An Investigation of Using Microchannel Condenser to Improve the Performance of Air Conditioning Unit

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ABSTRACT

In the recent years, many researches have been focused towards the new technology in heat exchangers. Besides, the economy of manufacturing takes up a great attention of scientific works. Through this work the use of microchannel condenser achieves increase in energy. The study was conducted on two air conditioning units utilize R-22 as a refrigerant and having different cooling capacities 3516 W and 5274 W. The results showed that the use of microchannel condenser achieves increase in Energy Efficiency Ratio (EER) which reflected on the total efficiency of the device and reduce energy cost. The obtained results indicated that, the use of microchannel condenser led to increase in EER by (4.3:11.3%) for the first unit and and (9.5:29%) for the second one compared with that unit use FTC. Also, the use of MChC led to decrease the pressure drop by (20%:30%) while the convective heat transfer coefficient is improved by (61.6:84.1%).

Keywords: Microchannel condenser; Microchannel heat exchanger; Window air conditioner; Refrigeration; Energy efficiency ratio.

1- Introduction

Todays with the increase of industries which results in increasing energy consumption, many researches have been conducted to develop the refrigeration system in order to reduce and save energy. One of these developments is focused on replacing the finned tube condenser with micro-channel one in air conditioning units. Besides, the heat transfer through microchannel has very potential of wide applications in cooling such as high-power density microchips in the computer processing unit (CPU). The micro power systems and even many other large scales thermal systems require effective cooling capacity. This is a result of the micro size of the cooling system which not only significantly reduces the weight load, but also enhances the capability of removing much greater amount of heat than any of large scales cooling systems.

Many efforts have been conducted in using microchannel condensers instead of finned tube one. The performance of refrigeration system using microchannel condenser, with R-134a, was analyzed experimentally and numerically by Patil et al. [1]. The results obtained were compared with round condenser having the same size. The tests were taken according to Air conditioning and Refrigeration Institute (ARI) condition. It was indicated that the microchannel condenser was more effective at various loads, operating conditions and refrigerants. The COP of microchannel condenser was around 7:8 % higher than the round tube one. The condensing temperature was decreased 2.5 °C when a microchannel condenser was used instead of classic condenser and the refrigerant pressure drop was also decreased from 166 kPa in the round-tube condenser to 57 kPa in the microchannel condenser. The seasonal energy efficiency ratio (SEER) of a residential air conditioning system having either a finned tube condenser or a microchannel condenser was experimentally investigated by Yun et al [2]. For that investigation, a commercially available 7 kW capacity residential air conditioning system having a finned tube condenser served as the base system. It was found that microchannel condensers offer a higher volumetric heat exchange. The test results showed that the system with a microchannel condenser has the coefficient of performance increased by 6:10% 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 condenser enhances the SEER of the residential air conditioning system by providing better heat transfer at reduced pressure drop.

The heat transfer coefficient for mini-channel aluminum tube condenser was studied by Fernando et al. [3]. It was reported that the overall heat transfer coefficient of the micro-channel condenser was 62% higher than that of a plate-type one. Chen et al. [4]

studied the performance of the liquid-vapor separation condenser (LSC) and compared it with that of the system having a baseline fin-and-tube one at various ambient temperatures from 29 to 43 °C. The results obtained, for R-22 as refrigerant, showed that both of the cooling capacity and energy efficiency ratio (EER) of the two systems were almost the same. The LSC is having just 67% of the heat transfer area of the baseline condenser.

Park and Hrnjak [5] studied the performance of different types condensers for the residential air conditioning systems charged with R-410A. Two air-conditioning systems, one with a microchannel condenser and the other with a round-tube condenser, were experimentally examined, while the other components of the two systems were identical except the condensers. The COP of the system with the microchannel condenser was 13.1% higher than that with the round tube condenser.

Roth et al. [6] indicated in their study that, the use of MChC instead of round tube one, with R-22 as a refrigerant, improves the EER from 11 to 12.2%. Hu and Chao [7] found that microchannel condensers exhibit higher heat transfer coefficients with the same Reynolds number. The condensation heat transfer coefficient was higher than that in the tubes with the diameter of centimeter. Pressure drops in the micro-condensers with smaller channels were higher due to the increased transition loss.

Bhatkar, et al [8] investigated experimentally the performance of compression refrigeration system using microchannel condenser with R134a and drop in substitute R152a with aluminum microchannel condenser at condensation temperature of 48 °C and evaporation temperature varied from -10 to 15 °C. It was found that, the product of overall heat transfer coefficient and surface area of the condenser (UA) were more for microchannel condenser than conventional condenser. The pressure drop was less in microchannel condenser than conventional condenser by around 0.1 to 0.5 bar. Also, the increasing of saturation temperature showed a significant effect on decreasing the pressure drop. Kandlikar [9] conducted a review for history, advances, and challenges in liquid flow and flow boiling heat transfer in microchannels. It was found that, the use of micro-scale passages improved heat transfer. It was recorded that, microchannels are gateway to flow modification through such Nano-scale surface enhancement configurations.

Jhariya, et al [10] studied two condensers in the same air-conditioning system. One with round tubes and the

second one with flat microchannel tubes. The window air conditioning systems was designed for R-22 and its alternative R-410A. The experiments were conducted at atmospheric temperatures ranged 35 °C to 41 °C. The performance of the air conditioner with R-410A was compared with the baseline performance with R-22. For R-22, the heat transfer in the round tube condenser was 0.75% of that of microchannel condenser; while for R-410A it was 2.39% higher than the microchannel condenser (The EERs of the system with microchannel condenser is higher than the system with the round tube condenser). The mass flow rate, heat rejected in the condenser and the compressor power of the system with microchannel condenser is lower than the system with the round tube condenser. The previous review indicates little attention of studying the performance of the Air Conditioning unit with microchannel condenser at various operating conditions.

2. Aim and Research Significance
The main objective of this study is to experimentally investigate the performance of microchannel condenser (MChC) in comparison with traditional round finned tube condenser (FTC) at different outdoor temperature and different capacities size of air conditioning unit, at steady state condition. Also, comparing the convective heat transfer coefficient for traditional finned tubes and the new microchannel condenser is performed.

3. Experimental Program
The experimental program was conducted on the system that shown schematically in Fig. 1. The system consists of split air conditioning units, two test rooms and experimental instrumentations. The air conditioning systems were charged with Freon R-22. Four split air conditioning units were taken into consideration in order to be experimentally tested. Two units of them were having 3516 W cooling capacity for each one while the other two units were 5274 W capacity for each one. The two indoor units for each cooling capacity are identical. They were finned tubes coils. For each two, the condenser was chosen of two types for the purpose of study. First type was finned tubes condenser while the other one was microchannel condenser as indicated in Table (1). The indoor and outdoor units are installed in air conditioned two rooms in order to carry out the experiments at variable conditions.
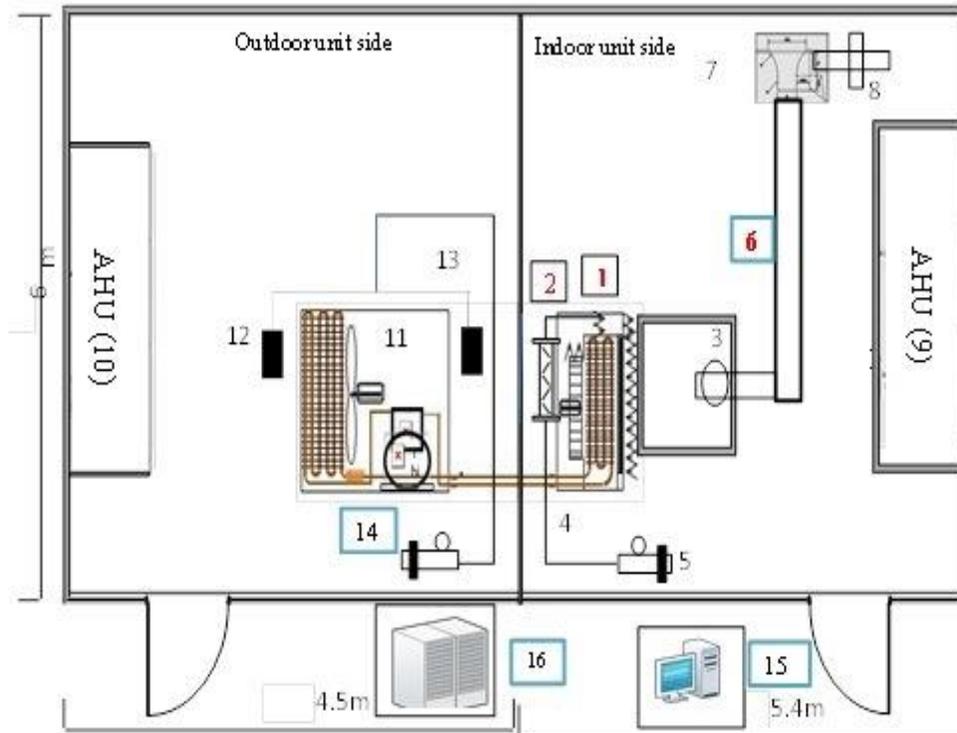
1- Indoor unit.
2- Air inlet samples hub for indoor unit.
3- Receiving chamber.
4- Air sample hose for indoor unit.
5- Psychometric box indoor unit.
6- Air outlet hose.
7- Air flow measurement tunnel.
8- Variable speed blower.

9- AHU air handling unit for indoor side.
10- AHU air handling unit for outdoor side.
11- Outdoor unit.
12 -Air sampling unit at air inlet to outdoor unit.
13- Air sampling unit at air outlet of outdoor.
14- Psychometric box outdoor unit.
15- Measuring instruments panel.
16- Data output unit.

Fig. 1 Schematic layout of the experimental system

Table 1 Tested and Geometric characteristics for the two condensers.

	Unit cooling capacity			
	3516 [W]		5217 [W]	
	FTC	MChC	FTC	MChC
Rated cooling capacity	3156	3156	5274	5274
Refrigerant	R-22	R-22	R-22	R-22
Face length	77cm	74.5 cm	78 cm	76cm
materials	Copper	Aluminum	Copper	Aluminum
Inside dimensions [mm]	d=4.8	1.3*1.3	d=7	1.3*1.3
tube	4	30	4	30
Indoor temperature	27 dry-bulb & 19 wet-bulb			
Compressor	Reciprocating			
Capillary tube	Straight capillary tube			

The indoor unit is installed on a stand frame as shown in Fig. 2 and the all are attached to an air receiving chamber. The received chamber is equipped with sensors for measuring the temperature and pressure. Air sampling hub is installed at air inlet to the indoor unit. Samples of air that enters to the indoor unit were taken through 80 holes distributed on a hub, as shown in Fig. 3, in order to obtain the mean value of the measured parameter. The collected air samples were taken through an insulated hose to a measuring psychometric box in order to measure the dry and wet bulb temperatures of air. The inside condition of the indoor unit test room was changed and adjusted by a central air handling unit in order to maintain the inside condition according to British standard (BS ISO 5151:2010).
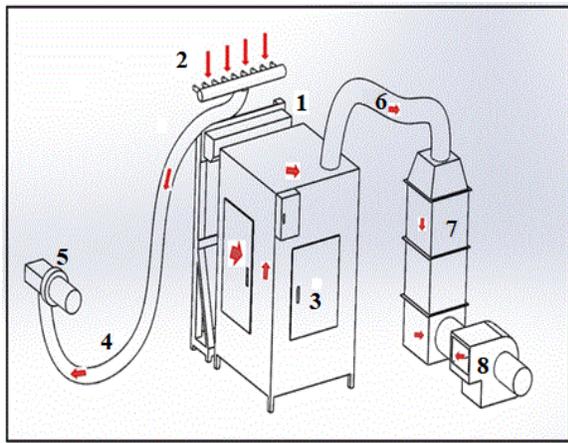
1-Indoor unit   2- Air inlet samples
3- Air chamber   4-Air sample hose,
5- Dry and wet bulb measuring,
6- Air outlet hose,
7- Air flow measurement tunnel,
8- Air outlet blower.
Fig. 2 Indoor unit test room
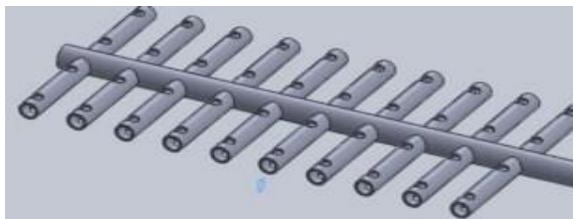
Fig. 3 Air sampling hub

The air which be cooled through the indoor unit was collected inside the receiving chamber in order to measure its pressure as well as its dry and wet bulb temperatures. The chamber was 0.4×0.4×0.7 m. It was covered with air tight lid and insulated with 50 mm thickness expanded polyurethane foam with covered steel faces. The pressure inside the chamber was measured by

the aid of a barometer. The dry and wet bulb temperature of the air was measured at chamber inlet and outlet. In order to measure the wet bulb temperature of the air, an automatic water recharge device and a water supply vessel was equipped, as shown in Fig. 4.
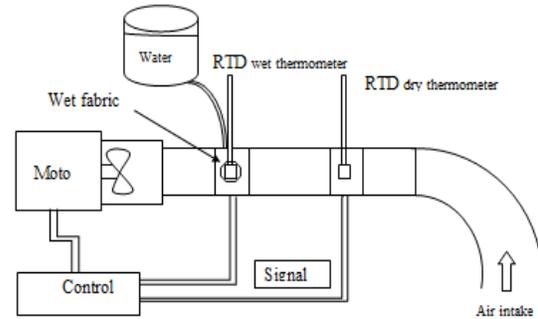
Fig. 4 Psychometric box

The cooled air carried out in air outlet hose enters the nozzles tunnel, Fig. (5). The tunnel was made of 2.3 mm thickness with inside cross section area of 450×500 mm and length (980 mm). Three nozzles were installed inside a steel tunnel as shown in the figure. They were compliant with JIS $\Phi50$, $\Phi70$, $\Phi100$ mm. Each nozzle is electronically opened according to the rated cooling capacity.
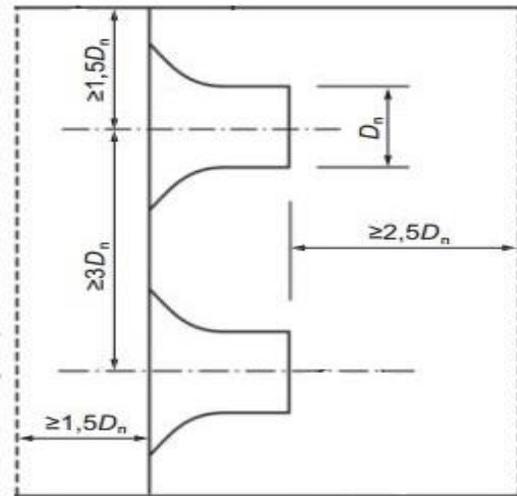
D_n =JIS $\Phi50$, $\Phi70$
Fig. 5 Air flow nozzle

The outdoor unit was also installed on a stand frame. The unit was installed in an insulated room. The room was supplied by air handling unit in order to maintain its inlet temperature and relative humidity at the required test conditions. Two sampling hubs were attached to the unit at air inlet and outlet for the purpose of measuring the dry and wet bulb

temperature. The power consumed by the compressor, as well as the electric current were measured by a wattmeter and an ammeter. The inlet volt was fixed at 220 V by the aid of stabilizer. Samples of air enters and leaves to the outdoor unit were taken to temperature measuring purposes as the same I indoor unit. The inside condition of the outdoor unit test room was changed and adjusted by a central air handling unit in order to be simulated to the change of ambient condition.

Platinum RTDs (Resistance Temperature Detectors) thermometers were used for measuring the temperature of the samples of air. These sensing elements were made of pure platinum wire coil (wire wound) encapsulated in ceramic or glass, or a thin film platinum deposited on a ceramic substrate. Platinum RTDs have a positive temperature coefficient. Their electrical resistance increases as temperature rises in a known and repeatable manner. Their linearity and unmatched long term stability firmly establish platinum RTD elements as an ideal sensor for most industrial applications. Thin film elements offer performance equal to standard wire wounds, but with improved cost, size and convenience. The thermometers measuring range was (-200 to 200 °C), Accuracy: $\pm$ (0.1% of reading + 0.2°C).

A control and measuring panel was located in the monitoring room for installing of meters and reading sensors and a computer interface, Fig. 6. The outputs of control or monitoring instruments were displayed. Those instruments involve part of a system that users could access, to see all of the indicators and parameters of the air conditioning unit such as (dbt, wbt, volt, ampere……etc.) on the screen. The measured data were transferred to a computer to be recorded. The control panels are often equipped with push buttons and analog instruments, digital devices in many cases touchscreens are used for monitoring and control purposes.

Digital power monitoring systems support simple monitoring tasks as well as more complex tasks, such as power trending, controlling and identification of sources of energy consumption and load profiles of power supply segments. The digital power meters were used to indicate the voltage difference and the current intensity, as well as the power of the air conditioning device, for calculating the air conditioning unit performance. The max input to digital power monitoring was: voltage: 1000 V, current: 20 A. as shown in, Fig. 7.

voltage and current accuracy	$\pm$(0.1% of reading+0.1% of range)

power accuracy	$\pm$ (0.1% +0.1% of range)
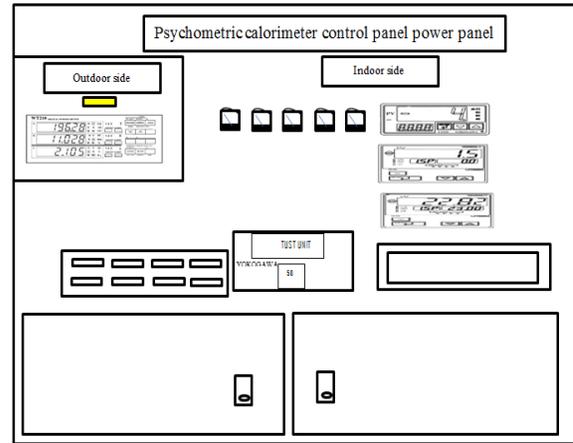
Fig. 6 control and measuring panel
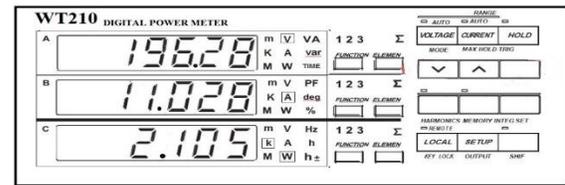
Fig. 7 Digital Power Meter

A computerized data Acquisition system was installed for collecting and recording data to test result process designed specifically for use with the test laboratory. The system was including a personal computer with necessary I/O hardware, printer, software and analog data acquisition instrument.

3- Results and Discussion
The obtained experimental data explain the effect of using micro-channel condenser in air conditioning unit. The HVAC systems may be affected by many parameters such as; energy efficiency ratio EER, condenser pressure drop, heat transfer coefficient calculation. So, the researches should be directed towards the improvement of environment in the various fields.

3. 1 Energy Efficiency Ratio EER
The Energy Efficiency Ratio "EER" is the ratio of the total cooling capacity $\dot{Q}$ achieved by the unit to the effective power input to the device at any given set of rating conditions. For cooling, it could be written as:

$$EER = \frac{\dot{Q}}{Power} = \frac{\dot{Q}}{Power_c + Power_{fan}} \qquad (1)$$

The EER is affected by the type of condenser as indicated in Fig. (8). It is clearly seen that the EER decreases with the increase of ambient condition. This is because the increase of ambient temperature results in an increase of the compressor work done, which in turn reflected on the power. It was observed that, for the unit has 3516 W rating cooling capacity, the increases in EER ranges between 0.42 and 0.6 when MChC was installed than the case of FTC at the testing temperature range. The percentage increase in EER. is between 4.3 % to 11.3% as indicated in Fig. (9). For the unit of 5274 W, EER increases by 0.9 to 1.55 in case of installing MChC. The corresponding percentage increase is between 9.47% and 29%. It is known that the increase in EER support and implement the requirements of the Egyptian organization for standardization and quality. Raise the classification of air-conditioning device with energy efficiency card, and consequently increasing the competitiveness of products in the market. When EER increasing given the same energy source and operating conditions, a higher EER heat pump will consume less purchased energy than one with a lower EER the overall environmental impact of a heating or air conditioning installation depends on the source of energy used as well as the EER of the equipment. The operating cost to the consumer depends on the cost of energy as well as the EER or efficiency of the unit.
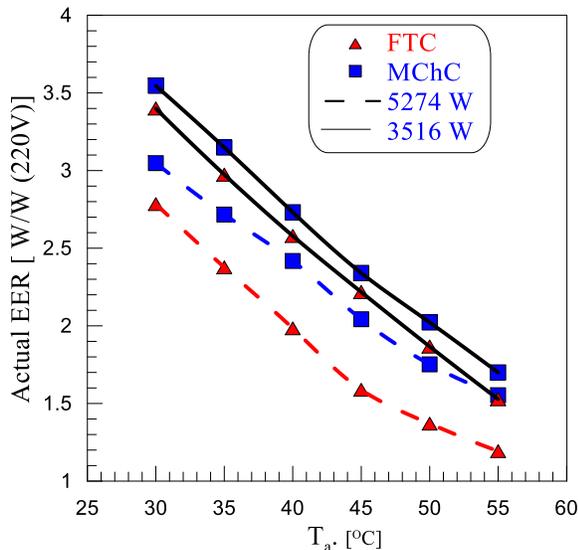
Fig. 8 Effect of ambient temperature on the EER at 220 V

The results obtained, for percentage increase in EER, in the present work were compared with others those work at nearly the same operating conditions and illustrated in Table (2) and Fig. (9). It is noticed that, the percentage increase achieved by Jhariya et al. [10]

was (6.6 and 6.8 %) for the two cases at which the tests were conducted. Chaikhumrop and Trutassanawin [12] achieved percentage increase of (1.14, 2.8, 3.08) % for the three test cases. The figure indicates that percentage increase in EER achieved by the present work is nearly close to that obtained by [10] and higher than that obtained by the other one.

Table 2: Comparison of percentage improvement in EER

Paper No.	Unit C. C. [W]	T_a [°C]	Freon	EER
10	3516	35, 41	R22	6.6: 6.8 %
12	5275	35, 38, 42	R22	1.14, 2.8, 3.08%
Present work	3516	30-55	R-22	4.3:11.3%
	5274			9.5:29%
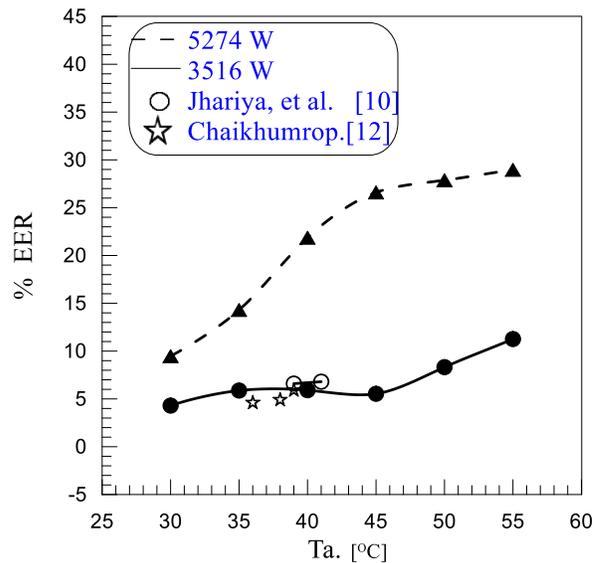
Fig. 9 Effect of ambient temperature on the percentage improvement in EER

3. 2 Condenser Pressure Drop:

The pressure drop in condenser is due to two reasons, friction loss and local loss. Theoretically, the friction loss is given by Darcy correlation [12] as indicated in equation (2):

$$\Delta p = f \left(\frac{l}{d}\right)\left(\frac{\rho u^2}{2}\right) \qquad (2)$$

The local loss is due to any of the following: pipe entrance or exit, sudden expansion or contraction, gradual expansion or contraction, bends, elbow, tee connection and other fittings. Equation (3) represents the local pressure drop [12]:

$$\Delta p = \frac{1}{2} K \rho u^2 \qquad (3)$$

In order to indicate the effect of using micro-channel condenser on the condenser pressure drop, the condenser inlet and outlet pressure was measured by digital electronic pressure transmitter at various ambient temperatures. Fig. (10) illustrates the effect of ambient temperature on the condenser inlet and outlet pressure respectively for 5274 W unit. Generally, it is indicated, that the condenser inlet pressure is ranged between 13.4 bar and 24.2 bar for the unit with finned round tube condenser. A small change in inlet pressure is observed for microchannel condenser (13.2 and 24.1 bar)**.** This small change in the recorded pressure is expected because this pressure is related to the compressor outlet pressure and has no relation to the condenser type. The figure indicates also the condenser outlet pressure for the same unit. The condenser outlet pressure for outdoor unit is ranged between 12.6 bar and 22.3 bar for the unit with finned round tube condenser while a small change 12.6 bar to 23 bar for the unit used microchannel condenser is observed.
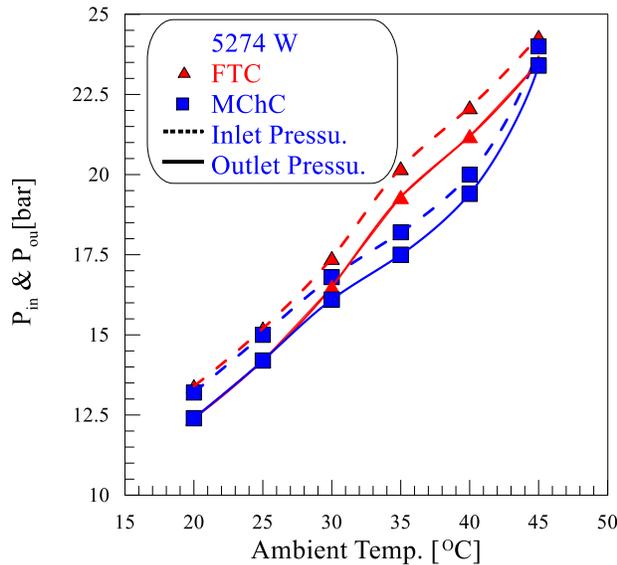
Fig. 10 Effect of ambient temperature on the actual condenser inlet & outlet pressure [bar]

Figure (11) indicates the experimental and theoretical condenser pressure drop for 5274 W cooling capacity unit for the two types of condenser (round tube and microchannel) at different ambient temperature. Experimentally, the condenser pressure drop is ranged between 0.8 and 1 bar for the unit with finned round tube condenser while it was change to 0.6 to 0.8 bar for the unit used microchannel condenser. For all tests,

the experimental pressure drop of the microchannel condenser is lower than that of the finned tube condenser by (20:33.3%).
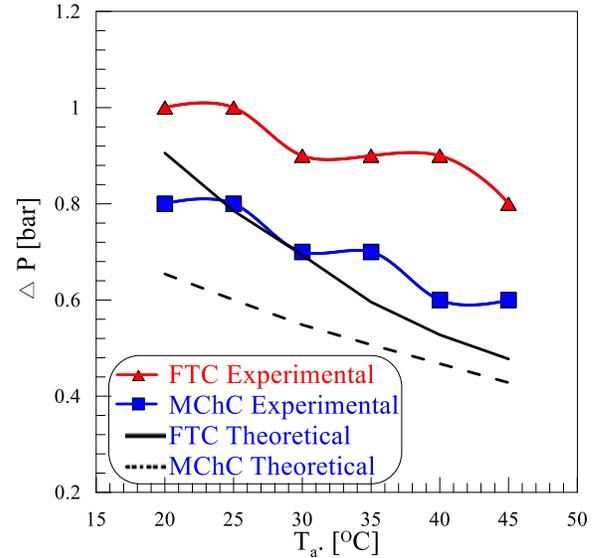
Fig. 11 Effect of ambient temperature on condenser pressure drop.

Theoretical calculation indicated that, the condenser pressure drop changes within the range from 0.48 bar to 0.91 bar for the unit with finned tube condenser while it was changed within the range from 0.43 bar to 0.65 bar for the unit used microchannel condenser. For all theoretical calculations, the pressure drop of the micro-channel condenser is lower than that of the fin-and-tube condenser by (10.3:27.8)%. 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 finned tube condenser. The other is shorter refrigerant path of the microchannel condenser than that of the fin-and-tube condenser.

Clearly, this explains the frequently noted misconception that condensers with small diameter tubes have a higher refrigerant side pressure drop than those with larger tubes. The effect of reducing the diameter is offset by the increased number of parallel channels. Certainly, the limit in controlling the refrigerant side pressure drop in a parallel flow microchannel condenser is a single-pass arrangement.

The results obtained, for percentage decrease in pressure drop, in the present work were compared with the others those work at nearly the same operating conditions and illustrated in Table (3) and Fig. (12). It is noticed that, the percentage increase achieved by

Patil et al. [1] was 65%, Yun et al [2] achieved percentage increase of 84% while Bhatkar, et al [8] achieved 18%. The figure indicates that percentage increase in pressure drop achieved by the present work is nearly close to that obtained by [8] and less than that obtained by the other two.

Table 3: Comparison of percentage improvement in condenser pressure drop.

Paper No.	Unit C. C. [W]	T_a [°C]	Freon	ΔP_c
1	1172.3	34	R134A	65 %
2	7000	35	R22	84.0%
8	3516	Tc = 48°C	R152a	0.5 bar (18%)
Present work	5274	30-55	R-22	0.2 bar (20:33 %)
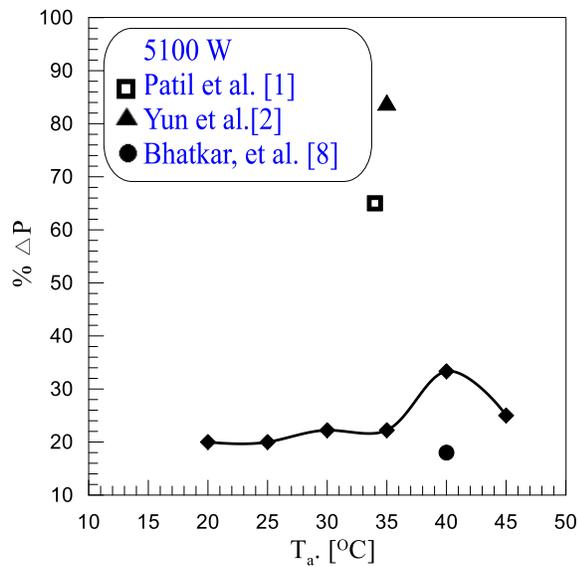
Fig. 12 Comparison of percentage savings in condenser pressure drops.

3. 3 Heat Transfer Coefficient
The convective heat transfer coefficient for condensation inside round tubes and in microchannel is given by Eqs. (4) and (5) respectively.
For round tube [17]:

$$\bar{h} = 0.555[\frac{g\,\rho_l\,(\rho_l - \rho_v)k_l^{\,3}i'_{av}}{\mu_l d_i\,(T_s - T_w)}]^{0.25} \quad (4)$$

For microchannel [18]:

$$\bar{h} = 0.943\frac{k}{d}[\frac{\rho(\rho_l - \rho_v)gi'_{av}L^3}{k_l\mu_l\,(T_{sat} - T_{wall})}]^{0.25} \quad (5)$$

The parameters in the two equations are obtained at condensing temperature corresponding to different ambient temperature for the case of using 5274 W cooling capacity unit. The convective heat transfer coefficient for both finned tube condenser and microchannel is calculated and represented in Fig. (13). The figure indicates that, the use of microchannel condenser instead of finned tube one leads to increase the heat transfer coefficient during condensation of R-22 inside tubes at the specified conditions in this work. The inside heat transfer coefficient is ranged between 409.5 and 486.7 W/m².K for the unit with finned round tube condenser while it was changed to be ranged from 852.7 to 1003.2 W/m².K for the unit used micro-channel condenser. For all tests, the internal heat transfer coefficient of the micro-channel condenser is (205:211%) greater than that of the finned tube condenser. The increase of heat transfer coefficient resulting in increasing of heat transfer rate and decreasing in the condenser size.
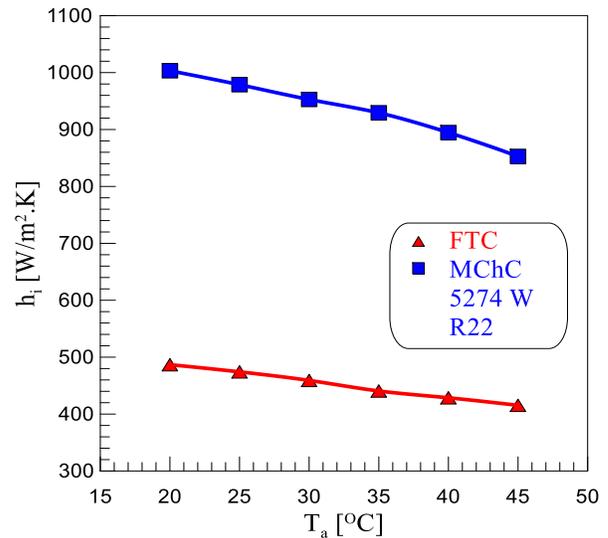
Fig. 13 Convective heat transfer coefficient

For theoretical calculation, the convective heat transfer coefficient by using other types of refrigerants [R134a, R404A and R410A] is calculated and illustrated in Fig. (14) for the two types of condenser. It is noticed that, the convective heat transfer coefficient that achieved by microchannel condenser at all types of refrigerants is (204% to 214%) greater than that of the fin-and-tube condenser. The figure indicates also the comparing of convective heat transfer coefficient "h_i" in the case of using microchannel condenser using R-22 with other refrigerants. It is seen that, Freon R134a achieved a percentage decrease in the heat transfer coefficient "h_i" by about (3.1:6.1%). Refrigerant 404A achieved a percentage decrease of (25:33.6 %) at specified

operating ambient temperature. For R410A, the heat convection coefficient at ambient temperature less than 23°C is nearly increased with small percentage than that of R-22, after that it decrease gradually until it achieves a percentage decrease of (10.6%) at ambient temperature of 45°C. The figure indicates that the value of convective heat transfer coefficient achieved by the present work is nearly close to that obtained by [R134a & R410a] and greater than that obtained R404A.
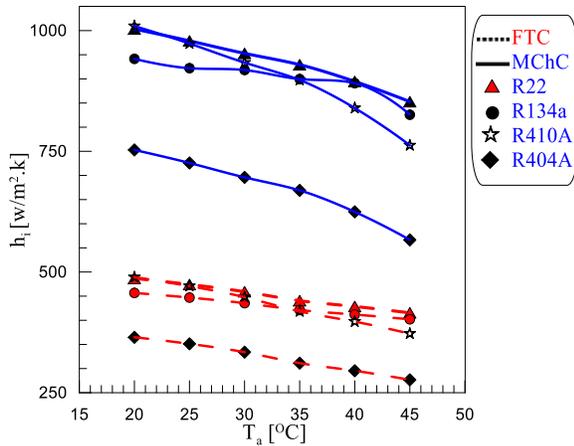
Fig. 14 Comparison of Convective transfer coefficient calculation between different Freon types

The convective heat transfer coefficient for microchannel condenser is compared also with other using different correlations of heat convection coefficient in parallel microchannel [13, 14, 15, 16]. The results obtained are shown in table 4 and illustrated in Fig. 15 while the percentage deviation between this work and the others are indicated in Fig. 16. It is clearly seen that, the heat convection coefficient obtained in this work is nearly close to the correlation of Kim [16], Eq. (6). However, the heat convection coefficient obtained by Kim correlation is less than that of this work by about (2.2:3.4 %).

$$\bar{h} = 5.03\,\text{Re}_{Lo}^{1.3}\,\text{Pr}_L^{0.333}\,k_L\,/\,D \qquad (6)$$

$$\text{Re} \le 50000,\ \ 60 \le \text{Nu.Pr}_L^{-0.333} \le 400$$

The values of (h_i) that obtained by Heun, [13], correlation (7), recorded a percentage increase ranged between (15.5:22.8 %) as indicated the tables and figures.

$$\bar{h} = 0.023\,\text{Re}_{LT}^{0.8}\,\text{Pr}^{0.4}\,k_L\,/\,D \qquad (7)$$

Fernando [14] and Shah [15] correlations (8) and (9) recorded a decrease of the convective heat transfer

coefficient ranged between (46.4:54.4 %) for Fernando and (29.35:31.15: %) for Shah.

$$\bar{h} = [0.0037\,\text{Re} + 5.532]\,\text{Pr}^{0.4}\,k_L\,/\,D \qquad (8)$$

$$\bar{h} = 1.32\,\text{Re}_{Lo}^{-1/3}\left[\frac{\rho(\rho_l - \rho_G)gk_l^3}{\mu_l^2}\right]^{1/3} \qquad (9)$$

Table (4): Comparison of value convective heat transfer coefficient after using other types of Condensation heat transfer correlations.

Paper No.	T_a [°C]	h_i [MChC] W/m².K	h_i [MChC] %
[13]	20-45	1299.2:1046.7	22.8: 15.5%.
[14]	20-45	537.3: 389	54.4: 46.4 %.
[15]	20-45	690.7: 602.4	31.15: 29.35 %.
[16]	20-45	980.5: 833.7	2.2: 3.4 %.
Present work	20-45	1003.2: 852.7	-
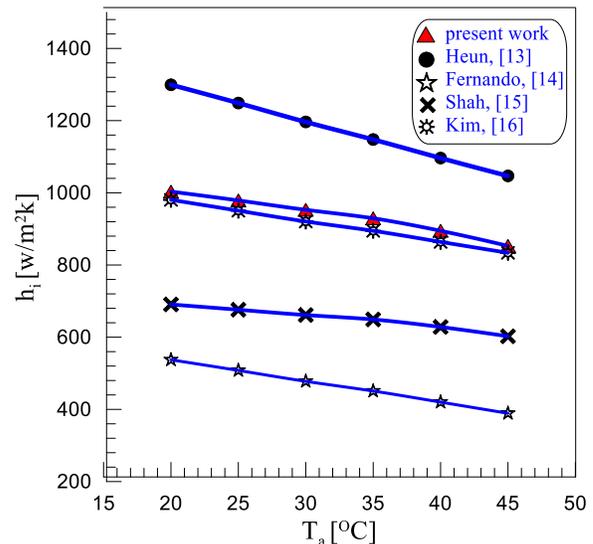
Fig. 15 Percentage deviation of Convective heat transfer between the present work and previous works
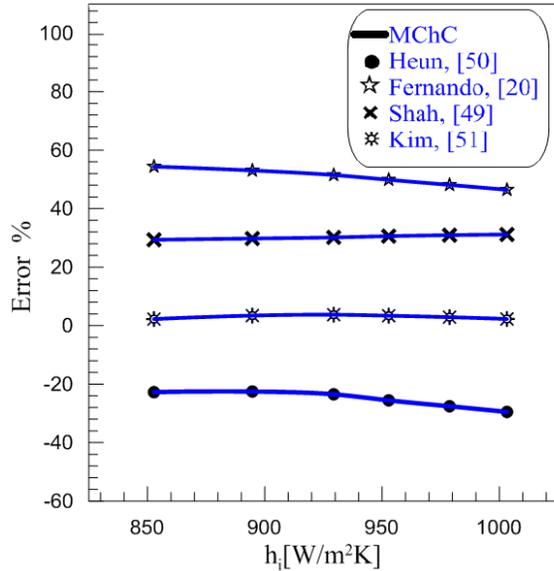
Fig. (16) Percentage deviation of convective heat transfer between the present work and some previous work

4- Conclusions

In this study, air conditioning systems with a micro channel condenser and finned tube one were experimentally and theoretically examined. The study was conducted on (3516 W and 5274 W) split air conditioners with refrigerant R-22. From the previous discussion, it can be concluded that:

1- For types of condenser systems (microchannel and a finned tube condenser), the EER and cooling capacity decrease with increasing outdoor temperatures.

2- The EER of the two units (3516 W and 5724 W) with the microchannel condenser is higher than that with the finned tube condenser by (4.3:11.3% and 9.5:29%) respectively.

3- The pressure drop in the microchannel condenser is less than that of (FTC) by around (20% to 30%) r, due to the increase in the cross sectional area and the decrease in the path that the refrigerant moves in.

4- The use of microchannel condenser instead of finned tube condenser leads to increase the convective heat transfer coefficient by (205:211%).

Nomenclature
Symbols

°C	Celsius degree.
d	Diameter, m
EER	Energy Efficiency ratio.
f	Friction coefficient.
h	Convective heat transfer coefficient, W/m^{2}K

$\bar{h}$	Mean convective heat transfer coefficient, W/m^{2}K
i	Enthalpy
K	Constant, depends on geometry.
k	Thermal conductivity of the fluid, W/m.K
L	Length, m
Nu	Nusselt number
p	Pressure, bar
Pr	Prandtl number, dimensionless
$\dot{Q}$	Cooling Capacity
Re	Reynolds number
Δp	Pressure drops.
U	Velocity, m/s
T	Temperature
wbt	Wet-bulb temperature, C.

Greek Symbol

ρ	Density, kg/m^3.
μ	Dynamic viscosity, kg/m.s.
υ	Kinematic viscosity, m^2/s.

Abbreviation

AHU.	Air handling unit
ARI	Air-Conditioning and Refrigeration Institute
CPU	Central Processing Unit
Dbt	Dry-bulb temperature, C
EER	Energy efficiency ratio
FTC	Finned tube condenser
HVAC	Heating, ventilation, and air conditioning.
JIS	Japanese Industrial Standards
MChC	Micro-channel heat condenser
R-134a	Refrigerant 134a.
R-152a	Refrigerant 152a.
R-22	Refrigerant 22.
R-410A	Refrigerant 410A.
RTD	Resistance Temperature Detectors
SEER	Seasonal energy efficiency ratio
T$_a$	Ambient Temperature. [°C]

Subscript:

av	Vaporization
a	Ambient
G	Gas
i	inside
in	Inlet
l	Liquid
Ou	Outlet
Sat	Saturation
W	Wall
υ	Vapor

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