Experimental study and mathematical modeling of a vapor compression refrigeration system

Dr. Shurooq Talib Remedhan Al-Hemeri *

Abstract

Vapor compression refrigeration systems are regularly utilized to give a cool atmosphere to space thermal condition control purposes in commercial and industrial applications, especially in dehydration of gases, and application of refrigeration in the petroleum industry including lubricating-oil purification in addition to the separation of volatile hydrocarbons. Keeping in mind to outline good-performance that can be implemented for a wide range of operation, mathematical modeling which allows the simulation of its behavior for changing input conditions, necessary to be developed. In this work we changed the volumetric flow rate of secondary fluid (water) in the condenser in the range of (16-40) liter/hr, to study the experimental and mathematical analysis of condenser and evaporator of the cycle. Experimental analysis was conducted using a test rig for a vapor compression refrigeration system with R-134a as a refrigerant. The theoretical model is based on the mathematical formulation of the refrigerant side and water side in the condenser and air side of the evaporator, and the simulation program is based on the steady state mathematical model of vapor compression refrigeration cycle components solved using Mat-lab program that is used to solve the algebraic equations. The model requires input parameters of mass flow rate of the secondary fluid (water) to obtain the model prediction of the outlet temperature of the secondary fluids at the condenser and evaporator, the condenser and evaporator thermal capacities, the power consumed by the compressor and the coefficient of performance. The experimental validation of the model has been carried out using R-134a as a working fluid, concluding that the model can predict the performance of the cycle with an error lower than ±10%.

Keyword: Refrigeration, COP, heat exchanger, R134a, refrigerant.

I. Introduction

Refrigeration processes are important in a variety of applications; where the refrigeration is used in large scale industries, especially in the manufacture of ice, dehydration of gases, domestic, commercial refrigerators and other commercial industrial services. The applications of refrigerants in the oil industry include the separation of Light components, in the reactions required low temperature and for the purification of lubricating-oil. The refrigeration system design depends on the properties of the refrigerants because evaporation and condensation processes in refrigeration systems are achieved in consider a result of the energy transfer from refrigerants and change its phase from the vapor into the liquid [1].

Due to the amount of energy required by the refrigeration sector, which is estimated to be around 15% of total energy consumption in the world, the scientific community is working hard to optimize the operation of refrigeration plants. To do so, mathematical models constitute one of the best tools both for analyzing and for controlling systems.

There are two types of mathematical models slated to the behavior of vapor compression refrigeration system: empirical or statistical models and physical models. Empirical models are based on mathematical routines which obtain the formulation of the system from experimental data, but these were only valid for a particular system or specific conditions. The physical models were based on detailed data of parts in the system and their modeling using equation derived from the physics laws [3].

Vapor compression refrigeration systems, which are frequently utilized to give chilled media to space warm condition control purposes in domestic, commercial and industrial applications, mainly comprise a compressor, a fixed-orifice expansion device, two of heat exchangers such as (condenser and evaporator), and the light secondary fluid known as (refrigerant) which evaporates and condenses readily. And the one since for the system was closed to never the refrigerant leaves the system [4].

The ratio of the cooling heat capacity of the compression work called as a coefficient of performance for refrigeration system; therefore COP can be increased by increasing the refrigerating effect or by decreasing the compression work. For a vapor compression refrigeration cycle, it is important to study how much heat is extracted and how little energy is spent. The ratio of heat absorbed to the work input is called the coefficient of performance. The ratio should be as high as possible [5].

\[
\text{COP} = \frac{Q_c}{W} = \frac{Q_c}{Q_0 - Q_c} 
\]  

(1)

Theoretical COP is the ratio of (theoretical refrigerating effect) to (the theoretical compressor work or isentropic compressor work). Theoretical refrigerating effect, calculated from enthalpies about evaporator is found from the pressure-enthalpy diagram or the temperature-enthalpy diagram and isentropic compressor work is calculated from enthalpies about compressor was also got from the
There are a many literature which deals with the mathematical statements of the vapor compression refrigeration systems such as Arora and Kanshik [7] who presented the theoretical analysis and the exergy analysis in detail to the real vapor compression refrigeration cycle for highest condensing efficiency in a condenser. Viahav et al. [8] studied the property of the refrigerant that dependent a thermodynamic model for the simple vapor compression refrigeration system using various refrigerants that simulate the performance of actual system used for a higher efficiency. Gustavo Maruthi et al. [11] performed the experimental analysis of vapor compression refrigeration system by using R134a and R404a also which also evaluated the performance of the cycle. Mitesh et al. [12] predicated a theoretical performance of vapor compression refrigeration system and refrigerating effect with alternative hydrocarbon refrigerants such as R290, R600a, R1270 as well as their blend mixtures in various ratios. Kedarnath and Patil [13] studied the performance enhancement of vapor compression refrigeration system using nano-fluids which increased the area available for exchanging heat and efficiency.

The experimental study of any refrigeration framework generally is considered complicated, essentially because of the cost and the vast number of factors included. The utilization of numerical models can lessen the costs and furthermore encourage understanding the phenomena related to the issue. Refrigeration systems models are partitioned into two wide classes: steady –state models and dynamic models [14].

There is a large number of researchers who studied a steady state condition for refrigeration system type of vapor compression ,for example Koury et al. [15], Jong et al. [16] and Sivak and Hroncova [17] suggested a refrigeration system model related to parameters for condenser and evaporator and simulated the compressor and expansion device using the refrigerant R134a.

The main objective of this work is to study the experimental data and present the mathematical analysis for the two heat exchanger “COP” as a ratio of actual refrigeration effect to the actual work supplied to the compressor [6].

II. Description of physical system:

A simple vapor compression refrigeration cycle is shown in Figure 1 which comprises of the four fundamental parts; evaporator, compressor, condenser and expansion valve. The refrigerant is the primary working fluid in the cycle. In the evaporator, the refrigerant takes heat from the low-temperature primary fluid by vaporization. The refrigerant in the evaporator is superheated and vapor is then moved to compressor. To get on the highest level of energy done by adding the mechanical work. On the discharge side of the compressor, the vapor at higher pressure and temperature is condensed in the condenser.

In the condenser, the refrigerant vapor gives the heat to the secondary fluid and then it condenses. The condensation is done in horizontal tubes which involves total condensation of the vapor. Close to the outlet of the horizontal level tubes, the vapor lessens to zero and the flow in the tube becomes liquid stream.

The condensed refrigerant then passed through an expansion valve which is lowering the pressure and control of the mass-flow rate of the refrigerant. The refrigerant leaving the extension gadget enters the evaporator; in which the refrigerant absorbs heat and boils. The refrigerant at that point comes back to the compressor and the cycle is returned again. The ideal vapor compression refrigeration cycle is shown on the pressure – enthalpy temperature-entropy diagrams in Figures 2,3 also the refrigeration cycle of vapor compression at actual state is shown in Figure 4 [14,18,19].
Process 1 – 2: Isentropic compression in the compressor.
Process 2 – 3: Constant pressure heat rejection in the condenser.
Process 3 – 4: Isenthalpic expansion in the expansion device.
Process 4 – 1: Constant pressure heat absorption in the evaporator.

Fig. 2: P-h diagram of ideal vapor compression refrigeration cycle [19].

Fig. 3: T-S diagram for the ideal vapor compression refrigeration cycle [19].

Fig. 4: P-h diagram for the actual vapor compression refrigeration cycle [18].
Table 1: Accuracy for the experimental measurements.

<table>
<thead>
<tr>
<th>Independent variables</th>
<th>Uncertainty interval</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure gauge</td>
<td>± 0.2 % ( bar )</td>
</tr>
<tr>
<td>Temperature readers</td>
<td>± 1 ( °C )</td>
</tr>
<tr>
<td>Thermometer</td>
<td>± 1 ( °C )</td>
</tr>
<tr>
<td>Flow meter</td>
<td>± 0.014 % ( kg/sec )</td>
</tr>
</tbody>
</table>

III. Experimental work:

Experimental rig

Test rig of refrigeration system was used to perform the experimental tests as shown in Figure 5 which consist of the following components: compressor, condenser, capillary tube, evaporator, and accessories like : (Pressure gages, thermocouples and readers, flow meters and sight glasses).

Compressor: A single 370W capacity hermetic compressor (Panasonic model QB77C16GAX5) was selected in the current work. The compressor was designed to work at a speed 2990rpm and R134a was used as a refrigerant.

Condenser: The water condenser used in this cycle is wire and tube type with exchange surface area equal to 600 m².

Expansion Device: In this cycle the capillary tube consists of a single copper tube with (2800 mm) length and (1.5 mm) diameter is located before the evaporator to make a pressure drop and control on the mass flow rate of refrigerant that enters the evaporator.

Evaporator: In order to get variable load on the evaporator, a shell and coil evaporator has been used, where the flow of refrigerant in the tube and air in a shell with exchange surface area equal to 600 m².

Sight Glass: Sight glasses are used for viewing the phase of refrigerant through the refrigeration system, and to view the liquid refrigerant level through tubes and other vessels in the system. It is used also as refrigerant moisture indicator to ensure that the refrigerant contains less moisture.

Pressure Gauges: Pressure gauges are used to measure the pressures at different locations in the cycle. Pressure gauge with a range of (0 to 20) bars is used to measure the low side pressure, and the high side pressure.

Temperature Sensors: Temperatures at different points in the test rig were measured by thermocouples connected to the temperature readers. Temperatures were measured by K-type thermocouple operating in the range of (-200 to 1250 °C).

Flow Meters: Two flow meters were used in the present work, the first one was used for measure the volumetric flow rate of the liquid refrigerant in the refrigeration cycle model (10C- R134a), with a range of (0.03 – 0.3) L/min. The second flow meter was used to measure the volumetric flow rate of the inlet water to the condenser model (Z-3002-liquid water) with a range of (14 –40) L/hr.

Experimental procedure:

The main object of this work is to estimate and analyze the performance of the vapor compression refrigeration system with R134a as a refrigerant by using refrigeration rig that is shown in Figure 5.

1- The test rig system was turned on and operated without a thermal load in the evaporator for 5 min. to ensure a proper operation.
2- Set the float in the water volumetric rot meter to any flow in the range (16-40) L / hr.
3- Turn on the compressor and the fan.
4- Check the system has reached to steady state by selecting the temperature and pressure measurements every minute until no further change is observed should be (10 to 15 )min.
5- At steady state, take the readings of the refrigerant flow rate, four pressures and eight temperatures .This step was repeated for each thermal load applied to the evaporator.
6- For shutdown, stop the compressor, wait for 5 minutes, then close the water valve .Then switch off the main power supply and thermo-couple device (to save batteries).

IV. Thermodynamic model for vapor compression refrigeration performance:

The mathematical models of the parts for the refrigeration circuit including the compressor, heat exchangers, expansion valve at steady state was include it in simulation program. To simulate a refrigeration system, we ought to be joined the parts models into a general model as indicated by the relationship for ingredients parameters. Figure 6 is a simplified flow chart for the vapor compression refrigeration system with assuming the pressure losses in condenser and evaporator are ignored. Considering the operation of the cyclic system at steady-state using refrigerant R134a vapor outlet from evaporator at point 1 and compression through the compressor to point 2. The sub-cooled refrigerant exits the condenser at point 3 then flows through the expansion valve to the evaporator inlet at point 4. The analysis of the cycle in Figure 6 in terms of:

1) Cooling capacity
2) Condenser heat capacity
3) Compressor power consumption
4) Mass flow rate of refrigerant
5) Coefficient of performance
6) Efficiency of vapor compression refrigeration cycle
In the present work, developed assumptions to specify thermodynamic analysis in the refrigeration system:

- Pressure losses in the pipelines are neglected.
- Kinetic and potential energy are not considered.
- Heat transfer from the system or to the system is not considered.

By applying the first law of thermodynamics with the change in internal energy for a cyclic process is zero, we get [8]:

\[(Q_{\text{cond}} - Q_{\text{evap}}) - W = 0\]  \hspace{1cm} (2)

Where, \(Q_{\text{cond}}\) is the rate of heat rejection in the condenser (kW), \(Q_{\text{evap}}\) is the rate of heat absorbed by the evaporator (kW), \(W\) is the power input to the compressor (kW).

The heat absorbed through the evaporator and exothermic from the condenser occurs by the working fluid “refrigerant” streams with mass flow rates and specific heats. Therefore, the heat-transfer rate to the cycle in the heat exchanger (evaporator) i.e, cooling capacity becomes:

\[Q_{\text{evap}} = m_{\text{ref}} (h_4 - h_1) = (m \cdot C_p)_{\text{air}} (T_7 - T_8)\]  \hspace{1cm} (3)

Where, \(C_p\) is the heat capacity for the external fluid (air) (kJ/kg·K), \(T_7\) is the external fluid (air) inlet temperature (K), \(T_8\) is the external fluid (air) outlet temperature (K), \(m_{\text{ref}}\) is the mass flow rate of refrigerant (kg/s) and \(h\) is the specific enthalpy of the refrigerant at state point (kJ/kg).

As the same, the heat-transfer rate between the refrigeration cycle and the sink in the condenser i.e, condenser heat capacity is:

\[Q_{\text{cond}} = m_{\text{ref}} (h_2 - h_3) = (m \cdot C_p)_{w} (T_5 - T_6)\]  \hspace{1cm} (4)

Where, \(T_5\) is the external fluid (water) inlet temperature (K) and \(T_6\) is the external fluid (water) outlet temperature (K).

And the power required by the compressor at isentropic efficiency, is given by:

\[W = m_{\text{ref}} (h_1 - h_2)\]  \hspace{1cm} (5)

We then calculate the circulation rate of refrigerant \(m_{\text{ref}}\) which is determined from the rate of heat absorption in the evaporator by the equation:

\[m_{\text{ref}} = \frac{Q_{\text{evap}}}{h_1 - h_4}\]  \hspace{1cm} (6)

The COP is defined as the refrigerating effect over the net work input, i.e.

\[\text{COP}_{\text{actual}} = \frac{Q_{\text{evap}}}{W} = \frac{(h_1 - h_4)}{(h_2 - h_1)}\]  \hspace{1cm} (7)
Performance of coefficient for Carnot ideal cycle:

\[ \omega = \text{COP}_{\text{carnot}} = \frac{T_c}{T_h - T_c} \]  

(8)

Where, \( T_c = T_{1}^{\text{sat}} \) is the saturated temperature of refrigerant in (K) at evaporator pressure, and \( T_h = T_{3}^{\text{sat}} \) is the refrigerant saturated temperature in (K) at condenser pressure.

Efficiency of vapor compression cycle (\( \eta \)):

\[ \eta = \frac{\text{COP}_{\text{actual}}}{\omega} \]  

(9)

Requirements for calculation the thermodynamic properties of R134a refrigerant in simulation of the refrigeration cycle:

To analysis the refrigeration system, the following must be availability to calculate the thermodynamic properties [20-21]:

Temperature Symbols | Define
--- | ---
T1 | Refrigerant inlet compressor temperature
T2 | Refrigerant discharge compressor temperature
T3 | Refrigerant outlet condenser temperature
T4 | Refrigerant inlet evaporator temperature
T5 | Water inlet condenser temperature
T6 | Water outlet condenser temperature
T7 | Air inlet evaporator temperature
T8 | Air outlet evaporator temperature

Fig. 6: Schematic diagram of the vapor compression refrigeration system.
The quick calculation: Because occur many calculations that related to the simulation field of thermodynamic properties for refrigerant, therefore, the quick calculation of refrigerant thermodynamic properties consider an important factor for simulation, and it has an effect on the selection of component model in the system.

Higher stability: Since there are many attempts for the calculation of refrigerant thermodynamic properties, the dissimilarity in calculation has happened regardless of the possibility that likelihood is low in a solitary estimation, so the requirement on the stability must be achieved.

The reversibility: There are a large number of refrigerant thermodynamic properties in the simulation of refrigeration systems which need to be converted to each other. In spite of the deviation is almost very little in a solitary conversion process however will lead to a large difference in the last figured outcomes due to a numerous iterations that required.

The continuity and simplify: The result of convergence is achieved only when there is iteration for continuous functions. And because of some refrigerant thermodynamic properties are used the differential coefficients, therefore the differential coefficients must be also continuous and its function curve should be smooth.

Cleland [22] and Wang et al. [23], presented a quick method to calculate the refrigerant thermodynamic properties in addition they gave the correlations for R134a for the saturation temperature of -40 to 70°C. For a reversible calculation the formula of saturation pressure and temperature is presented in a simple and practical model, consumes less computation time. The estimations of polynomial curve-fit equations for thermodynamic properties of refrigerant R134a are easy, and computationally quick; for usually refrigeration conditions anticipated properties generally a large concur with the source information to about ±0.4%. For the saturated temperature of the saturated pressure relationship, Antoine equation was adopted:

\[ T_{sat} = \left( \frac{2200.980}{21.512 - \ln P_{sat}} \right) - 246.61 \]  

Where, temperature in (°C) and the pressure in(KPa)

At point (1) outlet from evaporator: if vapor superheat that is given by:

\[ \Delta T_v = T_{sup} - T_{1, sat} \]  

Where, \( T_{sup} = T_1 \) and \( T_{1, sat} \) is calculated from eq. (10) at low pressure "P_{evap}".

\[ h' = 249455 + 606.163 \]  

\[ T_{1, sat} - 1.05644 \]  

\[ T_{2, sat} - 1.82426 \times 10^2 T_{3, sat} \]

Then, the enthalpy of saturated vapor at \( T_{sat} = 0 \)(kJ/kg).

\[ h_1 = h' (1 + 3.48186 \times 10^{-3} \Delta T_{sat} + 1.6886 \times 10^{-6} \Delta T_{sat}^2 + 9.2642 \times 10^{-6} \Delta T_{sat}^3 - 0.698 \times 10^{-8} \Delta T_{sat}^4) T_{1, sat} + 1.7070 \times 10^{-7} \Delta T_{sat}^2 + 1.2130 \times 10^{-9} \Delta T_{sat}^3 \]

\[ h_1 = \text{enthalpy of superheated vapor at } T_{sat} > 0 \text{ that inlet to compressor (kJ/kg).} \]

At point (2) outlet from condenser: Where \( \Delta T = \) outlet from condenser:

\[ T_{cond} = \text{enthalpy of saturated liquid at } \Delta T \text{ and is enthalpy of saturated liquid at } \Delta T_{sat} = 0. \]

For a continuous an smooth, the differential coefficients must be also continuous and its function curve should be smooth.

At point (3) outlet from condenser: The extend of liquid sub-cooling found by:

\[ \Delta T_b = T_{3, sat} - T_3 \]

Where: \( T_3 = T_{3, sat} \) and \( T_{3, sat} \) is calculated from eq. (10) at high pressure "P_{cond}.”.

\[ h_1 = h_1 T_{3, sat} \]

\[ \Delta h_{act} = \frac{\Delta h}{\eta_{comp}} \]

\( \eta_{comp} \): actual efficiency of compressor and equal 80%

\[ h_2 = \Delta h_{act} + h_1 \]

From the previous equations, the procedure of flow chart for the saving simulation is shown in Figure 7.
Start

Input: \( m_{\text{water}} \)

Read: \( \{ P_h, P_{\text{cond}}, P_{\text{evap}}, T_1, T_2, T_3, T_4, T_5, T_6, T_7, T_8, m_{\text{ref}} \} \) measured from experimental work

Calculation at point 3:
- Calculate \( T_{\text{sat}}^3 \) from eq.(10)
- Calculate \( \Delta T_6 \) from eq.(11)
- Calculate \( h^3 \) from eq.(12):

Calculation at point 4:
\( h_4 = h^3 \)

Calculation at point 1:
- Calculate \( T_{\text{sat}}^1 \) from eq.(10)
- Calculate \( \Delta T_3 \) from eq.(14)
- Calculate \( h^* \) from eq.(15)
- Calculate \( h_1 \) from eq.(16)

Calculation at point 2:
- Calculate \( \bar{u}_v \) from eq.(17)
- Calculate \( V_{\text{sup}} \) from eq.(18)
- Calculate \( \Delta T_c \) from eq.(19), \( C_1 \) from eq.(20) and \( C \) from eq.(21)
  - Calculate \( \Delta h^- \) from eq.(22)
  - Calculate \( \Delta h_{\text{act}} \) from eq.(23)
  - Calculate \( h_2 \) from eq.(24)

Outputs:
- cooling capacity eq.(3)
- condenser heat capacity eq.(4)
- compressor power consumption eq.(5)
- COP eq.(7)
- \( \omega \) eq.(8)
- Efficiency of vapor compression refrigeration cycle eq.(9)

End

Fig. 7: Schematic diagram describing the overall model for simple vapor compression refrigeration system with R134a
V. Results and Discussion

In this area, thermodynamic performance parameters are computed from the experimental data using refrigerant R134a by utilizing the different equations (2-24) and discussed. The outcomes acquired from this research may vary slightly relying on the refrigerant charge, experimental and the environmental conditions. Performance parameters of the vapor compression refrigeration system, for example, genuine work of compressor, coefficient of performance, cooling capacity and condenser capacity were analyzed for various volumetric flow rate of water inlet to the condenser that using R134a. Many investigations are executed into performance analysis of the vapor compression refrigeration system. But a limited research is implemented for effect of volumetric or mass flow rate of the secondary fluid in condenser on the refrigeration system. Some researchers such as Dalkic and Wongwises [24] and Elsayed and Hariri [25] studied the effect of air mass flow rate in shell evaporator on the heat pump system. The numerical results of the obtained simulation are presented in a graphical form, as shown in Figures 8-14.

It was seen that for the same inlet temperature (T_{in,cond}) of water which the condenser secondary fluid, as the water mass flow rate (m_w) increases, the condenser heat capacity (Q_{cond}) increased due to increase water mass flow rate as seen from Figure 8. Figure 9 shows the COP (coefficient of performance) variation with respect to the condenser heat capacity. As the cooling capacity (Q_{evap}) was the constant. It was observed that as, the condenser heat capacity increases, coefficient of performance increased as studied by Maruthi et al. [11], Bhatkar et al. [18] and Saidur et al. [26]. Figure 10 and Figure 11 show the effect of variation of condenser heat capacity on system temperatures. T_{in,cond}(T_s), T_{in,cond}(T_s), T_{out,cond}(T_o) and T_{out,cond}(T_o) data are independent of refrigerant temperature in two heat exchangers. The temperature (T_s) gradient outlet from condenser increases with increases in condenser heat capacity as shown in Figure 10. But there is a slight change in temperature of air outlet from evaporator (T_e) with variation for condenser heat capacity as shown in Figure 11. Variation of condenser pressure (P_c) and evaporator pressure (P_e) with condenser heat capacity is shown in Figure 12. The condenser pressure decreases whereas evaporator pressure increases with increase in condenser heat capacity. The results show the same trend as obtained by Akintunde et al. [27]. Figure 13 pointed out that as the pressure drop between condenser and evaporator increases for same cooling capacity (Q_{evap}) of the system, the work of compression increases due to decreased condenser heat capacity (refer Figure 12). Figure 14 shows the COP variation with respect to the rate of heat transfer in (kW). As the compressor work consumption reduced thus COP increases. And when condenser heat capacity increases, COP increases too. But cooling capacity is nearly the same at obtained increased in COP as studied by Yeunyongkul et al. [28]. Therefore, in this case, when consider in (equation 7), it was found that the compressor work decreases, resulting in the increase in COP with remaining the cooling capacity nearly constant.

![Fig. 8: Variation of condenser heat capacity at different water mass flow rate](image)

![Fig. 9: Variation of coefficient of performance at different condenser heat capacity](image)
Fig. 10: Variation of water inlet & outlet temperatures about condenser vs. condenser heat capacity

Fig. 12: Variation of condenser pressure & evaporator pressure vs. condenser heat capacity

Fig. 11: Variation of air inlet & outlet temperatures about evaporator vs. condenser heat capacity

Fig. 13: Variation of compressor power consumption with pressure drop between condenser & evaporator
VI. Conclusions

This experimental study and the mathematical model investigated the effect of the changing the water volumetric flow rate (16-40 )L/hr or by the mass flow rate (15.8 to 39.6 )kg/hr.The working fluid in the refrigeration system is R134a. From this study the following conclusion is deduced:-

1) The present study is allocated to develop a physical model that permits clarification the energy conduct of a vapor compression system utilizing input parameters that can be easily acquired from experimental work.

2) Based on the simulation of the refrigeration system, it can be said that the refrigeration system performance is improved by bringing down the power consumption for compressor, increasing the rejection of condenser heat capacity or decreasing the pressure difference amongst condenser and evaporator.

3) The mass flow rate of water, which is the secondary fluid in condenser are need to be considered in selecting the condenser, since it effect on the condenser heat capacity for vapor compression refrigeration system.

4) Given the requirement for energy preservation, it is desirable to enhance the execution of vapor compression system presenting new parameter designs and additionally applying a proficient operation. In this way, a mass flow rate of water of the refrigeration system can be used at higher value of the range that used to improve the condenser heat capacity and coefficient of performance of the system.

5) A validation of the model was carried out by a comparison between experimental information and model yields. The outcomes were demonstrating a comparison between measured mass flow rate values of refrigerant R134a and those obtained with the circulation rate of refrigerant that determined from the rate of heat absorption in the evaporator. It is observed that a good fit with relative error estimations that differ within±10%.

Acknowledgments

Dr. Shurooq T. Al-Hemeri thankful to the department of chemical engineering /University of Technology for providing Laboratory of thermodynamic and Dr. Zaidoon Mohsen Shakor their help during this study.

References:

4) Suresh B. and Satyanarayana V., ”Improving and comparing the coefficient of performance of domestic refrigerator by using refrigerants R134a and R600a”, International journal of scientific research and management studies2014,8,pp.267-277.
6)Neeraj U., ”To study the effect of sub-cooling and diffuser on the coefficient of performance of vapor compression refrigeration system”, International
journal of research in aeronautical and mechanical engineering 2015,6,pp.14-44.
Sh.T.Al-Hemeri: Shurooq.