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PERFORMANCE IMPROVEMENT OF AN AIR-CONDITIONING SYSTEM DURING REFRIGERANT EVAPORATION

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Abstract: In hot weather countries, the temperature and pressure of the air-conditioning system are increased considerably. This causes a decrease in the cooling capacity of the cycle and also causes an increase in the power consumption. In this work, an experimental and theoretical investigation has been done to improve the evaporator outlet fluid temperature. Engineering Equation Solver (EES) software has been used to analyze the performance of the experimental data. For this purpose, several ranges of evaporator water flow rate are considered tested. The amount of water flow rate is varied with respect to the evaporator load. The results indicate that the increase of water flow rate (160 to 190) L/h causes increase of COP by (11.1 %). The outlet cooling temperature from evaporator has been found to reduce by about (11.3%). However, (170 L/h) flow rate of water has been found as the best rate for improving evaporator temperature.

Keywords: *Evaporator, water flow rate, COP.*

تطوير كفاءة منظومة تكييف الهواء خلال عملية التبخير

الخلاصة: في البلدان ذات الظروف الجويه الحاره، فان درجة حرارة وضغط منظومة تكييف الهواء متزايد الى حد كبير. هذا يسبب نقصان في سعة تبريد دورة التبريد ويسبب زيادة ايضا في الاستهلاك المنظومة للطاقة الكهربائية. في هذا البحث تم التحقق عمليا ونظريا لتحسين درجة حرارة سائل التبريد الخارج من المبخر. تم استخدام البرنامج الهندسي لحل المعادلات المعروف ب (ESS) ويستعمل لتحليل اداء البيانات الماخوذة من الجزء العملي. لهذا الغرض، فان مجموعة من مديات معدل تدفق الماء بالنسبة للمبخر اختبرت واخذت بنظر الاعتبار. ان كمية معدل تدفق الماء مختلفة حسب اختلاف الحمل في المبخر. تشير النتائج الى ان زيادة معدل تدفق الماء بين (160 الى 190) L/h تسبب زيادة في معامل الاداء COP للمنظومة بنسبة (11.1%). ان درجة الحرارة الباردة والخارجة من المبخر انخفضت بحوالي (11.3%). على اية حال، فان كمية تدفق الماء بالمقدار (170 L/h) وجدت كافضل نسبة لتحسين درجة حرارة الهواء الخارجة من المبخر.

1. Introduction

Air conditioning systems are considered as an indispensable for life requirements. Most of them are based on vapor compression systems with window type, split units, chillers and heat pumps. Three main types of condensers used in these units are; air cooled, water cooled and evaporative cooling towers. Because the cooling medium (air) is a natural and free source, the air cooled condensers (fin and tube heat exchangers)

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Mostly used for low and average refrigeration capacities. In these air cooled condensers, the power consumption is a major issue in vapor compression cycle. The power consumption concern increased much more if the air-cooled condensers work in area

with very high ambient temperature (between 50-60 ^{0}C), as it happens in many Middle East countries. In this area, the temperature and pressure of the air-cooled condenser will increased considerably. This cause an increase in pressure ratio which increases the power consumption of the air conditioner. Also, when the pressure increases, the pressure limit in control system will shut down the compressor. The increase of the condenser temperature will decrease the cooling capacity of the cycle due to the reduction of liquid content in the evaporator. So, the high pressure and temperature of the air-cooled condenser will decrease the performance of the air conditioner considerably [1]. Chillers with water cycle are used for get a good performance comparing with air cooled systems. So, the water chiller is the best solution to the high temperature with variable water flow rate.

This field has been thoroughly investigated both experimentally and numerically, and the contributions are available in the literature. Payne and Domanski [2] explored the influence of increasing environmental temperature on the performance and pressure drop in the split air conditioning systems using R-22 and R-410a with no modifications. Michalis et al.[3] utilized an evaporative condenser working on water spray addition to the air side. Their conclusion is that this will be saving energy by 10%. Also they found that this will led to increase the duration life of the cooling unit. Hajidavalloo and Eghtedari [4] used cellulous media pad with water circulation ahead of the air condensation unit to develop evaporative condensation. They deduced that the evaporative condenser has higher performance compared to the air cooled condenser and that the higher the environmental temperature the more is the enhancement. Tissot et al.[5] studied sprayed air flow numerically.

They have explored droplet evaporation and air temperature under different loading conditions, solution and spray features. They found that using very fine droplets ranging between 25 and 50 μm results in a significant air cooling. Optimal conditions have between sought regarding their size, as too small droplets where found to flow in a concentrated manner with a poor dispersion ability resulting in a less effective mixing, despite their better expected capacities when considered as individual particles.

The counter flow injection is found to provide a more efficient cooling of the air. A study of heat transfer enhancement using an air flow containing water droplets across a heat exchanger has been carried out by Boulet et al. [6]. They showed that spray implementation increases the rate of heat exchange, and recommended further investigations to be carried out for predicting the gains. Moreover, recently Wen et al. [7] argued that evaporative cooling has not been covered properly in spite of the vast developments in split air conditioning systems and the advantages in power saving resulting by using air-cooled condensation.

Stevanovic et al. [8] numerical experiments are performed with design hanges of the evaporator header and the effectiveness of applied measures is demonstrated to eliminate the air exit temperature. They investigated by numerical simulations and analyses with the Computational Multi-Fluid Dynamic code. Under low operating loads a non-uniformity of the refrigerant liquid and gas phase distribution from the evaporator headers towards the evaporating channels leads to a decrease of the evaporator efficiency and a temperature stratification in the exit stream of cooled air. Wen et al. [9] Presents an experimental study for two-phase refrigerant distribution in the manifold with three different distributor evaporator configurations (Venturi tube connected with a smooth tube, Venturi tube connected with an internally–spirally micro-finned tube and distributor within a roller).

It shows that the Venturi tube connected with internally–spirally micro-finned tube has a larger COP and a lower C at a fixed mass flux than those of the distributor within a roller and the Venturi tube connected with the smooth tube. It means that the Venturi tube connected with internally–spirally micro-finned tube has the best distribution efficiency. Also, the best distribution efficiency results in the largest COP.

Somchai et al. [10] investigate an experimental data from the measurement of the tube-side heat transfer coefficient of HFC-134a evaporating inside a plate fin-and-tube evaporator with plain fin geometry. A new correlation based on the data gathered proposed for practical application.

Further, unlike the conventional air cooled evaporators, there is a lake for Performance data of water-Refrigerant (W-R) evaporator that subjected to high ambient temperature. Keeping the above in view, the main objectives of this work include;

- Design and Manufacture of a Water-Refrigerant (W-R) evaporator for comparing its effects on system performance with that of an air-cooled evaporator.
- Investigating the effects of different rates of evaporator water flow rate on the performance of an air-conditioner working at high ambient temperature with an air-cooled condenser.
- Using (EES) software to develop a computer program for predicting the system thermal performance for various system modifications.

2. Equipment Layout

The equipment layout of this work is shown in "Fig. 1". Two types of evaporators have been tested for enhancing the evaporative performance at high ambient temperature [11]. In this work, several computer programs in Engineering Equation Solver (EES) have been written for each type of the modifications. It should be noted that thermo-physical properties of R22 is already built into the software [12].

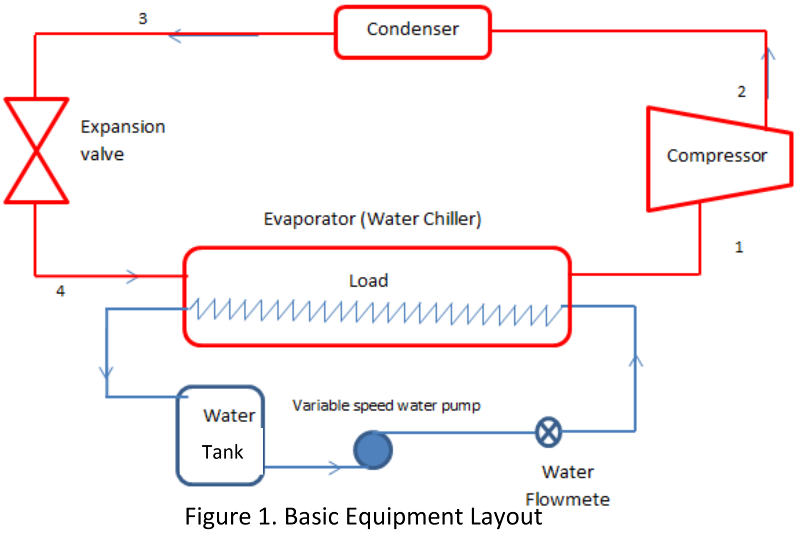

Figure 1. Basic Equipment Layout

A variable speed water pump is used to raise the water flow rate in the chiller and water flow meter is also used for measure the discharge of water with time. The volume flow rate of the water that used is fixed and measured continuously. A liquid water with antifreeze is used as a working fluid for this test rig. "Fig.2". "Fig.3" shows the (water-refrigerant) shell and tube evaporator that connected to the air-conditioning cycle.

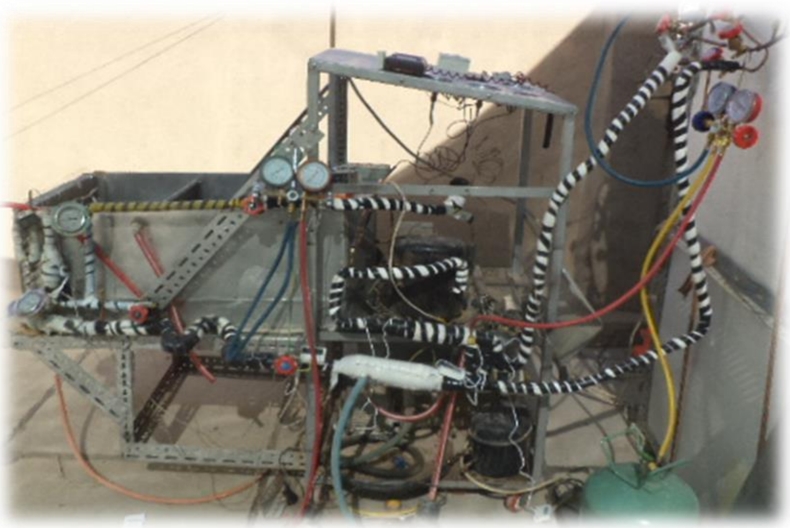

Figure 2. Test Rig with outdoor details

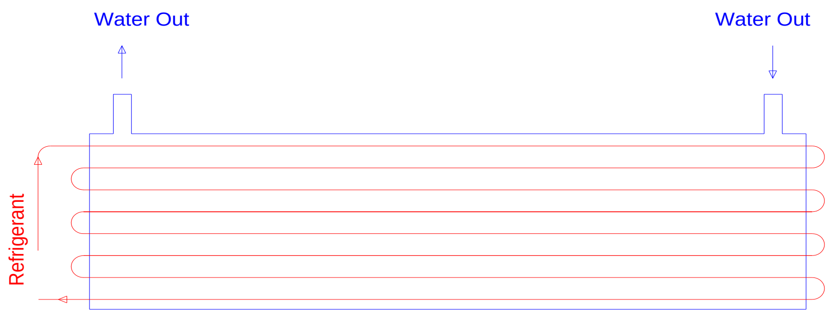

Figure 3. Water-Refrigerant Shell and Tube Evaporator

The length of the shell (W-R) evaporator is (1.2 m) of copper material. The shell pipe diameter is (2.15 inch) and includes nine copper tubes of (0.25 in) diameter. This model has been improved and modified to keep the same load of the original air type evaporator that is commonly used with this air-conditioning system. The (W-R) evaporator with the thermocouples and the pressure gauges that fixed in both water and refrigerant sides. The shell pipe is covered with (10 mm) insulation. "Fig. 4" shows the cross-section of the evaporator.

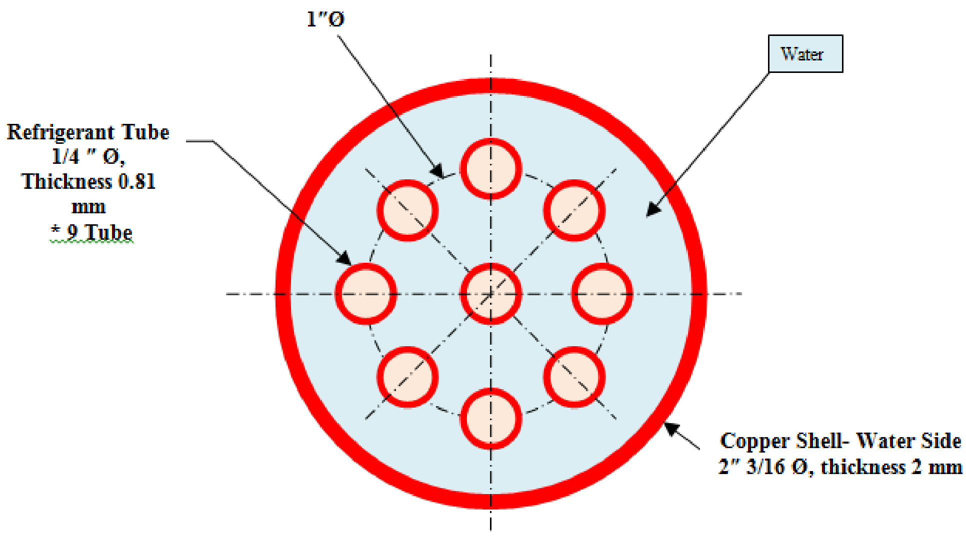

Figure 4. Cross Section of the Evaporator

The weather conditions during all tests in the present study were the same. Data were recorded after a steady state condition was established in the system and the properties of refrigerant and air remained constant. Five low pressure gauges and four high pressure gauges were fixed to measure the pressure drops along the refrigeration cycle. A total of twenty three thermocouples have been fixed in the air-conditioning system to record the temperature at different locations as shown in "Fig. 5" [11]. This Figure shows the location of all types of evaporators and the fixed sensors. A kilowatt-hour meter was used to measure the energy consumption. To measure the input power, voltage and electrical current, a clip-on power meter with accuracy of ±0.01% was used. All data were acquired by a computerized data logging system. The experimental data were recorded continuously with (180 s) intervals.

The capacity was determined by measuring the mass flow rate and enthalpy difference of both air-side and refrigerant-side.

For each evaporation type, all experiments were performed at least twice (on different days), to check the repeatability of the data, which was proved to be good. Because the data demonstrated repeatability, only results one of the test will be presented here. The uncertainties for important parameters and measurements made during the current research have been carried out on the basis of the method proposed by Moffat [13]. The maximum uncertainties are ±2.73% for refrigerant Reynolds number, ±2.14% for water

Reynolds number, ±1.88% for air Reynolds number, , ±0.78% for fluid temperature, and ±3.81 for rate of heat transfer.

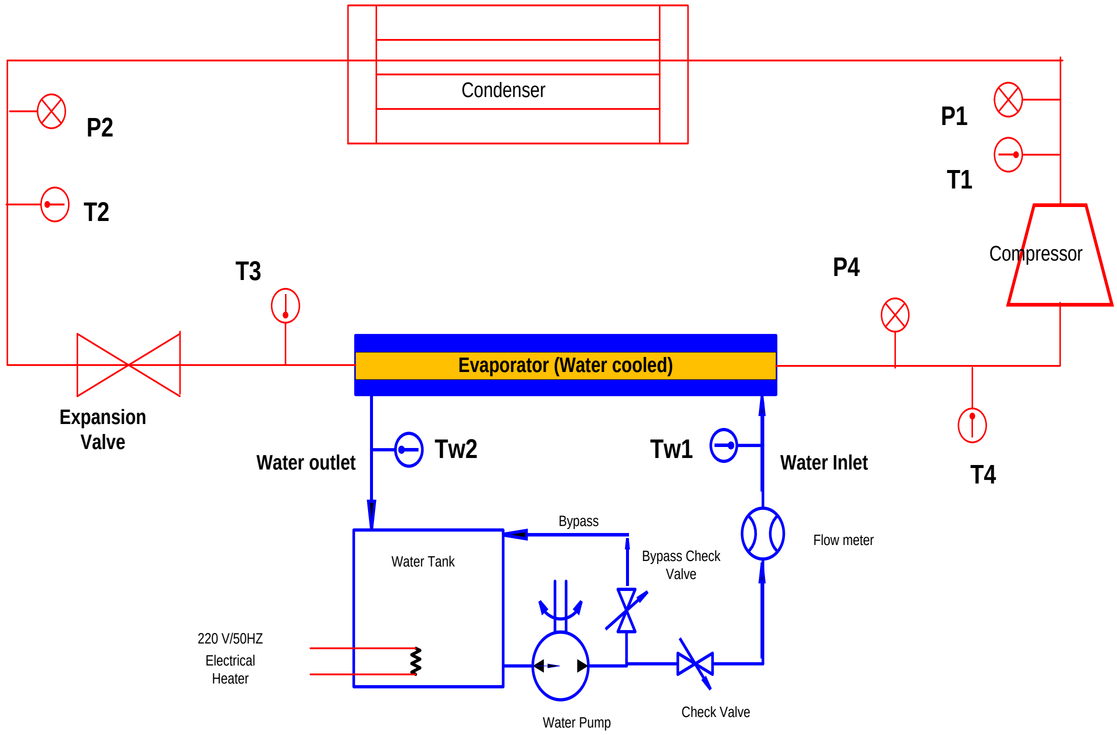

Figure 5. Schematic diagram of the air- conditioning system with modifications and sensors

3. Theoretical Model

Rotary type, positive displacement compressor, has been used in current study. The amount of specific, isentropic work done by an ideal compressor can be found by the energy equation:

$$w_{s,com} = \dot{m}_r(h_{2s} - h_1) \quad (1)$$

Since the finned-tube heat exchangers are most commonly used for residential air conditioning applications, a plate finned-tube type has been used in this work.

The condenser can be separated into three sections: superheated, saturated, and sub-cooled shown in "Fig. 6". The specific heat rejected from each section can be found by evaluating the refrigerant enthalpies at the inlets and outlets, so the equations (2 to 4) presented the heat rejection.

$$q_{c,sh}=h_2-h_{2a} \quad (2)$$

$$q_{c,sat}=h_{2a}-h_{2b} \quad (3)$$

$$q_{c,sc}=h_{2b}-h_3 \quad (4)$$

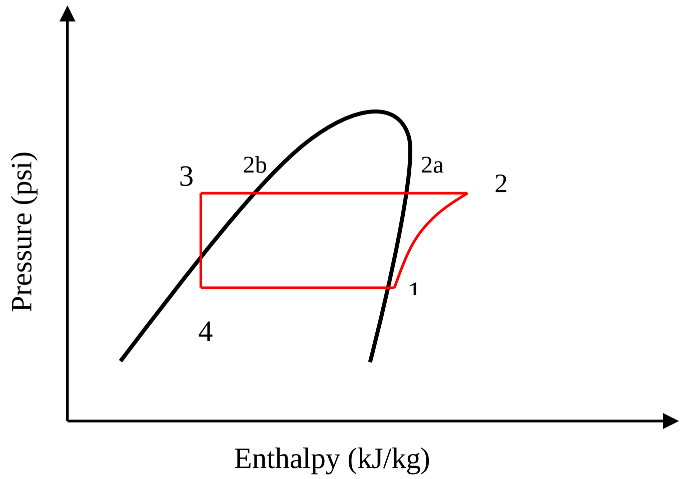

Figure 6: P-h Diagram of the cycle

Where h represent the enthalpy of the refrigerant at points 2,2a,3.

Then the heat transfer of the air across the condenser can be calculated according to the difference bulb temperature between the air inlet and air outlet temperatures as given in the following equation [8].

$$\dot{Q}_a = \dot{m}_a Cp_a \,\Delta Tb_\infty \qquad (7)$$

Where,

$$\Delta Tb_\infty = (T_{\infty 2} - T_{\infty 1}) \qquad (8)$$

Also the air mass flow rate that effect on the condenser coil can be calculated from the following expression:

$$\dot{m}_a = \rho_a V_\infty A_c \qquad (9)$$

While the surface temperature (T_s) is constant across the condenser, then the heat balance can be expressed by [8].

$$\dot{Q}_a = \dot{m}_a Cp_a \,\Delta Tb_\infty = \mathrm{h}_a \, A_{s,c}(T_s - Tb_m) \qquad (10)$$

Where,

$$A_{s,c} = N_c \pi D_c L_c \qquad (11)$$

Where, N_c = Number of refrigerant pipe in condenser.

All the air properties can be taken at Tb_m of the inlet and outlet of an air duct.

$$Tb_m = \frac{(T_{\infty 1} + T_{\infty 2})}{2} \qquad (12)$$

The heat transfer coefficient for the air side of the condenser can be calculated from the following equation (16) [9].

$$Q_a = h_{c,} N_c \pi D_c L_c (T_s - Tb_m) = \dot{m}_a cp \Delta Tb_\infty \qquad (13)$$

Also $\dot{m}_a$: is the mass flow rate of the air, could be calculated from the following equation:

$$\dot{m}_a = \rho V_a N_c S_n \qquad (14)$$

Also the heat transfer coefficient can be calculated using equation (18).

$$h_a = \frac{Nu.K}{D_c} \qquad (15)$$

If the number of rows less than 10, so the following equation can be calculates the Nusselt number [8]:

$$Nu|_{(N_L<10)} = C_2 . Nu|_{(N_L \geq 10)} \qquad (16)$$

The constants (C_1, n) are depends on the ratio of (S_p/D_c) & (S_n/D_c), that's shown in "Table 1", and (C_2) depends on the number of horizontal rows, as shown in "Table 2" below [10]:

Table 1 Constant (C_1, n) [10]

$b=S_p/D_c$	$a=S_n/D_c$							
	1.25		1.5		2.0		3.0	
	C_1	n	C_1	n	C_1	n	C_1	n
in-line								
1.25	0.386	0.592	0.305	0.608	0.111	0.704	0.703	0.752
1.5	0.407	0.586	0.278	0.620	0.112	0.702	0.0753	0.744
2.0	0.464	0.570	0.332	0.602	0.254	0.632	0.220	0.648
3.0	0.322	0.601	0.396	0.584	0.415	0.581	0.317	0.608
Staggered								
0.6	-	-	-	-	-	-	0.236	0.636
0.9	-	-	-	-	0.495	0.571	0.445	0.581
1.0	-	-	0.552	0.558	-	-	-	-
1.125	-	-	-	-	0.531	0.565	0.575	0.560
1.25	0.575	0.556	0.561	0.554	0.576	0.556	0.579	0.562
1.5	0.501	0.568	0.511	0.562	0.502	0.568	0.542	0.568
2.0	0.448	0.572	0.462	0.568	0.535	0.556	0.498	0.570
3.0	0.344	0.592	0.395	0.580	0.448	0.562	0.467	0.574

Table 2 Constant (C_2, n) [10]

n	1	2	3	4	5	6	7	8	9	10
In-line	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.0
staggered	0.68	0.75	0.83	0.89	0.92	0.95	0.97	0.98	0.99	1.0

Reynolds and Prandtle numbers can be calculated from equation (20 & 23), as the following:

$$\mathrm{Re} = \frac{\rho V_{\max} D_c}{\mu} \tag{17}$$

The maximum velocity can be found by the following equation (21) according to the dimensions shown in "Fig. 7".

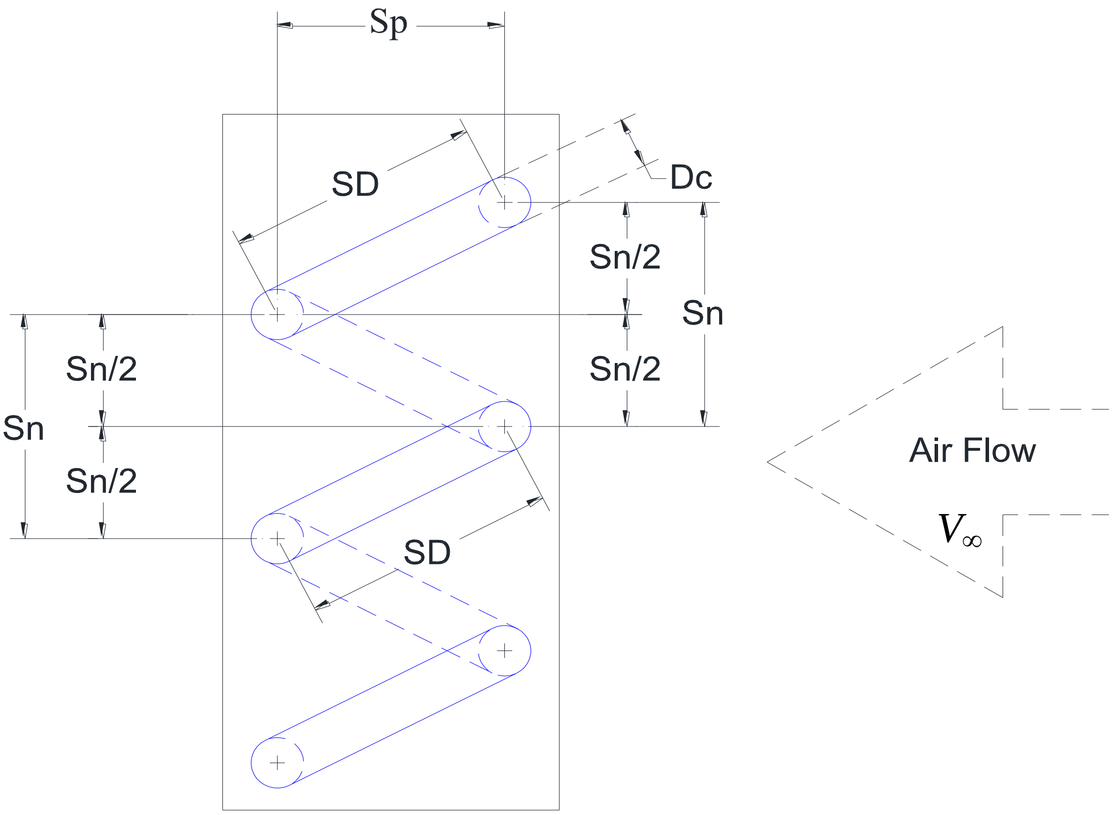

Figure 7. Z-type finned-tube dimensions

$$V_{\max} = V_\infty \frac{S_n}{2(S_D - D_c)} \tag{18}$$

$$S_D = \left[S_p^2 + \left(\frac{S_n}{2}\right)^2\right]^{1/2} \tag{19}$$

$$\mathrm{Pr} = \frac{\mu . C_p}{k} \tag{20}$$

Mass flow rate of the air to be assumed as a uniform distributed over the whole coil face regardless of the coil and fan respective locations. So each coil was associated with the same air mass flow rate. Also it was assumed that the air stream passing through the coil with a sufficient turbulence status so that the air with uniform properties was enters each tube bank.

For Turbulence flow, Dittus-Boelter correlation [9] has been used to calculate heat transfer coefficient for refrigerant side. The heat transfer coefficient is represented as:

$$h_r = 0.023\mathrm{Re}_f^{0.8}\ \mathrm{Pr}_f^{0.3}\ \left(\frac{k_f}{d_f}\right) \tag{21}$$

The heat transfer rate over the condenser of refrigerant side can be calculated from the following equation:

$$\dot{Q}_r = \dot{m}_r\ (h_2 - h_3) \tag{22}$$

3.1. Capillary Tube

Under normal operating conditions, there is a thermostat connected with capillary tube to maintain a fixed superheat exiting the evaporator.

The energy equation assumes that the enthalpy is constant across the expansion valve.

$$h_3 = h_4 \tag{23}$$

3.2. Evaporator

In this study an air-cooled evaporator was used. To keep the evaporator model simple, the coil is assumed to be dry, so the air-side heat transfer coefficient is not affected, but the specific heat is corrected to account for condensation. Because the air flowing over the evaporator is cooled below the wet bulb temperature, some of the heat rejected by the air results in condensing water out of the air rather than lowering the temperature. The total enthalpy change of the air is the sum of the enthalpy change due to temperature drop, or sensible heat, and the enthalpy change due to condensation, or latent heat.

The coefficient of performance (COP) is the ratio of the heat absorbed by the evaporator to the amount of compressor work as shown in the following equations:

$$COP = \frac{\dot{Q}_e}{\dot{W}_{com}} \tag{24}$$

$$\dot{W}_{com} = (h_2 - h_1) \tag{25}$$

$$\dot{Q}_e = (h_1 - h_4) \tag{26}$$

4. Results

"Fig. 6" shows the effect of increasing water flow rate on both exit water temperature from (W-R) evaporator. It is clear that there is an obvious drop in $(T_{w\ o})_{evap.}$ with increasing $(\dot{m}_{w.})$ by a range of (160 to 190 L/h). This behavior causes drop in both evaporator temperature and pressure.

Figure (7) shows comparison between system COP with (W-R) evaporator system with and without variable water flow rate by about (11.1%).

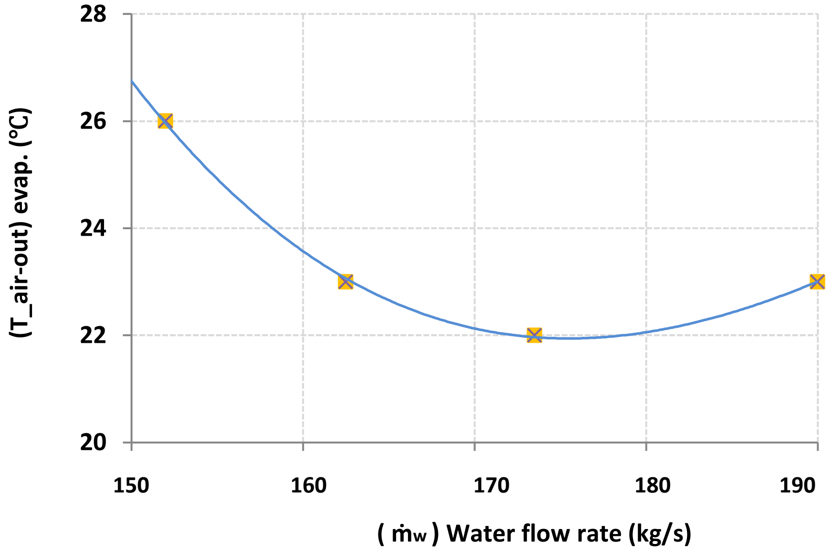

Figure 6. Effect of increasing water flow rate on the exit fluid temperature from evaporator.

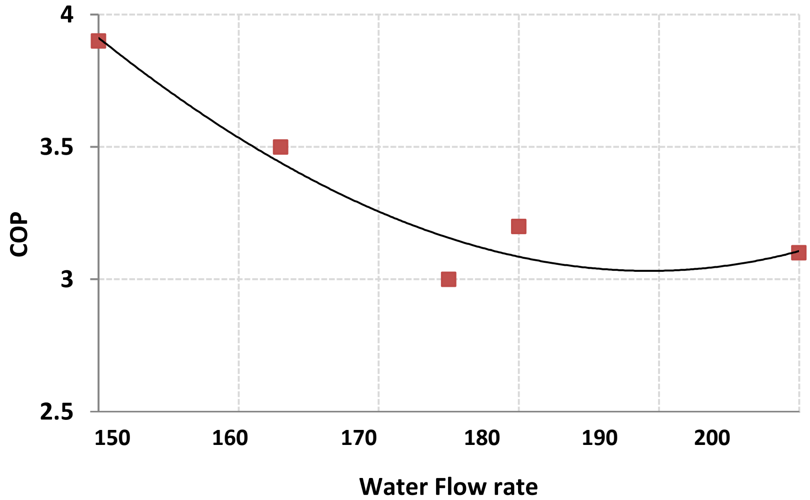

Figure 7. Effect of water flow rate on COP of system

However, more than 190 L/h of water flow rate gives an opposite effects. For this range of flow rate it is found that the consumed work of the compressor has been reduced bout (15.1%).

Figure (8) shows a comparison between (A-R) and (W-R) evaporators with variable speed for water flow rate.

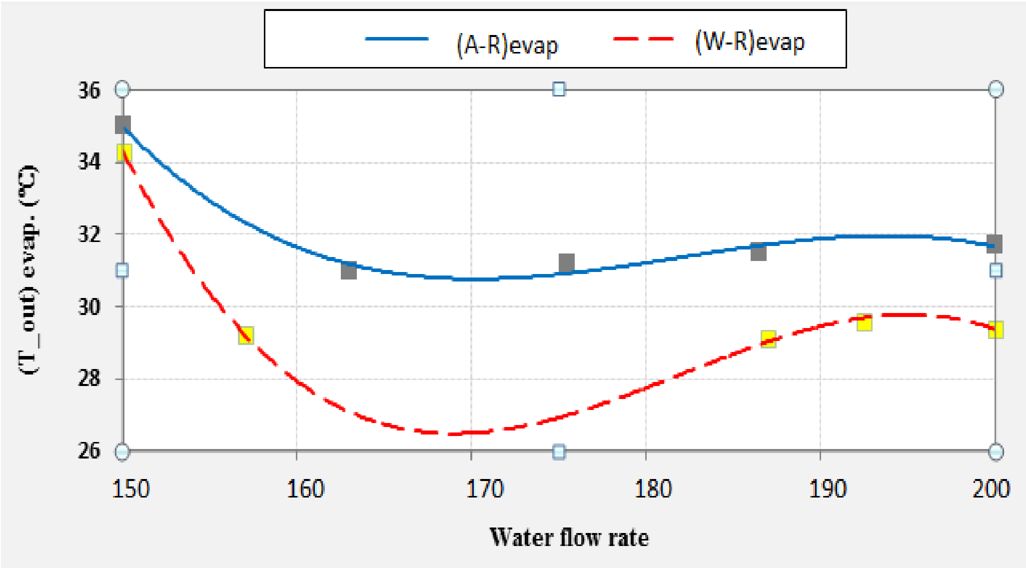

Figure 8. Effect of water flow rate on (A-R & W-R) evaporators

5. Conclusions

Within the limitations of materials and experimental results obtained in this work, the main conclusions may be summarized as follows;

At high ambient temperature, using (W-R) evaporator for enhancing system performance have been found to produce an obvious drop in outlet fluid temperature from evaporator. The increase in water flow rate from (160 to 190 L/h) has been found to increase the COP by (11.1 %) with a decrease in outlet air temperature of (11.3 %) for (W-R) evaporator.

Abbreviations

$(A\text{-}R)_{evap.}$	Air-Refrigerant Evaporator
COP	Coefficient Of Performance
EES	Engineering Equation Solver
HE	Heat Exchanger
L/h	Litter per hour
P	Pressure
T	Temperature
T_{w1}	Water temperature measured at the inlet
T_{w2}	Water temperature measured at the
T_{amb}	Ambient temperature
$(T_{w\,o})_{evap.}$	Outlet water temperature from
$(W\text{-}R)_{evap.}$	Water-Refrigerant evaporator
$\dot{m}_w$	Water flow rate

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