Theoritical Analysis For Miniature Vapor Compression Cycle Performance Using Microchannel And Finned Tube Condenser

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Abstract— **Theoretical and experimental study of miniature vapor compression cycle working with R-134a at 550 W cooling capacity, using microchannel and finned tube condenser was presented. Microchannel condenser that used in this work made from aluminum. This condenser have 12 rectangular channels with hydraulic diameter 1.07 mm, and condenser dimensions (25× 11.8× 2 cm) with 146 micro fins per tube. In theoretical work, miniature vapor compression cycle with microchannel and conventional condenser was analyzed. The effect of channel diameter change on refrigerant side heat transfer coefficient was taken into account. The theoretical result indicates that, using microchannel instead of finned tube condenser lead to enhancement in refrigerant side heat transfer coefficient by 31%, Increasing heat rejection by 36%, and coefficient of performance by 23%. Also it shows that refrigerant side heat transfer coefficient decreases by 29%, when hydraulic diameter increases from 0.85 to 1 mm. The result indicate that using microchannel condenser can reduce the total area of condenser by 53%, for the same amount of heat rejection.**

Keywords—microchannel condenser; miniature vapor compression cycle; hydraulic diameter

I. 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, theoretically tested of miniature vapor compression cycle with microchannel condenser have been proposed. Numerical simulation of micro channel condenser for domestic refrigerator was introduced by Zhang Huiyong et al. [4] in 2010, to evaluate design parameter for different tube length and tube diameter of micro channel condenser. Heat exchanger consisted of serpentine fin brazed to tube with internal channel in micro size was theoretically analyzed by J.R. Garcia-Cascales et al. [5] 2010, so they built model to studies the condensation of R134a and R410a in compact condenser and compared the results with number of other correlation. While S.M.Gaikwad et al. [6] in 2013, performed a numerical study to the fluid flow inside micro channel condenser by ANSYS Fluid flow (Fluent) to compare characteristics of micro channel condenser during counter and parallel flow. Condenser used with 1,1.5,2,2.5,3,3.5,4,4.5 m/s inlet velocity and w=0.4 mm H=0.4 mm L=10 mm geometry. In this work, an attempt was made to replace a plate finned tube condenser of a simple vapor compression cycle that shown in Figure (1), by microchannel condenser.

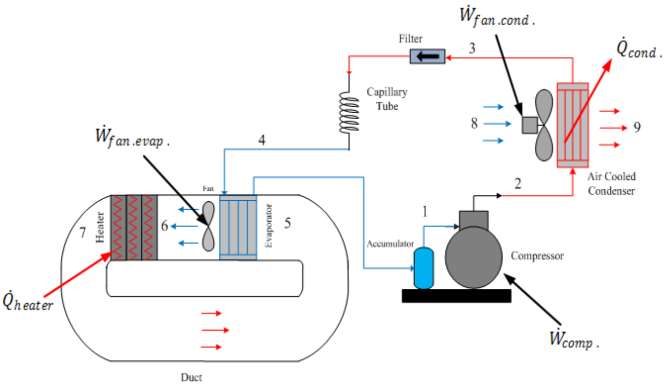

Fig. (1) Schematic Diagram of the Vapor compression cycle

II. THEORY

To evaluate the overall heat transfer coefficient for a tube with fins, first the air-side convective heat transfer coefficient should be calculated. Since the flow of air in both plate fin and micro channel condenser are over the tube and air forced by fan, so it can be said that are same for both types of condenser. Therefore, the work of (Mc Quiston and Parker, 1994) is used [7]. The convective heat transfer coefficient for dry outer surface of the condenser dependent on the Colburn j-factor is:

$$\bar{h}_a = \frac{j_c \cdot G_{max} \cdot cp_a}{{pr_a}^{2/3}} \quad (1)$$

While the Prandtl (pr_a) number can be expressed as:

$$\mathrm{Pr}_a = \frac{\mu_a \cdot cp_a}{k_a} \quad (2)$$

Because of channel size effect on heat transfer rate of condenser, heat transfer coefficient cannot be accounted by using models provided for condenser with conventional channel [8]. For this reason heat transfer coefficient for both types was analyzed by different correlations.

A. Micro Channel Condenser Refrigerant Side Model

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

$$\alpha = \frac{k_f}{d_h} Nu \quad (3)$$

$$Nu = \left({Nu_f}^2 + {Nu_b}^2\right)^{1/2} \quad (4)$$

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

$$Nu_f = 0.0112\, {pr_f}^{1.37} \left(\frac{\emptyset_g}{X_{tt}}\right) {Re_f}^{0.7} \quad (5)$$

$$Re_f = \frac{G(1-x)d_h}{\mu_f} \quad (6)$$

$$G = \frac{\dot{m}}{A_i} \quad (7)$$

$$A_i = [2 \times n_{rec}(a+b) + n_{cir}(a+\pi b)]L \quad (8)$$

$${\emptyset_g}^2 = 1 + 21(1 - e^{-0.319 d_h})X_{tt} + {X_{tt}}^2 \quad (9)$$

And X_{tt} is Lockhart-Martinelli parameter which expressed as

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1} \quad (10)$$

On the other hand, 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:

$$Nu_b = 0.725\left(1 - e^{-0.85\sqrt{Bo}}\right) H_{(\varepsilon)} \left(\frac{Ga_f pr_f}{ph_f}\right)^{1/4} \quad (11)$$

$$Bo = \frac{{d_h}^2 g(\rho_f - \rho_g)}{\sigma} \quad (12)$$

And $H_{(\varepsilon)}$ is the function of void fraction can be calculated as:

$$H_{(\varepsilon)} = \varepsilon + [10(1-\varepsilon)^{0.1} - 8.9]\sqrt{\varepsilon}\left(1 - \sqrt{\varepsilon}\right) \quad (13)$$

Where void fraction can be expressed as:

$$\varepsilon = \left(1 + \frac{\rho_g}{\rho_f}\frac{1-x}{x}\left(0.4 + 0.6\sqrt{\frac{\frac{\rho_f}{\rho_g} + 0.4\frac{1-x}{x}}{1 + 0.4\frac{1-x}{x}}}\right)\right)^{-1} \quad (14)$$

While Galileo number Ga_f can be calculated as follow:

$$Ga_f = \frac{g {\rho_f}^2 {d_h}^3}{{\mu_f}^2} \quad (15)$$

And ph_f is phase change number that expressed as:

$$ph_f = cp_f(T_r - T_w)/h_{fg} \quad (16)$$

B. conventional Condenser Refrigerant Side Model

The heat transfer in the single phase includes (super-heated and sub-cooled) region. For this study heat transfer coefficient are calculated by using the correlation of Kays and London (1984) [11] as:

$$\bar{h}_{r,sph} = a \cdot G_{r,sph} \cdot cp_{r,sph} \cdot {Re_{Di}}^{b} \cdot {pr_{r,sph}}^{-\frac{2}{3}} \quad (17)$$

Also, heat transfer coefficient in the sub-cooled region can be expressed as:

Also, heat transfer coefficient in the sub-cooled region can be expressed as:

$$\bar{h}_{r,sc} = 0.16 \cdot \frac{k_{sc}}{D_i} \cdot {Re_{r,sc}}^{0.89} \cdot {pr_{r,sc}}^{0.12} \quad (18)$$

While The heat transfer coefficient of two phase flow can be calculated by using Shah correlation (1979) [12] which are presented as follow:

$$\bar{h}_{r,tp} = \bar{h}_{r,l}\left[0.55 + \frac{2.09}{p*^{0.8}}\right] \quad (19)$$

While the heat transfer coefficient liquid region is calculated by using the DittusBoelter equation [13]:

$$\bar{h}_{r,l} = 0.023 \cdot \frac{k_{sc}}{D_i} \cdot {Re_{r,sc}}^{0.8} \cdot {pr_{r,sc}}^{0.4} \quad (20)$$

III. RESULT AND DISCUSSIONS

In this work, area of microchannel condenser was calculated by using equation represented in chapter three , while the area of finned tube condenser was calculated by using HTRI software. The result shows that the area was reduced by 53%, if conventional condenser was replaced by microchannel condenser.
The change in the predicted value of refrigerant side heat transfer coefficient for microchannel and finned tube condenser with ambient temperature is shown in Figure (2). The heat transfer coefficient of microchannel reduces from 347 to 250 W/m^2.k while in finned tube condenser decreases from 240 to 110 W/m^2.k, with increase in temperature from 30 to 40 ^{o}C. It is clear that heat transfer coefficient of microchannel is higher than finned tube condenser due to multiple rectangular microchannels within the flat tubes, which is increase refrigerant side heat transfer area.

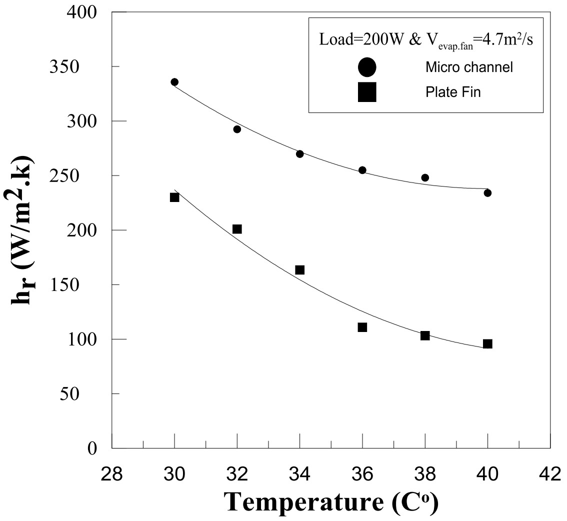

Fig (2) theoretical heat transfer coefficient vs. ambient temperature

While in Figure (3), it can observed the reduction in heat transfer coefficient of microchannel and finned tube condenser, with the increase in the evaporator load from 100 to 300 W, the heat transfer coefficient of microchannel decreased from 296 to 260 w/m^2.k, while for finned tube condenser reduced from 169 to 145 w/m^2.k. The heat transfer coefficient of microchannel is higher than that of finned tube condenser by 37% at load 100 W.

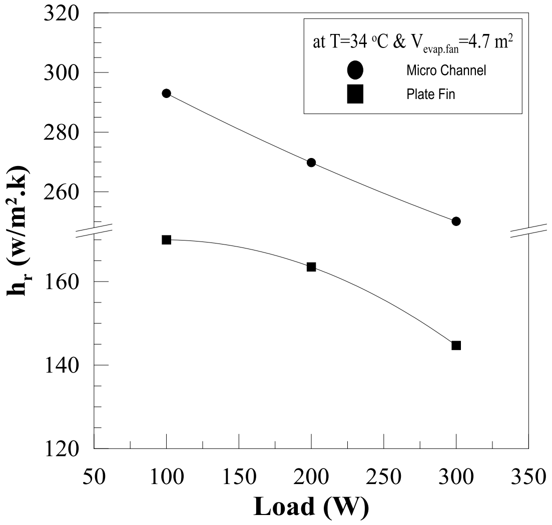

Fig. (3) theoretical heat transfer coefficient vs. load imposed on evaporator.

Figure (4), demonstrates the change in Nusselt number with ambient temperature. The result shows that enhancement in Nusselt number by about 34% using microchannel condenser instead of finned tube condenser.

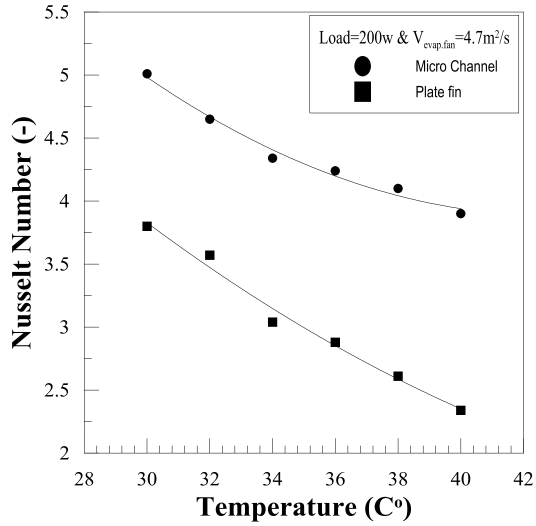

Fig. (4) Nusselt number vs. ambient temperature

Figure (5), depicts the variation of Nusselt number with load imposed on evaporator. It shows that increasing load from 100 to 300 W means decreasing in Nusselt number from 4.52 to 3.97 for microchannel and from 3 to 2.5 for finned tube condenser. Also the Nusselt number of microchannel is greater than that of finned tube condenser by about 33%.

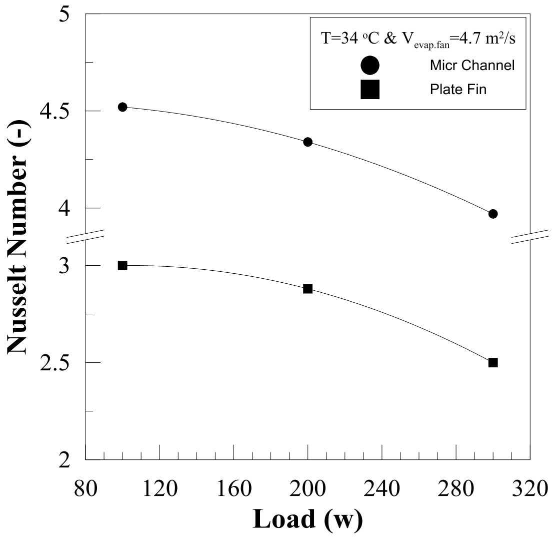

Fig. (5) Nusselt Number vs. load imposed on evaporator

The effect of ambient temperature and evaporator load on heat rejection from condenser are investigated in Figure (6) and (7).The heat rejection from condenser decreases with increase in ambient temperature and load on evaporator. Microchannel condenser heat rejection decreases from 1170 to 900 W, and reduces from 850 to 670 W for finned tube condenser with the increase in ambient temperature. While it decrease slightly by 2.9% for microchannel, and 2.3% for finned tube condenser when the evaporator load increased. Also it may be noted that the heat rejection from microchannel is greater than finned tube condenser by 36%.

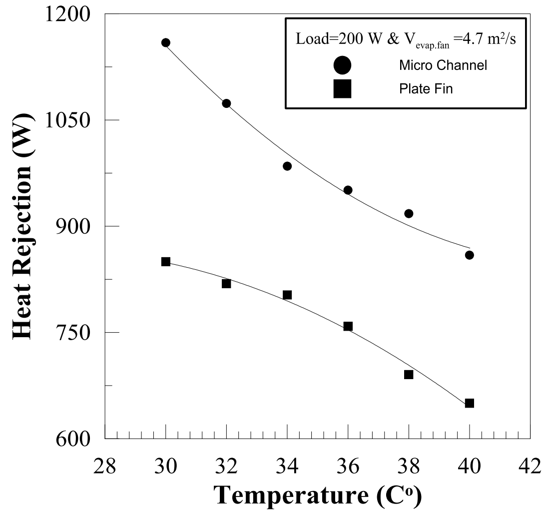

Fig. (6) heat rejection vs. ambient temperature

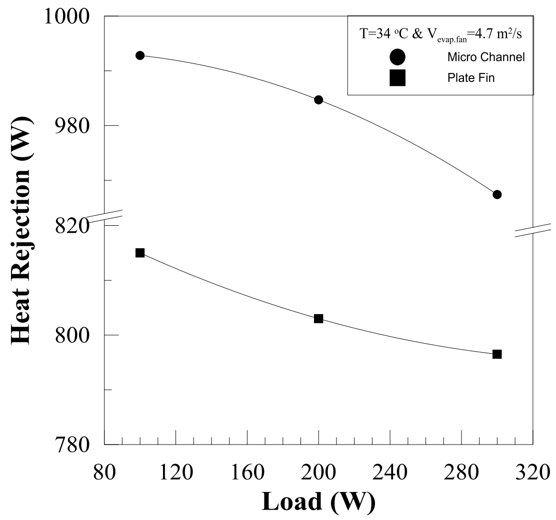

Fig. (7) heat rejection vs. load imposed on evaporator

Figure (8) illustrates that the coefficient of performance of miniature vapor compression cycle with microchannel and finned tube condenser, is inversely proportional with ambient temperature. It slightly decrease by 26% for microchannel and by 24% for finned tube condenser with increase in ambient temperature. Also it can be noted that the coefficient of performance of cycle with microchannel condenser is greater than that of cycle with finned tube condenser by 23%, at ambient temperature 30 °C and load 200w.

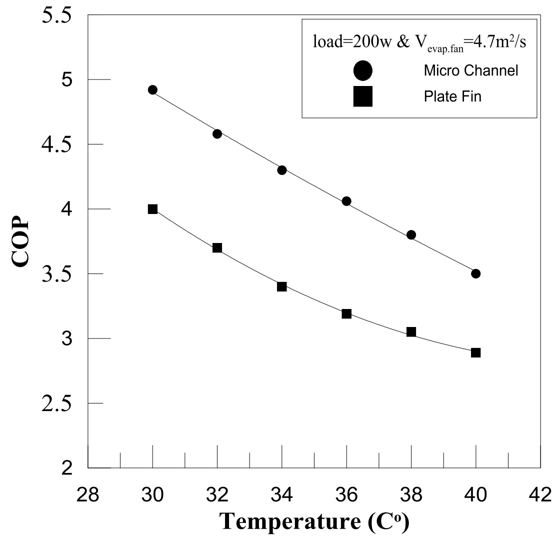

Fig. (8) coefficient of Performance vs. ambient temperature

The effect of increasing evaporator load on COP of the system is indicated in Figure (9). The COP of system with microchannel decreases from 4.5 to 4.12, while decreases from 3.53 to 3.27 with conventional condenser, when evaporator load increases from 100 to 300 W. Also it can be noted that the coefficient of performance of cycle with microchannel condenser is by 23%, at ambient temperature 30 °C and load 200w condenser.

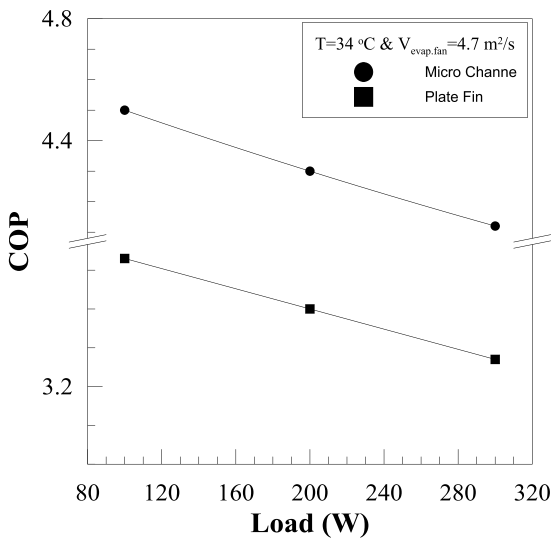

Fig. (9) coefficient of Performance vs. load imposed on evaporator

The change in value of refrigerant side heat transfer coefficient of microchannel condenser with respect to hydraulic diameter of condenser channel, is depicted in Figure (10). The increase in hydraulic diameter of channel leads to decrease in heat transfer coefficient of condenser. This is due to reduce in contact area between refrigerant and surface. From Figure (7) it can be observed that the heat transfer coefficient increases by 29%, when hydraulic diameter of channel decreases from 1mm to 0.85 mm.

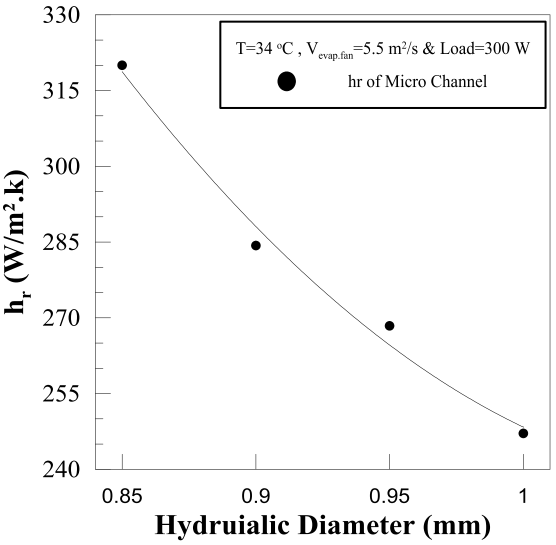

Fig. (10) variation of heat transfer coefficient vs. channel hydraulic diameter

IV. CONCLUSIONS:

1. Using microchannel instead of plate fin condenser can reduce the total area of condenser by 53%, for the same amount of heat rejection.
2. From theoretical results, it can be concluded that using microchannel instead of plate fin condenser leads to:
    a. Improvement in refrigerant side heat transfer coefficient by 31%.
    b. Nusselt number increased by 34%.
    c. Increasing in heat rejection by about 36%.
    d. The coefficient of performance for miniature vapor compression cycle increased by 23%.
3. Decrease in hydraulic channel diameter of microchannel condenser from 1 to 0.85 mm, leads to enhancement in refrigerant side heat transfer coefficient by 29%.

Symbols

j_c: Colburn j-factor
G_{max}: maximum mass flux of air (kg/m^2 s)
Cp_a: constant pressure specific heat for air (kJ/kg K)
d_h: is a hydraulic diameter of channels
$\dot{m}$: mass flow rate of refrigerant (kg/s)

A_i: is the cross section

n_{rec}: number of rectangular channel
a: is a height of channel
b: is the width of channel
n_{cir}: number of circular channel
g: gravity
ρ_f: density of refrigerant at liquid phase
ρ_g: density of refrigerant at vapor phase
σ: surface tension (N/m)
h_g: is enthalpy of refrigerant at vapor phase
h_f: is enthalpy of refrigerant at liquid phase
$\bar{h}_{r,sph}$: heat transfer coefficient in the super-heated region $(W/m2.k)$
$Cp_{r,sph}$: specific heat at constant pressure $(kJ/\,kg.K)$.

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