The Effect of the Capillary Tube Coil Number on the Refrigeration System Performance

Thamir K. Salim
Lecturer
Mechanical Engineering Department-University of Tikrit

Abstract

The capillary tube performance for (R134a) is experimentally investigated. The experimental setup is a real vapor compression refrigeration system. All properties of the refrigeration system are measured for various mass flow rate from (13 – 23 kg/hr) and capillary tube coil number (0-4) with fixed length (150 cm) and capillary diameter (2.5mm).

The results showed that the theoretical compression power increases by (65.8 %) as the condenser temperature increases by (2.71%), also the theoretical compression power decreases by (10.3 %) as the capillary tube coil number increases.

The study shows also that the cooling capacity increases by (65.3%) as the evaporator temperature increases by (8.4 %), and the cooling capacity increases by (1.6%)as the capillary tube coil number increases in the range (0-4).

The coefficient of performance decreases by (43.4 %), as the mass flow rate increases by (76.9%), also the coefficient of performance increases by (13.51 %) as the capillary tube coil number increases in the range (0-4).

Through this study, it was found that the best coil number in refrigeration cycle at the lowest mass flow rate (13 Kg/hr) and at high mass flow rate (23 Kg/hr) is (coil number = 4), this will give the highest performance, cooling capacity and lowest theoretical compression power.

An experimental relationship has been adopted between the coefficients of performance **(COP)** against $\mathbf{(-5.6032+\frac{e^{0.0413*nco.}}{0.0051*\dot{m}})}$.

Key Words: straight and coil Capillary tube, Refrigeration cycle Performance, R134a

تأثير عدد لفات الأنبوب الشعري على أداء منظومة التثليج الأنضغاطية

الخلاصة

تم دراسة أداء الأنبوب الشعري عمليا ل(R134a). إن معدات العمل التجريبية هي عبارة عن منظومة التثليج الحقيقية للانضغاط البخار. جميع خواص المنظومة تم قياسها لمعدلات تدفق مختلفة (13 – 23 kg/hr) وعدد لفاتِ الأنبوب الشعريِ (0–4) مع طولٍ ثابتٍ مقداره (150 سم) وقطر الأنبوب الشعري (2.5 ملم).

بينت النَتائجَ بأنّ قدرة الانضغاط النظرية تزداد بمقدار (65.8 %) مع زيادة درجاتَ حرارة المكثفَ بمقدار (2.71 %)، كذلك تقل قدرة الانضغاط النظرية بمقدار(10.3 %) مع زيادة عددَ لفات الأنبوب الشعريِ.

كما بينت نتائج الدراسةَ أيضاً بأنّ سعة التَبريد تزدادُ بمقدار (65.3 %) مع زيادة درجات حرارة المبخّرَ بمقدار (8.4 %)، وكذلك تَزِداد سعة التَبريد بمقدار(1.6 %)مع زيادة عدد لفات الأنبوب الشعريِ للمدى (0–4).

إن معامل الأداءِ يقل بمقدار(43.4 %)، مع زيادة معدل التدفق الكتلي بمقدار (76.9 %)، وكذلك يزدادُ معامل الأداء بمقدار (13.51 %) مع زيادة عددَ لفات الأنبوب الشعريِ للمدى (0–4).

خلال هذه الدراسةِ وُجِد أن أفضل عددِ لفات في دورةِ التبريدِ عند اقل معدل تدفق (13 kg /hr) و أعلى معدل تدفق (23 Kg /hr) هو (عدد اللفات = 4)، والتي أعطت أعلى أداء وسعة تبريد واقل قدرة انضغاط نظرية.من هذه الدراسة استنبطت علاقة تجريبية بين معامل الأداء (COP) يقابل $\left(-5.6032+\frac{e^{0.0413*nco.}}{0.0051*\dot{m}}\right)$

الكلمات الدالة: الأنبوب الشعري الحلزوني و المستقيم، أداء دورة التثليج،R134a

Nomenclature

A	cross section area	m^2
C	velocity	m/s
C_c	Compressor Clearance volume	-
C_L	Correction factor for leakage	-
d	Capillary tube internal diameter	m
D	Coil diameter	cm
E	Electrical power input	watt
fr	Friction factor	--
G	Mass flux	$kg.s^{-1}.m^{-2}$
h	Specific enthalpy	kJ/kg
I	current	A
L	Capillary tube length	m
$\dot{m}$	Refrigerant mass flow rate	kg/hr
n	Polytropic index	-
P	pressure	bar
Q	Volumetric flow rate	l/min
qe	Refrigerant effect	kJ/kg
Qe	Cooling capacity	watt
Qc	Rate of heat reject from condenser	watt
Rc	Comression ratio	_
Re	Reynolds number	-
T	temperature	^oC
TCP	Theoretical Compression Power	Watt
v	Specific volume	m^3/kg
V	volts	volt
Vcc	volumetric cooling capacity	kJ/m^3
We	Mechanical power output	watt
Wc	work of compression	kJ/kg
X	Dryness friction	-

Greek symbols

ρ	density	kg/m^3
μ	Viscosity	kg/m.s
η_e	Electrical efficiency	%
η_v	Volumetric efficiency	%

Subscripts

g	Gas or vapor	-
f	liquid	-
fg	Liquid vapor change	-
c	Condenser	-
e	Evaporator	-
nco.	Coil number	-

Introduction

Air conditioning and refrigeration systems play an important role in industry, infrastructure and households. The industrial sector includes the food industry, textiles, chemicals, printing, transport and others. Infrastructure includes banks, restaurants, schools, hotels and recreational facilities. Therefore, installation, repair and maintenance of equipment to function properly are important for the operations associated with those activities. At present reducing pressure valves used in air conditioning and refrigeration systems can be classified into two types: expansion valves and capillary tubes. The capillary tube is a copper pipe, with a diameter around 0.5 mm to 5 mm and length around 0.5 m to 5 m. Its usage depends on power and load capacity of the system. The capillary tube is often used with low cooling load and small changing load systems, such as refrigerators, water coolers and small air conditioners On the basis of geometrical shape the capillary tubes can be classified to straight and coiled capillary tube [1].

Kuehl&Goldsshmid[2], carried out an experimental studied on a set of capillary tube at different lengths and diameter account from (1.07-1.63mm) on refrigeration system using (R22). The work showed a practical relation between the inlet pressure and refrigerant mass flow rate at a certain degree of sub cooling temperature, and compute the friction factor at all diameters for wide range in Reynolds numbers. As follow its notice as increasing the capillary coils number causes an increase in the hamper by (5%) compared to straight capillary and due to decreases the refrigerant mass flow rate.

Bansal and Rupasinghe[3] developed an empirical model for sizing capillary tubes. The paper presented an empirical model that has been developed to size adiabatic and non-adiabatic capillary tubes for small vapour compression refrigeration systems, in particular, household refrigerators and freezers. The model is based on the assumption that the length of a capillary tube is dependent on five variables primary, namely the capillary tube inner diameter, the mass flow rate of the refrigerant in the capillary tube, the pressure difference between high side and low side, the refrigerant sub-cooling at capillary inlet and the relative roughness of the capillary tube material. The model is validated with previous studies over a range of operating conditions and is found to be agreement reasonably with the experimental data for RI34a.

the performance of capillary tubes for R-407C refrigerant has studied by Wei *et al*. [4]. In their study a total of nine capillary tubes were tested. The capillary tubes consisted straight and coiled configurations. Their results were compared with the correlations proposed by Bittle *et al*. and ASHRAE. The geometry of the capillary tubes that used: length (1 m), internal diameter (1.0 mm) and two coiled diameters of 52 and 130 mm. Comparing the flow rate of the coiled configuration with that of straight capillary tube, for the same inlet and out let pressures, tube diameter and length, the mass flow rate decreases with the decrease of the coiled diameter. The *decrease ratio* which was evaluated as $m_{coil}/m_{straigth}$ is relatively insensitive to the change of the inlet sub-cooling and the inlet pressure for both R-22 and R-407C.

Khan et al.[5], had also developed a numerical model flow through an adiabatic spiral capillary tube. An analytical model has been developed to predict the length of the adiabatic capillary tubes used in domestic refrigerators and low-capacity residential air conditioners. The model predicts the length of two types of tubes straight and spiral adiabatic capillary tubes. The proposed model is based on the homogenous two-phase flow model, which predicts the length of the adiabatic capillary tubes as a function of refrigerant mass flow rate, capillary tube diameter, degree of sub cooling at capillary inlet, internal surface roughness, and the pitch of the Archimedean spiral. The simulation results are validated with the experimental findings of previous researchers. The performance of the above two geometries of adiabatic capillary tube is compared, and it is established that for the same state of refrigerants at the inlet and exit of the adiabatic capillary, spiral capillary is found to have a shorter length. A much smaller capillary tube is required when the eco-friendly refrigerants R-134a and R 152a are used instead of refrigerant R-12 for similar conditions across the adiabatic spiral capillary tube.

Khan et al.[6], have presented a comprehensive review of the literature on the flow of various refrigerants through the capillary tubes of different geometries viz. straight and coiled and flow configurations viz. adiabatic and non-adiabatic. The literature has been presented in chronological order the experimental and numerical investigations systematically under different categories. Flow aspects like effect of coiling and effect of oil in the refrigerants on the mass flow rate through the capillary tube have been discussed. The paper also provides key information about the range of input parameters viz. tube diameter, tube length, surface roughness, coil pitch and coil diameter, inlet sub-cooling and condensing pressure or temperature. Other information including type of the used refrigerants, correlations proposed and methodology adopted in the analysis of flow through the capillary tubes of different geometries operating under adiabatic and non-adiabatic flow conditions. It has been found from the review of the literatures, anyway there are a lot more to investigate regarding the flow of various refrigerants through different capillary tube geometries.

Experimental Apparatus

Figure (1) is the schematic diagram of the experimental setup for the capillary tube test. The experimental setup is composed of compressor, condenser, evaporator, and the capillary tube as the vapor compression system is. The airtight compressor is the reciprocal type of (1/4) hp. A receiver of refrigerant is placed before the compressor in order to protect the compressor from the saturated liquid.

The air condenser with glass tube is a fin and tube type designed for this experiment. The thermal load of the condenser is controlled by the flow rate and temperature of the cooling air.

The air evaporator with glass tube is a fin and tube type. The thermal load of the evaporator is supplied from heated air and the capacity is from (0.7 to 1.2 kJ/s).A sight glass is installed before the refrigeration drier to see if the state at the inlet to the flow meter is liquid or not.

Several capillary tubes of different coil numbers are selected with uniform inner diameters and length. And this was measured using pin gauge. The specifications of the tubes are listed in Table (1).

A Roto-meter type effect volume flow meter is used to measure the volume flow rate from (0 to 0.35 L/min). In order to prevent vapor from flowing into the flow meter, it is controlled by valve and located between the condenser and the capillary tubes.

T-type thermocouples are used for temperature measurements. The thermocouples connected to the digital thermometer type (MTR6) with measuring range (-5 ^oC to 150 ^oC), it is located at the inlet and outlet tubes of the compressor, condenser, evaporator, and the capillary tubes. Particularly, several thermocouples are placed on the surface of the capillary tubes to measure the temperature variation along the tube. These thermocouples are installed on the outer surface of the tube in order to avoid the effect on the interruption of the refrigerant flow.

The pressure at the inlet capillary (condenser pressure) can be measured by using high pressure gauge or high pressure switches type (kp5) range by (0 to 35 bar), and the outlet capillary (evaporator pressure) is measured by using Low pressure gauge or Low pressure switches type (kp1) rang (-1 to 15 bar).

Experimental Procedures

The experimental procedures are:

1- After installing a new capillary tubes at different coil number from (0-4), the capillary tubes had been evacuated enough to remove any moisture and air.

2. Refrigerant (R-134a) is charged into the experimental system.

3. The auxiliary equipments such as constant temperature bath are run to get the required condition.

4. When the auxiliary equipments reach the required condition, then the refrigeration system is run.
5. The refrigerant system is run with its maximum efficiency until it reaches the steady state condition within (50 minute) time.
6. After reaching the steady state, temperature, pressure, mass flow rate, electrical power, the ampere and voltage current, are recorded, for different capillary tube coil numbers.
7. As the inlet condition of the capillary tube and its different (coils number), with uniform length and inner diameter. The above (1-6) procedures are repeated.
8. eighteen tests had been carried out for each of the following variables:-

- The coil number of capillary tubes has been varied within the range (0-4) and this was done three times.
- The refrigerant mass flow rate has been change by controller valve on the refrigerant amount pass through the capillary tubes to have different values. These values were (13,15,18,20,22 and 23 kg/hr).

The properties of refrigerant (R-134a) are computes depending on the change of the refrigerant temperature of the vapor compression system using the following equation [7]:

Saturation state

$$h_f = 199.7 + 1.4T + 0.0032T^2 - 6.8*10^{-5}T^3 + 7.1*10^{-7}T^4 \quad \text{.........(1)}$$

$$h_g = 399.3 + 0.6T - 0.004T^2 + 11.3*10^{-5}T^3 - 1.2*10^{-6}T^4 \quad \text{.........(2)}$$

$$\rho_f = 1296.6 - 3.5T - 0.025T^2 + 1.13*10^{-4}T^3 - 7.4*10^{-6}T^4 \quad \text{.........(3)}$$

$$v_g = 0.07 - 0.003T + 5*10^{-5}T^2 - 5.2*10^{-7}T^3 + 2.1*10^{-9}T^4 \quad \text{........(4)}$$

$$v_f = 7.8*10^{-4} + 2.6*10^{-6}T - 1.7*10^{-8}T^2 - 1.4*10^{-10}T^3 + 6.9*10^{-12}T^4 \quad \text{..........(5)}$$

$$\mu_g = (11.2 + 0.07T - 3.6*10^{-4}T^2 - 1.7*10^{-5}T^3 + 3*10^{-7}T^4)*10^{-6} \quad \text{......(6)}$$

$$\mu_f = (286.4 - 3.7T + 3.3*10^{-2}T^2 - 2.3*10^{-4}T^3 + 4.3*10^{-7}T^4)*10^{-6} \quad \text{..(7)}$$

Super- heated stat

$$h_1 = 400.9 + 0.52T + 0.0091T^2 - 7.4*10^{-5}T^3 + 2.2*10^{-7}T^4 \quad \text{........(8)}$$

$$v_1 = 0.05 + 2.8*10^{-4}T - 4.3*10^{-7}T^2 + 1.1*10^{-9}T^3 \quad \text{............(9)}$$

$$h_2 = 391.5 + 0.9123T + 0.00049T^2 \quad \text{..................(10)}$$

$$v_2 = 0.03 + 2.1*10^{-4}T - 3.2*10^{-7}T^2 + 6.2*10^{-10}T^3 \quad \text{..........(11)}$$

The refrigerant mass flow rate in (kg/hr) can be calculated by [8]:

$$\dot{m} = \rho_f * Q \quad \text{.......................(12)}$$

The refrigerant velocity is calculated by:-

$$C = \frac{\dot{m}}{A.\rho_f} \quad \text{............................(13)}$$

Where

$$A = \frac{\pi.d^2}{4}$$

Reynolds number (Re) is a quantity which the engineers use to estimate refrigerant flow is laminar or turbulent and can be calculated as follow [9]:

$$\mathrm{Re} = \frac{\dot{m}.d}{A.\mu} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(14)$$

For laminar flow the effect of friction factor is given by:

$$fr = 64/\mathrm{Re} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(15)$$

For turbulent flow $(\mathrm{Re} \le 10^5)$ the friction factor is given by:

$$fr = 0.32\,\mathrm{Re}^{-0.25} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(16)$$

The dryness fraction (x) can be determined depending on the quantity and energy equation:

$$(1000h_5 + \frac{v_5^2.G^2}{2} = 1000h_4 + \frac{C^2}{2}$$

$$1000h_{f5} + 1000(h_{g5} - h_{f5})$$

$$+ \frac{\left[v_{f5} + \left(v_{gf} - v_{f5}\right)X\right]^2}{2} * G^2$$

$$= 1000h_{f4} + \frac{C^2}{2}$$

$$X = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad \ldots\ldots\ldots\ldots\ldots(17)$$

Where

$$a = \frac{\left(v_{g5} - v_{f5}\right)G^2}{2}$$

$$b = 1000\left(h_{g5} - h_{f5}\right) + \left(v_{g5} - v_{f5}\right)v_{f5}.G^2$$

$$c = 1000\left(h_{f5} - h_{f4}\right) + \frac{v_{f5}^2.G^2}{2} - \frac{C^2}{2}$$

$$G = \frac{\dot{m}}{\mathbf{A}}$$

The enthalpy at capillary tube outlet can be calculated using the equation:

$$h = h_f\left(1 - X\right) + X.h_g$$
$$v = v_f\left(1 - X\right) + X.v_g \quad \ldots\ldots\ldots\ldots\ldots(18)$$

The conversion of the electrical energy to mechanical energy has losses due to wind age , friction, winding resistance and hysteresis which are calculated using the motor efficiency **[10]**:

$$\eta_e = \frac{We}{E} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(19)$$

Where

$$E = I * V$$

The compression process is assumed to be polytropic with the same polytropic index, n, following the equation:

$$Pv^n = \text{constant} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(20)$$

The volumetric efficiency is given by the following formula**[11]**:

$$\eta_v = Cl.\left[1 - Cc.\left[\left(Rc\right)^{\frac{1}{n}} - 1\right]\right] \quad \ldots\ldots(21)$$

Where

$$Rc = \frac{Pc}{Pe}$$

The main components of the basic refrigeration system is shown in fig (1).neglecting kinetic and potential energy the steady flow energy equations for the for components of the cycle will yield the following equations **[12]**:

Evaporator : (5-6)

$$Qe = \dot{m}.qe$$
$$qe = h_6 - h_5 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(22)$$

Compressor :(1-2)

$$TCP = \dot{m}.Wc$$
$$Wc = h_2 - h_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(23)$$

Condenser :(2-3)

$$Qc = \dot{m}.\left(h_2 - h_3\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots(24)$$

Capillary tube : (4-5)

$$h_4 = h_5 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(25)$$

The volumetric capacity in (kJ/m^3) can be calculated by:

$$Vcc = \frac{qe}{v_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(26)$$

Coefficient of performance is defined as (the refrigeration effect / the work required to produce it):

$$COP = \frac{Qe}{TCP} = \frac{h_6 - h_5}{h_2 - h_1} \quad \ldots\ldots\ldots\ldots..(27)$$

Results and discussion

The main goal of this work is to find the effect of the change in the capillary tube coil number from (0 to 4) with fixed capillary diameter and length. The performance of the refrigeration system for different flow rates from (13 to 23 kg/hr) using the refrigerant (R134a) is examined. Table (2) shows the comparison of the mathematical results for maximum and minimum refrigerant flow rates at different coils number.

The effect of the mass flow rate on the evaporator pressure at different coils number is shown in Fig.(2).As the mass flow rate increased by (76.9%) ,the evaporator pressure increased by (59.5%) for all capillary tube. Also ,the figure shows that evaporator pressure increases by (43.8%),as the capillary coils number increases as a result of increasing the hamper by (4.3%) due to decrease in the mass flow rate and the expand practical in capillary tube.

Fig.(3) shows the relation between the Reynolds number (Re) and the mass flow rate for all capillary tubes. This figure clearly indicates that Reynolds number increases, as the mass flow rate increases for all capillary tubes. At the same time, it can be observed that Reynolds number increases, as the capillary coils number decreases as a result of decreasing the hamper in capillary tube ,so it can be seen that at ($\dot{m}$= 20 kg/hr) the highest Reynolds number at (nco.=0) was (Re=9015.3),and lowest Renold number at (nco.=4) was (Re=8170.18).

The friction factor vs. the mass flow rate for all capillary tubes is presented in Fig.(4). As the mass flow rate increases, the friction factor decreases by (19.4%). Also, the figure shows that friction factor increases by (1.31%), as the capillary coils number increases, because the friction factor is reversely fit with the Reynolds number.

While Fig.(5) shows the degree of subcooling vs. the mass flow rate for all capillary tubes. As shown in the figure, the degree of subcooling decreases by (84.8%) as the mass flow rate increases for all capillary tubes. Also, shows that the degree of subcooling decreases by (24.8%) as the capillary coils number increases. This is because of decreasing in the theoretical compression power, therefore the refrigeration system needs minimum degree of subcooling.

The coefficient of performance (COP) vs. the mass flow rate for all capillary tubes is shown in Fig.(6). As the mass flow rate increases, the coefficient of performance decreases for all capillary tubes, while for the same mass flow rate, the coefficient of performance increases as the capillary coils number increases. This is because of decreasing the theoretical compression power. It can be seen that at ($\dot{m}$=16 kg/hr) the highest COP at (nco.=4) is (COP=6.9191), and lowest COP at (nco.=0) is (COP=6.2033).

Fig.(7) shows the pressure ratio vs. the condensing temperatures for all capillary tubes. As the condensing temperatures increases by (2.71%), the pressure ratio increases by (75.81 %). Also, this figure shows that pressure ratio increases by (37.1 %), as the capillary coils number decreases. This is because decreasing the capillary coil numbers will increase the expansion practical in capillary tubes.

Fig.(8) represents the theoretical compression power (TCP) vs. the condensing temperatures. As the condensing temperatures increases, the TCP increases by (65.8 %). And the TCP decreases by (10.3 %), as the capillary coils number increases, because of increasing the capillary coils number willed decrease the pressure ratio.

The effect of condensing temperatures on the coefficient of performance (COP) for all capillary tubes is shown in Fig. (9). It is clear that the coefficient of performance decreases by (43.4%), as the condensing temperatures increases for all capillary tubes. Also, shows that the COP increases by (13.51%), as the capillary coils number increases. When the capillary coils number increases, the losses will increase because of increasing the friction between the refrigerant and internal surface for capillary tube which means reducing the theoretical compression power.

Fig.(10) shows the electrical efficiency vs. the evaporator pressure for all capillary tubes. As the inlet pressure increases, the electrical efficiency increases by (33.5%). also, from the same figure it can be seen that the electrical efficiency increases by (6.2%), as the capillary coils number increases. Increases the capillary coils number will increase the suction pressure which will lead to increase the mass flow rate through the compressor.

Fig.(11) shows the effect of evaporator pressure on the volumetric efficiency for all capillary tubes. As shown in the figure, the volumetric efficiency increases by (57,6%), as the evaporator pressure increases by (59.5%). At the same time, it can be found that the volumetric efficiency increases by (13.8%), as the capillary coils number increases. This happen because increases the capillary coils number will reduce the pressure ratio which is reversely fit with the volumetric efficiency as indicated in equation (21).

Fig.(12) represents the cooling capacity vs. the evaporator temperature for all capillary tubes. As the evaporator temperature increases by (8.4%), the cooling capacity increases by (65.3%), because of increasing the inlet pressure. Also, shows that the cooling capacity increases, as the capillary coils number increases. This is because of increasing the super heated region and due to enthalpy increasing at suction region. So it can be found that at ($\dot{m}$ =16 kg/hr) the lowest cooling capacity at (nco.=0) is (Qe=769.43 watt) and highest cooling capacity at (nco.=4) is (Qe=825.797 watt).

The system scheme on (P-h) diagram at ($\dot{m}$ =18 kg/hr) for all capillary tubes is shown in Fig.(13). As shown in the figure, as the capillary coils number increases from (0 to 4), the theoretical compression power decreases by (12.67%), the cooling capacity increases by (0.72 %) and the coefficient of performance increases by (15.34 %).

Conclusions

This experimental study investigated the effect of the changing the coil number of capillary tubes (0 to 4) and the mass flow rate (13 to 23 kg/hr). The working fluid is R134a. From this study the following conclusion is deduced:-

1- Theoretical compression power (TCP) changes in the opposite direction with coil number of capillary tube. When the capillary tube coil number increases, the compression power decreases by (10.3%).

2- Cooling capacity (Qe) will be changed when compared with the capillary tube coil number. When the coil number increases, refrigerant effect increases by (1.6%).

3- Coefficient of performance (COP) has changed direction to capillary tube coil number. When the coil number of capillary tube increases, COP trends to be higher by (13.51%).

4- This study shows that when increasing the capillary coils number from (0-4), the hamper of mass flow rate will increase by (4.3%). The best coil number depending on the mass flow rate change is (nco.=4), since it provide higher performance and cooling capacity and lower theoretical compression power at minimum and maximum mass flow rate.

5- Correlation was proposed to evaluate the correlation for the coefficient of performance with capillary tube coil number from (0 to 4), and the mass flow rate from (13-23 kg/hr) as:-

$$COP = -5.6032 + \frac{e^{0.0413*nco.}}{0.0051*\dot{m}}$$

References

1.Poolkrajang, A., “Refrigerationand Air Conditioning”. Promote Vocational Center, Bangkok, Thailand (in Thai), 2004.

2. Kuehl S.J. and Goldsshimdt V.W., ” Steady flow of R22 through capillary tube :Test data”, ASHRAE Trans.,Vol.96,Pt.1,pp.(719-728),1990.

3. Bansal, P.K. and Rupasinghe, A.S. ,”An Empirical Model for Sizing Capillary Tubes”, Int. J. Refrigeration, 19, p 497 –505,1996.

4. Wei, C.Z.; Lin, Y.T.; Wang, C.C.; and Lev, J.S. ,”An experimental study of the performance of capillary tubes for R-407c refrigerant”. ASHRAE Trans. 27: 634-8,2001.

5. Khan, M.K., Kumar, R., Sahoo, P.K.,”A Homogenous Flow Model for Adiabatic Helical Capillary Tube”, ASHRAE Transactions, Vol. 114, Part 1, pp. 239-249,2007.

6. Khan, M.K., Kumar, R.and Sahoo, P.K.,” Flow Characteristics of Refrigerants Flowing through Capillary Tubes - A Review”, Applied Thermal Engineering,(available online on http://sciencedirect.com),2008.

7. ASHRAE Fundamental Handbook, 1997.

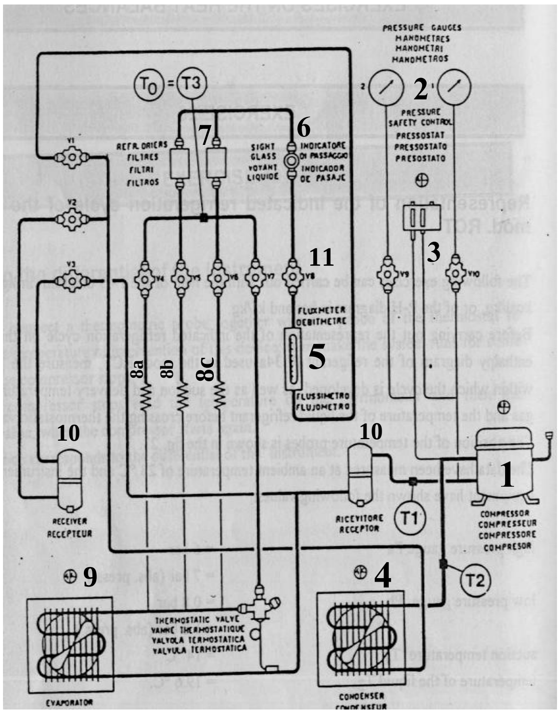
8. W.F.stoeker, “Refrigeration and Air condition”, Mc Graw-Hill puplishing company ltd, second Edition,1982.

9. IIT Kharagpur,”Air Conditioing”, Lesson24”,verasion1ME,pp(1-18),2010. www.Ebookbrowse.com/search/version1

10. احمد عبد النبي عمران, “مقارنة أداء مائع التثليج R22 وR290 في منظومة تثليج انضغاطية ودراسة تأثرها على الأنبوب الشعري " رسالة ماجستير، قسم هندسة المكائن والمعدات، الجامعة التكنولوجية، 2001.

11.Hussain R.M.,"A numerical simulation of vapour compression refrigeration cycle using alternative refrigerants", ph.D Thesis, Baghdad university, Mech. Eng. Dept,1998.

12. الدكتور خالد الجودي ، "مبادئ هندسة تكييف الهواء والتثليج"، جامعة البصرة، 1986.



1	Airtight compressor	
2	High & Low pressure gauge	
3	High & Low pressure switches	
4	Air condenser with glass tube	
5	Flow meter	
6	Sight glass	
7	Refrigeration drier	
8	8a	Capillary tube (nco.=0)
	8b	Capillary tube (nco.=2)
	8c	Capillary tube (nco.=4)
9	Air evaporator with glass tube	
10	Receiver of refrigerant	
11	Flow meter valve	

Fig. (1) Schematic diagram of experimental setup.

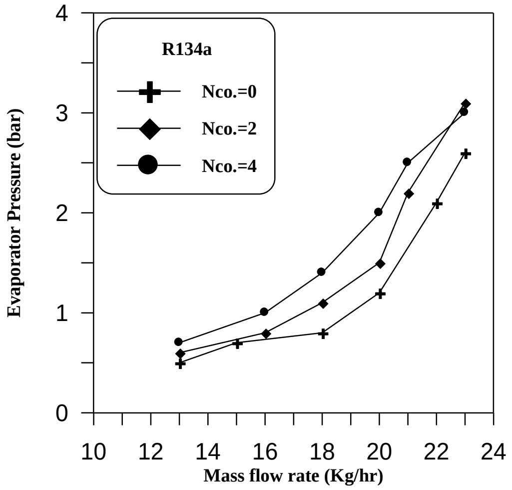


Fig (2) variation of evaporator pressure with mass flow rate

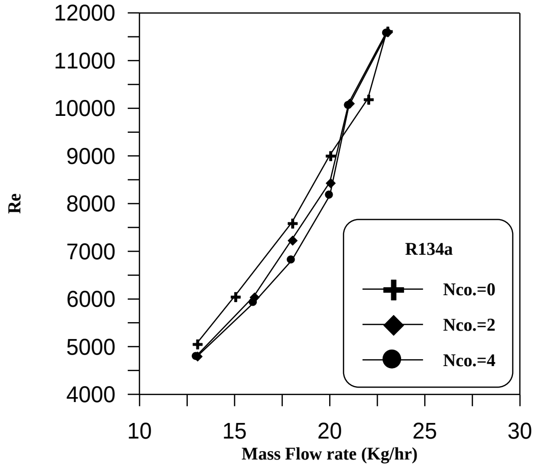


Fig (3) variation of Reynolds number with mass flow rate

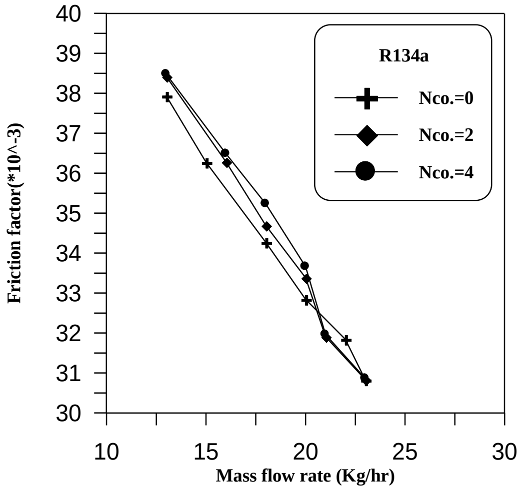


Fig (4) variation of friction factor with mass flow rate

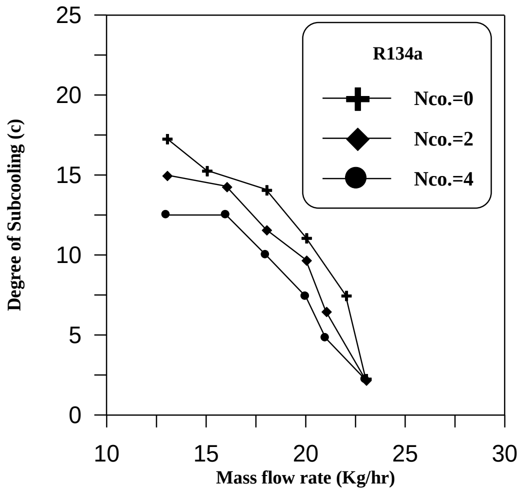


Fig (5) variation of degree of sub-cooling with mass flow rate

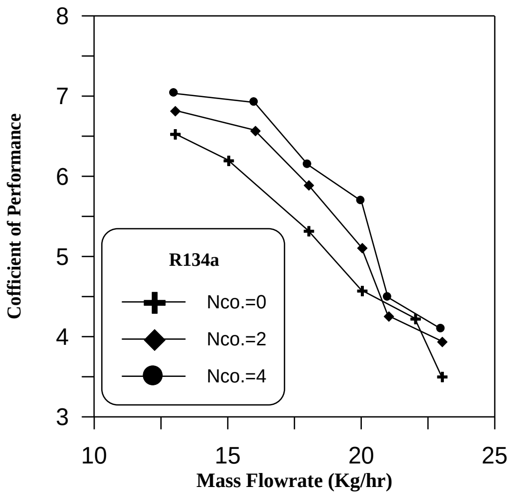


Fig (6) variation of COP with mass flow rate

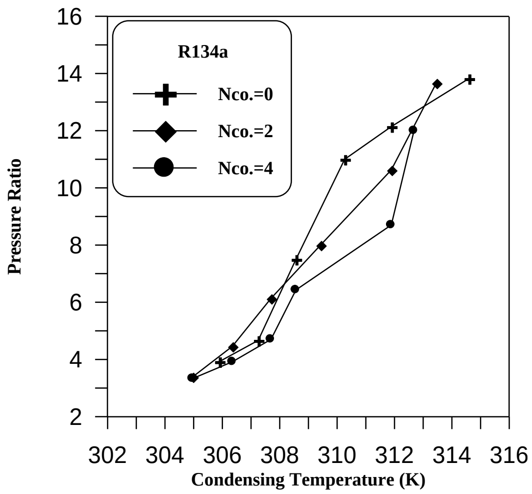


Fig (7) variation of pressure ratio with condensing temperature

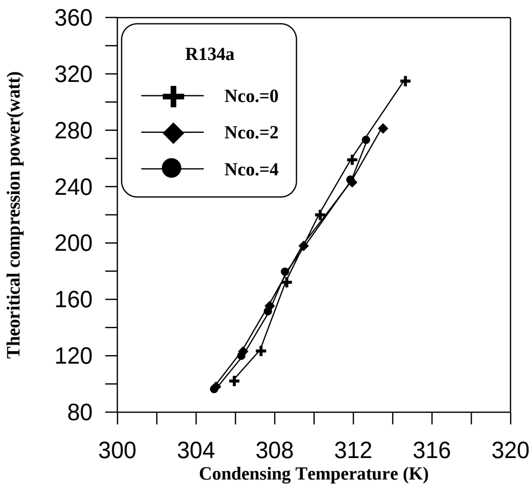


Fig (8) variation of TCP with condensing temperature

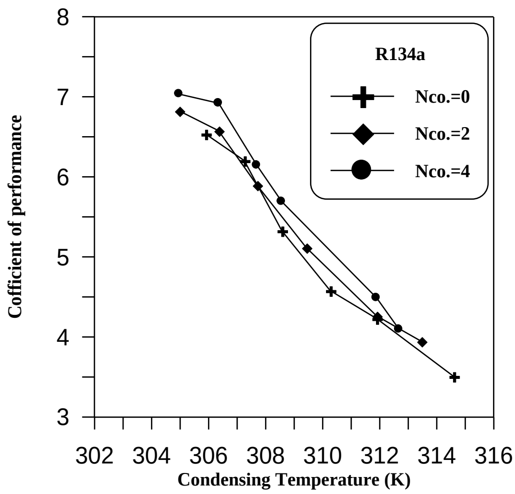


Fig (9) variation of COP with condensing temperature

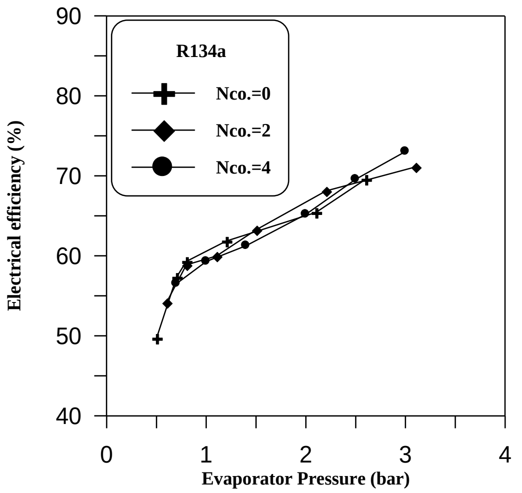


Fig (10) variation of Electrical efficiency with evaporator pressure

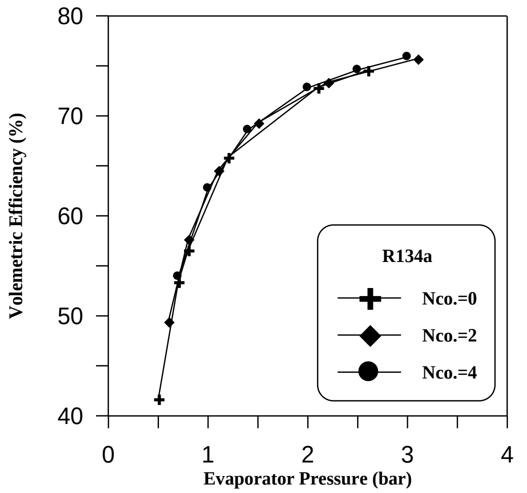


Fig (11) variation of volumetric efficiency with evaporator pressure

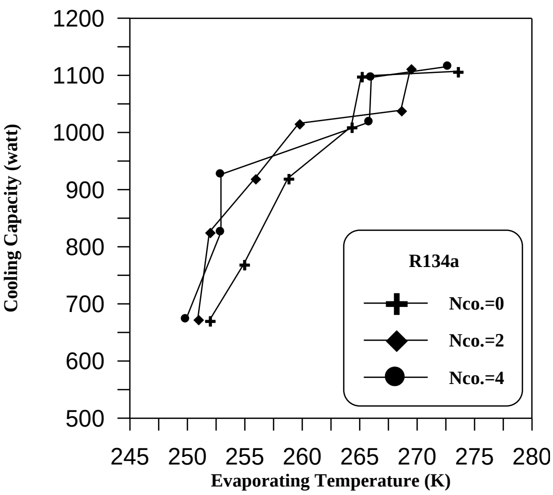


Fig (12) variation of cooling capacity with evaporator temperature

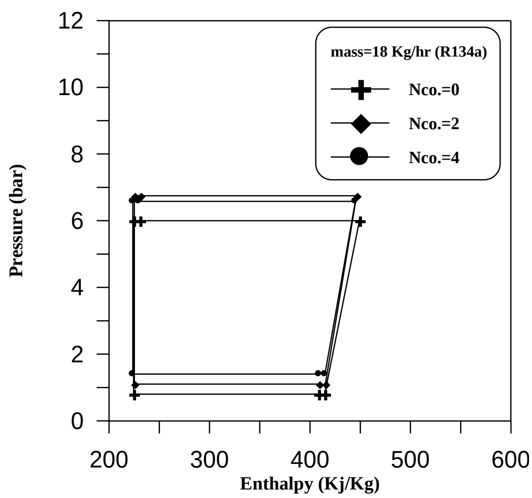


Fig (13) The system scheme on (P-h) diagram with different coils number

Table (1) Specifications of the capillary tubes.

	Internal diameter (mm)	Length (mm)	Coil diameter (mm)	Coil number
Capillary tube	2.5	1500	75	0
				2
				4

Table (2) Comparison of computational results for minimum and maximum mass flow rate with different capillary tubes coil numbers.

Mass flow rate	**13 kg/hr**			**23 kg/hr**		
Coil number (nco.)	**0**	**2**	**4**	**0**	**2**	**4**
TCP (watt)	102.695	98.6776	95.687	315.576	281.896	272.4422
Vcc (kJ/m^3)	3418.78	3420.448	3420.6	3210.92	3238.686	3264.443
COP (-)	6.5341	6.823	7.0332	3.5078	3.94546	4.0947
Qe (watt)	671.02	673.277	672.986	1106.97	1112.21	1115.577
X ($*10^{-1}$) (-)	2.53878	2.542801	2.33764	1.82274	1.741288	1.709431
η_v **(%)**	41.692	49.426	53.9129	74.558	75.721	75.892
η_e **(%)**	49.689	54.1586	56.522	69.565	71.111	73.0435
n (-)	1.37578	1.436954	1.45793	1.09152	1.109662	1.122364
Tsub-cooling(c^o)	17.3	15	12.5	2.3	2.2	2.2
Rc (-)	3.923	3.3871	3.33	13.821	13.667	12
Re (-)	5064.29	4812.367	4784.96	11626.2	11611.19	11566.32
Fr ($*10^{-2}$) (-)	3.793	3.842	3.8475	3.082	3.083	3.086