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EXPERIMENTAL STUDY AND MINIMIZATION OF (LMTD) OF A REFRIGATOR BY BALANCING COOLING WATER

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ABSTRACT

This work involves an experimental investigation of the refrigerator equipment. To complete of the study model, an inside and outside temperatures of refrigerant (R12) and cooling water through heat exchanger is assumed to be steady state flow. Special tables of refrigerant (R12) and cooling water are used to know properties at specified temperature , by using refrigeration system basically which work between two pressures, and specific enthalpy is one of the most useful properties which it is need in energy equation application to determine condenser and evaporator tube length of system , which takes as carefully to compression system design for giving temperature difference in cooling or heating systems. Also, tube length of condenser and evaporator are calculated (18 , 9) m respectively. A pressure - specific enthalpy or (p-h) chart of refrigerant R12 is depended on to determine specific enthalpy for each run at present pressure. This investigation is to study the influence of the volume flow rate of cooling water as range (2,4,6,and 8) liter /min which it supplies by control valve to the heat exchangers of system on logarithm mean temperature difference (LMTDe) and cooling coefficient of performance. Also, this work investigates the minimization of the (LMTDe) evaporator by using valve controlled of cooling water. The present results show that, the degree (LMTDe) is decrease as the volume flow rate of cooling water increasing then, due to increase of refrigerating effect at same time cooling coefficient of performance $(COP)_{RF}$ is increasing. Also, in the compression cooling system has been get temperature difference about(6)°C. Comparison between the obtained experimental performance results and the previous theoretical work shows a good agreement with (±3.8) percentage.

دراسة عملية لتقليل المعدل أللوغاريتمي لفرق درجات الحرارة لمنظومة تثليج بالتحكم بماء التبريد

الخلاصة:

في هذا البحث تم عمل دراسة عملية لجهاز تثليج، ولإتمام هذه الدراسة تم افتراض درجات الحرارة الداخلة والخارجة لوسيط التبريد (R12) وماء التبريد للمبادل الحراري كحالة مستقرة. تم استخدام جداول الخاصة بغاز التبريد(R12) و الماء لمعرفة الخصائص عند درجات حرارة معينة و باستخدام منظومة التثليج الأساسية التي تعمل بين ضغطيين ، والمحتوى الانثالبي النوعي هو احد الخواص المفيدة التي نحتاج تطبيقها في معادلة الطاقة لإيجاد طول أنبوب المكثف والمبخر للمنظومة ، التي تؤخذ بنظر الاعتبار في عملية تصميم منظومة انضغاطية نوفر منها فرق في درجات الحرارة سواء تبريد أو تدفئة. تم حساب طول أنبوب المكثف والمبخر (١٨، ٩) متر على التوالي . مخطط ضغط - محتوى انثالبي نوعي لوسيط التبريد (R12) تم اعتماده لاستخراج المحتوى الأنثالبي عند ضغط معين لكل قراءة. وتم دراسة تأثير معدل ألحجمي لماء التبريد بكميات مختلفة (8,٦,٤,٢) لتر\دقيقة من خلال صمام محكم مجهز لمبادلات الحرارية المنظومة على المعدل ألوغاريتمي للفرق بدرجة الحرارة للمبخر ومعامل أداء التبريد للمنظومة عمليا. كذلك ، في هذا الشغل درس تقليل المعدل ألوغاريتمي للفرق بدرجة الحرارة للمبخر بواسطة موازنة ماء التبريد.

وقد أظهرت النتائج العملية أن المعدل ألوغاريتمي للفرق بدرجات الحرارة للمبخر يقل مع زيادة معدل ألحجمي لماء التبريد وهذا يؤدي إلى زيادة حمل التبريد وفي نفس الوقت معامل أداء منظومة التبريد يتحسن مع نقصان المعدل ألوغاريتمي للفرق بدرجة الحرارة للمبخر. كذلك تم الحصول فرق بدرجات الحرارة لمنظومة التبريد الأنظغاطية حوالي (6 °C). وللتأكد من صلاحية النموذج العملي تم إجراء مقارنة بين كل من النتائج العملية الحالية ونتائج نظرية من أبحاث أخرى سابقة وكانت نتيجة هذه المقارنة مرضية بنسبة (±3.8%) .

Nomenclature:

COP- *Coefficient of Performance.*

$\dot{m}$ - *refrigerant mass flow rate (kg/sec).*

$\dot{V}$ - *volume flow rate (liter/min).*

Q - *heat exchanger capacity (kW).*

W - *compression power (kW).*

h - *specific enthalpy (kJ/kg) .*

LMTD - log mean temperature difference (K).

T - *temperature (K).*

ΔT- *difference temperature (K).*

P - *pressure (kpa).*

Subscripts:

r - *refrigerant.*

RF- refrigerator.

Hp –heat pump.

w - *water .*

eo - evaporator outlet.

ei - *evaporator inlet .*

co - condenser outlet.

ci - *condenser inlet .*

cpo-compressor outlet.

cpi - compressor inlet .

1- INTRODUCTION :

A heat pump is a machine or device that moves **heat** from one location (the 'source') to another location ('heat sink') mainly electrical power is used for the compression of the working fluid**.** Most heat pump technology moves heat from a low temperature heat source to a higher temperature heat sink.

One common type of heat pump works by exploiting the physical properties of an evaporating and **condensing** fluid known as a **refrigerant.** Due to the variations required in temperatures and pressures, many different refrigerants are available. Refrigerators and air conditioners are common applications that use this technology, Sheldon [1].

In order to control the amount of refrigerants that enters into the evaporator of a vapor compression refrigeration system, many expansion devices could be employed. Some of these are: thermostatic expansion valves (TEVs), solenoid valves, high-or low-side float valves, capillary tubes, or discharge bypass. In the use of any form of expansion device it has been noticed that evaporators exhibit an undesirable behavior known as "hunting" under certain operating conditions, Mithraratne and Wijeysundera [2]. The systems variables such as the refrigerant flow rate, evaporator pressure, and superheat temperature oscillate in a sustained manner when hunting takes place.

Many researchers had worked on this same effect using capillary tube and other expansion devices with evaporators. These can be found in the works of Meyer and Dunn [3], Motta *et al.* [4], Kim *et al.* [5], Wolf and Pate [6], and Wijaya [7]. The results of their theoretical and experimental investigations were similar to those of TEV-controlled evaporators.

As pointed out by Ibrahim [8], with zero superheat, the evaporator may be flooded and liquid refrigerant might pass to the compressor if no protection is used and may result in compressor damage. Hence, a certain degree of superheat between the evaporator and the compressor may be beneficial, but this actually leads to the instability of the evaporator performance. Such instability results in unstable cooling capacity and fluctuation of evaporator temperature for a certain period of time, which may have a direct effect on the refrigerated products.

Air source heat pumps are relatively easy (and inexpensive) to install and have therefore historically been the most widely used heat pump type. However, they suffer limitations due to their use of the outside air as a heat source or sink. The higher temperature differential during periods of extreme cold or heat leads to declining efficiency, as explained above. In mild weather, (COP) may be around (4.0), while at temperatures below around (−8°C) an air-source heat pump can achieve a (COP) of (2.5) or better, which is considerably more than the (COP) that may be achieved by conventional heating systems. The average (COP) over seasonal variation is typically (2.5-2.8), sheng [9] .

The air-conditioning and commercial refrigeration industries also rely heavily on both (R-11 and R-12) as working fluids, Suzai [10]. The next section disc the success of achieving these new goals with some existing alterative refrigerants.

(R-134a) is becoming widely accepted as the replacement for (R-12) in domestic refrigerator/freezer and automotive air conditioning applications. The refrigeration capacity and coefficient of performance (COP) of alternative refrigerants must also be established. Numerous investigations have been conducted to determine the capacity and performance of alternatives relative to their (CFC) counter parts. In a test conducted for a household refrigerator/freezer**,** (R-134a) was shown to consume approximately (8%) more power than (R-12) and require more run-time, resulting in an energy consumption (45%) greater than (R-12),[10,11].

The above factors have been mentioned in a number of studies and many equations predicting the actual behavior presented the effects of the evaporator hunting minimization.

The present work involves an experimental study of the refrigerator equipment . Thus, the object of this investigation is to study the influence of the volume flow rate of water which it supplies by control valve to the heat exchangers of refrigerator on the logarithm mean temperature difference (LMTDe) and cooling coefficient of performance . Also, this work investigates the minimization of the (LMTDe) evaporator by using controlled valve of cooling water.

2- EXPERIMENTAL APPARATUS:

Fig.(1) shows the experimental apparatus, the main components of the refrigerator and its instrumentation were mounted on a wood signboard. The refrigerator consisted of a piston compressor was rated at (0.25 kW) , an insulated coiled concentric tube water cooled condenser , a liquid receiver , a thermostatically controlled expansion valve by using capillary tube and water heated evaporator and high and low pressure gauges. The components were clearly but compactly arranged in a manner similar to that used for many domestic air-water heat pumps and all were visible from the front of the unit .

The flow rates of water were measured by flow rate meter. Water and refrigerant(R-12) temperatures were measured by copper constantan thermocouples. Before each test, the system was kept working for about half hour, to reach steady state condition. At each test run, the flow rates of water was varied by using control valves. Then, gauges readings of the instruments were recorded. The data recorded included temperature, pressure, flow rate and power consumption. All the required properties were evaluated from those parameters.

The experimental of a simple refrigeration unit, shown in Fig.1. In this study, it is taking a compressor power about a (0.25 kW) cooling system with a double tube condenser. From design calculations of system , the evaporator is made up of (9m) copper tube each of length (0.98 m), arranged in a winding manner in a horizontal plane. The refrigerant flows in the inner tube of the evaporator, while the secondary fluid (water in this case) flows in the direction in the annulus.

The specification along with the uncertainties of the parameters are summarized in Table(1). Because refrigerants work in the liquid/vapor phases must use appropriate property charts or tables for saturated refrigerant (R-12). The geometric parameters of the tested coils are listed in Table(2).

3- THERMODYNAMIC MODEL:

The refrigeration cycle most commonly used in heating, ventilation, air conditioning, refrigeration industry systems are the basic vapor compression cycle, it is consisting of compressor, condenser, thermostatic expansion valve, and evaporator. This refrigeration cycle is shown in Fig.(2a).

In this theoretical vapor compression cycle, the refrigerant enters the compressor at state (1) at low pressure, low temperature, and saturated vapor state. From state (1) to (2), the refrigerant is compressed by the compressor and is discharged at state (2) at high pressure, high temperature, and superheated vapor condition. At state (2), it enters the condenser where it rejects heat to the cooling water.

It leaves the condenser at state (3) at high pressure and saturated liquid state. From state (3), the refrigerant enters the expansion valve where its pressure is reduced in a throttling process from high pressure (condenser pressure) to low pressure (evaporator pressure). After this it is at state (4) and enters the evaporator where it absorbs heat from the refrigerated space; and it leaves the evaporator at low pressure, low temperature, and saturated vapor state.

Figure (2b) shows a pressure-enthalpy diagram with the above described states for the theoretic simple vapor compression refrigeration cycle.

The following parameters: rate of heat absorption (Q_e), rate of heat rejection(Q_c), coefficient of performance (COP) and compression power (W_{cp}) were evaluated with the aid of the measured parameters, and equations (1) to (6) suggested by Jabardo, *et al.* [12]. By knowing two variables it was possible to obtain the value of the third unknown and for this purpose the pressure-enthalpy diagram was used . The experiment was made upon three assumptions, *Kilicarslan and Muller [13]*:

1-It was assumed that the refrigerant flow rate and the property values were steady, so steady flow energy equation was valid.

2- Reasonably it was assumed that changes in kinetic, potential energies were negligible and equal to zero.

3- Assumed that heat transfer in compressor was zero and negligible, no pressure drop throughout the cycle, ie the steady state energy equation is applicable;

$$Q_e = \dot{m}_R (h_{eo} - h_{ei}) \qquad (1)$$

$$Q_c = \dot{m}_R (h_{co} - h_{ci}) \qquad (2)$$

$$W_{cp} = (Q_c - Q_e) \qquad (3)$$

For operation as a refrigerator, a measure of system performance is the amount of heat absorbed per unit work supplied to drive the system. This is known as the Coefficient of Performance *(COP_{RF})* of the experimental plant was achieved by the following equations:

$$COP_{RF} = \frac{q_e}{W_{cp}} = \frac{h_{eo} - h_{ei}}{h_{cpo} - h_{cpi}} \qquad (4)$$

For operation as a heat pump**,** a measure of system performance is the amount of heat delivered per unit work supplied to drive the system. This is known as the Coefficient of Performance (COP_{HP}.)

$$COP_{HP} = \frac{q_c}{W_{cp}} = \frac{h_{co} - h_{ci}}{h_{cpo} - h_{cpi}} \qquad (5)$$

$$LMTD = \frac{\Delta T_i - \Delta T_o}{\ln \frac{\Delta T_o}{\Delta T_i}} \qquad (6)$$

4- RESULTS AND DISCUSSIONS:

The following parameters were measured during the experiment: inlet and outlet evaporator temperatures, inlet and outlet circulating water temperatures. In order to determine the above need to know the specific enthalpy values. Because refrigeration systems basically work between two pressures, and specific enthalpy is one of the most useful properties need, refrigerant thermodynamic properties are normally presented in the form of a **pressure - specific enthalpy** (or p-h) chart of R12. Before each test, the system was kept working for about half hour, to reach steady state condition.

As the degree of superheating increases, the heat flow of refrigerating effects and rejected heat of condenser are decreasing, as shown in Fig. (3). As the degree of superheat increases, the rate at which the refrigerant in the evaporator flashes into gas increases. This thereby increases the volume of gas in the evaporator since the condensing temperature is kept constant, thus resulting into a decrease in the volume flow rate , see Fig.(4).

Normally, if volume flow rate of water though evaporator increasing, then the refrigerating effect is increased, it increases almost linearly with the water volume flow rate. This means that more heat transfer area will be transferred for water to the evaporator for an effective rejection of the heat absorbed by the refrigerant. This is shown in Fig. (5).

Figure (6) depicts the effect of the volume flow rate of water though evaporator and condenser against the log mean temperature difference. It was observed that, as the water volume flow rate though evaporator and condenser increase, the degree (LMTDe) decreases, because of evaporator effect is increasing. But, the degree (LMTDc) increases, because of rejected heat of condenser is increasing.

Figures (7 and 8), where the cooling coefficient of performance (COP) and refrigerating effects are plotted against the degree (LMTDe), which showed to be any decrease in the degree (LMTDe) causes increase in refrigerating effects, also at same time cooling and heating coefficient of performance is raising.

Generally, this study compared with the works of other researchers (Wijaya [7], Ibrahim [8], Mithraratne and Wijeysundera[14], Wedekind[15]). This showed the effect of balancing the system components. Also, variation in the mode of heat transfer may result from the variation in the atmospheric condition (ambient temperature) and this may cause slight anomalies (Ibrahim [8]), as shown in Table (3). It is seen again that the present values of (*COP*) are in very good agreement with that obtained by different authors, such as Wijaya [7], Mithraratne and Wijeysundera[14].

Figure (9) shows the coefficient of performance comparison with other study of Wijaya [7]. This demonstrates that the present experimental study is a good agreement of a comparison for predicting the coefficient of performance with (±4) percentage error .

5- CONCLUSION:

From the above experiment, the thermodynamic cycle of vapor – compression refrigerator for the refrigerant (R-12) was studied . The main conclusions of the present study are summarized as following :

1- As the degree of superheating decreases, refrigerating effect increases, at the same time the volume flow rate is increasing .

2- The effect of the volume flow rate of water though evaporator against the log mean temperature difference. It was observed that, as the water volume flow rate though evaporator increases, the degree (LMTDe) decreases, then evaporator effect is increasing. Also, the cooling and heating coefficient of performance (COP) are increasing.

3-The compression cooling system design has been get temperature difference about (6) $^{\circ}C$.

4. Errors raised from instruments used in the experiment .The error bounds for temperature , pressures, and using the (P-h) chart, the effect of the above errors on the enthalpy values for the superheated vapor , as shown in table(1) . Therefore, the enthalpy values determined in the experiment were correct to within (± 2) enthalpy reading error . As well , human error in reading the measurements contributed to the other source of errors .

6- REFERENCES:

[1] Sheldon, Theodore L. " Tva's operational experiences with heat pump water heaters. " Proceedings of the 7th Heat Pump Technology Conference , pp5, 1984.

[2] Mithraratne, P. and Wijeysundera, N.E. " Dynamic simulation of a thermostatically- controlled counter-flow evaporator." Int. J. Refrig. 23, pp 74-89, 2002.

[3] Meyer, J.J. and Dunn, W.E. "New insight into the behaviour of a metastable region operating capillary tube." Int. J. Refrig., pp 10-15,1998.

[4] Motta, S.F.Y; Parise, J.A.R.; and Braga, S.L. "A visual study of r-404a/oil flow through adiabatic capillary tubes." Int. J. Refrig. 25, pp 86-96, 2002.

[5] Kim, S.G.; Kim, M.S. and Ro, S.T. "Experimental investigations of the performance of R22, R407 and R410A in several capillary tubes for air-conditioning." Int. J. Refrig. 25, pp 21-31, 2002.

[6] Wolf, D.A.; and Pate, M.B. "Capillary tube-suction line heat exchanger performance with alternative refrigerants." ASHRAE Trans. 97, pp 39-49, 2001.

[7] Wijaya, H. "Adiabatic capillary tube test data for HF-134a." Int. J. Refrig. 11, pp 63-71, 1992.

[8] Ibrahim, G.A. "Effect of sudden changes in evaporator external parameters on a refrigeration system with an evaporator controlled by a thermostatic expansion valve." Int. J. Refrig. 24, pp 66-76, 2001.

[9] Li lian-sheng " Development of refrigerating and air-conditioning compressor." Int. J. of refrigerating and air-conditioning technology ,pp31-37, 2005.

[10] Suzai T, Sato A, Tadano M, et al. "Development of a carbon dioxide compressor for refrigerators and air conditioners." . Conference of the Japan Society of Refrigerating and Air Conditioning Engineers, pp 43-54, 1999.

[11] Guo pei , "Review on the development of carbon dioxide heat pump water heater system and its compressor. "[J] . Science and Technology of Household Electric Appliance, pp42-44, 2005

[12] Jabardo, J.M.S.; Mamani, W.G.; and Ianalla, M.R. " Modeling and experimental evaluation of an automotive air-conditioning system with a variable capacity compressor." Int. J. Refrig. 25, pp 57-72, 2002.

[13] A. Kilicarslan and N.Muller, " COPs of R718 in Compression with other Modern Refrigerants " Proceedings of the First Cappadocia International Mechanical Engineering Symposium , Turkey, pp 317-323, 2004.

[14] Mithraratne, P. and Wijeysundera, N.E. "An experimental and numerical study of hunting in thermostatic-expansion-valve-controlled evaporator." Int. J. Refrig. 25, pp 45-51 , 2003.

[15] Wedekind, G.L. "An experimental investigation into the oscillatory motion of mixture-vapor transition point in horizontal evaporator flow." J. Heat Transfer 93, pp 47-54, 1997.

Table(1): Characteristics of instrumentation employed in the refrigerator test.

Parameter	Uncertainty	Full scale
Temperature ((K- type thermometers)	±0.1 ^{o}C	(-40 to 150)^{o}C
Pressure gauges	0.25 bar	(1_15) bar
Volume flow rate meter	±0.2%	(1_5) ltr/min
Enthalpy values for the superheated vapor	± 2 kJ.kg^{-1}	(200_400) kJ.kg^{-1}

Table(2): Coil geometric parameters.

Geometric parameters	Condenser		Evaporator
Tube diameter, mm		7.8	
Number of circuits	18		9
Tube material		Copper	
Inner tube surface		Smooth	
Refrigerating capacity(R12), kg		0.250	
Capacity cylinder(water tank), $m^3 \times 10^{-3}$		13.5	

Table 3: Coefficient of performance (COP) for the optimal solution for a (5/16") tube condenser and evaporator comparison for the refrigerator system of past researchers at condenser temperature 46°C.

	Theoretical Study Wijaya [7]	Experimental Study Ibrahim [8]	Experimental and Numerical Study Mithraratne and Wijeysundera [14]	Experimental Study Wedekind[15]	Experimental Present work
COP	4	3.28	4.12	4.86	4.32

Fig.1 (a): Picture of mechanical refrigerator

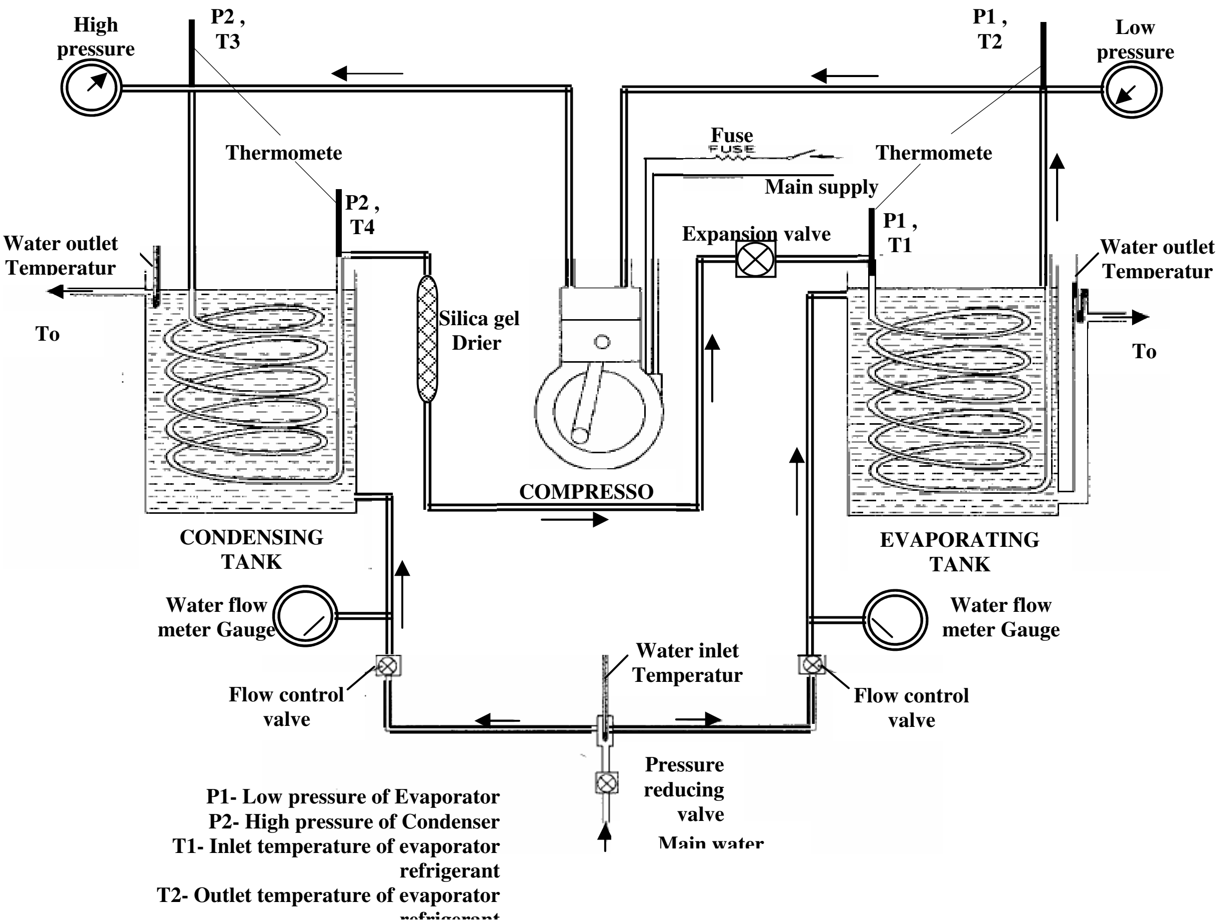

Fig.1(b): Schematic of the mechanical refrigerator apparatus

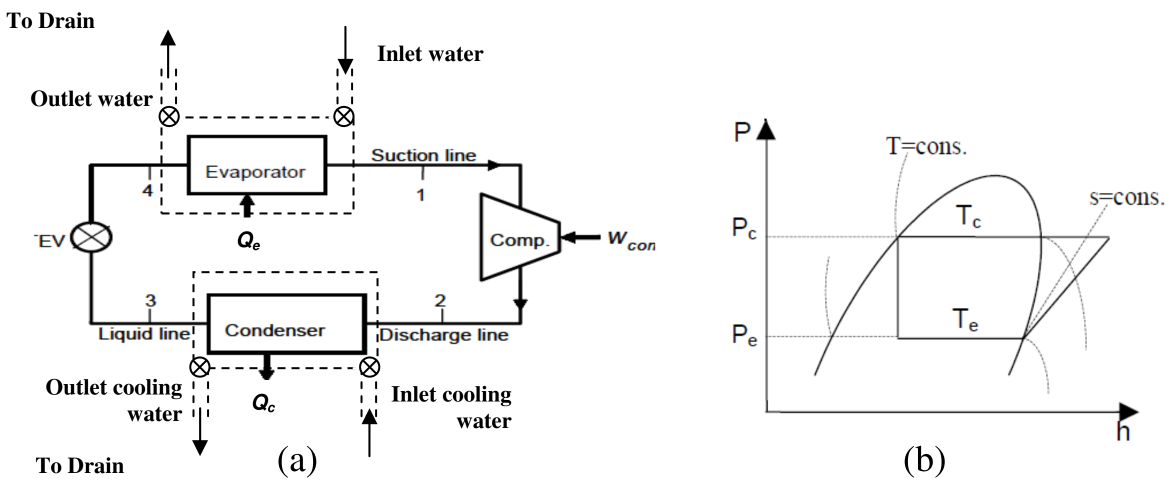

Fig.(2): (a) Schematic of one stage vapor compression refrigeration system; (b) P-h diagram of a simple vapor compression refrigeration cycle

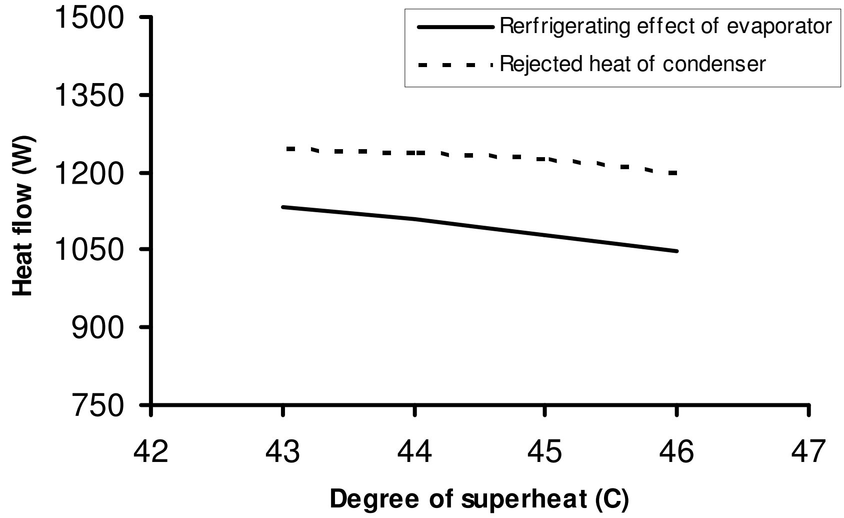

Fig. (3): Variation of heat flow with degree of superheat

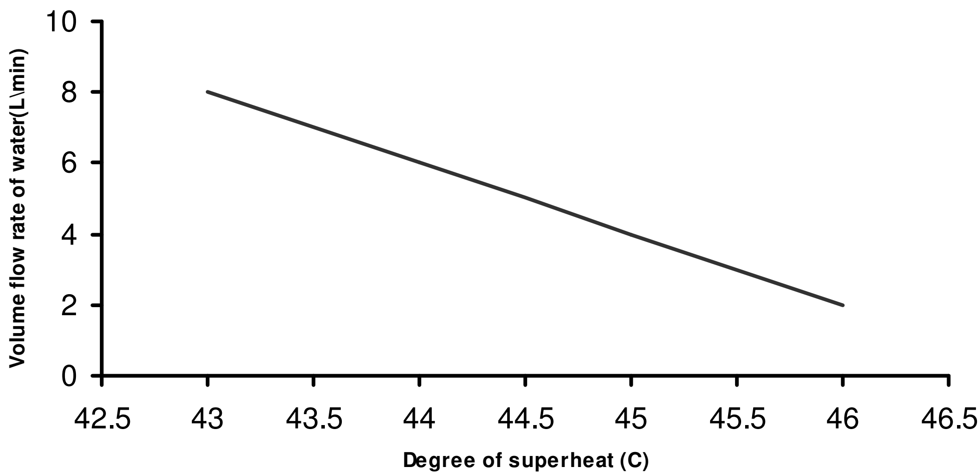

Fig.(4): Variation of volume flow rate of water with degree of superheat

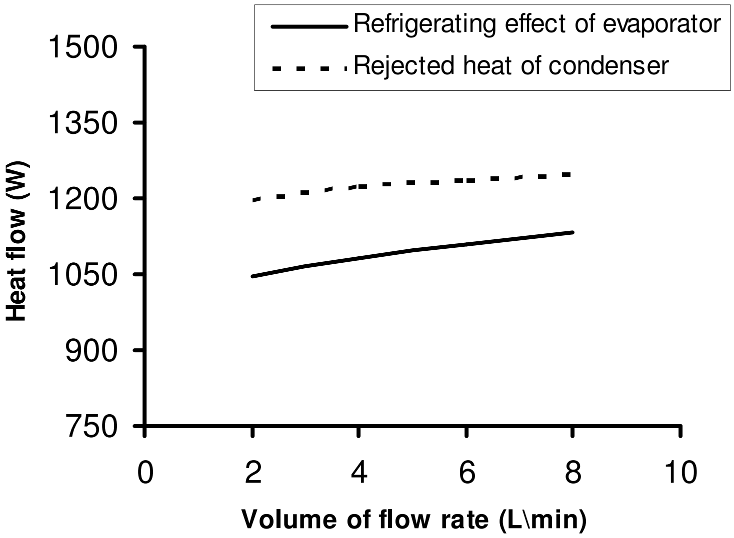

Fig. (5): Variation of volume flow rate of water with heat flow.

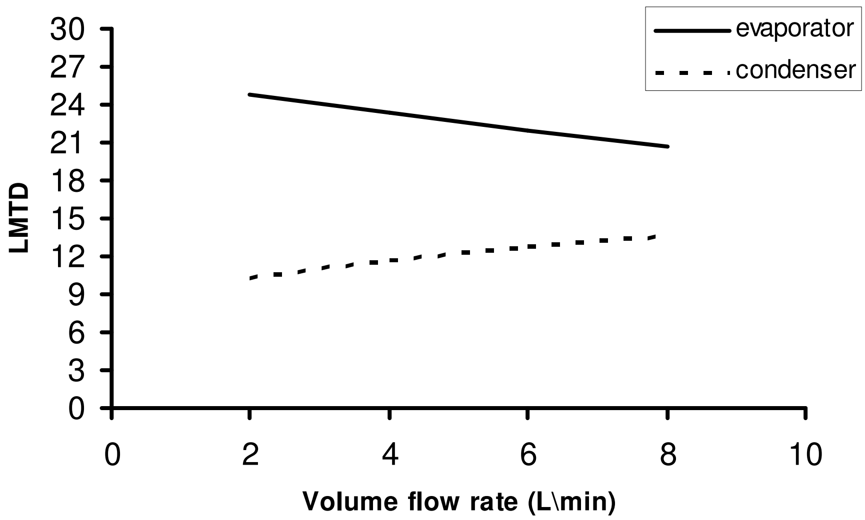

Fig.(6): Variation of volume flow rate of water with log mean temperature

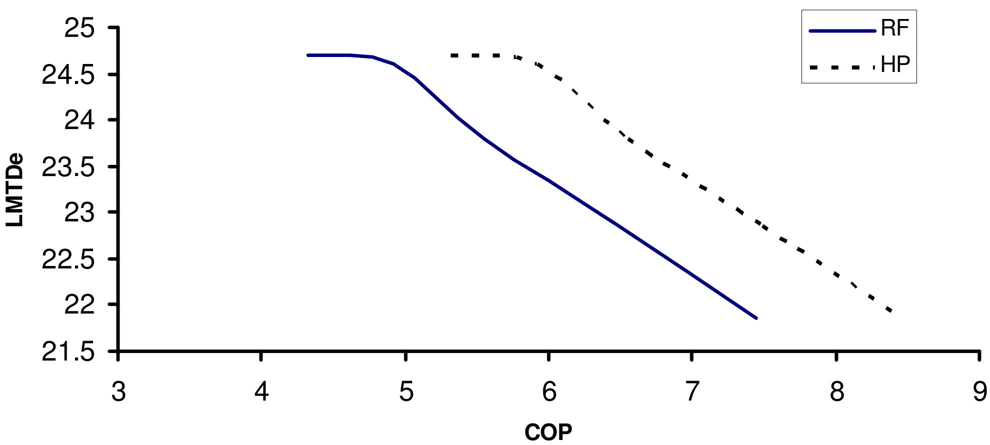

Fig.(7): Variation of Coefficient of performance with log mean temperature difference.

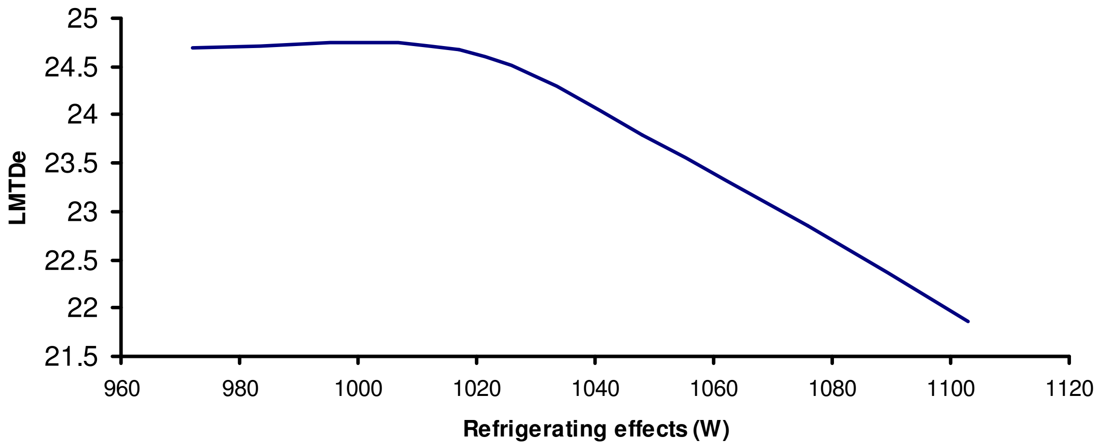

Fig.(8): Variation of refrigerating effects with logarithm mean temperature.

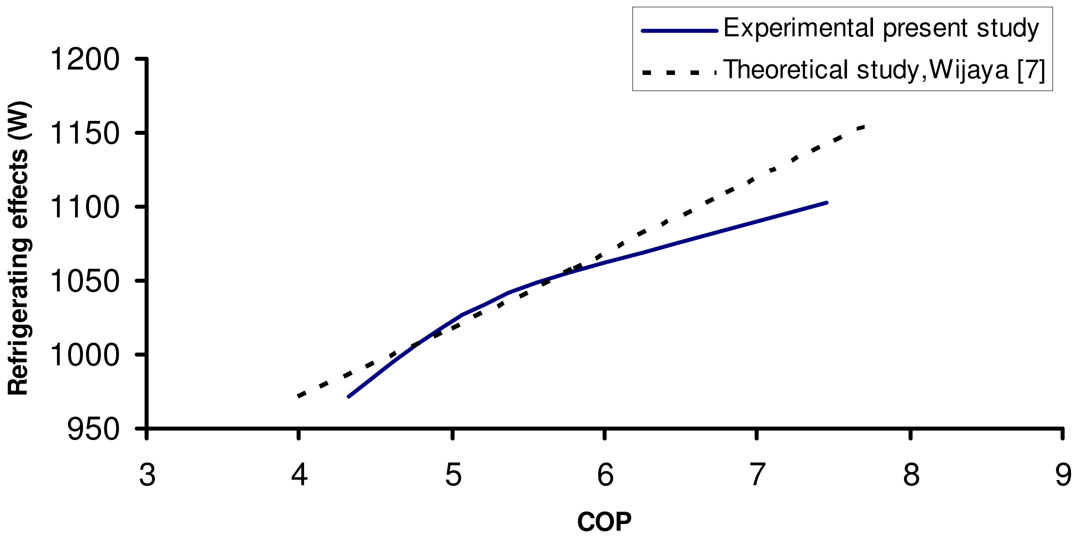

Fig.(9): The coefficient of performance calculated in this study versus that of previous study, Wijaya [7].