}_{lat} + \dot{Q}_{sens} + \dot{Q}_{loss} = \dot{m}_w \Delta h_{lv} + \dot{m}_a c_p (T_{in} - T_{out}) + UA_{duct}(T_a - T_{amb}) \quad (1)$$

$$Energy\ balance = \frac{\dot{Q}_a - \dot{Q}_r}{\dot{Q}_a} \times 100 \quad (2)$$

$$\dot{Q}_r = \dot{m}_r (h_{eva,out} - h_{eva,in}) \quad (3)$$

The coefficient of performance (COP) was calculated by eq. (4), and the uncertainty of COP of present tests was found to be ± 4%. The calculation of the seasonal energy efficiency ratio (SEER) followed the calculation methods for single-speed compressor and single-speed condenser fan unit as shown in eqs. (5), (6), and (7). (ARI, 1989) The results of test C and D shall be used to calculate a degradation coefficient of C_D.

$$COP_c = \frac{\dot{Q}_{c,a}}{W_{fan,id} + W_{fan,od} + W_{compressor}} \quad (4)$$

$$C_D = \frac{1 - EER_{cyc,dry} / EER_{ss,dry}}{1 - CLF} \quad (5)$$

$$PLF(0.5) = 1 - 0.5 \times C_D \quad (6)$$

$$SEER = PLF(0.5) \times EER \quad (7)$$

3. RESULTS AND DISCUSSION

3.1 Charge Optimization

The comparison of the performances of both systems has to be done at optimum refrigerant charge conditions. The optimum charge was determined by comparing the COP and the degree of superheating and subcooling of the system by changing the charge amount under "test A" condition. Fig. 3 shows the variations of COP with refrigerant charge amount. Fig. 4 shows the degree of superheating and subcooling of the system. It was observed that the superheating decreases and the subcooling increases with increase of charge amount as expected. In case of the baseline system, the maximum COP was found at around 1.5 kg of charge amount. However, the superheating is less than 1.0^{o}C at that refrigerant charge. Although the COP at 1.35 kg charge is less than the COP at 1.5 kg refrigerant charge by 1%, the optimum charge was decided to 1.35 kg for system stability since the superheating of 5^{o}C was found at that charge amount. For the microchannel condenser system, the optimum charge was decided to 1.21 kg. The maximum COP and the superheating of 6.8^{o}C were found at that charge amount. The optimum refrigerant charge amount of the microchannel condenser system is reduced by 10% due to the smaller internal volume of the new condenser. The internal volume of the microchannel is less than 49% that of the fin-and-tube condenser.

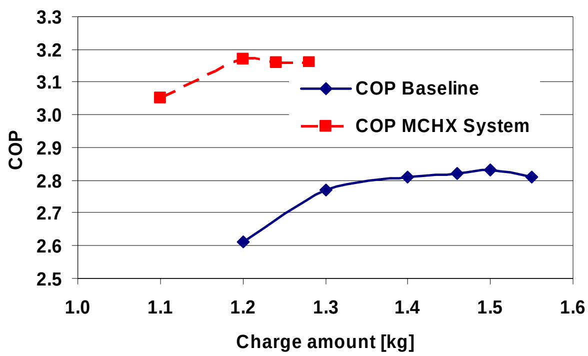


Figure 3 COP vs. Refrigerant Charge Amount

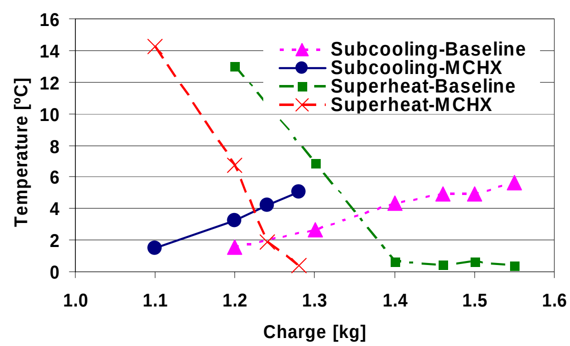
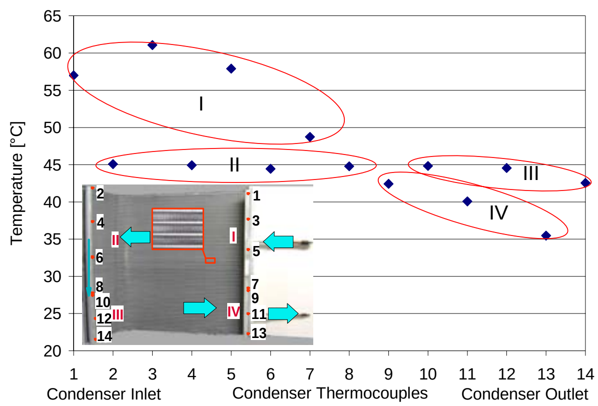


Figure 4 Degree of Superheating and Subcooling vs. Refrigerant Charge Amount

3.2 System Operating Characteristics

Fig. 5 shows the surface temperatures of the microchannel condenser during system operation. The thermocouples are grouped and named as I, II, III and IV. The temperatures of group I show the state of refrigerant is superheated vapor. During the refrigerant flow through the upper part of 48 tubes, condensation of refrigerant occurs. Thermocouple groups II and III show constant temperatures during this condensation. Around the condenser outlet part, the refrigerant can be found to be subcooled (group IV).

Fig. 6 shows cycle characteristics of the systems with a fin-and-tube condenser and a microchannel condenser. The big difference between two systems is the lower condensing pressure of the microchannel condenser system than that of the baseline system. Also, a decrease of the enthalpy difference between the suction and discharge of the compression process for the microchannel condenser system can be observed. Details of pressure variations and temperature variations for both systems are shown in Fig. 7 and Fig. 8, respectively. The condensing pressure of the microchannel condenser system is 6.3% lower than that of the baseline system for the condition "test A." This decrease of the condensing pressure can be explained by the increase of the heat transfer capacity in the microchannel condenser. On the other hand, the average evaporation pressure is about the same for both systems within 1.4% variation for the condition "test A." As a result, the pressure ratio between the condensing and evaporating pressures for the microchannel condenser system is 4.5% less than that of the baseline system, which contributes to the reduction in the compressor power. Table 3 shows the comparison of the compressor power consumption between baseline system and microchannel condenser system. The compressor power consumption decreased by 6.8% for conditions "test A" and "test B", and by 8.2% for "test C." It is also observed that the compressor discharge temperature of the microchannel condenser system is 4.8% lower than that of the fin-and-tube system as shown in Fig. 8.



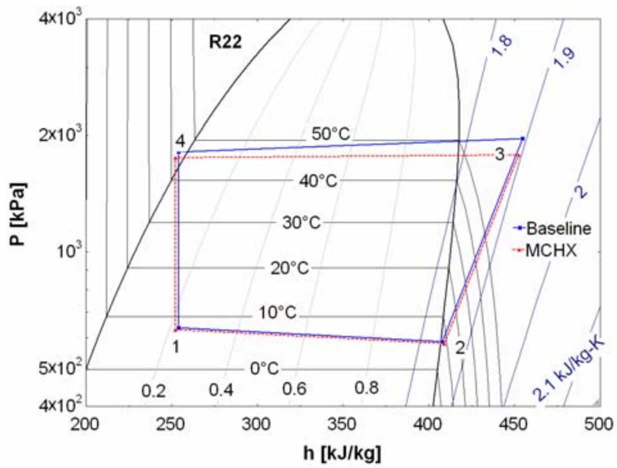


Figure 5 Average Surface Temperatures of Microchannel Condenser

Figure 6 Cycles in P-h Diagram

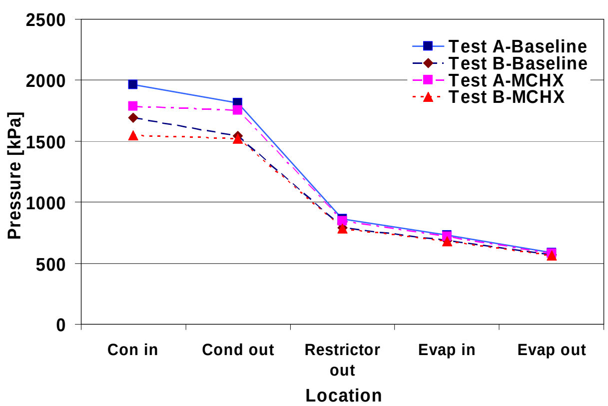


Figure 7 Comparison of Cycle Pressures

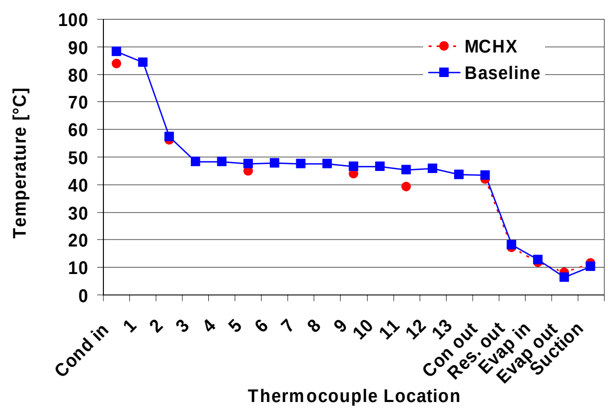


Figure 8 Comparison of Cycle Temperatures

Table 3 Compressor Power consumption

Test conditions	Fin-and-tube condenser	Microchannel condenser	Change (%)
Test A	2,036	1,898	-6.8
Test B	1,806	1,683	-6.8
Test C	1,776	1,630	-8.2

Table 4 shows the comparison of the refrigerant pressure drop across the condenser between the baseline and the microchannel condenser system. For all tests, the pressure drop of the microchannel condenser is 84% lower than that of the fin-and-tube condenser. This lower pressure drop of the microchannel condenser can be explained by two reasons. One is the increase of the refrigerant flow cross sectional area of microchannel condenser by 48/23 parallel tubes than that of the fin-and-tube condenser. The other is shorter refrigerant distance of the microchannel heat exchanger than that of the fin-and-tube condenser.

Table 4 Refrigerant-Side Pressure Drop Across The Condenser

Test	Fin-and-tube condenser (kPa)	Microchannel condenser (kPa)	Change (%)
A	130.1	20.5	-84
B	148.7	23.6	-84
C/D	142.3	24.5	-83

3.3 COP and SEER

Fig. 9 shows the cooling capacities and COPs of each test for both systems. The COPs of the microchannel condenser system are higher than those of the baseline system for all tests. The COP of the microchannel condenser system is 9.5%, 6.1% and 8.7% higher than that of the baseline system for conditions "test A", test B" and "test C", respectively. As compared to the COP change, the change of the cooling capacity between the baseline and the microchannel condenser system is smaller, and for "test B" the cooling capacity of both systems is almost same. Therefore, the increase of COP for the microchannel condenser system can be attributed to the decrease of the compressor power consumption rather than the increase of the cooling capacity.

Fig. 10 shows the variations of the cooling capacity and compressor power consumption during the cyclic tests for "test D." The system was turned on for 6 minutes and then off for 24 minutes. The cooling capacity of the microchannel condenser system much quickly approached to its steady state cooling capacity than that of the baseline system. On the other hand, the compressor power consumption of the microchannel condenser system is less than that of the baseline system. It means that the increase in the condenser heat transfer capacity is beneficial to the cooling capacity during the cyclic conditions. The calculated SEER of the baseline system and the microchannel condenser system is 10.4 and 11.2 Btu/kW-h, respectively. This means that the SEER of the microchannel condenser system is 7.7% higher than that of the baseline system.

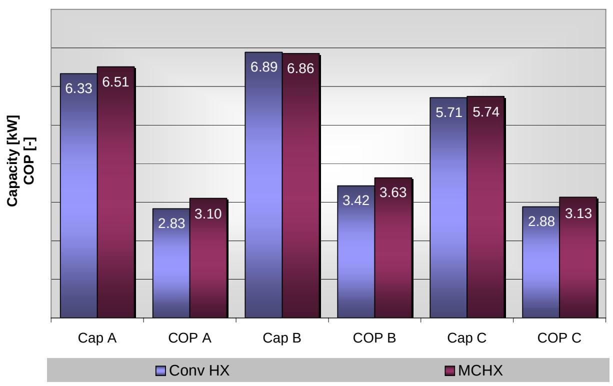


Fig. 9 Comparison of Steady State Performance

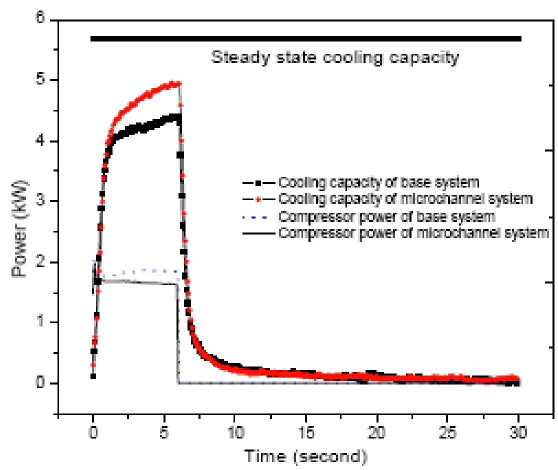


Fig. 10 Comparison of Cyclic Performance

4. CONCLUSIONS

The seasonal performance of an air conditioning system having a microchannel condenser was experimentally evaluated, and was compared with that having a fin-and-tube condenser system. The tested microchannel condenser has the same face area as that of the fin-and-tube condenser, but its internal refrigerant flow volume is 49% smaller than that of the baseline condenser. Due to its smaller internal volume, the optimum refrigerant charge amount of the microchannel condenser system is 10% lower than that of the baseline system. The refrigerant pressure drop across the microchannel condenser is 84% lower than that of the fin-and-tube condenser due to the increase in the cross sectional area of the refrigerant flow and the shorter refrigerant flow distance. The steady state COP of the microchannel condenser system is increased by 6 to 10% as compared with that of the baseline system. This is attributed to the decrease in the compressor power consumption rather than the increase in the cooling capacity. Decrease of the power consumption for the microchannel condenser system is caused by the lower pressure ratio between the condensing and evaporation pressure, which is caused by the increased condenser heat capacity. In the cyclic operation, the cooling capacity of the microchannel condenser system is higher and its compressor power is lower than those of the baseline system. These benefits in the cycle operation are originated from the increased condenser heat capacity. When both the steady state and cyclic performances are considered, it was found that the SEER of the microchannel condenser system having the same frontal area is 7.7% higher than that of the baseline system.

NOMENCLATURE

A	area	(m^2)
C_D	degradation coefficient	
CLF	cooling load factor	
COP	coefficient of performance	
C_p	specific heat	(kJ/kgK)
DB	dry bulb temperature	(^oC)
fpi	number of fins per inch	
h	enthalpy	(kJ/kg)
Δh_{lv}	specific enthalpy of vaporization	(kJ/kg)
$MCHX$	microchannel heat exchanger	
$\dot{m}$	mass flow rate	(kg/s)
$\dot{Q}$	capacity	(kW)
PLF	part load factor	
$SEER$	seasonal energy efficiency ratio	(Btu/kWh)
T	temperature	(K)
U	overall heat transfer coefficient	(W/m^2K)
$\dot{W}$	power	(kW)
WB	wet bulb temperature	(^oC)

subscript

a	air
amb	ambient
c	cooling
cyc	cyclic
eva	evaporator
id	indoor
in	inlet
lat	latent
out	outlet
od	outdoor
r	refrigerant
sens	sensible
ss	steady state
w	water

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