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Effect of Throttling Variation on the Performance of Vapour Compression Refrigeration System

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Authors' contributions

This work was carried out in collaboration between all authors. Author PKF designed the study. Author TJE performed the experiment and obtained the necessary data. Author TSM analyzed the experimental data and wrote the first draft of the manuscript. Author PKF supervised over all research and finalized the manuscript. All authors read and approved the final manuscript.

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ABSTRACT

In this paper, the effect of throttling variation on the performance of vapour compression refrigeration system using R134a refrigerant as working fluid was investigated and reported. The investigation was carried out using a refrigeration test rig with capillary line as throttling device which was split into three capillary lines of the same length under a room temperature condition of 29 – 32°C. The test rig was equipped with four sigma manifold gauges which were installed at the inlet and exit of the condensing and evaporating units to take the pressure and temperature readings of the working fluid. Experimental trials were conducted by taking the pressure-temperature readings when one-, two- and all the three capillary lines were engaged for the throttling process. The enthalpy condition of the refrigerant at the inlet and exit of the condenser and evaporator were obtained from a standard COOLPACK saturation table. The performance of the refrigeration system was analysed based on the experimental data. Results showed that the temperature difference between the inlet and exit of the evaporating unit was found to increase from 8.66°C to 24.65°C, refrigerating effect increase s from 141.2 kJ/kg to 144.6 kJ/kg while work done by compressor decreases from 24.8 kJ/kg to 21.6 kJ/kg as the number of capillary lines

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employed increases from one to three. The coefficient of performance of the system was found to increase from 5.69±0.04 to 6.71±0.04 which was 17.9% and 7.5% higher than the COP of the system if only one- and two capillary lines were respectively employed for the throttling process. The system behaviour has been attributed to earlier development of the two-phase mixture from the saturated liquid as the number of capillary line increases, thus improving the system performance.

Keywords: Vapour compression refrigeration system; capillary; throttling; refrigeration effect; coefficient of performance; compressor work; R134a.

1. INTRODUCTION

Refrigeration may be defined as that branch of science which deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surrounding [1]. This is achieved by continuous extraction of heat from the enclosed space so that the temperature is maintained below that of the surrounding temperature. The medium with which this heat is removed is known as a refrigerant otherwise referred to as working fluid. This working fluid picks up heat from the space to be cooled, by evaporating at low temperature and low pressure, and dissipate it by condensing at high temperature and high pressure. Refrigeration has many applications which may include: preservation of household perishable food items and drinks, industrial cooling, cryogenics and air conditioning amongst others. There are three (3) main classifications of refrigeration system according to their working principle, these include: vapour compression-, vapour absorption-, and gas cycle refrigeration systems. However, vapour compression refrigeration system (VCRS) is the most frequently used amongst others classified and the major components of VCRS include: compressor, condenser, expansion valve universally known as throttling device and evaporator.

Moreover, it is desirable to have a highly efficient VCRS, and the efficiency of this refrigeration system is dependent on some factors which include: the type of refrigerant used, the effective surface area of condenser for heat transfer and the power of the compressor amongst others. The efficiency of a VCRS is measured by determining its coefficient of performance (COP) which is the ratio of the system refrigerating effect to the work done by the system compressor [2,3]. Successful attempts have been made by different researchers to evaluate the COP of VCRS using different refrigerants. Mogaji [4] carried out a study on the simulation and comparative analysis of single stage VCRS performance for comparison of refrigerant fluids and reported that the system COP and overall efficiency for R134a are similar to that of R12 and R22 when subjected to similar operating conditions. Thus, R134a can act as good substitute for R12 and R22 to help alleviate ozone depletion potential and global warming potential environmental problems. Bolaji [5] discussed the process of selecting environmental-friendly refrigerants that have zero ozone depletion potential and low global warming potential (GWP). It was observed that R23and R32 refrigerants from methane derivatives and R152a, R143a, R134a and R125 refrigerants from ethane derivatives are the emerging refrigerants that are non-toxic, have low flammability and environmental-friendly. It was further recommended that these refrigerants should be subjected to both theoretical and experimental analysis to investigate their performance in VCRS. Mogaji and Yinusa [6] investigated the performance of a VCRS using double effect condensing unit and observed that the system COP and refrigerating effect, subject to test conditions, improved when compared to basic VCRS. A performance improvement of 11% and 11.7% were achieved for the refrigerating effect and COP respectively with evaporation temperature of 15℃ and condensation temperature of 35℃. Upadhyay [7] investigated the effect of the use of diffuser at the condenser inlet and subcooling system on the performance of VCRS and reported a decrease in the work of compressor with similar refrigerating effect achieved which made the COP to improve from 2.65 to 3.38 when compared to conventional VCRS. In another similar study, the effect of introducing diffusers of varying divergence angle at the condenser inlet was investigate and it was observed that the rate of heat transfer at the condenser was found to increase with the diffuser installed thus reducing the work of the compressor and allowing for the size of the condenser to be reduced in VCRS [8].

Mishra [9] evaluated the thermal modelling of the performance of VCRS using R134a as refrigerant in primary circuit and water based nano particles in secondary circuit and reported that with similar VCR system configuration, the system performance was found to increase from 17% to 20% as a result of the incorporation of the secondary fluid. In another study according to Sulaimon et al. [10] which aimed at developing a model predict the mass flow rate of hydrocarbon refrigerant mixture flowing through a capillary tube, it was noted that the correlation between the values obtained from the model developed and measured values was as high as 86.67%. Vasanthi and Yadav [11] investigated the performance of VCRS using R401C and R134a refrigerants by installing a low pressure receiver between the evaporator and compressor so as to flood the evaporator with liquid refrigerant, and noted that the COP of the system improved with the receiver installed and that R401C refrigerant gave the optimum performance. In another study, Nagalakshmi and Yadav [12] designed and carried out performance evaluation of a VCRS using R12 and R134a refrigerants and observed that the COP of the VCRS designed was similar for both refrigerants. It was however noted that R12 be replaced with R134a due to zero ozone depletion potential and low GWP of 0.25. In another similar study, Parmesh and Sharma [13] reported on the comparative performance analysis of refrigeration test rig for R12 and R134a and noted that R134a be substituted for R12 which has got a high ozone depleting potential and a high GWP of 1300 [14].

With the review of work that has been done to evaluate the efficiency of VCRS using different refrigerants, it was noted that R134a is one of the refrigerants that is promising as it makes VCRS to be more efficient when compared to other working fluids, and it is environmental friendly and has low global warming potential. Though extensive research has been done in determining the refrigerating effect and COP of VCRS operating with different refrigerants employed. However, the effect of throttling variation on refrigerating effect, system compressor work and COP is yet to be established. Hence this research work is set to investigate the effect of throttling variation, by splitting up the expansion device into a number of subsystems connected in parallel, on the performance of a VCRS with R134a serving as the working fluid.

2. MATERIALS AND METHODS

2.1 Experimental Setup

Fig. 1 shows a schematic diagram of the experimental setup to investigate the effect of throttling variation on the performance of the VCRS. The setup has three capillary tube lines of the same length which were connected in parallel and each of the capillary tube line has a valve on it to allow or shut off the flow of refrigerant through it into the evaporating unit. The heart of the VCRS is represented by the Process 1 – 2, which indicates the work done by the compressor to circulate the system working fluid by isentropically compressing the vaporised refrigerant sucked through the suction line and discharging it through the discharge line into the condenser.

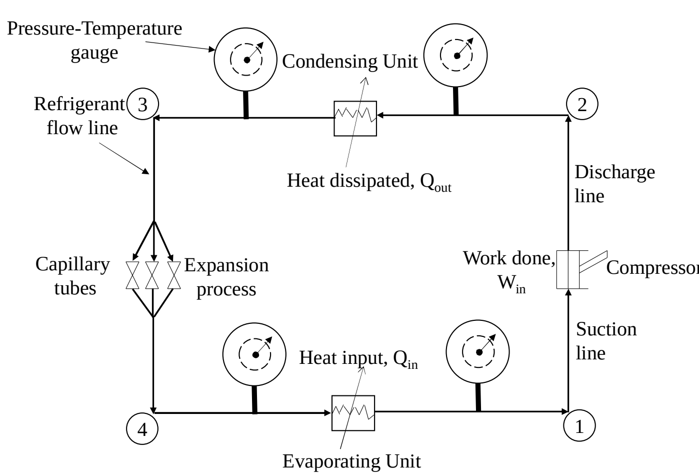

Fig. 1A. Schematic diagram of VCRS with three capillary lines to evaluate effect of throttling variation on its COP

Process 2 – 3 represents the condensation process as latent heat is removed from the vaporised refrigerant which undergo phase change to liquid refrigerant. As the liquid refrigerant leaves the condensing unit at high pressure, it is made to undergo an isenthalpic expansion, as shown in Fig. 2, which reduces the pressure of the refrigerant flow as represented by Process 3 – 4. In this work, the expansion device which is in form of capillary coil tube is split into three (3) lines, which allow the effect of throttling to be evaluated on the VCRS by either employing one, two or all of the capillary tubes for the throttling process. The refrigerant leaves the capillary tube at a low pressure state to enter into the evaporating unit to pick up latent heat by boiling and vaporising at low temperature to achieve the refrigerating effect (cooling) in the unit represented by Process 4 – 1. The vaporised refrigerant is sucked into the compressor again and the cycle continues.

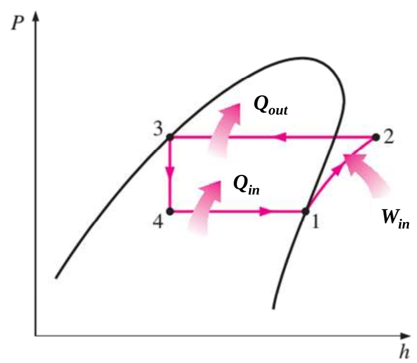

Fig. 2. Pressure-Enthalpy diagram of the developed VCRS

In order to investigate the effect of throttling variation on the performance of the designed VCRS, four pieces of Sigma testing manifold gauges (model number CT-536-G) were incorporated into the refrigeration system to simultaneously measure pressure and temperature of the refrigerant at the inlet and the exit of the condenser and the evaporator, as shown in Fig. 3. Before running the experiment, the rig was vacuumed and charged with R134a refrigerant. The refrigeration rig was powered and allowed to run for twenty (20) minutes. Leakage test was conducted on the rig while in operation using the soap-bubble method.

Having confirmed that there was no leakage and the rig is running at a steady state condition, the first experimental test on the system was conducted by opening only one of the three (3) capillary tubes to perform the throttling process while the other two valves of the three (3) capillary tubes remain shut. Subsequently, second experimental test was carried out while the two valves of the three (3) capillary tubes remain opened and only one of the three (3) capillary tubes remain shut. The third experimental test was conducted while all the three valves remained opened. The experiments were performed under a room condition with temperature ranging from 29 – 32℃. Each of the experiment was conducted for fifty (50) minutes and reading were recorded at an interval of ten (10) minutes resulting into five (5) observatory recordings for pressure and temperature readings obtained through the manifold pressure-temperature gauges installed on the rig respectively.

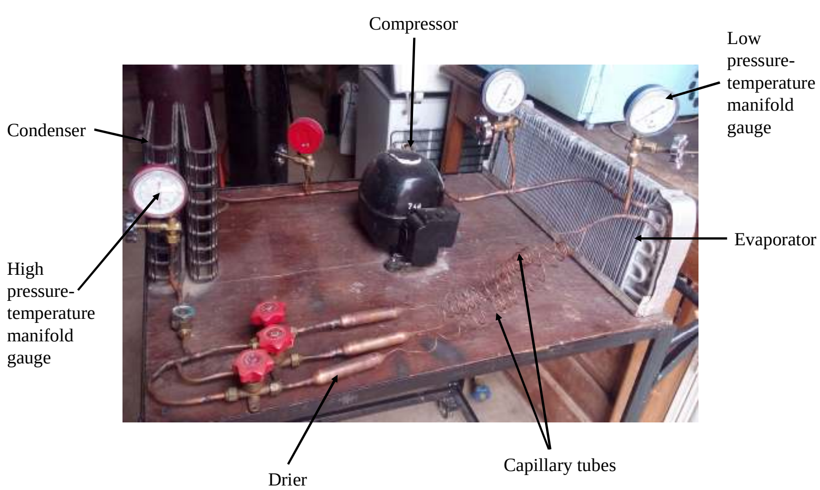

Fig. 3. Experimental vapour compression refrigeration system test rig employed to assess the effect of throttling variation

2.2 Performance Evaluation

The performance of the system is evaluated based on the experimental data obtained. The values of enthalpy (kJ/kg) of the refrigerant R134a for each data set for the three experimental tests conditions were obtained from a standard COOLPACK saturation table [15]. The evaluation of the VCRS are characterized by refrigerating effect in terms of cooling capacity, work done by the compressor and coefficient of performance (COP) of the system.

The coefficient of performance (COP) of the system is calculated from:

$$COP_{Ref} = \frac{Q_e}{W_c} \tag{1}$$

Where; Q_c is the cooling capacity of the VCRS otherwise known as refrigerating effect at the evaporator and W_c is the work done by the VCRS compressor, which are both evaluated using:

$$Q_e = \dot{m}(h_1 - h_4) \tag{2}$$

$$W_c = \dot{m}(h_2 - h_1) \tag{3}$$

Where; $\dot{m}$ is the mass flow rate of refrigerant in the VCRS, h_1 is the enthalpy of the refrigerant at the evaporator outlet, h_2 is the enthalpy of the refrigerant at the condenser inlet, h_3 is the enthalpy of the refrigerant at the condenser outlet, h_4 is the enthalpy of the refrigerant at the evaporator inlet. Since the mass flow rate is the same at any cross section along the VCRS line, (h_1 – h_4) is the refrigerating effect taking place in the evaporating unit (kJ/kg), and (h_2 – h_1) is known as the work of compression (kJ/kg). Hence,

$$COP_{\mathrm{Re}\,f} = \frac{h_1 - h_4}{h_2 - h_1} \tag{4}$$

3. RESULTS AND DISCUSSION

3.1 EFFECT OF THROTTLING VARIATION ON PRESSURE AND TEMPERATURE OF REFRIGERANT

Tables 1, 2 and 3 present the observed values of pressure and temperature of refrigerant recorded at the inlet and the exit of the condenser and the evaporator as one (1), two (2) and three (3) capillary lines respectively were opened during the course of experimentation. In all cases, as expected, the highest values of pressure and temperature of the refrigerant was observed at the inlet of the condenser which is due to the work done by the compressor to increase these properties of the working fluid before it is being discharged into the condensing unit. At the exit of the condenser, the values of the pressure and temperature decrease and more significantly as the refrigerant passes through the throttling device, which made the values of the pressure and temperature at the inlet of the evaporator to be lower. However, the lowest value of pressure and temperature was observed at the exit of the evaporator after the latent heat has been absorbed by the working fluid to create the refrigerating effect. The standard error for all values of experimental observations was less than 0.65 which is indicative of low variability in the data collected. It is noteworthy that the values of the pressure and temperature at the inlet and outlet of the evaporator increase as the number of the capillary tube lines increases. Most significantly, the inlet temperature of the refrigerant to the evaporator increases from 4.7 – 26.3℃ and the difference between the evaporator inlet and exit temperatures (T_4 – T_1) are 8.66℃, 20.16℃ and 24.65℃ when one, two and three capillary lines are respectively engaged for the throttling process. This behaviour indicates that there is an increase in the rate of heat transfer as the number of capillary lines engaged increases thus increasing the system refrigerating effect. This may be attributed to an earlier transition of the working fluid from saturated liquid to two phase flow mixture at the expansion valve exit as the number of the capillary tube lines employed from the throttling increases.

3.2 Effect of Throttling Variation on Coefficient of Performance

Tables 4, 5 and 6 present the values of the enthalpy at the inlet and outlet of the condenser and evaporator obtained from COOLPACK saturation table [15] based on the measured experimental temperature values and the calculated values of COP using equation (4). The results obtained indicate that the COP of the refrigeration system increases as the number of capillary line employed for the throttling process increases. According to equation (1), the COP is the ratio of the refrigerating effect (equation (2)) to work done by the compressor (equation (3)), and it was found that as the number of the capillary line employed for the expansion process was increasing, the refrigerating effect was increasing, while the work done by the compressor was decreasing which yielded an increase in the system performance.

Table 1. Pressure and temperature reading at inlet and exit of condenser and evaporator when one capillary line was opened

S/N	Condenser				Evaporator			
	Inlet		Outlet		Inlet		Outlet	
	P_2(bar)	T_2 (°C)	P_3(bar)	T_3 (°C)	P_4(bar)	T_4 (°C)	P_1(bar)	T_1 (°C)
1	11.50	44.67	10.01	39.42	3.52	5.19	2.52	-4.09
2	11.20	43.70	9.80	38.64	3.38	4.03	2.55	-3.76
3	11.00	42.97	10.07	39.64	3.48	4.86	2.48	-4.04
4	11.30	44.00	10.01	39.42	3.45	4.61	2.48	-4.04
5	11.30	44.00	10.01	39.42	3.45	4.61	2.52	-4.09
Average	11.26	43.87	9.98	39.31	3.46	4.66	2.51	-4.00
Std.Error	0.08	0.28	0.05	0.17	0.02	0.19	0.01	0.06

Table 2. Pressure and temperature reading at inlet and exit of condenser and evaporator when two capillary lines were opened

S/N	Condenser				Evaporator			
	Inlet		Outlet		Inlet		Outlet	
	P_2(bar)	T_2 (°C)	P_3(bar)	T_3 (°C)	P_4(bar)	T_4 (°C)	P_1(bar)	T_1 (°C)
1	10.90	42.62	9.66	38.11	5.35	17.87	2.76	-1.62
2	11.04	43.11	9.80	38.64	5.45	18.46	2.73	-1.92
3	10.70	41.92	9.59	37.84	5.45	18.46	2.69	-2.32
4	10.90	42.62	9.66	38.11	5.66	19.67	2.79	-1.33
5	10.35	40.67	9.66	38.11	5.18	16.85	2.69	-2.32
Average	10.78	42.19	9.67	38.16	5.42	18.26	2.73	-1.90
Std.Error	0.12	0.42	0.03	0.13	0.08	0.46	0.02	0.19

Table 3. Pressure and temperature reading at inlet and exit of condenser and evaporator when three capillary lines were opened

S/N	Condenser				Evaporator			
	Inlet		Outlet		Inlet		Outlet	
	P_2(bar)	T_2 (°C)	P_3(bar)	T_3 (°C)	P_4(bar)	T_4 (°C)	P_1(bar)	T_1 (°C)
1	11.40	44.33	9.66	38.11	6.62	24.83	3.14	1.94
2	11.04	43.11	10.01	39.42	6.97	26.56	3.11	1.67
3	11.73	45.43	10.01	39.42	7.11	27.24	3.14	1.94
4	10.76	42.13	10.07	39.64	6.97	26.56	2.97	0.39
5	11.59	44.97	10.01	39.42	6.90	26.22	3.17	2.21
Average	11.30	43.99	9.95	39.20	6.91	26.28	3.11	1.63
Std.Error	0.18	0.61	0.07	0.28	0.08	0.40	0.04	0.32

Table 4. Values of refrigerating effect, compressor work and COP with one capillary line

S/N	T_1 (°C)	T_2 (°C)	T_3 (°C)	h_1 (kJ/kg)	h_2 (kJ/kg)	$h_3 = h_4$ (kJ/kg)	$h_1 - h_4$ (kJ/kg)	$h_2 - h_1$ (kJ/kg)	COP
1	-4.09	44.67	39.42	396.20	421.38	255.54	140.66	25.18	5.59
2	-3.76	43.70	38.64	396.39	420.99	254.38	142.01	24.60	5.77
3	-4.04	42.97	39.64	396.23	420.69	254.38	141.85	24.46	5.80
4	-4.04	44.00	39.42	396.23	421.11	255.54	140.69	24.88	5.65
5	-4.09	44.00	39.42	396.20	421.11	255.54	140.66	24.91	5.65
Average	-4.00	43.87	39.31	396.25	421.06	255.08	141.17	24.81	5.69
Std.Error	0.06	0.28	0.17	0.04	0.11	0.28	0.31	0.13	0.04

Table 5. Values of refrigerating effect, compressor work and COP with two capillary lines

S/N	T_1 (℃)	T_2 (℃)	T_3 (℃)	h_1 (kJ/kg)	h_2 (kJ/kg)	$h_3 = h_4$ (kJ/kg)	$h_1 - h_4$ (kJ/kg)	$h_2 - h_1$ (kJ/kg)	COP
1	-1.62	42.62	38.11	397.65	420.54	253.60	144.05	22.89	6.29
2	-1.92	43.11	38.64	397.48	420.75	254.38	143.10	23.27	6.15
3	-2.32	41.92	37.84	397.24	420.25	253.31	143.93	23.01	6.26
4	-1.33	42.62	38.11	397.83	420.54	253.60	144.23	22.71	6.35
5	-2.32	40.67	38.11	397.24	420.56	253.60	143.64	23.32	6.16
Average	-1.90	42.19	38.16	397.49	420.53	253.70	143.79	23.04	6.24
Std.Error	0.19	0.42	0.13	0.12	0.08	0.18	0.20	0.11	0.04

Fig. 4 gives a summary of the effect of increasing the number of throttling on the refrigerating effect and the work done by compression. It is noteworthy that the decrease in the work done by the compressor brings about a decrease in energy consumption which makes the system to be energy saving. Moreover, the increase in the refrigerating effect increases the efficiency and effectiveness of the refrigeration system. Thus, having a system with increased refrigerating effect and its compressor work decreased is desirable for higher efficiency.

Fig. 5 shows an increasing trend for the COP as the number of capillary lines employed for throttling increases. With only one capillary tube employed for the throttling process, a COP of 5.69±0.04, when two capillary lines was employed, the COP increased to 6.24±0.04 which is 9.7% higher than the COP when only one capillary line was employed. Moreover, when three capillary lines were employed for the expansion process, the COP of the system was evaluated as 6.71±0.04 which is 17.9% and 7.5% higher than when only one capillary line and when two capillary lines were respectively employed.

3.3 Discussion

Theoretically applying the equation of continuity of flow to the Process line 3 – 4, where the isenthalpic expansion of the refrigerant is taking place, the volumetric flow rate of refrigerant at different cross sections A-A and B-B, as shown in Fig. 6, should always be the same as expressed in equation 5.

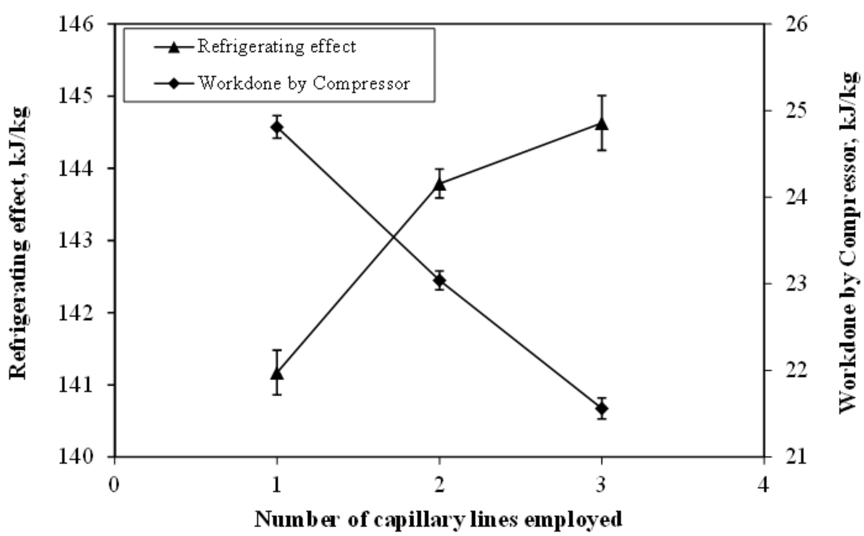

Fig. 4A. Graph showing increase in refrigerating effect and decrease in work done by compressor as the number of capillary lines engaged for throttling increases

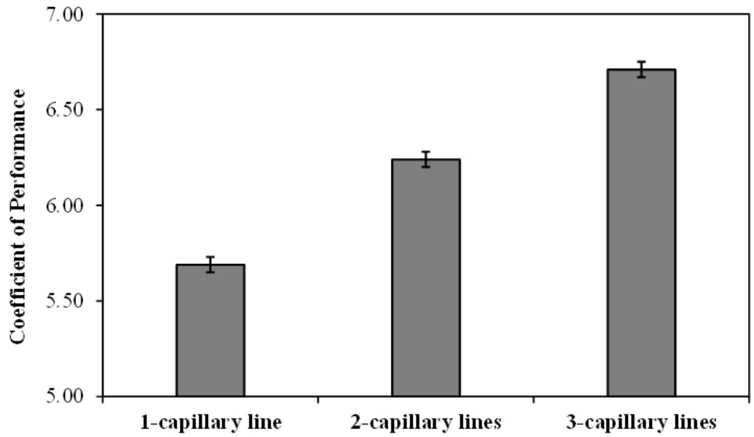

Fig. 5. Variation of COP against the number of capillary lines employed for the throttling process

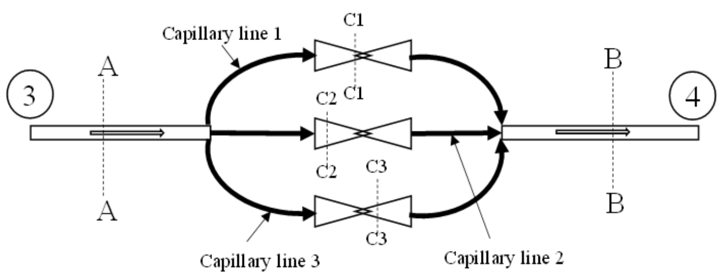

Fig. 6. Flow diagram of the throttling process 3 – 4 (Fig. 1) showings the equivalent volumetric flow rate of refrigerant at section A-A and B-B

Table 6. Values of refrigerating effect, compressor work and COP with three capillary lines

S/N	T_1 (°C)	T_2 (°C)	T_3 (°C)	h_1 (kJ/kg)	h_2 (kJ/kg)	$h_3 = h_4$ (kJ/kg)	$h_1 - h_4$ (kJ/kg)	$h_2 - h_1$ (kJ/kg)	COP
1	1.94	44.33	38.11	399.73	421.25	253.6	146.13	21.52	6.79
2	1.67	43.11	39.42	399.57	420.75	255.54	144.03	21.18	6.80
3	1.94	45.43	39.42	399.73	421.69	255.54	144.19	21.96	6.57
4	0.39	42.13	39.64	398.83	420.34	254.38	144.45	21.51	6.72
5	2.21	44.97	39.42	399.89	421.51	255.54	144.35	21.62	6.68
Average	1.63	43.99	39.20	399.55	421.11	254.92	144.63	21.56	6.71
Std.Error	0.32	0.61	0.28	0.19	0.25	0.40	0.38	0.12	0.04

Mathematically, the relationship of the refrigerant discharge at various section can be expressed as:

$$\dot{Q}_A = \dot{Q}_B = \underbrace{\dot{Q}_{C1}}_{\text{When only one capillary line is used}} = \underbrace{\left(\dot{Q}_{C1} + \dot{Q}_{C2}\right)}_{\text{When two capillary lines are used}} = \underbrace{\left(\dot{Q}_{C1} + \dot{Q}_{C2} + \dot{Q}_{C3}\right)}_{\text{When three capillary lines are used}} \tag{5}$$

Knowing that $\dot{Q}$ = AV; then equation (5) can be expressed as:

$$A_A V_A = A_B V_B = A_{C1} V_{C1} = \left(A_{C1} V_{C1} + A_{C2} V_{C2}\right) = \left(A_{C1} V_{C1} + A_{C2} V_{C2} + A_{C3} V_{C3}\right) \tag{6}$$

Where; $\dot{Q}_A$ is the refrigerant flow discharge at cross section A-A, $\dot{Q}_B$ is the refrigerant flow discharge a cross section B-B, $\dot{Q}_{C1}, \dot{Q}_{C2}, \dot{Q}_{C3}$ are the refrigerant discharge across the capillary lines 1, 2, and 3 respectively, A_A, A_B are the cross sectional bore area at cross section A-A and B-B respectively, A_{C1}, A_{C2}, A_{C3} are the cross sectional bore area of the capillary line tube 1, 2 and 3 respectively, V_A, V_B are velocity of refrigerant flow across the cross sections A-A and B-B respectively and V_{C1}, V_{C2}, V_{C3} are the velocity of refrigerant flow across the cross sectional bore of the capillary line 1, 2 and 3 respectively.

Since, the diameters of the bore at section A-A and B-B are the same, then the velocity of flow across these sections will be the same. Also, for the capillary tubes with the same bore diameter, the velocity of flow across each capillary line is the same. Therefore, equation (6) can be rewritten as:

$$A_A V_A = A_{C1} V_{C1} = 2A_{C1}\left(\frac{1}{2}V_{C1}\right) = 3A_{C1}\left(\frac{1}{3}V_{C1}\right) \quad (7)$$

Thus, equation (7) shows that when two capillary lines are employed which are connected in parallel, the velocity of flow in each of them would be half of the velocity of flow if only one capillary had been employed. Also employing the three capillary lines indicate that the velocity of flow in each line would be one third of the expected velocity if only one capillary line had been used for the throttling process. Therefore, the velocity of flow across the capillary lines is a factor that contributes to the improvement of the system COP. The increase in refrigerant pressure at the inlet to the evaporator as the number of capillary increases as presented in Table 1, 2 and 3 can be justified owing to the fact that pressure drop in capillary tube is achieved by friction due to fluid viscosity and momentum due to flashing of liquid refrigerant to vapour. Thus, with increasing number of the capillary lines, friction and momentum of fluid are reduced due to the decrease in the velocity of flow, and this made the pressure drop across the capillary section to decrease as well. Increasing the number of the capillary lines engaged for throttling allows an earlier development of the two-phase flow mixture at the inlet of the evaporator thus increasing the rate of heat transfer, refrigerating effect in this unit and the overall system COP.

With the three capillary coil tubes employed in this work, the effect of them arranged in parallel has been identified, as the work done by the compressor is decreasing and the refrigerating effect increasing as the number of the capillary lines engaged increases, thus increasing the COP of the refrigeration system. However, the subject for investigation in the future is to know if the COP of the system will continue to increase or it will reach a threshold if the numbers of capillary lines employed keep increasing.

4. CONCLUSION

In this study, the effect of throttling variation on the performance of vapour compression refrigeration system (VCRS) has been successfully investigated with the following deductions:

i. The temperature difference between the inlet and exit of the evaporating unit was found to increase in the order of 8.66°C, 20.16°C and 24.65°C when one-, two- and three capillary lines were respectively engaged for the throttling process. This indicates an increase in rate of heat transfer and refrigerating effect as the number of capillary coil lines used for throttling was increasing.
ii. As the number of capillary lines employed for the throttling process was increasing from one to three, the refrigerating effect was found to increase from 141.2 kJ/kg to 144.6 kJ/kg, while the work done by the system compressor decreased from 24.8 kJ/kg to 21.6 kJ/kg.
iii. The COP of the system was found to increase in the order of 5.69±0.04, 6.24±0.04 and 6.71±0.04 when one-, two- and three capillary lines were respectively engaged for the throttling process; and the COP of 6.71±0.04 was 17.9% and 7.5% higher than the COP of the system if only one- and two capillary lines were respectively employed for the throttling.
iv. The behaviour of the VCRS with modification made to the throttling process has been attributed to an earlier transition of the working fluid from the saturated liquid state to two-phase flow mixture at the exit of the expansion valve. Also, the velocity of flow is reduced with increasing number of capillary lines, thus reducing the

friction and momentum of the refrigerant before it enters into the evaporating unit.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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