IMPACT OF REFRIGERANT CHARGE OVER THE PERFORMANCE CHARACTERISTICS OF A SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

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ABSTRACT

Experimental investigation was done to find the role of capillary tube length and amount of refrigerant charge on the overall heat transfer coefficient in condenser and evaporator and actual COP of a simple vapour compression refrigeration system. It was concluded that increasing of the refrigerant charge in the system largely enhances the overall heat transfer coefficient in the evaporator by increasing the part of space occupied by liquid refrigerant in the evaporator. Capillary tube length is important as it decides the evaporator temperature and pressure directly but also affects the tendency of refilling of evaporator with liquid refrigerant after initial start up and alters the amount of optimum charge in the system. A simple refrigeration system should be designed with minimum possible length of capillary tube to satisfy the refrigeration conditions and maximum amount of refrigerant charged in the system limited by unwanted condition of refrigerant liquid entering the compressor.

KEYWORDS: *Vapour Compression Refrigeration, Refrigerant charge, Capillary tube, heat transfer coefficient, coefficient of performance*

I. INTRODUCTION

A simple vapour compression refrigeration system with simplest expansion device as capillary tube is used in numerous of small or medium refrigeration applications like domestic refrigerator, deep freezer, water cooler, room air conditioners, cooling cabinets and many more all over the world. The small scale refrigeration machines are produced in large numbers and have substantial contribution to energy consumption. [1] Energy conservation in refrigeration, air conditioning and heat pump systems has a large potential. The working conditions for a refrigerating system in steady operation depend on several factors: boundary conditions (ambient temperature, cold room temperature, compressor speed, and control settings), refrigerant type and refrigerant charge, system architecture and size, thermal loads. [2] The performance is influenced by matching of all these factors. Theoretical performance of the system deteriorates in real conditions due to internal and external irreversibility in the system. [3, 4, 5] Internal irreversibility is due to non isentropic compression, friction and entropy generation in the system components. [6, 7]

NOMENCLATURE

A	surface area of tubes	***Greek Symbols***	
c	specific heat	Δ	difference
COP	coefficient of performance	ρ	density
i	specific enthalpy	***Subscripts***	

m	mass flow rate	ac	actual
Q	heat transfer rate	c	condenser
t	temperature	e	evaporator
U	overall heat transfer coefficient	i	inlet/ inside
v	specific volume	isen	isentropic
V	total volume	liq	liquid
VCR	vapour compression refrigeration	m	mean
W	power consumption of compressor	o	outlet/ outer
		r	refrigerant
		th	theoretical
		vap	vapour
		w	water

Minimization of internal irreversibility depends mainly on the design and selection of compressor which is not in the scope of this study. External irreversibility losses occur over the condenser and evaporator due to finite rate of heat exchange against finite values of temperature difference and heat capacities of external fluids. These losses can be minimized by maximizing the heat transfer coefficient over condenser and evaporator. [8, 9, 10] Considering the internal and external irreversibility, a vapour compression refrigeration system can be theoretically optimized and balanced using finite time thermodynamics. [11, 12, 13] But a correct estimate of the parameters causing irreversibility i.e. finite value of heat transfer coefficients is a real challenge.

Heat transfer coefficient on the external fluid (air/water) side in the evaporator and condenser can be enhanced optimized and managed easily. But the condensation heat transfer coefficient over the refrigerant side in condenser and boiling heat transfer coefficient over the refrigerant side in evaporator are quite difficult to estimate and manage because these are associated with change of phase of refrigerant and the two phase flow behavior is quite difficult to estimate through inside space of condenser and evaporator due to non availability of exact void fraction correlations. [14, 15, 16] Boiling coefficient in evaporator is even more difficult to estimate as compared to condensing coefficient in condenser. [17, 18]

Condensing coefficient depends on how the condensate film forms, flows and is pierced through by condensing vapours and finally accumulate at the bottom section of the condensing coil under the influence of gravity, mean velocity of refrigerant vapours and geometry of condensing coil. The boiling heat transfer characteristics on refrigerant side in evaporator are quite different than that of condenser. In small refrigeration systems, generally dry expansion tubular type evaporator without any accumulator is used in which some portion is used for boiling of refrigerant (where nucleate boiling dominates) and rest is used for superheating of vapours (where forced convection dominates). [19, 20, 21] Superheating is necessary to safeguard the compressor from damage by suction of incompressible refrigerant liquid. [22] Heat transfer coefficient in the boiling zone before dry out point is much higher than in the superheating zone beyond dry out point. Thus a correct estimation of average heat transfer coefficient on the refrigerant side both in the evaporator and condenser is not possible analytically and mostly empirical approach is used. Simulation techniques have been used by researchers for design of vapour compression refrigeration system under steady state conditions. [23, 24, 25] Design of evaporator and condenser depends mainly on two design parameters as heat transfer coefficients and corresponding pressures, which further depend on other conditions in the system. Among these, two main conditions are size and length of capillary tube and refrigerant charge which can also most easily be altered in a given system. Capillary tube length is very important as it directly decides the pressures of system. [26] The present experimental study is about to find the actual values of refrigeration rate, overall heat transfer coefficient in the evaporator and condenser and COP of a simple vapour compression refrigeration system under real steady state conditions for different combinations of capillary tube size and refrigerant charge in the system and to find the impact of refrigerant charge with different lengths of capillary tube over the performance of system under same constant boundary conditions.

In the following sections 2 and 3, description of experimental set-up used and the detailed procedure adopted is given. Thereafter the results have been plotted in the form of bar charts and the detailed analysis is given in section 4. The results and future scope of work are concluded in section 5.

II. EXPERIMENTAL SET-UP AND PROCEDURES

The experimental facility as shown in "figure 1" consists of a simple vapour compression refrigeration system charged with HFC-134a refrigerant. The evaporator and condenser are shell and tube type adiabatic heat exchangers. Refrigerant flows through copper tubes of outside and inside diameters as 9.5 mm and 8.5 mm throughout the condenser, evaporator and connecting lines. All connecting tubes of refrigerant are well insulated by polyurethane cellular foam. Water can flow through the insulated shell of each of the evaporator and condenser and there is an arrangement for control and measurement of water flow rate through each. Compressor used is Kirloskar Copeland model no KCE444HAG (1/3 HP). Hand operated valves and connectors are provided before and after the capillary tube to facilitate its replacement. The temperature of refrigerant at various points is measured with RTDs (Pt 100 Ω at 0^oC) strongly insulated along length of tubes by means of polyurethane cellular foam. (axial heat conduction was hence neglected). Pressure of refrigerant is measured and indicated by separate dial gauges at four points before and after each of the evaporator and condenser. Mass flow rate of refrigerant liquid after condenser is indicated by a glass tube rotameter fitted in the refrigerant line after condenser. A digital wattmeter gives the instant value of power consumption of compressor and also the total energy consumed during whole trial.

The total inside space of the closed refrigeration system is calculated as 1825 cm^3 out of which 673 cm^3 is of evaporator, 777 cm^3 is of condenser, and 200 cm^3 is of 'liquid' line from condenser to evaporator. The total mass of refrigerant charged in the system is given by equation (1)

$$m_r = V_{liq}\rho_{liq} + \frac{V_{vap}}{v_{vap}} \qquad \text{Eq. (1)}$$

$$V_{liq} + V_{vap} = V \qquad \text{Eq. (2)}$$

From the equations (1) and (2), V_{liq} and V_{vap} (volume occupied by liquid and vapour phase of refrigerant charge) can be calculated if we know the total weight of refrigerant charged in the system (m_c) and ρ_{liq} and v_{vap} at the corresponding pressures. Equation (1) & (2) can also be employed separately for evaporator and condenser.

It is hard to find the exact inventory of liquid and vapour refrigerant in different components of system during its working. But to an approximation, only to calculate appropriate refrigerant charge in the system, it may be taken that 10% of total volume of condenser, full liquid line and 40 % of evaporator space is occupied by liquid refrigerant during working of the system. Rest of the inside space is occupied by vapour phase. With this approximation the total charge calculated from equation (1) & (2) is 700 g. Trials were conducted however with three different amounts of refrigerant charge as 500 g, 700 g and 1000 g i.e. one estimated correct value, one less than this and one more than this value. Refrigerant charge filled in the system is weighed by keeping the charging cylinder on weighing balance and taking readings before and after fresh charging each time. From the previous experience three different capillary tube sets chosen are twin capillary tubes, each of diameter 0.044" (1.1176 mm) but lengths of 30" (0.762 m) , 42" (1.067 m) and 54" (1.372 m). In this way with three different capillary tubes and three different amounts of charge a total of 9 trials (3*3) were conducted in repetition.

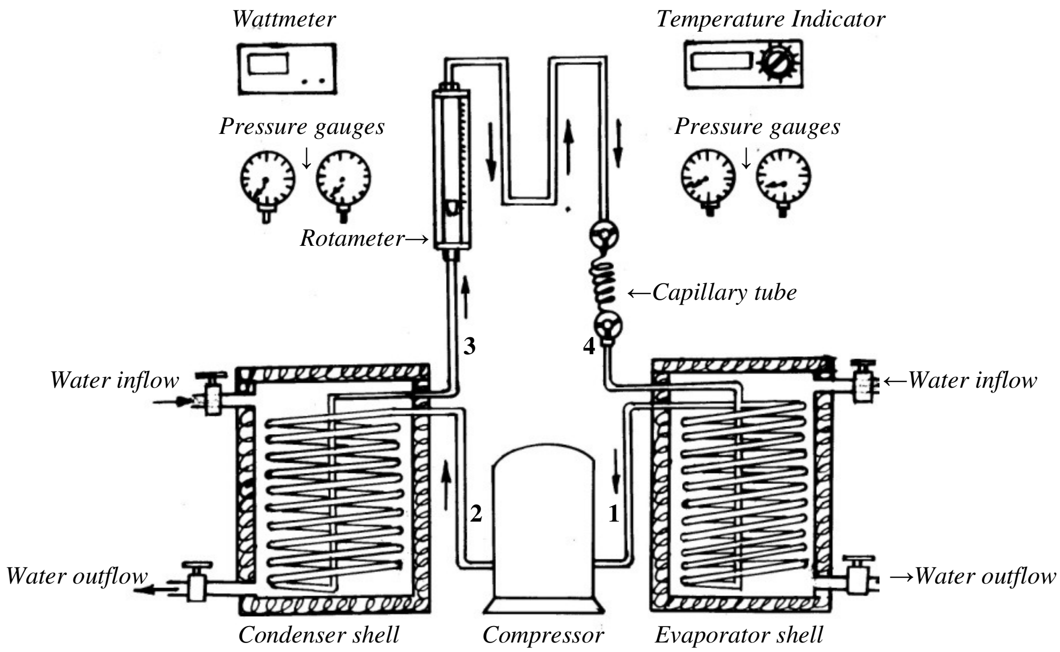

Figure 1. Experimental set up of simple vapour compression refrigeration system

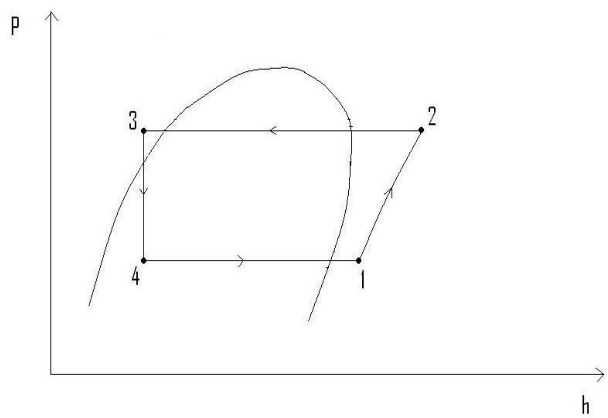

Figure 2. Theoretical Vapour Compression Refrigeration Cycle

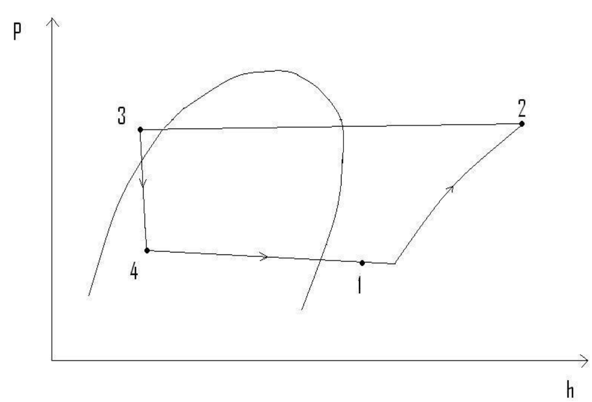

Figure 3. Actual Vapour Compression Refrigeration Cycle

III. DATA REDUCTION

The pressure and temperature readings of refrigerant were taken at four strategic points 1, 2, 3 & 4 as indicated in "figure 1" and "figure 3". Temperature of cooling water at the inlet and outlet of condenser shell and evaporator shell are also recorded in the same way. Mass flow rates of refrigerant liquid at condenser outlet and of water entering the condenser and the evaporator are recorded with the help of corresponding glass tube type rotameter. Actual reading in Wattmeter is also recorded regularly. All this data was uploaded in MS Excel worksheets and the properties of refrigerant were calculated for each of the observation by using computer subroutines for calculating refrigerant properties. [27] This data was reduced to useful performance parameters as described below:
Refrigeration rate of evaporator,

$$Q_e = m_{w,e} c_w (t_{w.e,i} - t_{w,e,o}) = m_r (i_1 - i_4) \qquad \text{Eq. (3)}$$

Heat transfer rate in condenser

$$Q_c = m_{w,c} c_w (t_{w.c,o} - t_{w,c,i}) = m_r (i_2 - i_3) \qquad \text{Eq. (4)}$$

Theoretical COP

$$COP_{th} = \frac{i_1 - i_4}{\Delta i_{isen}} \qquad \text{Eq. (5)}$$

Actual COP

$$COP_{ac} = \frac{Q_e}{W_{ac}} \qquad \text{Eq. (6)}$$

Overall heat transfer coefficient over evaporator

$$U_e = \frac{Q_e}{\Delta t_{m,e} A_{e,o}} \qquad \text{Eq. (7)}$$

$$\text{Where, } \Delta t_{m,e} = \frac{t_{w,e,i} - t_{w,e,o}}{\log_e\left(\dfrac{t_{w,e,i} - t_{r,e}}{t_{w,e,o} - t_{r,e}}\right)}$$

Overall heat transfer coefficient over condenser

$$U_c = \frac{Q_c}{\Delta t_{m,c} A_{c,o}} \qquad \text{Eq. (8)}$$

$$\text{Where, } \Delta t_{m,c} = \frac{t_{w,c,o} - t_{w,c,i}}{\log_e\left(\dfrac{t_{r,c} - t_{w,c,i}}{t_{r,c} - t_{w,c,o}}\right)}$$

IV. ANALYSIS AND RESULT

Most of the refrigerant in liquid form will accumulate in the evaporator during pressure equalization period whereas the condenser and capillary tube contain superheated gas only. A typical course of events at the start of compressor is as follows: on start of compressor, boiling of liquid refrigerant in evaporator starts at a fast pace and initial mass flow rate through compressor is high due to higher evaporator pressure and temperature. On the other side mass flow rate through capillary tube is least initially due to superheated gas and least pressure difference across it. The result is that in a very short period, refrigerant mass is displaced towards the condenser and evaporator becomes more or less starved of liquid refrigerant. By this, evaporator pressure falls and condenser pressure rises. The displaced mass of refrigerant condenses in the condenser by forming liquid layer inside the condensing tubes and more liquid accumulates at the inlet of capillary tube. With this the evaporator starts to refill with refrigerant liquid. This refilling process is accelerated with sub cooling of liquid backed up in the condenser and so increase in mass flow rate of capillary tube. At the start, mass flow rate of refrigerant through compressor is highest and through capillary tube is least. This difference is adjusted by initial displacement of refrigerant from evaporator to condenser. Thus, once most of the liquid is displaced to condenser but again it starts coming back to evaporator with the effective condensation and increase of pressure difference across capillary tube. This refilling of evaporator with refrigerant liquid again activates the heat exchange and evaporation process in the evaporator and opposes the decline of evaporator pressure. A natural balance between the individual working of components of the system is established after some time if the boundary conditions of the system are not changing. Under these steady state conditions, the impact of different combinations of capillary tube length and refrigerant charge in the system on various performance parameters is analyzed as follows:

4.1 General working parameters of VCR system:

As shown by "Table 1", the evaporator pressure and condenser pressure both have a higher value for the higher amount of refrigerant charge in the system. More is the initial charge, more liquid is there in the evaporator activating the heat exchange and evaporation process and so increasing the evaporator pressure, which also results in higher condenser pressure due to increased compressor discharge. Increase in the value of condenser pressure is however less because simultaneously the condensation becomes more effective with the increase in discharge of compressor. Therefore the pressure ratio is least in case of highest refrigerant charge for a given length of capillary tube and obviously it is least for the shortest length of capillary tube. Superheating of vapours at the suction of compressor is more in case of larger length capillary tube due to lower evaporator temperature. For a given length of capillary tube however the superheating of vapours decreases with the increase in refrigerant charge because the dry out point moves downstream in the evaporator due to more liquid charge at a time. Sub-cooling of refrigerant liquid in condenser has opposite trend. With more superheating of vapours at the suction of compressor, more is the temperature of vapours entering the condenser so less sub-cooling and vice-versa.

Table 1 Actual conditions of VCR system under steady state conditions

Double capillary tube length (m)	Mass of refrigerant charge (g)	Condenser Pressure (bar)	Evaporator Pressure (bar)	Pressure Ratio of compressor	Superheating of vapours at suction of compressor (^oC)	Subcooling of liquid in condenser (^oC)
0.762	500	10.103	4.69	2.15	13.1	6.5
	700	10.586	5.345	1.98	8.2	6.3
	1000	11.413	5.828	1.96	1.7	10.4
1.067	500	9.69	3.897	2.49	19.4	5.9
	700	10.241	4.793	2.14	12.4	6.7
	1000	11.62	5.62	2.07	2.49	12.5
1.372	500	8.655	1.862	4.65	41.2	3.9
	700	8.724	2.276	3.83	35.1	4.2
	1000	11.275	3.207	3.52	27.8	11.2

4.2 Refrigerant mass flow rate and Refrigeration capacity:

Mass flow rate of refrigerant through the system under steady state conditions has a clear trend with the change of capillary tube length and refrigerant charge as shown in “figure 4”. It increases sharply with the decrease in length of capillary tube and moderately with the increase of refrigerant charge in the system because of increased evaporation rate in evaporator with more filling of it with liquid refrigerant. Refrigeration rate is directly proportional to mass flow rate of refrigerant and hence follows the same trend as shown in “figure 5”.

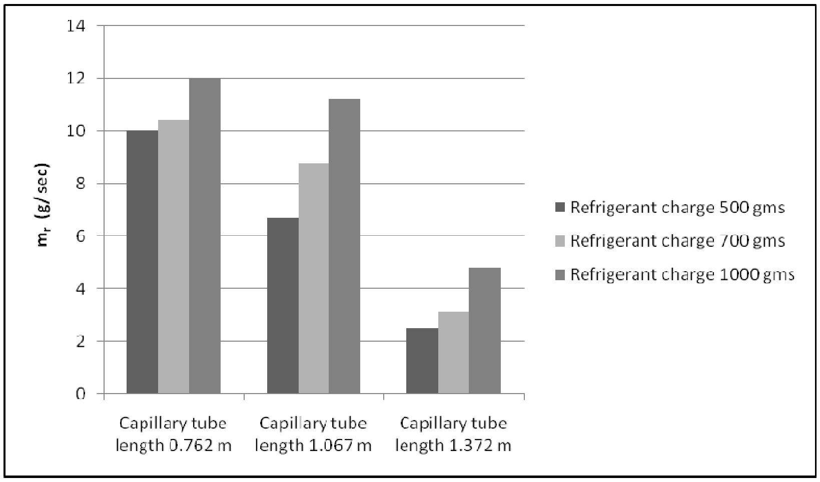

Figure 4. Mass flow rate of refrigerant (m_r) for different combinations of “capillary tube length” and “amount of refrigerant charged in the system”

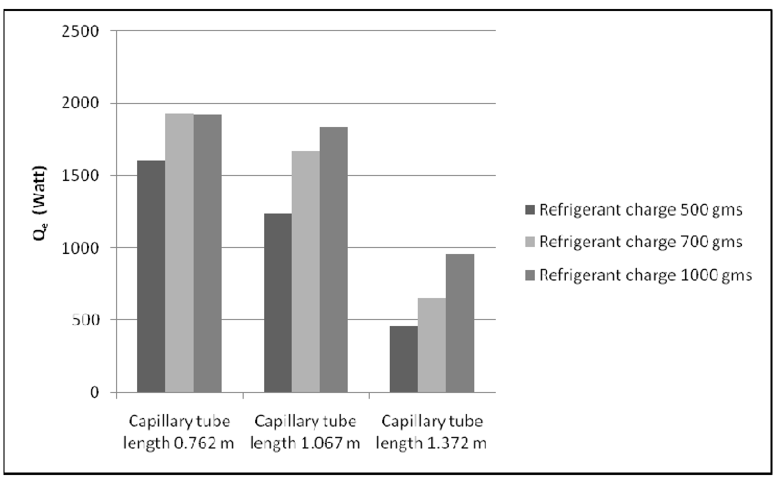

Figure 5. Rate of refrigeration (Q_e) for different combinations of “capillary tube length” and “amount of refrigerant charged in the system”

4.3 Overall heat transfer coefficient in the evaporator:

With the same size capillary tube, the overall heat transfer coefficient in the evaporator is raised considerably on increasing the refrigerant charge in the system as is clear from “figure 6”. This is solely because of increase in heat transfer coefficient on refrigerant side due to increased liquid fraction in the evaporator at a time. On filling of evaporator with liquid, pool boiling and nucleate boiling conditions prevail in maximum part of evaporator, which enhance the heat transfer multi times. Great decrease in heat transfer coefficient takes place with increase in length of capillary tube due to decreased mass flow capacity of compressor with increase in pressure ratio and decreased tendency of refilling of evaporator with liquid refrigerant through longer capillary tube. A wide variation in the data of overall heat transfer coefficient was noted while the water side coefficient and conduction resistance of wall are approximately constant in each trial. Therefore this variation is only in the heat transfer coefficient on refrigerant side. Highest value of overall heat transfer coefficient is in the case of shortest capillary tube with highest refrigerant charge in the system because here evaporator is expected most filled by liquid refrigerant. Lowest value of overall heat transfer coefficient (30 times less than the highest value) is in case of largest capillary tube with minimum refrigerant charge because here evaporator is expected most dry.

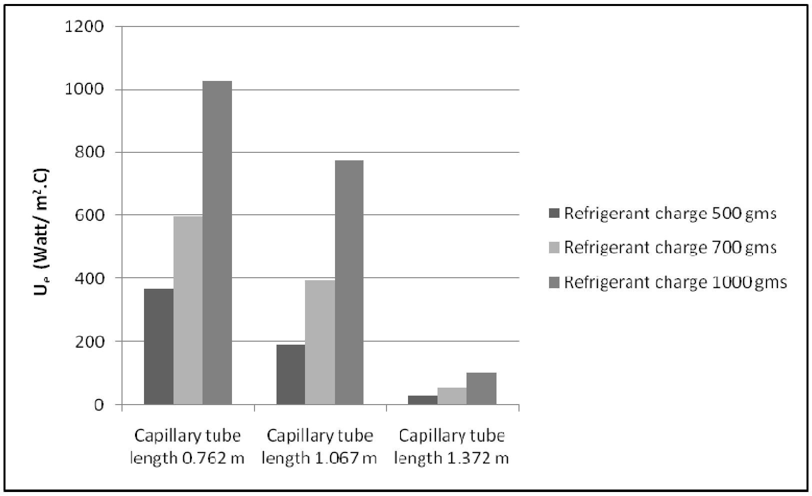

Figure 6. Overall heat transfer coefficient in the evaporator (U_e) for different combinations of capillary tube length" and "amount of refrigerant charged in the system"

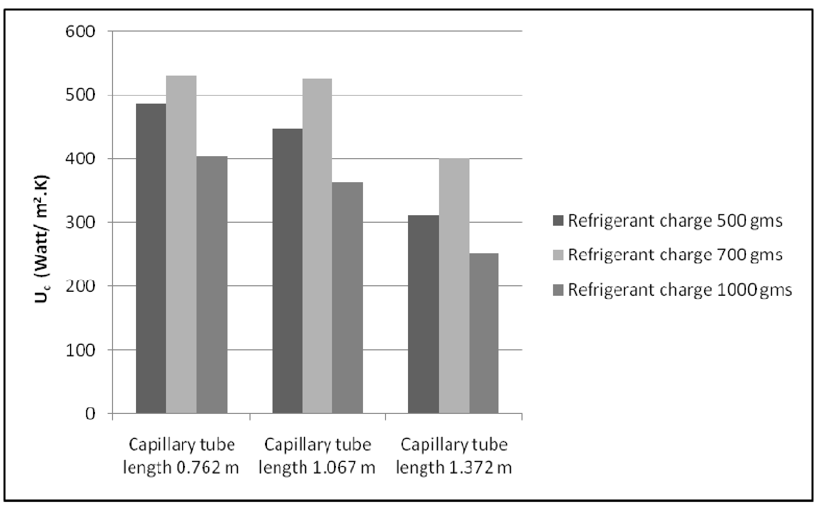

Figure 7 Overall heat transfer coefficient in the condenser (U_c) for different combinations of "capillary tube length" and "amount of refrigerant charged in the system"

4.4 Overall heat transfer coefficient in the condenser:

With the same amount of refrigerant charge, the overall heat transfer coefficient in the condenser decreases with increase in length of capillary tube as shown in "figure 7". It is obviously because of the sharp decrease in mass flow rate through condenser. This decrease is however not sharp because of opposite effect of simultaneous decrease in condenser pressure. Due to decrease in condenser pressure, temperature difference across condensing layer also decreases and latent heat increases, which increase the condensing coefficient as per Nusselt's well known equation for film wise condensation. With the same length capillary tube, value of condensing coefficient rises on increase of refrigerant charge in the system from 500 g to 700 g, but falls on increasing the charge from 700 g to 1000 g. First rise is due to increase in mass flow rate through condenser (as discussed before) which enhance the heat transfer coefficient. But simultaneously pressure in the condenser also increases exponentially, which poses the opposite effect and decreases the condensation coefficient in case of 1000 g refrigerant charge. So the correct combination of capillary tube length and refrigerant charge in the system is important rather selecting one individually.

4.5 Coefficient of Performance:

COP of a vapour compression refrigeration system is the single most important parameter which has to be optimized in a given refrigeration application for maximum conservation of energy. Highest COP is coming in case of smallest capillary tube of length 0.762 m and 700 gms of refrigerant charge as shown in "figure 8". Length of capillary tube decides evaporator pressure and temperature directly. Lesser is the length of capillary tube, higher are the evaporator temperature and pressure and so the COP if simultaneously these satisfy the required refrigeration capacity. But the role of refrigerant charge is also very important. More charge means more filling of evaporator with liquid so more refrigeration capacity until the limiting condition of liquid sucking by compressor is reached. In this way the refrigerant charge is very critical and its optimum value depends primarily on the length of capillary tube in a refrigeration system.

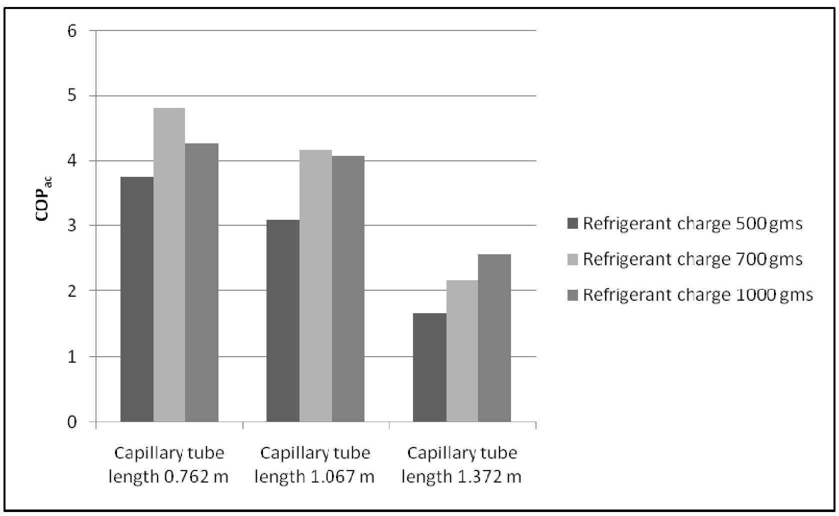

Figure 8 Coefficient of performance (COP) for different combinations of "capillary tube length" and "amount of refrigerant charged in the system"

V. CONCLUSION

This study offers some insight into the role of capillary tube length and refrigerant charge over the performance characteristics of a simple vapour compression refrigeration system. It was found that as the compressor is started, fast shifting of charge from evaporator to condenser via compressor and in a short while, again the comparatively slow refilling of this in the evaporator through capillary tube takes place. Refilling of evaporator with more and more liquid refrigerant causes multifold increase in heat transfer coefficient, which ultimately enhances the overall COP of the system but is limited by the undesirable condition of liquid sucked by compressor. The refilling of evaporator and the inventory of liquid charge in evaporator and condenser during working depends greatly on the refrigerant charged in the system and length of capillary tube. Dry out point in the evaporator can be shifted downstream by allowing more liquid to stay in the evaporator either by increasing the refrigerant charge or by cutting short the length of capillary tube. But simultaneously the length of capillary tube also decides the desired pressure in evaporator and other dependable parameters of the system and so cannot be altered much for the gain in heat transfer. In this way by managing the distribution of refrigerant liquid in condenser and evaporator by choosing optimum value of refrigerant charge, heat transfer coefficients on refrigerant side should be optimized. Based on the optimum value of overall heat transfer coefficients and required refrigeration capacity at given conditions, the evaporator and condenser can be designed on fixing the appropriate value of evaporator and condenser pressures. Thereafter the capillary tube should be designed and right compressor should be selected based on the designed value of pressures and mass flow rate of refrigerant. In the last, correct amount of refrigerant should be charged in the system to ensure optimum values of heat transfer coefficients and overall performance of the system. Further research work can be extended in the direction of correct estimation of refrigerant liquid inventory separately in the evaporator and condenser, for given amount of initial charge, during working of the system and establishing appropriate correlations of average value of refrigerant side heat transfer coefficients based on the known fraction of liquid and vapour refrigerant in the evaporator and condenser.

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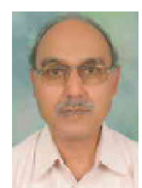

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