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EXPERIMENTAL AND THEORETICAL STUDY OF MINIATURE VAPOR COMPRESSION CYCLE USING MICROCHANNEL CONDENSER

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ABSTRACT
Experimental study for miniature vapor compression cycle working at R-134a with 550 W cooling capacity, using microchannel condenser was presented. Microchannel condenser that used in this work made from aluminum, have 12 rectangular channel with hydraulic diameter 1.07 mm, and dimensions (25×11.8×2 cm) with 146 micro fins per tube. This unit consist of tiny compressor, microchannel condenser, and capillary tube and finned tube evaporator.

System with microchannel condenser was analyzed. The variation of refrigerant-side heat transfer coefficient of condenser and compressor work were studied under the following key variables; ambient temperature increased from 30 to 40 ^{o}C. Load applied to evaporator increased from 100 to 300 W and evaporator fan velocity different from 4 to 5.5 m/s.

Results showed that, increasing in ambient temperature lead to 31% reduction in heat transfer coefficient, and 19% increasing in compressor work. While increasing in evaporator load cause 4% decreasing in heat transfer coefficient, finally compressor work increased by 8.9%. This result have a good agreement with numerical data. The standard deviation was 8.4% for heat transfer, 6.9% for heat rejection.

INTRODUCTION
Vapor Compression Cycle (VCR) provide a progress means for removing large amounts of heat and is a compelling technology to improvement the performance and reliability of electronics [1]. miniature vapor compression cycle have small size, it can provide high cooling capacity by using microchannel condenser, which reject large amount of heat to the ambient. Due to the space limitation, system that proposed should contain micro channel evaporator or condenser and small scale compressor like one that used in this work. This type of condenser can be effectively used as energy efficient system. Micro and mini channel condenser are different from conventional channel in term of channel hydraulic diameters. Micro channel condenser is used to effectively absorb and dissipate heat from the surroundings more efficient than conventional condenser [2]. Micro or mini channel condenser have been widely used in cooling system of automotive and also being used in stationary heating, ventilation, air conditioning and refrigeration industry. this type have rectangular tubes divided into many micro channel in which refrigerant flow. Multi louver fins sandwiched between this tubes to dissipate heat by cooled air flow across this fins The flat tubes allow maximization of the air side heat transfer surface area, and the multiple micro refrigerant channels within the flat tubes increase refrigerant side heat transfer [3].

Through much research efforts, experimentally and theoretically tested of miniature vapor compression cycle with microchannel condenser have been proposed. Air conditioning system with micro channel condenser and R410a as working fluid was experimentally tested to studies the effect of micro channel condenser on the performance of the system. C.Y. Park et al. [4], used three type of condenser (micro channel, round tube and plat fin condensers). By comparison between those three type under the same condition (temperature, humidity and air flow rates), the result indicated that coefficient of performance (COP) of the system with micro channel improve under all condition, and reduce in refrigerant charge by 9% when using micro channel condenser. Srinivas Garimella et al. [5], used horizontal micro channel condenser with different hydraulic diameter $0.5 < Dh < 1.5$

mm over mass flux as 150 < G < 750 kg/m2.s, to provide mathematical model for condensation heat transfer coefficient in circular micro channel condenser. From result they are concluded that heat transfer coefficient inside microchannel increase with increasing in mass flux and quality while decrease with increasing in channel diameter. Experimentally characteristic of condensation of R134a in mini channel condenser contain multiport with hydraulic diameter 1.2 and 1.1mm are performed by Kittipong Sakamatapan et al. [6], these testes are performed with heat flux from 20 to 25 kw/m2, mass flux between 340 to 680 kg/m2.s and saturation temperature from 35 to 45 oC. From result we can see that HTC increase with increasing vapor quality, heat flux and mass flux while reduce with increasing saturation temperature. The result also showed that HTC increase about 5-15% when hydraulic diameter reduce from 1.2 to 1.1 mm. Thanhtrung Dang et al. [7], are experimentally investigated HTC in micro channel condenser. The result referred to that HTC obtained in counter flow higher than that obtained during parallel flow by 1.04 to 1.05 time, also from result indicated that HTC increase with decreasing inlet cooling water temperature. While a model for prediction condensation heat transfer in noncircular micro channel was achieved by Akhil Agarwal et al. [8], they used R134a as refrigerant in condensers with different hydraulic diameter (0.422< Dh< 0.839 mm) and different shape such as barrel-shaped, N-shaped, W-shaped, rectangular, square and triangular extruded tubes. Result showed that higher heat transfer coefficient provided by condenser with N-shape, W-shape and rectangular shape channel.

TEST APPARATUS AND METHOD

A miniature vapor compression cycle model RAC550DC24-D is used in this work. This unit operate with 24 V DC source, have 250 mm length, 160 mm width, and 170 mm height. Figure (1) illustrates schematic diagram for the test rig showing the locations of all unit component, while Figure (2). shows the photograph of the test rig

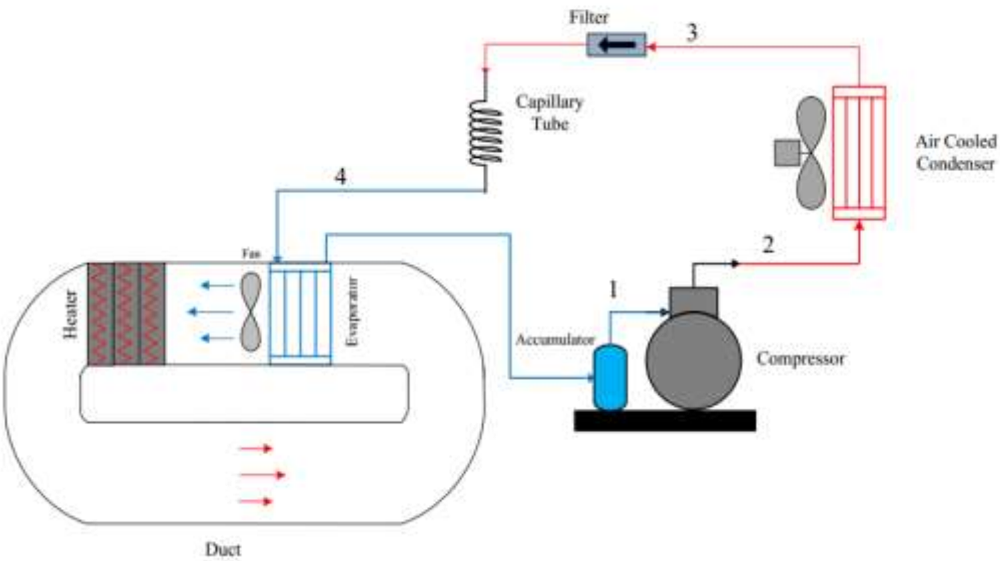

Figure (1): schematic diagram

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Figure (2): photograph of the test rig

A miniature vapor compression cycle model RAC550DC24-D is used in this work. This unit operate with 24 V DC source, have 250 mm length, 160 mm width, and 170 mm height. Miniature vapor compression consist of reciprocating tiny compressor have model (HA27CRDC24) is used. This compressor works on refrigerant R134a, and have capacity 168W. Operating on the power supply of DC 24V. Compressor displacement is 2.7 cc/rev. The compressor dimensions are 105 mm height, 50 mm diameter, and 1.3 kg weight. This compressor also have a small accumulator connect to it. Small finned and tube evaporator have 12 tube made from copper with diameter 8 mm connected to fins made from aluminum was used, temperature controller, and expansion valve. While microchannel condenser that used in this work is micro channel condenser with dimension of (25×11.8×2 cm) as shown in Figure (3). This type have tubes and fins made from aluminum, and have 16 horizontal tubes. Each tube have 12 parallel and rectangular channel. Micro channel condenser specifications are listed in table (1).

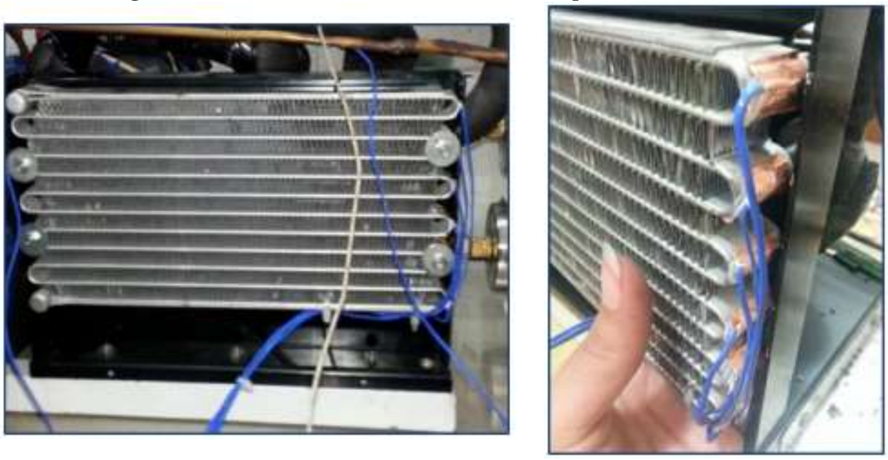

Figure (3): Microchannel condenser

Table (1): Microchannel condenser specification

Specification	
Tube height (mm)	2
Tube depth (mm)	20

Number of channel	12
hydraulic diameter (mm)	1.07
Tube length (mm)	250
Wall thickness (mm)	0.5
Fin number per tube	146
Fin thickness (mm)	1
Fin length (mm)	6.6
Fin pitch (mm)	1.2

Heat transfer coefficient method

Koyama et al. suggested simplified correlation to calculate heat transfer coefficient of R-13a inside rectangular channel as expressed below [9]

$$\alpha = \frac{k_f}{d_h} Nu \quad (1)$$

$$Nu = \left(Nu_f^{\,2} + Nu_b^{\,2}\right)^{1/2} \quad (2)$$

Where

Nu_f: is forced convection condensation term and can be expressed as

$$Nu_f = 0.0112\, pr_f^{\,1.37} \left(\frac{\phi_g}{X_{tt}}\right) Re_f^{\,0.7} \quad (3)$$

And Re_f is Reynolds number based on the hydraulic diameter of channel which defined as:

$$Re_f = \frac{G(1-x)d_h}{\mu_f} \quad (4)$$

G is mass velocity of refrigerant $(kg/m^2.s)$ and calculated by using equation [11]:

$$G = \frac{\dot{m}}{A_i} \quad (5)$$

Where

$\dot{m}$: mass flow rate of refrigerant (kg/s)

A_i: is the cross section area which expressed as [6]

$$A_i = [2 \times n_{rec}(a+b) + n_{cir}(a+\pi b)]L \quad (6)$$

Where

n_{rec}: number of rectangular channel

a: is a height of channel

b: is the width of channel

n_{cir}: number of circular channel

And L is the total length of condenser tube.

while, Nu_b is the gravity controlled convection condensation [10] which is calculated depending on Galileo number Ga_f, void fraction ε, and Bond number Bo as follow:

$$Nu_b = 0.725\left(1 - e^{-0.85\sqrt{Bo}}\right) H_{(\varepsilon)} \left(\frac{Ga_f pr_f}{ph_f}\right)^{1/4} \quad (7)$$

Where

Bond number Bo calculated by using equation:

$$Bo = \frac{d_h^{\,2} g(\rho_f - \rho_g)}{\sigma} \quad (8)$$

And Galileo number Ga_f can be calculated as follow:

$$Ga_f = \frac{g\rho_f^{\,2} d_h^{\,3}}{\mu_f^{\,2}} \quad (9)$$

While ph_f is phase change number that expressed as:

$$ph_f = cp_f(T_r - T_w)/h_{fg} \quad (10)$$

RESULTS AND DISCUSSION

In this study refrigerant side heat transfer coefficient was determined experimentally. The present result are plotted in Figure (4), which represented the variation of predicted value of heat transfer coefficient of refrigerant side with ambient temperature and evaporator fan velocity. Heat transfer coefficient proportional adversely with ambient temperature, it reduce from 324 to 223 W/m^2.k when temperature increase from 30 to 40 ^{o}C at evaporator load and fan velocity are 200 W and 4 m/s respectively. Also in this figure it can observe the slightly reduction in heat transfer coefficient during increasing in evaporator fan velocity. Figure (5) demonstrated the effect of evaporator loads on heat transfer coefficient. The increasing of load effected inversely on heat transfer coefficient which decrease slightly by (4%) when load imposed increase from 100 to 300 W at 34 ^{o}C and fan evaporator velocity is 4 m/s.

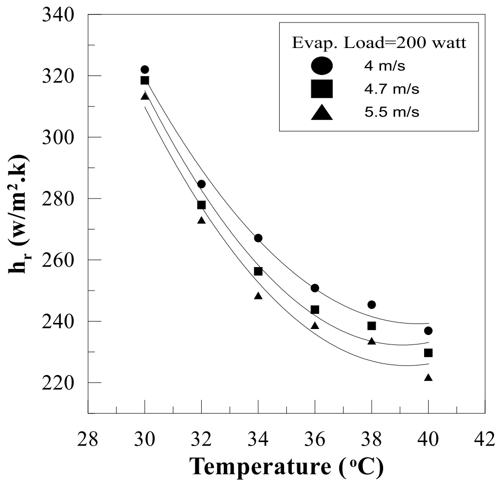

Figure (4): Variation of heat transfer coefficient of refrigerant side with ambient temperature

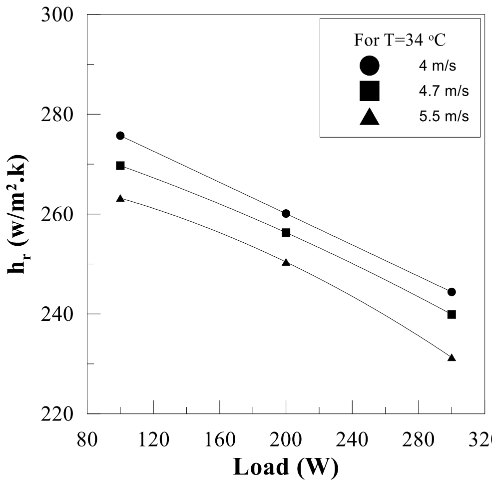

Figure (5): Variation of heat transfer coefficient of refrigerant side with evaporator load

The effect step change in ambient temperatures and fan evaporator velocity with compressor work, was plotted in Figure (6). Compressor work increase by 19% with raising ambient temperature from 30 to 40 ^{o}C, this due to reduction in heat rejected from condenser, which lead to increase in discharge temperature and then increasing in power consumption of compressor. While Figure (7) showed the effect of load applied to evaporator on compressor work. It can observed that the compressor work increase from 178 to 194 W, according to increasing in thermal load imposed to evaporator from 100 to 300 W. In both cases the compressor work increase with increase in velocity of evaporator fan.

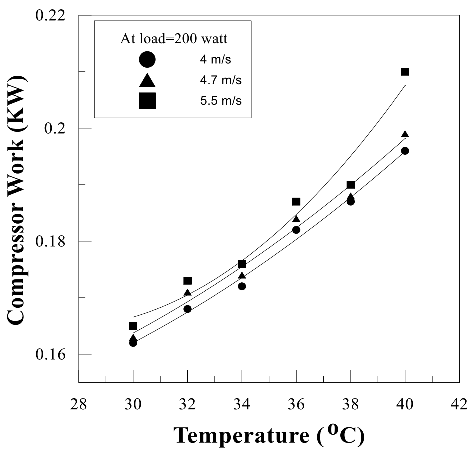

Figure (6): Variation of Compressor work with ambient temperature

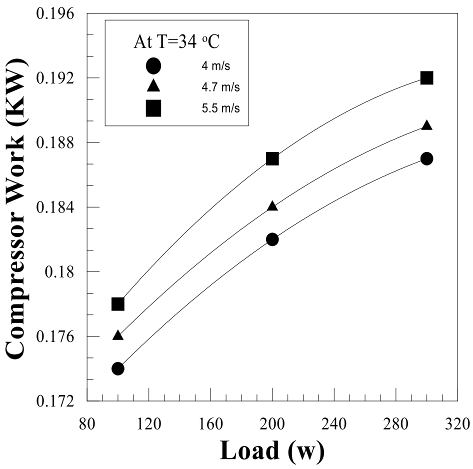

Figure (7): Variation of Compressor work with load imposed on evaporator the experimental result was obtained by using data based on experimental measurement, and theoretical result determined by using the mathematical model that solved by the Engineering Equations Solver (EES) software. The variation between theoretical and experimental results for heat transfer coefficient illustrate in figure (8) and (9). The standard deviation 8.4, and the relative deviation is 1.8%, for different temperatures. At ambient temperature 30 ^{o}C and load change from 100 to 300 w the standard deviation is 8.8, while relative deviation is 2%.

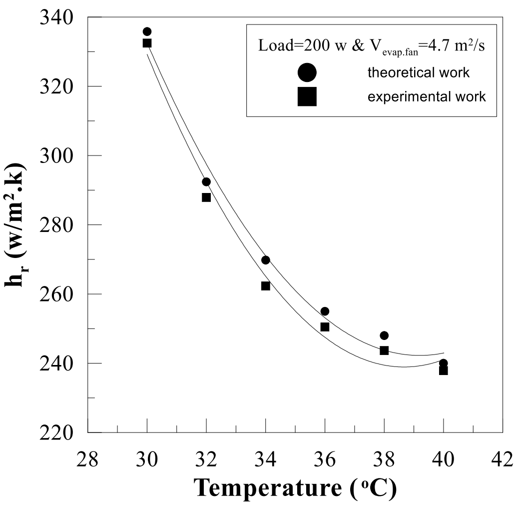

Figure (8): Experimental and theoretical transfer coefficient through microchannel condenser at different temperature

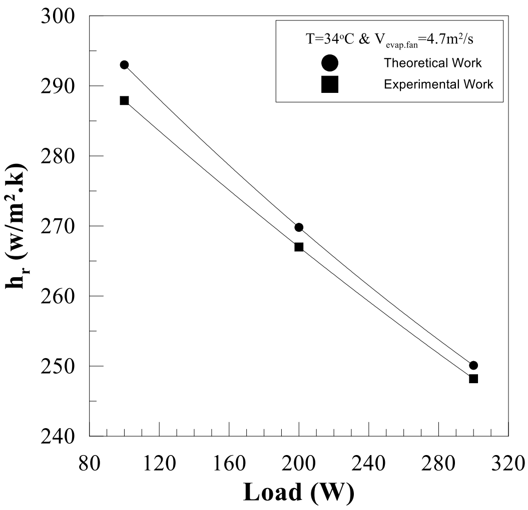

Figure (9): Experimental and theoretical heat heat transfer coefficient through microchannel condenser at different load

CONCLUSION
Miniature vapor compression cycle was experimentally tested under different condition:

- Ambient temperature increase from 30 to 40 ^{o}C.
- Load applied to evaporator increase from 100 to 300 W.
- Evaporator fan velocity increased from 4 to 5.5 m/s.

The conclusion that can be derived from above tests as follows:

1. Refrigerant side heat transfer coefficient of microchannel condenser reduced by 31% with increasing in ambient temperature. And 4% by increasing in evaporator load.
2. Heat rejection from micro channel condenser is decreased by 18% with increasing in ambient temperature. While it decrease by 3% when evaporator load was increased.
3. Nusselt number also decrease by 18% when ambient temperature increased. And by 13% with increasing in evaporator load.
4. Compressor work increase by 19% during increasing in ambient temperature. But it increased by 8.9% with increasing in evaporator load.
5. The theoretical was identical with experimental results. The maximum standard deviation was 8.4% for heat transfer, 6.9% for heat rejection, and 7% for Nusselt number.

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