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The Effect of the Cooling Water Loop on the Exergy Destruction Components of Split Air Conditioning Systems

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ABSTRACT

The present study is an experimental investigation to improve the performance of the air conditioning unit (ACU) by precooling the condenser with a cooling water loop using a water heat exchanger with the employment of energy and exergy analyses under various ambient temperatures between 30 – 45℃ at different inlet water temperatures; 15, 19, 24 and 30℃ and water flowrates; 9, 11, 14 and 15.8 L/min. The results indicated that the cooling water loop have a large effect on the exergy destructions as compared with those of the no water case. When a cooling water loop is used, the compressor, evaporator, and expansion valve irreversibilities reduced. The exergy efficiency of the unit decreases as Tamb increases to about 23%; while, the exergy efficiency increases when cooling water loop is used depending on the inlet water temperature and water flowrates. The maximum enhancement in the exergy efficiency is obtained at high water flowrate for Twi = 15℃ with a percentage value about 13% as compared with those of the conventional ACU.

Keywords:

Exergy, Refrigeration, COP, ACU, Energy

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1. INTRODUCTION

Energy consumption has increased as a result of the rise in demand for cooling and refrigeration systems. [1]. Power consumption is a significant issue during the vapor compression cycles of air conditioning systems that use air-cooled condensers [2]. Many techniques have been used to reduce the energy consumption of the air conditioning system. One of them is cooling the refrigerant using cooling water loop before entering the condensing unit by using a refrigerant heat exchanger [3]. Exergy is a concept which is used in engineering systems and its related to the energy. It determines the true efficiency level of engineering systems [4].

Exergy is described as the maximum amount of mechanical work that can be generated from a given flux (or quantity) of energy [5]. Exergy analyses typically aim to determine the system's maximum performance as well as the locations where exergy is destroyed[6]. The exergy analysis based on the second law of thermodynamics is regarded as an effective tool for identifying irreversible loss in an energy system[7]. Various studies on vapor compression refrigeration systems have been published [8].

Karakurt et al [9], studied the exergy and economic effects of sub cooling and superheating on vapor compression refrigeration systems. Variations in condenser and evaporator temperatures, pressure losses in heat exchangers, and compressor isentropic efficiency are all highly related to performance and economic parameters. Kílíc [10], used Solkane software to calculate the thermodynamic properties of the refrigerants; R507, R407c, R404a containing hydrocarbon blends. The energy and exergy performance of the whole system were investigated under different operating conditions.

Ahamed et al [11], investigated the vapor-compression refrigeration cycles, which are used in many different industries, undergone exergy analyses. As alternatives to R-134a, they conducted their studies using R-407A, R-600A and

R-410A. They have determined that the compressor is where the majority of the exergy is destroyed. Xia Song et al [6], used energy and exergy analyses to determine the cooling effectiveness of a transcritical CO2 air conditioning system for an electric bus.

According to test results, the cooling capacity was 31.4 kW and the coefficient of performance (COP) was 2.88 at a temperature of 35 °C/27 °F. KARAKURT et al [7], analyzed and investigated the exergetic and economic effects of subcooling and superheating on VCRS. The results indicated that the variations in the condenser and evaporator temperatures, pressure losses in heat exchangers, and airflow are closely related. Yang et al [12], used the exergy analysis method, and the VCRS using R22, R134a, R410A, and R717 were performed. The COP, heat-exchanger area, irreversibility, and friction loss in this system were calculated using the first law of thermodynamics. The optimal subcooling obtained by applying the second law of thermodynamics is always greater than the first law. Saravanakumar et al [13], conducted an exergy analysis of a domestic refrigerator using hydrocarbon refrigerant mixture R290/R600a as an alternative to R134a. When the mixture refrigerant was used in place of R134A, a evaporator temperature of 263°K produced the highest average exergic efficiency of the system (42.1%) (-10°C). Yasin Ust et al[14], an analysis of a cascade refrigeration system's theoretical performance using the exergetic performance coefficient (EPC) criterion has been done. The study's results show that in terms of availability and coefficient of performance, the refrigerant couples R23-R717 performed as the best one. Anand et al [15], conducted an experimental analysis of 2TR for a vapor compression refrigeration cycle using different percentage of refrigerant charge via the exergy analysis method. The total exergy destruction was high when the system is 100% charged, and low when it is 25% charged. Losses in the compressor are more pronounced. Chandrasekharan [16], studied the vapor compression refrigeration cycle with R12 and R134a as refrigerants, based on a computational model, using an energy and exergy analysis.

According to the literature review, the exergy analysis method may be used to analyze the VCRC system. In previous studies, it was shown that both theoretical and experimental research had been done to modify the system's performance by using different techniques. The current work is an experimental investigation to ascertain whether the use of a water heat exchanger to pre-cool the condenser with a cooling water loop as opposed to a conventional split AC unit will improve the exergy destructions for the main mechanical parts of the split air conditioning unit in terms of evaporator, compres sor, expansion valve, and condenser and the exergy efficiency under various ambient temperatures around the condenser, different inlet water temperatures and different water flowrates.

2. EXPERIMENTAL WORK

2.1. Experimental rig

In the laboratory of the Zakho Technical Institute, the split-type AC unit and its accessories were designed, manufactured, and installed. The experimental rig is displayed in Figure 1 with its components which displays in table 1.

No	Name of component	No	Name of component
1	Controlling board	10	Inlet water
2	AC out door unit	11	Refrigeration heat exchanger
3	Water heater	12	Inlet Ref
4	Water tank	13	Outlet water
5	Water flow meter (L/min)	14	AC remote control
6	Water pump	15	Outlet Ref
7	Air heater	16	Heater's thermostat
8	Thermometers		
9	Pressure gauge of refrigerant		

Table 1: Experimental apparatus components

Figure 1 shows a schematic diagram of the experimental apparatus displaying the main components of a split air conditioning unit with a capacity of one ton refrigeration manufactured by CHRANI Company. The split unit consisting of the rotary compressor, a finned tube air-cooled condenser, an expansion valve (capillary tube), and an air-cooled evaporator compact finned tube. The cooling water loop is made up of a shell and coil tube heat exchanger that is connected between the condenser and compressor (discharge line). This equipment's working fluid is R22, and is located in a room with dimensions of 4 x 3 x 2.9 m. As illustrated in Figure 2, which displays the water heat exchanger and its schematic figure, the superheated refrigerant vapor flows from the inner tube into the heat exchanger tube, while cooling water flows from the outer tube. The measurement instruments control many parameters. The thermocouples are used to measure the ambient temperature, outlet and inlet water temperatures, and outlet and inlet air condenser temperatures, as shown in figure 1, are measured using the thermocouples. Figure 1 shows the pressure and temperature of the compressor's refrigerant outlet and inlet, refrigeration heat exchanger, and expansion valve. A water flow meter (rotameter) is used to measure the volumetric water flow rate that enters the heat exchanger.

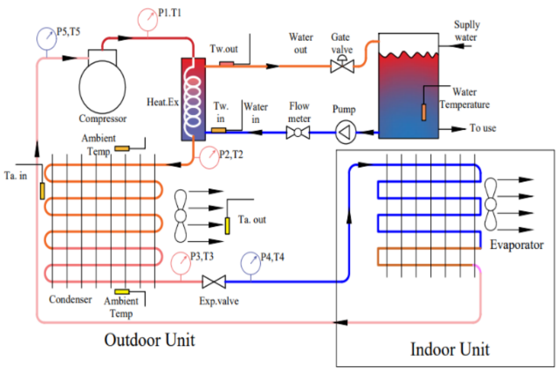
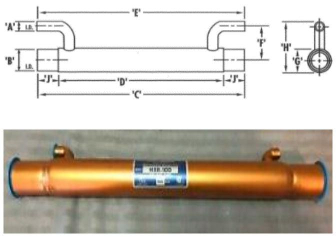


Fig 1. The experimental apparatus's schematic diagram

To precool the discharge line between the compressor and the air condenser, a heat exchanger between the discharge refrigerant line at the condenser's inlet and cold water is used. These heat exchangers are suitable for all HC, HFO, HFC, HCFC, and CFC refrigerants. For the present experimental study, the HXR-100 refrigerant heat exchanger is used to transfer heat from the liquid refrigerant line to the cold water. Figure 3 displays the heat exchanger specifications.



Model	HXR-100
Capacity	2 hp
Outer Shell Material	Copper
Inner Tube Material	Red Brass
Connection Material	Copper
Dimension A	3/8 in (ID)
Dimension B	1 1/8 in (ID)
Dimension C (Length)	14 1/8 in
Dimension D	11 in
Dimension E	12 3/4 in
Dimension F	1 7/8 in
Dimension G (Width)	1 3/8 in
Dimension H (Height)	2 13/16 in
Dimension J	1 9/16 in
Maximum Working Pressure	450 psig
Burst Pressure	2250 psig
Paint	None

Fig 2. The refrigerant heat exchanger's schematic diagram and property table

2.2. Operating Parameter Ranges:

The experiments were carried out over a 10-hour period (8 a.m. to 6 p.m.), with measurements taken every 10 minutes when the operation reached a steady state. The range of the temperature for the condenser's surrounding air was between 30 – 45°C. The temperature range of the inlet water that flows into the heat exchanger was 15 – 30°C, and the flowrate range of the water that flows inside the heat exchanger was 9 – 15.8 L/min. The temperature and pressure measurements are taken at steady state with the condenser's outlet air temperature set to 30°C, inlet water temperature set to 15°C, and water flowrate set to 9 L/min; then the same measurements are taken with other volume flowrates and inlet water temperatures at different ambient temperatures.

2.3. Reduction of data

Using the conventional and improvement vapor compression refrigeration cycle, the refrigerant mass flowrate can be calculated based on the various refrigerant properties and flow rates in split air conditioners for two cases, figure 3:

vapor compression refrigeration cycle without water cool loop [3]:

$$W_{in,without\ cool} = m_r(h_2 - h_1) = IxVxcos\phi \quad \ldots\ldots\ldots \ (1)$$

and vapor compression refrigeration cycle with water cool loop:

$$W_{in,with\ cool} = m_r(\acute{h}_2 - \acute{h}_1) = IyVycos\phi \quad \ldots\ldots\ldots\ldots \ (2)$$

Where the enthalpies of refrigerant at the compressor's inlet and outlet, without and with a cold - water heat exchanger, respectively, are represented by the letters h_1, h_2, $\acute{h}_1$, and $\acute{h}_2$ in the equation where the refrigerant mass flow rate is defined by the mr.

The rate of heat transfer in the air-cooled condenser can be expressed both without and with a cooling water loop [3].

$$Q_{c\ without\ water} = m_r(h_2 - h_3) \ \ldots\ldots\ldots \ (3)$$

$$Q_{c\ with\ water} = m_r(\acute{h}_2 - \acute{h}_3) \ \ldots\ldots\ldots\ldots \ (4)$$

Where h_2, h_3, h_2', and h_3' respectively are the refrigerant enthalpies at the condenser's inlet and outlet, both with and without a heat exchanger for refrigerant and water.

The heat transfer rate in shell and coiled tube heat exchangers can be calculated using the formula below [3]:

$$Q_{HE} = m_r(\dot{h}_2 - \ddot{h}_2) \quad \ldots\ldots\ (5)$$

Where the heat exchanger's inlet and outlet enthalpies are $\dot{h}_2$ and $\ddot{h}_2$, respectively.

As follows are the terms for the evaporator's cooling capacity [3]:

$$Q_{e\ without\ water} = m_r(h_1 - h_4) \quad \ldots\ldots \ (6)$$

$$Q_{e\ with\ water} = m_r(\acute{h}_1 - \acute{h}_4) \quad \ldots\ldots\ (7)$$

Where, respectively, h_4, h_1, h_4', and h_1' represent the refrigerant enthalpies at the inlet and outlet of the evaporator with and without a refrigerant heat exchanger.

This information can be used to calculate the performance coefficient with and without a heat exchanger [3].

$$COP_{\text{without water}} = \frac{Q_{e-\text{without water}}}{W_{in-\text{ without water}}} \quad ….(8)$$

$$COP_{\text{with water}} = \frac{Q_{e-\text{ with water}}}{W_{in-\text{ with water}}} \quad ….(9)$$

The exergy balance can be expressed as follows in general [17]:

$$Ex_{in} - Ex_{out} = Ex_{dest} \quad ….(10)$$

where Ex_{in} - Ex_{out} represents the net exergy transfer rate by heat, work, and mass which shows the net exergy destruction [17]. The same equation can be written as:

$$Ex_{heat} - Ex_{work} + Ex_{mass,in} - Ex_{mass,out} = Ex_{dest} \quad ….. (11)$$

The exergy rate is calculated as follows:

$$Ex = \dot{m}_r \Psi \quad ….. (12)$$

For each steady-flow device, the exergy efficiency or second law of thermodynamics efficiency is defined as [17]:

$$\eta = Ex_{out} / Ex_{in} \quad ….(13)$$

The exergy destructions for main vapor compression system are as follows [17]:

- **For compressor:**

$$\dot{m}_r\, ex_{in} + \dot{W}_{in} = \dot{m}_r ex_{out} + \dot{E}x_{dcom}$$

$$\dot{E}x_{dcom} = \dot{m}_r (ex_{in} - ex_{out}) + \dot{W}_{in}$$

$$\dot{E}x_{dcom} = \dot{m}_r (h_1 - h_2 - T_o (s_1 - s_2)) + \dot{m}_r (h_2 - h_1)$$

$$\dot{E}x_{dcom} = \dot{m}_r T_o (s_2 - s_1) \quad …… (14)$$

- **For condenser:**

$$\dot{m}_r\, ex_{in} = \dot{m}_r ex_{out} + \dot{Q}_c \left(1 - \frac{T_o}{T_c}\right) + \dot{E}x_{dcon}$$

$$\dot{E}x_{dcon} = \dot{m}_r (ex_{in} - ex_{out}) - \dot{Q}_c \left(1 - \frac{T_o}{T_c}\right)$$

$$\dot{E}x_{dcon} = \dot{m}_r (h_2 - h_3 - T_o (s_2 - s_3)) - \dot{Q}_c \left(1 - \frac{T_o}{T_c}\right) \quad …..(15)$$

- **For capillary tube:**

$$\dot{m}_r\, ex_{in} = \dot{m}_r ex_{out} + \dot{E}x_{dexp}$$

$$\dot{E}x_{dexp} = \dot{m}_r (ex_{in} - ex_{out})$$

$$\dot{E}x_{dexp} = \dot{m}_r (h_3 - h_4 - T_o (s_3 - s_4)) \quad …..(16)$$

- **For evaporator:**

$$\dot{m}_r\, ex_{in} + \dot{Q}_e \left(1 - \frac{T_o}{T_e}\right) = \dot{m}_r ex_{out} - \dot{E}x_{devp}$$

$$\dot{E}x_{devp} = \dot{m}_r (ex_{out} - ex_{in}) - \dot{Q}_e \left(1 - \frac{T_o}{T_e}\right)$$

$$\dot{E}x_{devp} = \dot{m}_r (h_1 - h_4 - T_o (s_1 - s_4)) - \dot{Q}_e \left(1 - \frac{T_o}{T_e}\right) \quad ………… (17)$$

As a result, the second law of exergy efficiency for refrigerators is the ratio of total output exergy to input exergy [17].

$$EX_{DT} = EX_{in} - EX_{out}$$

$$\eta_{exergy} = 1 - \frac{EX_{DT}}{W_C} \quad ……… (18)$$

The inlet and outlet exergies, respectively, are denoted by the symbols Ex_{in} and Ex_{out}. The exergy has been destroyed in the compressor, condenser, expansion valve, and evaporator are designated as Ex_{dcop}, Ex_{dcon}, Ex_{dexp} and Ex_{devp}, respectively. Q_c and Q_e represent the heat rejected by the condenser and evaporator, respectively, while m_r denotes the mass flow rate of the refrigerant.

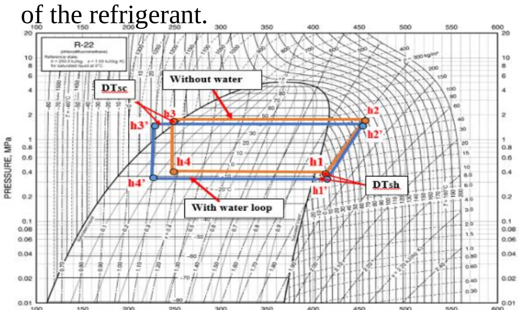


Fig 3. P-h diagram of refrigerant R22 with and without cooling water loop at vol= 11 L/min, Twi= 19°C and Tamb= 30 °C

2.4. Uncertainty analysis:

The uncertainty analysis in the current work can be calculated, based on instrument errors, measurement variance, and calibration errors[3]. This method is founded on the precise characteristics of the uncertainties in the preliminary experimental measurements. The result is the (COP), which is a function of the independent variables h_1, h_2, h_3, and h_4, which are determined by temperature and pressure measurements. The measured variables' uncertainty values are shown in Table 2.

No	Instrument,	Accuracy (%),	Uncertainly,
1	Thermometer, °C	0.1	± 0.1
2	Pressure Gages, pa	0.020	± 0.2
3	Ammeter, Amp	0.1	± 0.2
4	Rotameter, L/min	0.1	± 0.1

Table 2: Measurement Accuracy and Uncertainty

Uncertainly of COP

$$= \sqrt{\left(\frac{\partial COP}{\partial h_1}\Delta h_1\right)^2 + \left(\frac{\partial COP}{\partial h_2}\Delta h_2\right)^2 + \left(\frac{\partial COP}{\partial h_3}\Delta h_3\right)^2 + \left(\frac{\partial COP}{\partial h_4}\Delta h_4\right)^2}$$

The total uncertainty value of COP is about ± 4%

2.4. Validation of the present work

The validation of the present work is obtained when the COP of the vapor compression refrigeration cycle is compared with those of the previous experimental work for the case when cooling water loop is used [18]. Figure 4. displayed the comparison between the two cases. The trends are similar for two cases, but the agreements between two cases are fair with a maximum percentage error about 13 %. This error is due to the diffrerence in the refrigeration capacities of the ACU.

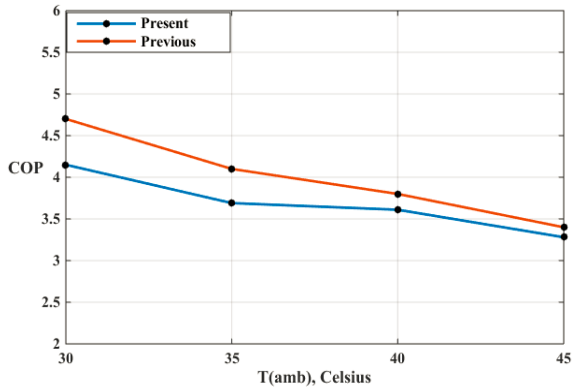


Figure 4. Comparison between the present work and previous work [18]

3. RESULT AND DESCUSION:

The purpose of this study is to conduct an experimental investigation due to the effect of the cooling water loop in the condenser on the exergy performance of a split air conditioning unit, using a range of ambient temperatures, inlet water temperature and water volume flowrate values as compared to those of a typical split air conditioner. The split unit's condenser was put to the test at ambient temperatures of 30, 35, 40, and 45°C without and with a water loop, for various inlet water temperatures which enter the heat exchanger at temperatures of 15, 19, 24, and 30 °C. Additionally, various flow rates of water of 9, 11, 14 and 15.8 L/min were used. Figure 3 illustrates the difference between traditional and cooling water loop split air conditioning. The P-h diagram represents a simple vapor compression refrigeration cycle (VCRC). Figure 3 displays with cooling water loops (1', 2', 3', and 4') and without cooling water loops (1, 2, 3, and 4). When compared to conventional ACUs, it should be noted that ACUs with water loops, the compressor power consumption, increased COP, and reduced evaporator cooling capacity (no water loop). The degree of superheating and sub-cooling is increased.

The effects of ambient temperature on evaporator exergy destructions are shown in figure 5, with various water flowrates. Figure 5 illustrates how, in the case of no water, the evaporator's exergy destruction rises with a small increase in ambient temperature. When the water heat exchanger is utilized in all cases, the evaporator's exergy destructions are reduced. For the cases Twi = 15, 19, and 24 °C, there is a large gap between the exergy destruction of the no water case and when a water heat exchanger is used, and these differences become low at Twi = 30 °C. At Twi = 15 °C and Tamb = 30 °C for a flowrate of 11 L/min with a percentage of approximately 37%, the maximum difference in the evaporator's exergy destruction is indicated.

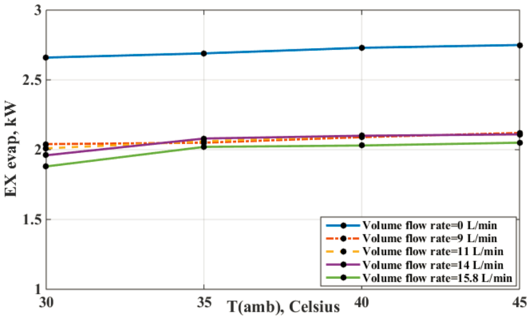


Fig 5. Effect of ambient temperature on evaporator exergy destruction with different volume flowrate at inlet water temperature Twi=19 °C

The relationship between ambient temperature and evaporator irreversibility is shown in figure 6, with the variations in the temperatures and flowrates of the inlet water. Due to high inlet water temperature, the irreversibility values reduce for Twi = 30 °C which have minimum values for all flowrates. With a smaller difference between these values, the reduction increases for other Twi values. For all flowrates, the maximum reduction in the irreversibility values is indicated for Twi = 15 °C.

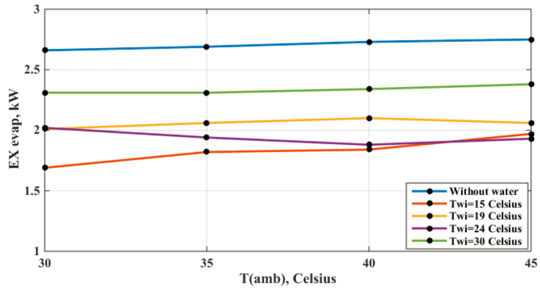


Fig 6. Effect of ambient temperature on evaporator exergy destruction with different inlet water temperature at volume flowrate Vol=11 L/min

Figures 7 shows the effects of ambient temperature variation on the exergy destruction of the expansion valve for different volumetric flowrates and inlet water temperatures. The results show that, in the case of no water heat exchanger, the irreversibility in the expansion valve increases as the ambient temperature increases. When a water heat exchanger is used, the exergy destruction of the expansion valve for Twi ranges of 15 to 24°C decreases more than for Twi = 30°C. All flowrates have reduced irreversibility values that are closer to one another.

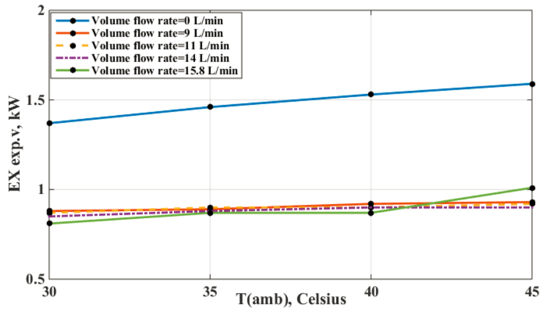


Fig 7. Effect of ambient temperature on expansion valve exergy destruction with different volume flowrate at inlet water temperature Twi=19 °C

The exergy destructions of the expansion valve for different inlet water temperatures and different flowrates shown in figure 8 are due to effect of ambient temperature. When using a water heat exchanger for flowrates of 9 and 11 L/min, the results show that the Twi has an effect on the irreversibility values. The difference in the irreversibility of the expansion valve for various Twi increases with flowrate.

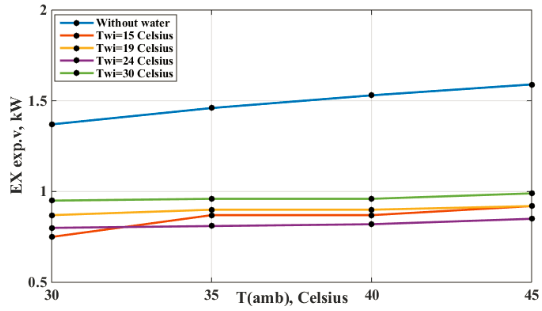


Fig 8. Effect of ambient temperature on expansion valve exergy destruction with different inlet water temperature at volume flowrate Vol= 11 L/min

The effect of ambient temperatures on condenser exergy destructions for various water flowrates and inlet water temperatures is shown in figure 9. With an increase in Tamb, the irreversibility of the condenser decreases until it reaches its lowest point at Tamb = 45°C. The exergy destruction increases with ambient temperature as a water heat exchanger is used. Additionally, for all water flowrates, the condenser irreversibility is higher than it is in the case with no water.

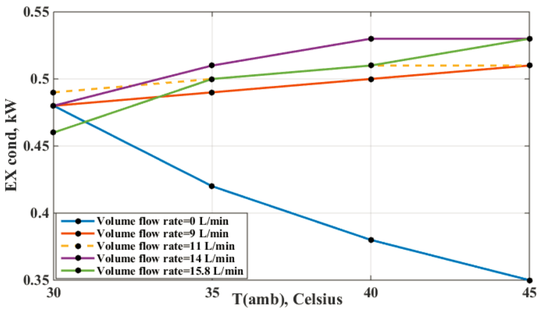


Fig 9. Effect of ambient temperature on condenser exergy destruction with different volume flowrate at inlet water temperature Twi=19 °C

Figure 10 shows how different inlet water temperatures and water flowrates affect the condenser's irreversibility as a function of ambient temperature. The condenser irreversibility at Twi = 30 °C is higher than that of other water temperatures for flowrates of 9, 11, and 14 L/min. The variation in energy destructions decreases with flowrate.

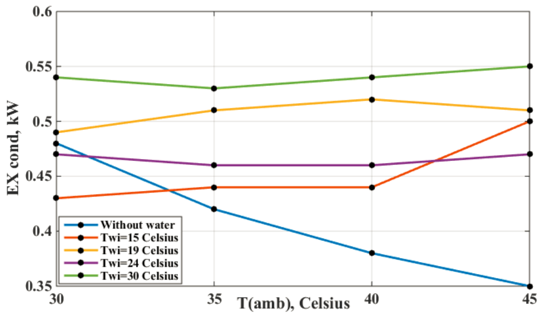


Fig 10 Effect of ambient temperature on condenser exergy with different inlet water temperature at volume flowrate Vol = 11 L/min

Figure 11 shows the impact of ambient temperature on compressor exergy destruction at various volume flowrates of 9, 11, 14, and 15.8 L/min and inlet water temperatures of 15 to 30°C. The compressor irreversibility significantly increases as ambient temperature rises for the no water case. When a cooling water loop is in use, the exergy destruction slightly increases as the ambient temperature increases. Furthermore, in comparison to those of the conventional ACU, the compressor exergy destruction is reduced by 12 to 70%. For all flowrate values, Twi causes a convergence of the exergy destruction of the compressor and the reduction in the enhancement percentage of using a water heat exchanger.

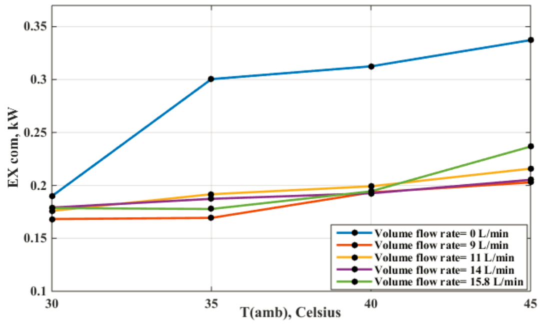


Fig 11. Effect of ambient temperature on compressor exergy destruction with different volume flowrate at inlet water temperatures Twi=19 °C

In Figure 12, the exergy destruction of the compressor is shown in relation to the variation in inlet water temperature and water flowrates under the influence of ambient temperature variation. When a water heat exchanger is used in conjunction with an ACU as opposed to a conventional ACU, the compressor irreversibility is reduced. When using a cooling water loop, the compressor's exergy destruction is typically reduced to a maximum of about 70% less than when using no water, which is typically achieved at Tamb = 45°C.

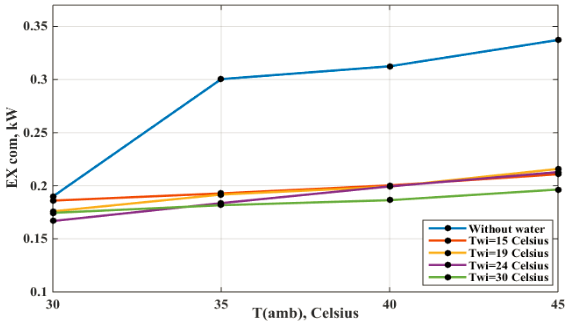


Fig 12. Effect of ambient temperature on compressor exergy destruction with different inlet water temperature at volume flowrate Vol= 11 L/min

Figure 13 shows the effect of ambient temperature on exergy efficiency at various volume flowrates of 9, 11, 14, and 15.8 L/min and with inlet water temperatures ranging from 15 to 30°C. In all cases, the exergy efficiency decreases as ambient temperature increases. As the flowrate increases for Twi = 15°C, the exergy efficiency increases in comparison to values for the no water case. With a percentage of approximately 13%, flowrates of 14 and 15.8 L/min result in the greatest increase in exergy efficiency. As Twi increases, the use of water heat exchangers becomes less prevalent and the exergy efficiency for all flowrates approaches unity.

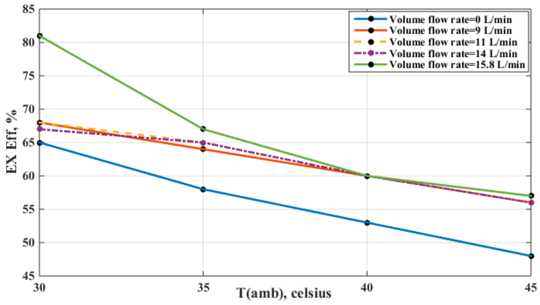


Fig 13. Effect of ambient temperature on exergy compressor efficiency with different volume flowrate at inlet water temperature Twi=19 °C

Figure 14 shows the exergy efficiency under the influence of ambient temperature variation with the variation of inlet water temperature and water flowrates. The exergy efficiency decrease as the Tamb increases for constant Twi or without water. The efficiency can decrease by a maximum of about 23%. When using a water heat exchanger versus the no water case, the exergy efficiency increases for all flowrates. The most of the time, Twi = 19 °C is where the exergy efficiency reaches a maximum. The maximum value at 15.8 l/min of water flow, Tamb = 30 °C, and Twi = 19 °C.

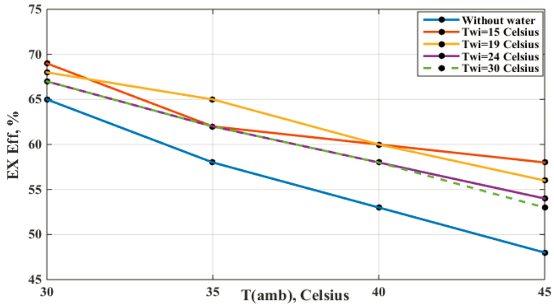


Fig 14. Effect of ambient temperature on compressor efficiency with different inlet water temperature at volume flowrate Vol= 11 L/min

4. CONCLUSION

The current study is an experimental investigation of the performance of an air conditioning unit (ACU) due to the precooling of the condenser with a cooling water loop and a water heat exchanger. The performance is based on exergy analyses at different inlet water temperatures of (15, 19, 24, and 30°C) and water flowrates of (9, 11, 14, and 15.8 L/min) under a range of ambient temperatures of (30, 35, 40, and 45°C). The experimental test was conducted at the Zakho Technical Institute in Zakho city. The following conclusions are drawn:

1. The results indicate that the relation between the energy and exergy results.
2. The results show the impact of the water-cooling loop on the exergy destructions for the Vapor compression refrigeration system components and the exergy efficiency.
3. For no water case, the exergy destruction of the evaporator and expansion valve increases slightly with increasing ambient temperature, whereas it decreases dramatically with increasing ambient temperature for condenser.
4. As the ambient temperature increases, with and without usingcooling water loop, the irreversibilities of the main components of the VCRS increase.
5. When cooling water loop is used, the exergy destructions of the evaporator, expansion valve and compressor reduce by using the cooling water loop, while the condenser exergy destruction increases with the use of the water loop.
6. The variation of the inlet water temperature and water flowrate have an impact on the exergy destructions of the VCRS components.
7. The exergy efficiency decreases as the ambient temperature increases in all cases, with and without cooling water. The cooling water enhances the efficiency of exergy with a percentage reach to about 13% as maximum value.

Nomenclature

Symbol	Meaning	Unit
$\dot{W}$	Compressor power	Watt
$\dot{m}$	mass flow rate	kg/s
$h_1, h2, h3, h4$	Enthalpy without water	kJ/kg
$\acute{h}_1, \acute{h}_2, \acute{h}_3, \acute{h}_4$	Enthalpy with water	kJ/kg
I	Power current	A
V	Power voltage	V
$\dot{Q}$	Heat rate	Kw

$\dot{h}_2$	Enthalpy at the inlet of the heat exchanger	kJ/kg
$\ddot{h}_2$	Enthalpy at the outlet of the heat exchanger	kJ/kg
COP	Coefficient Of Performance	
Ex	Exergy	kJ/s
T	Temperature	C°
s_1, s_2, s_3, s_4	Entropy	kJ/kg.k
P	Pressure	kPa
W	Power	Watt
VCRC	Vapor Compression Refrigeration Cycle	

Greek symbols

Symbol	Meaning	Unit
φ	Power factor	
η	Efficiency	%

Subscripts

Symbol	Meaning
amb	Ambient
in	Inlet
out	Outlet
r	Refrigerant
Com	Compressor
c	Condenser
e	Evaporator
DT	Temperature difference
HE	Heat exchanger

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تأثير دوران مياه التبريد على مكونات تدمير الطاقة لأنظمة تكييف الهواء المنفصلة

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الملخص

الدراسة الحالية عبارة عن تحقيق تجريبي لتحسين وحدة تكييف هواء منفصلة والتي تعتمد على دورة تثليج انضغاطي بخاري (VCRC) مع دوران مياه التبريد باستخدام المبادل الحراري للماء المبرد. يتم استخدام طريقة تحليل exergy لتحديد تدمير exergy للمكونات الرئيسية لـ VCRC وكفاءة exergy للدورة. تم فحص وحدة تبريد الهواء المحسّن في نطاق درجة الحرارة المحيطة بين 30-45 درجة مئوية ودرجات حرارة الماء الداخل عند 15 و 19 و 24 و 30 درجة مئوية ومعدلات تدفق المياه عند 9 و 11 و 14 و 15.8 لتر / دقيقة. أشارت النتائج إلى أن دوران مياه التبريد لها تأثير كبير على تدمير الطاقة مقارنة مع تلك الموجودة في حالة عدم وجود الماء. عند استخدام حلقة مياه التبريد ، يتم تقليل خسائر الضاغط والمبخر وصمام التمدد. تم تحسين كفاءة exergy لوحدة ACU باستخدام دوران مياه التبريد بالمقارنة مع تلك الخاصة بالوحدة التقليدية. يتم تحقيق أقصى قدر من التحسن في كفاءة الطاقة المفيدة في معدلات تدفق المياه العالية عندما يكون $Twi = 15$ درجة مئوية.

الكلمات الداله :

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