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Experimental and Analytical Investigation of Heat Transfer Coefficient of a Water Cooled Condenser for Different Water Flows and Condensation Pressures

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Abstract

In the phase change process, latent heat is transferred and the amount of heat transferred will be excessive high compared to the amount of sensible heat transfer. For that reason, condensation and evaporation processes are the main steps in the refrigeration cycle in order to increase the amount of heat transferred. The main objective of this study is to experimentally and analytically examine overall heat transfer coefficient and to present the effect of water flow and refrigerant pressure on condensation process. For this purpose, a refrigeration system where a water cooled condenser with a heat transfer surface area of 0.075 m^2 was installed. R134a was used as a refrigerant and condenses on the outer surface of the pipe that water circulates through. In this study, experiments were repeated for water mass flow rates of 15, 20, 25, 30 and 35 g/s at constant 7.0 bar condensation pressure. Then, condensation pressures were changed to 6.5, 6.75, 7.0, 7.25 and 7.5 bar at constant water flow rates of 25 g/s. Correlation of condensation heat transfer coefficient has been applied to refrigerant side of the condenser unit. On the other side, logarithmic mean temperature difference (LMTD) and number of transfer unit (ε-NTU) methods have been applied to experimental results in order to calculate heat transfer coefficient. Experimental results for different water flow rates at constant refrigerant pressure and for different condensation pressures at constant water flow rate are taken from the refrigeration machine unit and have been compared with calculated values that are obtained from condensation heat transfer correlation.

Keywords: Refrigeration cycle, Condensation, Logarithmic mean temperature difference (LMTD) method, Number of transfer unit (ε-NTU) method, Heat transfer coefficient

Değişen Su Debisi ve Yoğuşma Basıncı İçin Bir Su Soğutmalı Yoğuşturucunun Isı Transferi Katsayısının Deneysel ve Analitik İncelenmesi

Öz

Faz değişiminde gizli ısı transfer edilir ve transfer edilen ısı miktarı, duyulur ısı transferi miktarına kıyasla aşırı yüksek olur. Bu nedenle, transfer edilen ısı miktarını artırmak için yoğunlaşma ve buharlaşma işlemleri, soğutma çevrimindeki ana basamaklardır. Bu çalışmanın temel amacı toplam ısı

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transferi katsayısını deneysel ve analitik olarak incelemek ve su debisi ile soğutucu akışkan basıncının yoğuşma hal değişimi üzerindeki etkisini ortaya koymaktır. Bu amaçla, ısı transferi yüzey alanı 0.075 m^2 olan su soğutmalı yoğuşturuculu bir soğutma sistemi kurulmuştur. Soğutucu akışkan olarak R134a kullanılmış ve soğutucu akışkan suyun içinde dolaştığı borunun dış yüzeyinde yoğuşmaktadır. Bu çalışmada, sabit 7,0 bar yoğuşma basıncında 15, 20, 25, 30 ve 35 g/s su debileri için deneyler tekrarlanmıştır. Daha sonra, 25 g/s su debisinde yoğuşma basınçları 6,5, 6,75, 7,0, 7,25 ve 7,5 bar olacak şekilde değiştirilmiştir. Yoğuşma ısı transfer katsayısı bağıntısı yoğuşturucu ünitesinin soğutucu tarafına uygulanmıştır. Öte yandan ısı transfer katsayısını hesaplamak için logaritmik ortalama sıcaklık farkı (LMTD) ve geçiş birimi sayısı (ε-NTU) metotları deneysel sonuçlara uygulanmıştır. Sabit yoğuşma basıncında farklı su debileri ve sabit su debisinde farlı yoğuşma basınçları için soğutma makinası ünitesinden elde edilen deneysel sonuçlar yoğuşma ısı transferi katsayısı bağıntısından elde edilen sonuçlarla kıyaslanmıştır.

Anahtar Kelimeler: Soğutma çevrimi, Yoğuşma, Logaritmik ortalama sıcaklık farkı (LMTD) metodu, Geçiş birimi sayısı (ε-NTU) metodu, Isı transfer katsayısı

1. INTRODUCTION

Phase change processes like condensation and evaporation play an important role in refrigeration, air conditioning and heat pump applications [1]. In the phase change process, latent heat is transferred and the amount of heat transferred will be excessive high compared to the amount of sensible heat transfer.

Phase change occurs when the temperature difference between fluid with solid surface in contact with each other acceptably large [2]. When vapor contacts with a solid surface whose temperature is below the saturation temperature of the vapor, condensation occurs [3].

Rahman et al. [4] installed a new experimental apparatus in order to obtain explicit local condensation heat transfer coefficient measurements. They developed a new correlation for condensation heat transfer in horizontal rectangular multiport minichannel with and without fins using R134a. They compared experimental results with the well-known condensation heat transfer models available in the literature and they have reached the point that all the correlations that exist are unsuccessful.

Jung et al. [5] developed an experimental apparatus in order to measure flow condensation heat transfer coefficients of R12, R22, R32, R123, R125, R134a, and R142b experimentally on a horizontal plain tube. They developed a new correlation by modifying Dobson and Chato's correlation with an introduction of a heat and mass flux ratio combined with latent heat of condensation.

Ebisu and Torikoshi [6] developed 'herringbone heat transfer tube' in order to enhance the heat transfer performance of R-407C. They investigated the heat transfer characteristics of R-407C experimentally. Their experimental results showed that the heat transfer coefficients for the herringbone tube were about 90% higher during evaporation and 200% (maximum) higher during condensation than those for the inner grooved tube.

Asker and Turgut [7] discussed the accuracy of the correlations used in the calculation of the coefficient of heat transfer for condensing in the pipe. They applied some correlations found in the literature to experimental data obtained from various researchers using R134a, R717 and R600A as the test fluid. They found that the correlations they used were given the best results for R134a.

Dalkılıç and Demir [8] designed an experimental setup in order to determine condensation heat transfer coefficient in the case of refrigerant flowing downward inside a smooth and microchannel vertical tube. They determined two phase flow pattern from the sight glasses which are at the entrance and exit of the test tube and Hewitt and Robertson flow pattern map. Then average condensation heat transfer coefficient were given.

Kumar et al. [9] performed experimental investigation on two different experimental set-ups for water and R134a in order to study the heat transfer augmentation during condensation of water and R134a vapor on horizontal integral-fin tubes. They developed an empirical equation that predicts the condensing heat transfer coefficient from their own experimental data for the condensation.

In this study, condensation on horizontal spiral pipes is examined on a water cooled refrigeration system. Experiments were repeated for cooling water mass flow rates of 15, 20, 25, 30 and 35 g/s under conditions of 7.0 bar condensation pressure. Then, condensation pressures were changed to 6.50, 6.75, 7.00, 7.25 and 7.50 bar while experiments were repeated for water mass flow rate of 25 g/s.

Correlation of condensation heat transfer coefficient has been applied to refrigerant side of the condenser unit. On the other side, LMTD and ε-NTU methods have been applied to experimental results in order to calculate heat transfer coefficient. Experimental results for different water flow rates at constant refrigerant pressure and for different condensation pressures at constant water flow rate are taken from the refrigeration machine unit and have been compared with calculated values that are obtained from condensation heat transfer correlation.

2. HEAT EXCHANGER DESIGN

Heat transfer coefficient can be determined experimentally by LMTD and effectiveness-NTU methods. Besides, analytical investigation will be applied to the heat exchanger by using correlations given in the literature.

2.1. Performance Analysis with Number of Transfer Unit Method

If only the inlet temperatures of both cold and hot fluids are known, ε −NTU method was used in order to design or to predict the performance of a heat exchanger experimentally [10].

The ratio of the actual heat transfer rate to the thermodynamically possible maximum amount of heat transfer is called an effectiveness [11]. Heat transfer effectiveness of heat exchanger is

$$\varepsilon = \frac{\dot{Q}_{actual}}{\dot{Q}_{max}} \tag{1}$$

where $\dot{Q}_{actual}$ is actual heat transfer rate between refrigerant and cooling water and $\dot{Q}_{max}$ is the thermodynamically limited maximum possible heat transfer rate.

The actual heat transfer rate is the amount of heat released to cooling water circulates in coaxial heat exchanger.

$$\dot{Q}_w = \dot{m}_w C_{p,w}(T_6 - T_5) \tag{2}$$

or

$$\dot{Q}_r = \dot{m}_r\left[h_{fg} + C_{p,f}(T_{sat} - T_3)\right]_r \tag{3}$$

where $\dot{m}_w$ is water mass flow rate, $C_{p,w}$ is the specific heat of water at arithmetic average water temperature and T_5, T_6 are inlet and outlet water temperatures, respectively. Also, $\dot{m}_r$ is refrigerant mass flow rate, ${h_{fg}}_r$ is the enthalpy difference of enthalpies of saturation vapor and saturation liquid phases, ${C_{p,f}}_r$ is the specific heat for saturated liquid phase of refrigerant at saturation temperature and T_{sat}, T_3 are saturation and outlet refrigerant temperatures, respectively.

Heat removed from refrigerant, $\dot{Q}_r$, includes not only sensible but also latent heat since the refrigerant condenses in the heat exchanger at high pressure.

Thermodynamically limited maximum possible heat transfer rate is

$$\dot{Q}_{max} = \left(\dot{m}C_p\right)_w (T_{\text{sat}} - T_5) \qquad (4)$$

where T_{sat} is the saturation temperature of the refrigerant and T_5 is the water inlet temperature.

Number of transfer unit for the heat exchanger which phase change occurs

$$NTU = -ln(1 - \varepsilon) \qquad (5)$$

From the definition of the number of transfer unit, the overall heat transfer coefficient is obtained by

$$U = \frac{\left(\dot{m}C_p\right)_w}{A} NTU \qquad (6)$$

where A is the total heat transfer surface area.

Suppose that the condensation pressure is 7.0 bar at a refrigerant flow rate of 3.5 g/s. The saturation temperature and enthalpy at that pressure is 29.08 °C and 174.1 kj/kg, respectively. Therefore, by ignoring sensible heat, the heat transfer rate can be calculated by equation (3).

$$\dot{Q}_r = \dot{m}_r h_{fg} = 3.5\text{g/s} \cdot 174.1\, kj/kg = 609.35\text{W}$$

The maximum possible heat transfer rate is determined by equation (4).

$$\dot{Q}_{max} = (40\, g/s \cdot 4.18\, kj/kgK)(29.08 - 20)\text{°C}$$

$$\dot{Q}_{max} = 1518.18\text{ W}$$

Heat transfer effectiveness of heat exchanger is obtained by equation (1).

$$\varepsilon = \frac{609.35\text{ W}}{1518.18\text{ W}} = 0.4014$$

Number of transfer unit is computed for the condensation process by equation of (5).

$$NTU = -l\,n(1 - 0.4014) = 0.513$$

The overall heat transfer coefficient is estimated by using equation of (6).

$$U = \frac{(40\, g/s) \cdot (4.18\, kj/kgK)}{(0.075\, m^2)} \cdot (0.513)$$

$$U = 1144\, W/m^2K$$

2.2. Performance Analysis with Logarithmic Mean Temperature Difference Method

If inlet and outlet temperatures of both cold and hot fluids are known, LMTD method provides simplicity to do a performance analysis [12]. In this study, this method has been used in order to determine the overall heat transfer coefficient of the water cooled condenser unit.

In the heat exchanger, amount of heat transfer from hot fluid to cold fluid can be calculated as

$$Q = UA\Delta T_{lm} \qquad (7)$$

where U is overall heat transfer coefficient, A is the total heat transfer surface area and ΔT_{lm} is logarithmic mean temperature difference of hot and cold fluids.

The logarithmic mean temperature difference of the fluids

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)} \qquad (8)$$

where ΔT_1 and ΔT_2 are temperature differences. For counter–flow heat exchanger, the temperature differences are defined as

and $\Delta T_1 = T_{h,i} - T_{c,o}$, $\Delta T_2 = T_{h,o} - T_{c,i}$ (9)

where subscripts “h”, “c”, “i” and “o” are hot fluid, cold fluid, inlet and outlet, respectively.

Overall heat transfer coefficient can be obtained when substituting equations (2) and (8) into equation (7).

$$U = \frac{Q}{A \Delta T_{lm}} \tag{10}$$

The heat removed by the water in the condenser unit must be equal the amount of heat lost by the refrigerant. Heat transfer rate was found 609.35W in previous calculation. If the specific heat of water is taken as 4.18 kj/kgK, for water at 20 °C with flow rate of 40 g/s, the exit temperature of the water can be calculated as follows:

$$609.35\ \mathrm{W} = 40\ g/s \cdot 4.18\ kj/kgK \cdot (T_6 - 20°\mathrm{C})$$

$$T_6 = 23.64\ °\mathrm{C}$$

It is assumed that refrigerant leaves the condenser in the saturated vapor phase which means both inlet and outlet temperatures of the refrigerant are 29.08 °C.

Since inlet and outlet temperatures of the water and refrigerant are known, the logarithmic mean temperature difference is determined as

$$\Delta T_{lm} = \frac{(T_{sat} - T_6) - (T_{sat} - T_5)}{ln\left(\frac{T_{sat} - T_6}{T_{sat} - T_5}\right)} = \frac{(29.08\ - 23.64) - (29.08\ - 20)}{ln\left(\frac{29.08\ - 23.64}{29.08\ - 20}\right)} = 7.105\ K$$

Substituting ΔT_{lm}=7.105 K, Q=609.35 W and U=1144 W/m^{2}K into equation (10) gives the total heat transfer surface area of the condenser unit under assumed conditions.

$$1144\ W/m^2K = \frac{609.35\ \mathrm{W}}{A(7.105\ K)}$$

$$A = 0.074965\ m^2 \cong 0.075\ m^2$$

3. MATERIAL AND METHODS

In order to investigate the effect of water mass flow and refrigerant pressure on condensation process a refrigeration system where a shell and tube exchanger as a condenser with a heat transfer surface area of 0.075 m^2 was installed (Figure 1). Municipal water is circulated in tube side of the condenser in order to cool the refrigerant and the amount of water flow is controlled by control valve. Refrigerant is R134a and flows on shell side of the exchanger. The condensation occurs on the outer surface of the pipe that water circulates through.

In the refrigeration system, there are two electric heaters in the evaporator where the cooling load can be adjusted. Thus, the evaporation pressure, therefore the condensation pressure, will be controlled.

The bulb of the thermostatic expansion valve is

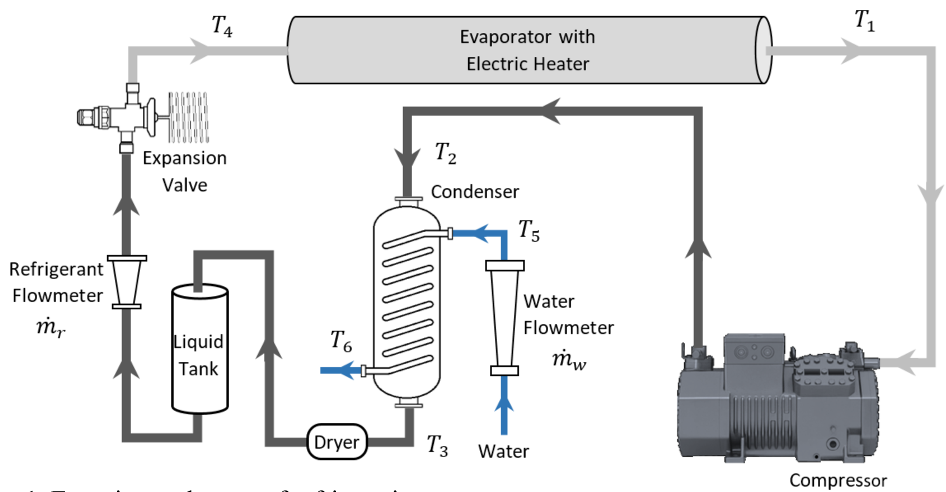

Figure 1. Experimental setup of refrigeration system

placed at the outlet of the evaporator so that the amount of superheat of the refrigerant is controlled.

The refrigerant, R134a which left from the evaporator outlet in the vapor phase is absorbed by the compressor. The pressure and temperature of the refrigerant are increased by the work input in the compressor. At this high temperature, the refrigerant at high pressure is sent to the water-cooled condenser. The condenser cooling water flows through the sealed copper tube coils in the condenser.

4. INVESTIGATION OF THE OVERALL HEAT TRANSFER COEFFICIENT

Heat transfer coefficient can be determined analytically by using correlations given in the literature.

Overall heat transfer coefficient is calculated by the following equation [13,14].

$$U = \frac{1}{\frac{1}{h_r} + \frac{t}{k} + \frac{1}{h_w} + R_f} \tag{11}$$

where h is the coefficient of heat convection, k is the the coefficient of heat conduction, t is the pipe thickness and R_f is fouling factor. Also, subscripts "r" and "w" indicate refrigerant side and water side, respectively.

In the heat exchanger, heat is transferred by forced convection for both the water flow in the pipe and the flow outside the pipe. For this reason, the heat transfer coefficient on both sides is calculated with the help of Nusselt number [14].

$$h = \frac{k}{d} Nu \tag{12}$$

In circular pipes, for single phase, turbulent and fully developed flow, Nusselt number is computed by Gnielinski [15] correlation;

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)} \tag{13}$$

where f is friction factor, the dimensionless numbers, Re and Pr are the Reynolds number and the Prandtl number, respectively. This equation is valid for the conditions of $0.5 \leq Pr \leq 2000$, $3000 \leq Re \leq 5x10^6$ and $l/d \geq 10$. The friction factor is obtained from the Moody diagram or from the following equation for smooth pipes [10]:

$$f = (1.82\, lnRe - 1.64)^{-2} \tag{14}$$

For single phase liquid water, the Reynolds number and Prandtl number are respectively

$$Re = \frac{u_{m,w} d_i}{\nu_f} = \frac{\dot{m}_w d_i}{A_c \mu_f} \tag{15}$$

$$Pr = \frac{\mu_f c_{p,f}}{k_f} \tag{16}$$

where $u_{m,w}$ is mean water velocity, d_i is inlet diameter of the pipe, ν_f is kinematic viscosity of liquid phase, $\dot{m}_w$ is water mass flow rate, A_c is cross-sectional area, μ_f dynamic viscosity of liquid phase, $c_{p,f}$ is specific heat of liquid phase and k_f is thermal conductivity of liquid phase. All thermophysical properties of the water (μ_f, $c_{p,f}$, k_f) are taken at the saturation temperature of the fluid.

Refrigerant condenses outside the horizontal pipe in the case of laminar film condensation. Therefore, the Nusselt theory [16] yields the following correlation:

$$Nu = 0.728 \left[\frac{\rho_f (\rho_f - \rho_g) g h_{fg} {d_o}^3}{\mu_f [T_{sat} - T_{wall}] k_f} \right]^{1/4} \tag{17}$$

where ρ is density of refrigerant, g is gravitational acceleration, h_{fg} is saturation enthalpy of refrigerant, d_o is outside diameter of pipe, μ is dynamic viscosity of refrigerant, T_{sat} is saturation temperature of refrigerant, T_{wall} is wall temperature of horizontal pipe and k is thermal conductivity of refrigerant. Also, subscripts "f" and "g" indicate liquid and vapor phase, respectively. All thermophysical properties of the refrigerant are taken at the saturation temperature of the refrigerant.

Besides, overall heat transfer coefficient can be also investigated by applying methods of LMTD and ε-NTU on experimental results.

5. RESULTS AND DISCUSSIONS

Firstly, experiments were done under condition of 7.0 bar condensation pressure by changing water mass flow rates as 15, 20, 25, 30 and 35 g/s. The results are presented in Table 1. When cooling water flow rate increases from 15 g/s to 35 g/s at constant condensation pressure, refrigerant flow rate increased by 106.7% while water outlet temperature decreased by 8.5%.

Table 1. Experimental results at 7.0 bar condensation pressure with water flow rates of 15, 20, 25, 30 and 35 g/s

Refrigerant Side				Water Side		
P_{con} bar	T_2 ℃	T_3 ℃	$\dot{m}_r$ g/s	T_5 ℃	T_6 ℃	$\dot{m}_w$ g/s
7.00	34.5	15.3	1.5	19	24.8	15
7.00	45.8	16.2	2.0	19	24.2	20
7.00	48.8	17.3	2.6	19	23.4	25
7.00	52.0	17.7	2.7	19	23.2	30
7.00	53.3	17.6	3.1	19	22.7	35

Then, condensation pressures were changed to 6.5, 6.75, 7.0, 7.25 and 7.5 bar at constant water flow rate of 25 g/s. The results are presented in Table 2. Experimental results show that at constant water flow rate, increasing condensation pressure from 6.5 bar to 7.5 bar caused an 83.3% increase in refrigerant flow rate. Likewise, the condenser outlet temperature has slightly increased by 11.4%.

Table 2. Experimental results at 25 g/s water flow rate at condensation pressures of 6.5, 6.75, 7.0, 7.25 and 7.5 bar

Refrigerant Side				Water Side		
P_{con} bar	T_2 ℃	T_3 ℃	$\dot{m}_r$ g/s	T_5 ℃	T_6 ℃	$\dot{m}_w$ g/s
6.50	55.8	13.1	1.8	18	22.0	25
6.75	55.5	15.3	2.2	18	22.7	25
7.00	48.8	17.3	2.6	18	23.4	25
7.25	56.5	18.3	3.1	18	24.3	25
7.50	56.5	19.7	3.3	18	24.5	25

ε-NTU and LMTD methods are applied to these experimental results. Calculation results are given in Table 3 and Table 4 for different flow rates with constant refrigerant pressure and water temperature and for different condensation pressures with constant water flow rate and temperature, respectively.

Table 3. Calculation results for ε-NTU and LMTD methods with different water flow rates

	ε-NTU			LMTD	
$\dot{m}_w$ g/s	ε $-$	NTU $-$	U W/m^2K	ΔT_{lm} K	U W/m^2K
15	0.724	1.288	1077.7	3.895	1078.3
20	0.660	1.077	1202.0	4.534	1203.3
25	0.601	0.920	1282.9	5.461	1282.2
30	0.558	0.816	1365.5	5.438	1363.6
35	0.518	0.729	1423.2	5.901	1422.9

Table 4. Calculation results for ε-NTU and LMTD methods with different condenser pressures

	ε-NTU			LMTD	
P_{con} bar	ε $-$	NTU $-$	U W/m^2K	ΔT_{lm} K	U W/m^2K
6.50	0.617	0.960	1339.8	3.742	1338.1
6.75	0.609	0.940	1310.5	4.627	1309.3
7.00	0.601	0.920	1282.9	5.461	1282.2
7.25	0.591	0.894	1246.9	6.803	1246.4
7.50	0.587	0.883	1232.1	7.292	1231.7

Calculated results show that, heat transfer coefficients obtained by both methods are close to each other. Increasing water flow rate from 15 g/s to 35 g/s at constant pressure, logarithmic mean temperature difference increased by 51.5%, while the effectiveness of heat exchanger and number of transfer unit decreased by 28.5% and 43.4%, respectively. On the other hand, with increasing condensation pressure from 6.5 bar to 7.5 bar at constant water flow rate, logarithmic mean temperature difference increased by 94.9%, while the effectiveness of heat exchanger and number of transfer unit decreased by 4.9% and 8.0%, respectively.

Equation 17 is solved as correlation of condensing heat transfer coefficient for experimental results. Results are given in Table 5 and Table 6 for different flow rates with constant refrigerant pressure and water temperature and for different condensation pressures with constant water flow rate and temperature, respectively.

Table 5. Analytical calculation results with different water flow rates

$\dot{m}_w$	Nu_w	h_w	Nu_r	h_r	U
g/s	$-$	W/m^2K	$-$	W/m^2K	W/m^2K
15	23.63	1877.5	297.29	2600.3	1087.9
20	29.59	2346.7	288.77	2525.8	1213.5
25	35.05	2773.2	277.29	2425.3	1290.4
30	40.60	3213.2	278.55	2436.4	1381.8
35	45.73	3614.5	273.61	2393.2	1435.7

Table 6. Calculation results for analytical investigation with different condenser pressures

P_{con}	Nu_w	h_w	Nu_r	h_r	U
bar	$-$	W/m^2K	$-$	W/m^2K	W/m^2K
6.50	34.81	2748.5	303.30	2688.2	1355.3
6.75	34.93	2760.3	288.35	2538.7	1318.9
7.00	35.05	2773.2	277.29	2425.3	1290.4
7.25	35.19	2787.5	263.29	2280.8	1251.2
7.50	35.21	2789.7	259.04	2237.1	1238.4

For constant pressure and constant flow rate conditions, the approximate relative error between the experimental results and the analytically calculated results are 0.93% and 0.72%, respectively. Increasing water flow rate from 15 g/s to 35 g/s at constant pressure, condensation heat transfer coefficient slightly decreased by 8.0%, although the water heat convection coefficient increased by 92.5%. On the other hand, with increasing condensation pressure from 6.5 bar to 7.5 bar at constant water flow rate, condensation heat transfer coefficient decreased by 16.8%, although the water heat convection heat transfer coefficient increased negligibly small as 1.5%.

Effects of cooling water flow rate on amount of condensate flow rate is shown in Figure 2.

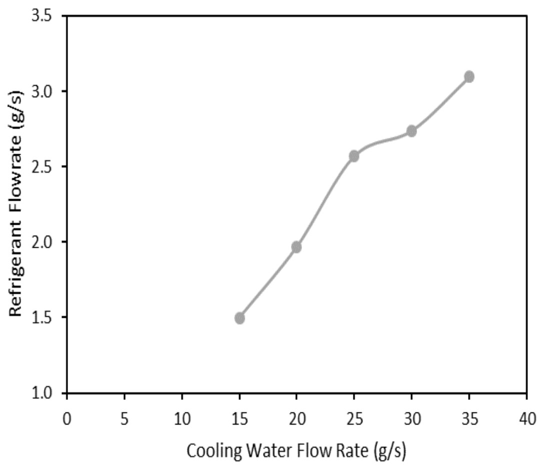

Figure 2. Effect of condensation pressure and water flow on condensate flow rate

Increasing in cooling water flow rate increases the refrigerant flow rate. Effects of condensation pressure and cooling water flow rate on temperatures of cooling water exit, saturation and refrigerant exit are plotted in Figure 3 and 4, respectively. Saturation temperature is proportional to the refrigerant pressure, then increasing in condensation pressure provides the rise in temperatures of water exit, saturation and refrigerant exit.

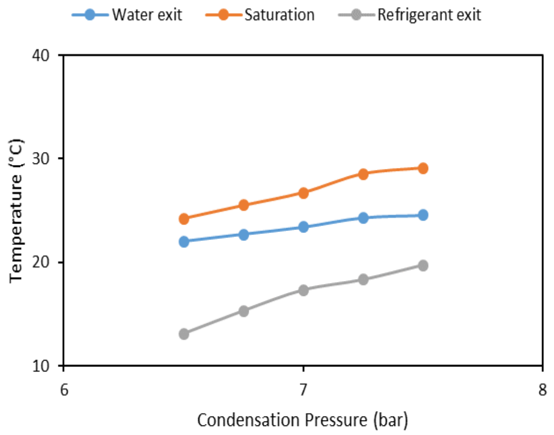

Figure 3. Effect of condensation pressure on temperatures of cooling water exit, saturation and refrigerant exit

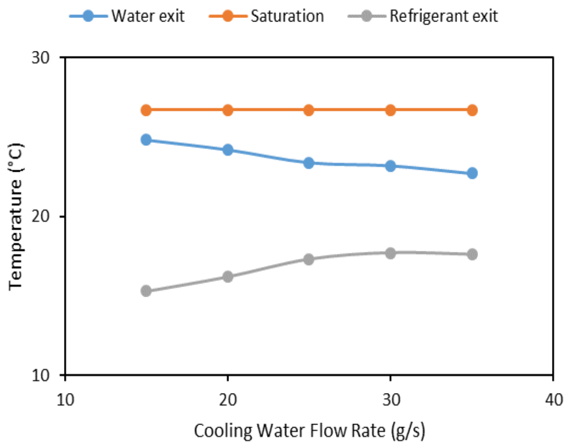

Figure 4. Effect of cooling water flow rate on temperatures of cooling water exit, saturation and refrigerant exit

Effects of condensation pressure and cooling water flow rate on amount of heat transfer to water, from refrigerant and thermodynamically possible maximum heat transfer rates are illustrated in Figure 5 and 6, respectively. Amount of the heat transferred from refrigerant and amount of the heat transferred to the cooling water must be equal. For that reason, these curves are coincided on the graph below. According to graphs, the amount of heat transferred increases with increasing condensation pressure and water flow rate.

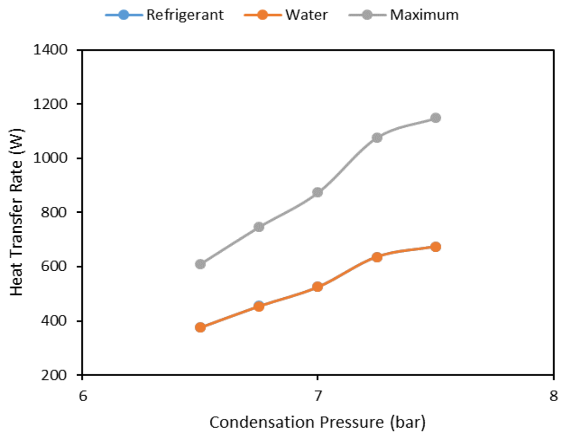

Figure 5. Effect of the condensation pressure on amount of heat transfer rate

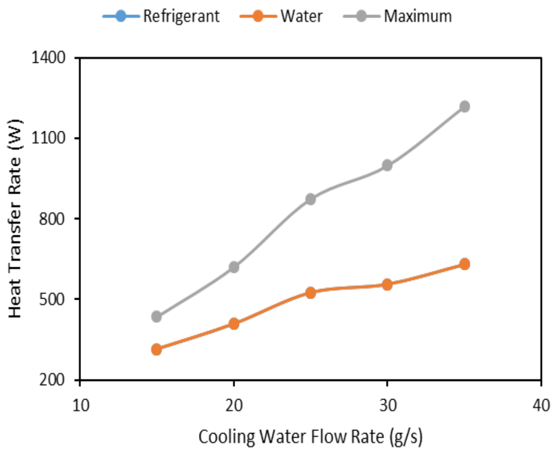

Figure 6. Effect of the cooling water flow rate on amount of heat transfer rate

Effects of condensation pressure and cooling water flow rate on overall heat transfer coefficients obtained from LMTD method, ε-NTU method and analytical method are highlighted in Figure 7 and 8, respectively. The relative error between the experimental results and the analytically calculated results for condensation pressures of 6.5, 6.75, 7.0, 7.25 and 7.5 bar are 1.3%, 0.7%, 0.6%, 0.4% and 0.5%, respectively. The relative error between the experimental results and the analytically calculated results for water flow rates of 15, 20, 25, 30 and 35 g/s are 0.9%, 0.8%, 0.6%, 1.3% and 0.9%, respectively.

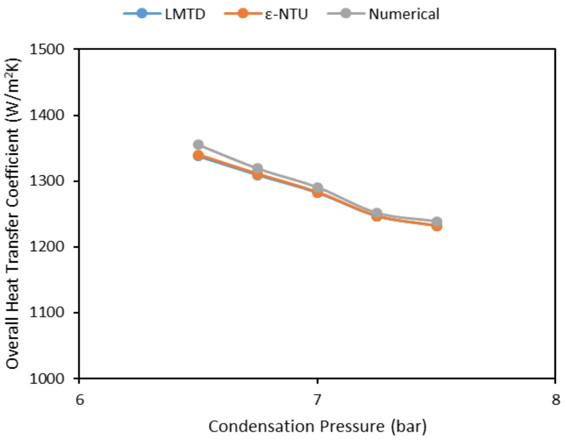

Figure 7. Effect of the condensation pressure on overall heat transfer coefficient

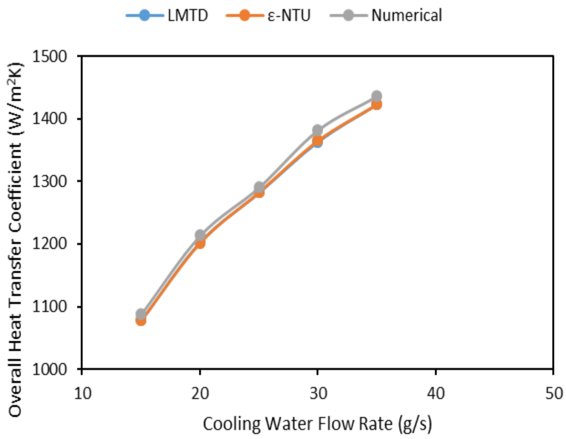

Figure 8. Effect of the cooling water flow rate on overall heat transfer coefficient

6. CONCLUSIONS

Condensation and evaporation, phase change processes, are the most important processes in the refrigeration cycle since latent heat is transferred. The amount of heat transfer rate is enormous high compared to the amount of sensible heat transfer.

Increasing condensation pressure from 6.5 bar to 7.5 bar at constant water flow rate caused an 83.3% increase in refrigerant flow rate, 79.3% increase in heat transfer rate and 16.8% decrease in condensation heat transfer coefficient were observed. Although there is a negligibly small increase in the water heat convection heat transfer coefficient.

On the other hand, increasing water flow rate from 15 g/s to 35 g/s at constant pressure resulted in an 106.7% increase in refrigerant flow rate, 100.0% increase in heat transfer rate and slight decrease in condensation heat transfer coefficient as -8.0%, although there is a considerable increase in the water heat convection heat transfer coefficient increased as 92.5%.

Heat transfer coefficient was calculated by using correlation of the Nusselt theory and calculated results have been compared with experimental results obtained from LMTD and ε-NTU methods. When cooling water flow rate increases 2.3 times, the overall heat transfer coefficient obtained from the correlation has increased by 32.0%. Otherwise, the overall heat transfer coefficient decreased by 8.6% with 13.3% increase in condensation pressure. The relative error of the overall heat transfer coefficients obtained from LMTD and ε-NTU methods are approximately 0.8%.

For constant pressure and constant flow rate conditions, the approximate relative error between the experimental results and the analytically calculated results are 0.93% and 0.72%, respectively.

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