Performance Characteristics of "Vapour Compression Refrigeration System" Under Real Transient Conditions

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Abstract

The behavior of performance parameters of a simple vapour compression refrigeration system were studied while its working under transient conditions occurred during cooling of a fixed mass of brine from initial room temperature to sub-zero refrigeration temperature. The effects of different lengths of capillary tube over these characteristics have also been investigated. It was concluded that with the constantly falling temperature over evaporator, refilling of it with more and more liquid refrigerant causes multifold increase in heat transfer coefficient which helps in maintaining refrigeration rate at falling temperature. Larger capillary tube decreases the tendency of refilling of evaporator but offers less 'evaporator temperature' effective in lower range of refrigeration temperature. Shorter capillary tube ensures higher COP initially but which deteriorates at a faster rate in lower temperature range. Capillary tube length must be optimized for maximum overall average COP of the system for the complete specified cooling job.

Keywords: *Vapour compression refrigeration, transient, capillary tube, heat transfer coefficient, coefficient of performance*

1. Introduction

The performance of a simple vapour compression refrigeration system, used in numerous of small refrigeration applications all over the world, deteriorates in actual conditions due to internal and external irreversibility [1, 2, 3, 4] and further due to transient conditions over evaporator and condenser. [5] Internal irreversibility losses occur in the system mainly in the compressor, where entropy increases due to friction. These losses are taken into account in the form of isentropic efficiency of the compressor. Minimization of internal irreversibility is linked to the type and design of compressor selected. External irreversibility occurs over the heat exchangers (condenser & evaporator) due to finite rate of heat exchange against finite value of temperature difference and heat capacities. [6] More is the temperature difference between external fluid and refrigerant across evaporator and condenser, better would be heat transfer in evaporator and condenser and

Nomenclature:

A	surface area of tubes
C	specific heat
COP	coefficient of performance
h	specific enthalpy
m	mass flow rate
M	mass
P	pressure
Q	heat transfer rate
t	temperature
U	overall heat transfer coefficient
W	power consumption of compressor

Greek Symbols

τ	time point
η	efficiency
Δ	difference

Subscripts

ac	actual
b	brine
c	condenser
e	evaporator
i	inlet
isen	isentropic
m	mean
o	outlet
r	refrigerant
th	theoretical
w	water

More would be the cooling capacity of system. [7] But simultaneous decrease in theoretical COP of the system occurs due to wider gap between condenser and evaporator temperature because of higher value of temperature gradient across evaporator and condenser. [8] A vapour compression refrigeration system can be theoretically optimized and balanced under steady state conditions for its main design parameters as evaporator and condenser pressure based on the given refrigeration duty using finite time thermodynamics. [9] Simulation techniques have been used by researchers for design of vapour compression refrigeration system under steady state conditions. [10, 11] But practically a refrigeration system has to work under transient conditions. The conditions over condenser, however, may be considered steady state for a while but over evaporator, these remain transient in most of the refrigeration applications like domestic refrigerator, water cooler, cooling cabinet, cold storage, food storage locker and many more. In these applications, a fixed mass or space has to be cooled from some initial room temperature to final refrigeration temperature by direct or indirect contact with evaporator coil. Due to continuous fall of temperature of external fluid over evaporator, conditions surrounding evaporator does not remain steady and whole system works under transient conditions. With the decrease in difference of temperature of refrigerant and external fluid across evaporator w.r.t. time, the heat transfer across the evaporator tubes should also get reduced deteriorating the performance of system In this way a condition may come before the end of cooling job i.e. reaching of required refrigeration temperature, when the difference of temperature of refrigerant and external fluid over evaporator is too less to sustain a reasonable amount of heat transfer and evaporation rate in evaporator. The length of capillary tube mainly decides the evaporator and condenser pressure. Lesser the length higher is the evaporator pressure and vice versa. [12] The length of capillary tube may be theoretically optimized if the heat transfer conditions over evaporator and condenser and so the evaporation and condensation pressures are steady state. But under transient conditions, it will not work for the whole range of temperature in a given cooling job. During the transient operation, the heat exchangers (evaporator and condenser) are important in the system behavior because they retain initially almost all the system refrigerant charge. Generally, to investigate the dynamic behavior of refrigeration systems, usually the heat exchangers are treated by transient models while the expansion valve as well as the compressor is considered in steady state. [13] But need was felt to observe experimental values of refrigeration rate, power consumption, condenser duty, COP and overall heat transfer coefficient in condenser and evaporator and their variance with time and also to find the role of capillary tube length, while working of system under real transient conditions. This study is helpful in the design and balancing of components of a "vapour compression refrigeration system" for optimization of its performance in real life conditions.

2. Experimental set-up and procedures

The experimental set up of a simple vapour compression refrigeration system used in study is shown in "Fig.1". The evaporator and condenser are fabricated as shell and tube type adiabatic (insulated shell) heat exchangers. Refrigerant flows through tubes and external fluid remain in the shell of both of evaporator and condenser. Refrigerant HFC-134a flows through copper tubes of outside and inside diameters as 9.5 mm and 8.5 mm throughout including condenser and evaporator both. Supply water can flow through the shell of both of evaporator and condenser and there is arrangement for control and measurement of water flow rate. However, inlet and outlet valve of evaporator shell are closed and a fixed mass of brine is filled in the shell, which is cooled from initial room temperature to final refrigeration temperature during each of the trial. An agitator is also fitted in the evaporator shell to agitate brine for uniformity of its temperature through the shell.

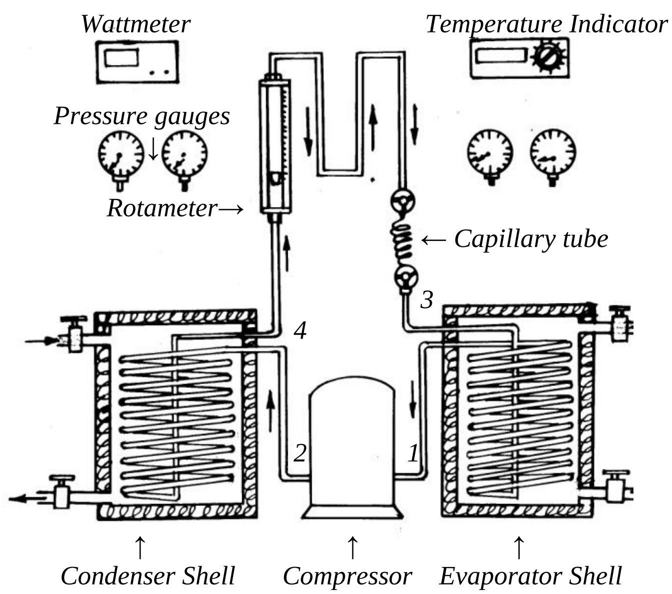

Figure 1 Experimental set up of simple vapour compression refrigeration system

Compressor used is Kirloskar Copeland model no. KCE444HAG (1/3 HP). Hand operated valves are provided before and after the capillary tube to facilitate easy replacement of different size and length of capillary tubes. The pressure and temperature readings of refrigerant were taken at four strategic points 1, 2, 3 & 4 as indicated in the theoretical and actual vapour compression cycle shown on pressure enthalpy chart in "Fig.2 and Fig 3" [11] and also in "Fig.1" of the actual experimental set up. The temperature 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). Temperature of cooling water flowing at the inlet and outlet of condenser shell and temperature of fix mass of brine filled in evaporator shell are also recorded in the same way. Pressure of refrigerant is indicated by pressure gauges. Mass flow rates of refrigerant liquid after condenser and of water through condenser are measured by respective rotameters. A digital wattmeter gives the instant value of power consumption of compressor. Many trials of cooling a fix quantity of brine in the evaporator from initial room temperature to final refrigeration temperature were taken with different size capillary tubes. Readings of various parameters at different positions in the system were taken at a number of time points, separated by a fix time interval during one working period of the machine between consecutive start and stop.

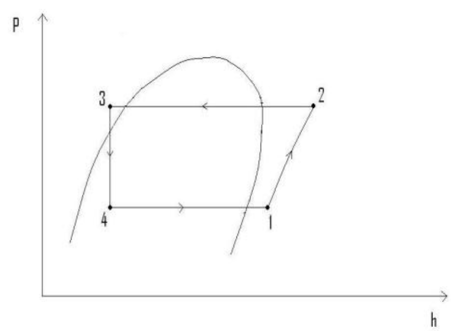

Figure 2 : Theoretical vapour compression cycle

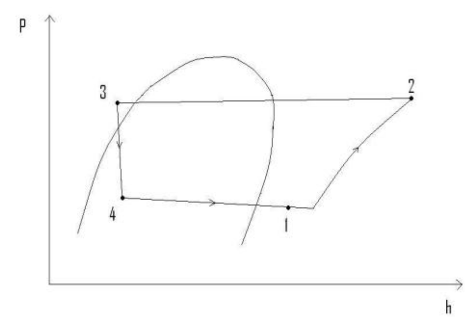

Figure 3: Actual vapour compression cycle

3. Data reduction

All the readings of pressure, temperature, flow rates, power etc. as suggested before are noted down at each time point 'τ' separated by fixed time interval 'Δτ' through the total time period of one trial. In one trial of system for cooling a fixed mass of brine from initial room temperature to final refrigeration temperature taking place under transient conditions, a large data was recorded. Many trials were conducted with different capillary tubes. All this data was transformed in the Excel worksheets. The properties of refrigerant are calculated using computer subroutines. [12, 13] Other performance parameters of the system are also calculated for each time point as per the procedure given below:

Heat transfer rate or refrigeration rate in evaporator at a time point 'τ' is given as:

$$Q_{e,\tau} = M_b c_b (t_{b,\tau-\Delta\tau} - t_{b,\tau}) = m_{r,\tau}(h_{1\tau} - h_{4\tau}) \quad \text{--- (1)}$$

Heat transfer rate in condenser at a time point 'τ' is given as:

$$Q_{c,\tau} = m_{w,\tau} c_w (t_{w,o,\tau} - t_{w,i,\tau}) = m_{r,\tau} (h_{2\tau} - h_{3\tau}) \quad \text{--- (2)}$$

Isentropic power consumption of compressor at a time point ‘τ’

$$W_{isen,\tau} = m_{r,\tau} (h_{2\tau} - h_{1\tau}) \quad \text{--- (3)}$$

Isentropic efficiency of compressor

$$\eta_{isen,\tau} = \frac{W_{isen,\tau}}{W_{ac,\tau}} \quad \text{--- (4)}$$

Theoretical COP at a time point ‘τ’ is

$$COP_{th,\tau} = \frac{h_{1\tau} - h_{4\tau}}{h_{2\tau} - h_{1\tau}} \quad \text{--- (5)}$$

Actual COP at a time point ‘τ’ is

$$COP_{ac,\tau} = \frac{Q_{e,\tau}}{W_{ac,\tau}} \quad \text{--- (6)}$$

Overall heat transfer coefficient over evaporator at a time point ‘τ’ is

$$U_{e,\tau} = \frac{Q_{e,\tau}}{(t_{b,\tau} - t_{r,e,\tau}) A_e} \quad \text{--- (7)}$$

Overall heat transfer coefficient over condenser at a time point ‘τ’ is

$$U_{c,\tau} = \frac{Q_{c,\tau}}{\Delta t_{m,c} A_c} \quad \text{--- (8)}$$

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

4. Analysis and result

4.1 Refrigeration Rate, Power Consumption and Condenser Duty

The variation in refrigeration rate of evaporator, work consumption of compressor and heat rejection rate of condenser with time from the start to stop of the VCR system is shown in “Fig. 4”. The refrigeration rate of evaporator and heat rejection rate of condenser drops almost parallel to each other, both with a decreasing rate with time. But the power consumption does not drop much. Cooling rate drops due to decreasing of temperature gradient across evaporator with more and more cooling of brine with the working of system itself.

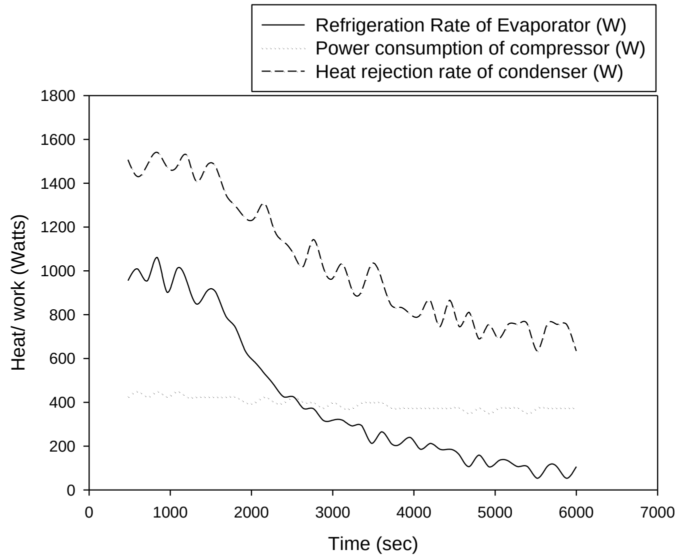

Figure 4 : Performance of VCR system under transient conditions

Due to this the evaporation rate of refrigerant in the evaporator and so the evaporator pressure also drops with time. But the compressor power consumption does not drop proportionately. It is simply because with the pressure drop in evaporator, the specific work consumption of compressor increases with the increase in pressure ratio. However, condenser pressure also tends to drop due to drop in refrigerant mass flow rate through compressor but this drop is counter balanced somewhat by decrease in condensation rate due to decrease in mass flow rate of compressor. Thus pressure drop with time in condenser is not as much significant as in the evaporator and pressure ratio rises. Refrigerant mass flow rate through compressor and capillary tube remain unbalanced under transient conditions and slowly the condenser becomes scarce and evaporator becomes flooded of liquid refrigerant which is evident from the data of enhancement of heat transfer coefficient in evaporator. The system must be stopped in this situation to avoid liquid suction by compressor. But by the time the required refrigeration temperature is also to be achieved. So here comes the role of capillary tube which decides the evaporator pressure and temperature in the system and so the temperature gradient necessary for heat transfer across evaporator and condenser.

4.2 Overall Heat Transfer Coefficient

The variation in value of overall heat transfer co-efficient, U_c and U_e in condenser and evaporator respectively with time from the start to stop of the vapour compression refrigeration system is shown in Fig. 5. The value of U_c slowly decreases with time but the value of U_e increases at a higher rate with time. The inlet temperature and mass flow rate of external fluid i.e. water, flowing over condenser remains same and also in evaporator, the external fluid conditions except temperature remain same throughout the working of system. But the refrigerant flow

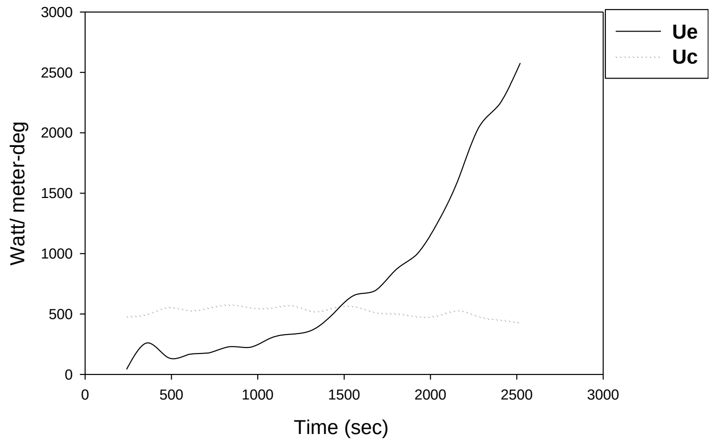

Figure 5 Variation in overall heat transfer coefficients, "Uc "over condenser and "Ue "over evaporator

conditions in evaporator and condenser are continuously changing under transient conditions as already discussed above. So, it can easily be judged that the variation in Uc & Ue is mainly because of the variation in "refrigerant side heat transfer coefficients" both in evaporator and condenser. In the condenser, refrigerant side heat transfer coefficient deteriorates because of reduction in compressor discharge as already mentioned. Refrigerant mass flow rate through compressor and capillary tube remain unbalanced under transient conditions and slowly the condenser becomes scarce and evaporator becomes flooded of liquid refrigerant. With more and more liquid accumulating in the evaporator, pool boiling conditions prevail in more space of evaporator. Multifold increase in refrigerant side heat transfer coefficient in evaporator proves this inference. The condenser should be designed as per the possible reduction in U_c in the later portion of one running cycle between start (cut-in) and stop (cut-out) points. The benefit of increase in U_e due to increasing level of liquid in evaporator may be taken only up to a limit because of simultaneous reduction in ΔT over the evaporator. Also the accumulation of liquid refrigerant in evaporator is limited by unwanted state of liquid entering the compressor.

4.3 COP variation with time and with different size capillary tubes

The comparison of variation in COP values of vapour compression refrigeration system with different size capillary tubes under transient conditions is shown in "Fig. 6". The sizes of four different capillary tubes used are given in table 1.

Table 1 Sizes of capillary tubes used in "Vapour compression refrigeration system".

Capillary Tubes	Available Sizes		Conversion in SI Units	
	Internal Diameter (Inches)	Length (Inches)	Internal Diameter (mm)	Length (m)
Tube 1	0.044	30	1.1176	0.762
Tube 2	0.044	33	1.1176	0.838
Tube 3	0.044	54	1.1176	1.372
Tube 4	0.05	48	1.27	1.219

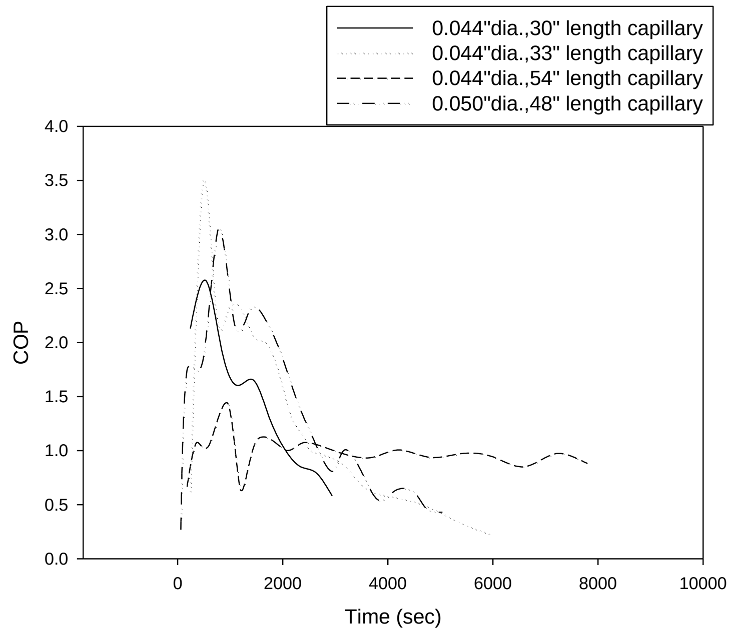

Figure 6 Variation in actual COP with different sizes of capillary tube under transient conditions

It is evident from the graph that initially at the start of refrigeration system, tube 2 gives higher value of COP, but it deteriorates much faster, later on due to much decrease in temperature gradient across the wall of evaporator tube. On the other side, capillary tube of larger length i.e. tube 3 gives however lesser COP value initially due to lower evaporator pressure but this value does not deteriorate much, because temperature gradient across evaporator is still maintained until the end of cooling job. Tube 1 gives cyclic average COP as 1.44 but the system is not able to achieve -10^{o}C temperature and had to be stopped at -3^{o}C as shown in figure7.

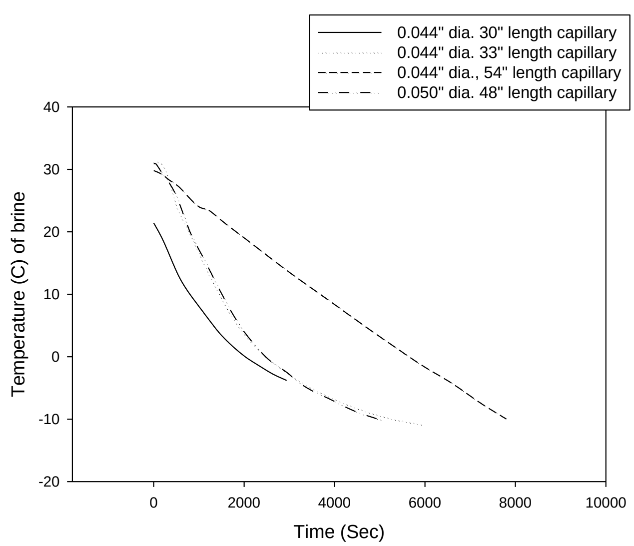

Figure 7 Fall of temperature of brine in evaporator shell with time

With tube 2, system is able to achieve the required temperature at a faster rate but has average cyclic COP as 1.17. With tube 3, system is achieving required low temperature at a constant rate and at almost constant COP value. But average COP value is 0.97 only. So the selection of capillary tube size and length is critical based on the specific cooling job i.e. range of fall in temperature of a fixed mass from given initial value to required refrigeration temperature value. The capillary tube size should be chosen for the system to give maximum value of overall average COP for complete working period between consecutive start and stop of the system.

5. Conclusion

In most of the practical refrigeration applications, a vapour compression refrigeration system works in transient conditions because the temperature surrounding evaporator and so the temperature difference across the wall of evaporator tubes falls continuously with the working of system itself. With this the heat transfer rate in evaporator or the refrigeration rate also falls but not with same pace. One reason behind is that as the heat transfer rate reduces, evaporation rate and so the pressure and temperature of refrigerant in the evaporator also reduces, which maintain the temperature difference causing heat transfer in evaporator. Another reason is that with the constant reduction in evaporator pressure during cooling of a fixed mass, more and more refrigerant liquid accumulates in the evaporator causing multifold rise in overall heat transfer coefficient. In this way the refrigeration sustains even at very low temperature of brine surrounding evaporator tubes because the system keeps on adjusting itself naturally under the transient conditions over evaporator. The pressure in evaporator and mass flow rate of refrigerant keeps on decreasing and the specific compressor work keeps on increasing. Thus the performance of system deteriorates continuously. The limiting condition is when the refrigerant flow rate reduces too much due to reduction in evaporation rate and evaporator tends to fill with refrigerant liquid. In this condition there may be the chances of liquid suction by compressor. The compressor must be stopped before this limiting condition but simultaneously the required temperature of brine must also be achieved. A capillary tube of shorter length ensure higher COP but the limiting condition would reach fast while increase in length of capillary tube delays the limiting condition but gives lesser COP. A vapour compression refrigeration system with given internal and external irreversibility can be balanced and designed optimally between static values of condenser and evaporator temperature. But in the transient condition stated above, system performance deteriorates from the optimum value. The capillary tube should be of minimum length to give maximum COP but it should also be sufficient to maintain the required low evaporator pressure and minimum value of condenser pressure in the later portion of cooling job. A refrigeration system should be designed more accurately by its simulation in real life transient conditions. However to find accurate transient models for each of the component is a challenging task. An alternate suggestion is to use a lesser length of capillary tube initially and then shifting to larger length later on with the help of some automatic temperature sensing device or valve.

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