Effects Analysis of Incorporating Suction Line Heat Exchanger (SLHX) Combined with Adiabatic Capillary Tube in Vapor Compression Refrigeration (VCR) System Using R-600a

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Abstract: This article experimentally analyses the effects of the incorporated Suction Line Heat Exchanger (SLHX) on the performance of vapor compression refrigerators to validate the theoretical relations. The set up uses concentric coiled SLHX combined with adiabatic Capillary tube. The model is based on mass, momentum and energy conservation equations on finite element procedure using R-600a as working fluid. This paper emphasizes on the target design of Domestic refrigerators incorporating such heat exchangers of any refrigeration capacity considering the effect of various geometric and operating parameters on performance of VCR. Here the eco-friendly refrigerant R-600a has additional advantage of low operating pressure range of 0.7 bar to 6 bar claiming for a small size compressor. In the present work the effect of incorporated SLHX on COP has been analyzed theoretically and experimentally and fond to increase by 7% and 3% respectively.

Keywords: Adiabatic capillary tube, COP, Fanno line, Heat load.

I. Introduction

In the ongoing competition of energy saving and high performance equipment design enormous works have came into reality in the service of the society in the fields of refrigeration and cryogenic applications by different researchers. This article emphasizes on the development and implementation of suction line heat exchanger in combination with adiabatic capillary tube in domestic refrigerators and expects the appreciation for competitive idea to the designers and manufacturers.

In most domestic refrigerators a capillary tube is used between the condenser outlet and evaporator inlet working as simple expansion device in Vapour Compression Refrigeration System (VCRS) to drop the pressure and the corresponding saturation temperature of refrigerant from condenser condition to the evaporator condition. Here as the liquid refrigerant from the condenser flows through the capillary tube, causes the expansion following the flash vaporization due to the drop of internal energy and hence enthalpy, so also with change of other state properties. During this expansion as the refrigerant temperature falls much below the ambient temperature, so the outer heat has the possibility to flow into the refrigerant reducing the heat extraction capacity from the refrigerating space. So the capillary tube is made adiabatic through thermal insulation.

Primarily the fresh condensate at high pressure and saturation temperature if expanded directly in the capillary tube it will require of very large length and also much flash vaporization which reduces the heat extraction capacity from the refrigerating space. Conversely the fresh vapour from the evaporator at very low temperature and corresponding saturation pressure enters into the compressor; it may claim for higher compression work and consequently reduces the COP. So this requires the heating and corresponding rise of pressure at the suction line of the compressor to reduce the compression pressure ratio and hence work requirement. This is compensated by the use of a suction line heat exchanger in between the evaporator and condenser to heat up the low temperature vapour by the consumption of heat from the fresh condensate. As a result the condensate is sub cooled that reduces the flash vaporisation and enhances the heat extraction capacity. In the overall the reduction of compression works and increases of heat extraction capacity improves largely the COP of the refrigerator.

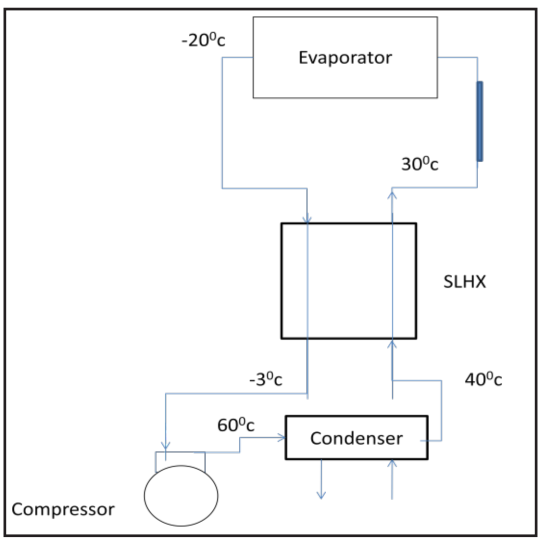
The article also includes an experimental analysis for validation of the theoretical result.

A. *Concept of Adiabatic Capillary Tube & Non-Adiabatic Suction Line Heat Exchanger (SLHX)*

A capillary tube used as expansion device for liquid refrigerant in VCRS if made completely insulated from the effects of hot surrounding so that no ambient heat is absorbed to prevent the desired expansion and saturation temperature drop, is called adiabatic capillary tube. It may be installed just after the condenser outlet line or after the Suction Line Heat Exchanger (SLHX) discharge.

Here the SLHX is a heat exchanger immediate between the high temperature condenser and low temperature for evaporator as shown in the Fig. 1 below and has the following functions.

1. It sub cools the condensed refrigerant at high constant pressure and delays the subsequent flashing during expansion in the capillary.
2. It also increases the heat extraction capacity of the refrigerant and hence the refrigeration capacity.
3. It superheats the vapor refrigerant from evaporator at very low pressure in compressor suction line and eliminates liquid deposition corrosion on the compressor.
4. As the vapor refrigerant passes through the confined suction line and simultaneously gains heat in the exchanger, its specific volume almost remains constant but the inlet pressure P_1 to the compressor increases. This reduces the compression pressure ratio (P_2/P_1) and increases the COP of refrigerator.



Schematic Diagram of VCR system with SLHX using R-600a

Fig. 1: Schematic Diagram for the Position of Adiabatic Capillary Tube & SLHX in VCRS

B. Classification of Non-Adiabatic Heat Exchangers (SLHX)

These are broadly classified into two types.

(a) Lateral type
(b) Concentric type

The transverse and longitudinal sections are as following:

(a) *Lateral Heat Exchangers:-* Here the capillary tube is soldered to the suction line pipe of the compressor. The vast majority of previous non-adiabatic capillary tube performance work was carried out with lateral capillary tube-suction line heat exchangers (Pate and Tree, 1984), (Bittle *et al.*, 1995) in despite of the widespread use of concentric heat exchangers in many parts of the world and proved less efficient than concentric heat exchangers due to less heat transfer area.

(b) *Concentric Heat Exchangers:* In concentric heat exchangers, the capillary tube passes inside the suction line pipe and brazed or soldered at the ends. The study presented herein focuses on a theoretical design and experimental evaluation of concentric capillary tube-suction line heat exchanger performance with R-600a.

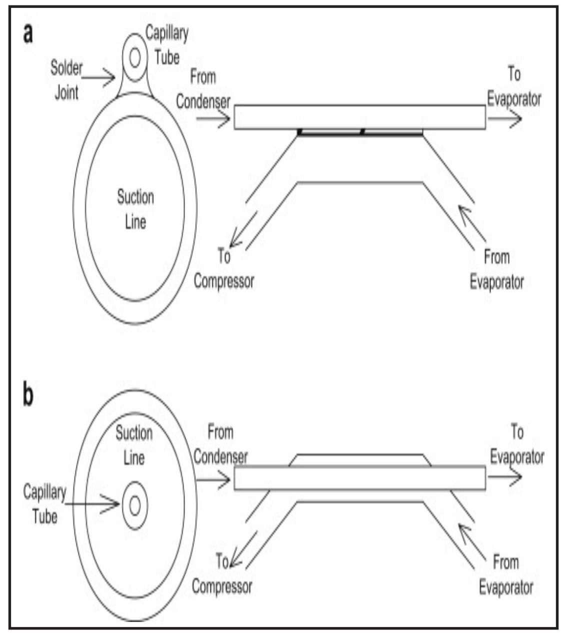


Fig. 2: Types of SLHX

II. Function Analysis of SLHX

The following figure shows the expansion of refrigerant in the capillary tube with suction line heat exchanger in which the condensate is sub cooled from saturation temperature 40^0C at point 3 to 30^0C at point 3’ by loosing heat to the suction line vapor of the compressor from the evaporator. Here the suction line vapor in the first sight is assumed to be super heated at constant pressure from point 1 to 1’ in the SLHX. But actually the annular space of the SLHX will act like a vapor reservoir for the compressor near its suction valve. Here its volume remaining constant as the saturated vapor gains heat, there is possibility of rise of pressure according to Amagat’s relation of gases. At very close to the vapor inlet of SLHX, the heat gain is negligible and the vapor enters from a narrow bored tube with high momentum into the large bore annular space causing considerable pressure drop due to sudden expansion. This prevents the reverse flow of vapor near the inlet. Again strong vacuum of the compressor pulls the vapor towards the outlet, although the heat gain and the pressure rise occurs there. This pressure rise causes a natural shifting of vapor condition towards the saturation line to the point 1’’. The line 1-2 represents the compression without using SLHX and that the line 1'- 2' represents compression considering without pressure rise in SLHX. But the actual compression is along1"– 2". So the actual compression work along 1"– 2" is expected to be lower than along 1-2 or 1'- 2'.

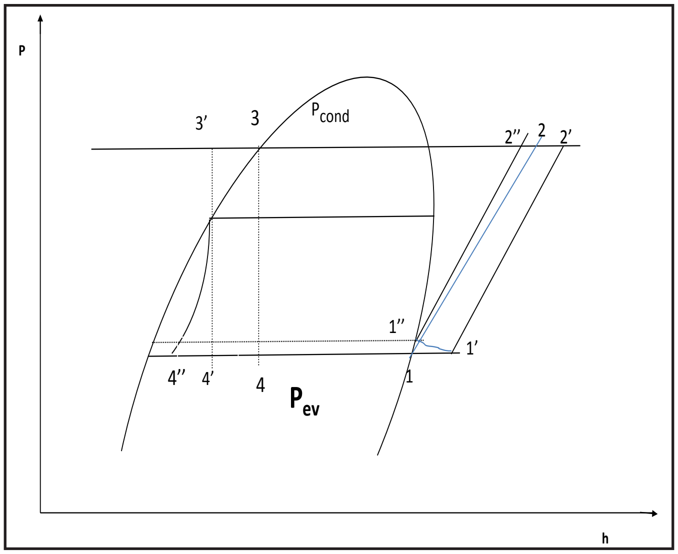


Fig. 3: Ph-chart Showing Expansion in Capillary with and without Subcooling in SLHX

Conversely the hot liquid refrigerant from the condenser as gets sub cooled from state 3 to 3' in the SLHX and then enter into the capillary tube. Here the iso-enthalpic expansion is expected to be along 3'- 4' instead of being along 3 - 4 and hence increases the heat extraction capacity as difference of h_1 and h'_4 than the difference of h_1 and h_4. Further the actual expansion along the Fanno line 3'- 4", considering momentum gain through consumption of own enthalpy still increases heat extraction between 1 and 4". So in overall the increase of heat extraction and decrease of compression work causes the rise of COP of the refrigerator.

A. *Heat Load Estimation for Concentric SLHX*

This paper aims to analyse a concentric SLHX for 0.1-tonne refrigerator maintaining -5^{0}C in the chilling chamber using R-600a refrigerant. Here the saturated vapor refrigerant from evaporator at about -20^{0}C as cold fluid enters the annular space of the concentric SLHX. The condensate at about 40^{0}C as hot fluid passes through the tube in counter current to the cold vapor and sub cools to 30^{0}C at saturation pressure corresponding to inlet 40^{0}C. From the energy balance between the hot and cold fluids it is found, the cold saturated vapor to be superheated to nearly -3^{0}C.

The necessary specifications for the heat exchanger are as following:
Refrigeration capacity, Q r = 0.1 - Tonne = 0.35 kW
Refrigerant is R-600a
Evaporator temp. T_{ev} = -50^{0}C

Refrigerant vapor temp. from evaporator,

Tci = 20^{0}C

Vapor outlet temp. of SLHX, Tco = -3^{0}C

Condensate temperature at inlet of SLHX,

Thi = 40^{0}C

This hot fluid outlet temp. Tho = 30^{0}C

The theoretical heat load for any fluid can be determined from the relation,

$Q = m \times Cpm \times \Delta T$

Where Cpm = Mean sp. heat within ΔT

(a) *Liquid Side Heat Load Calculation*:- For R-600a the liquid specific heat collected from Data book (VDI Heat Atlas page-416) and the liquid side heat load is calculated as in the following Table I.

TABLE I

Temp (°C)	Sp. Heat (kj./kg-k)	Mean Sp. Heat	Heat Exchange (kj./kg)
40	2.535		
30	2.469	2.502	25.02

(b) *Vapor Side Heat Load Calculation*:- For R-600a, a standard vapor specific heat relation at different temperatures was collected from PDF -Adobe Acrobat.

$$C_p = a + b \times T + c \times T^2 + d \times T^3$$

where $a = -7.913$, $b = 41.60 \times 10^{-2}$
$c = -23.01 \times 10^{-5}$ $d = 49.91 \times 10^{-9}$.

Using this relation, the vapor specific heats at different temperatures and vapor side heat load is calculated as in the Table II.

TABLE II

Temp (°C)	Sp. Heat (kj/kg-k)	Mean Sp. Heat	Heat Exchange (kj/kg)
-20	1.438		
-3	1.528	1.482	25.19

Taking the greater side heat exchange, the actual heat load of refrigerator corresponding to the actual mass flow rate of 0.00126 kg/s can be determined as:

$Q_E = 0.00126 \times 25.19 = 0.032$ kW = 32 W

So the heat load of SLHX = Q_E = 32 W

B. *Estimation of COP*

Assuming the use of reciprocating compressor, its work requirement:-

$W_c = (\mathbf{Y}/ Y\text{-}1)\ P_1\mathbf{v}_1\ [(P_2/P_1)^{Y-1/Y} - 1]$

For R-600a, characteristic gas constant

$R = R_u/M = 8.314/58 = 0.0744$ kj/kg-k

C. Considering no Sub-Cooling

The suction vapor temperature is -20^0C and corresponding inlet entropy s_1 =2.3530 kj/kg-k. Assuming isentropic compression the final temperature corresponding to the condenser saturation condition is found by interpolation from data Table as 45^0C. The mean compression temperature between -20^0C and 45^0C.

$T_m = 12.5^0C = 285.5$ k. At this mean temperature

$Cp = a + b \times T + c \times T^2 + d \times T^3$

$= -7.913 + 41.6 \times 10^{-2} \times 285.5 - 23.01 \times 10^{-5} \times 285.5^2 + 49.91 \times 10^{-9} \times 285.5^3 = 93.26$ kj/kmol-k = 1.60 kj/kg-k

Now $Cv = Cp - R = 1.608 - 0.0744 = 1.534$ kj/kg-k

$\mathbf{Y} = Cp/Cv = 1.608/1.534 = 1.049$

Here the condensed refrigerant directly enters the capillary at 40^0C and expands to -20^0C approaching the Fanno line enthalpy of 272.04 kj/kg. So refrigeration effect

$Q_r = 529.46 - 272.04 = 257.42$ kj/kg.

At suction temperature of -20^0c sp. Volume = $\nu_1 = 0.4895\ m^3/kg$

So $Wc = (1.049/0.049) \times 0.728 \times 10^5 \times 0.4895\ [(5.361/0.728)^{0.049/1.049} - 1]$

= 74.57 kj/kg

Now COP = Qr/ Wc = 257.42/74.57 = 3.45

D. Considering Sub Cooling by SLHX

Here, Qr = 529.46-252.24 = 277.22 kj/kg

Since the saturated vapor is superheated in the suction line from -20^0C to

-3^0C, the corresponding specific volume = $\nu_1 = 0.5071\ m^3/kg$

Entropy = s_1 = 2.4099 kj/kg-k

Assuming isentropic compression, the final superheating temperature corresponding to the saturation pressure at 40^0C can be found by data interpolation $T_2 = 54^0C$

The mean compression temperature between -3^0C and 54^0C

$Tm = 28.5^0C = 301.5^0k$, At this mean temperature

$Cp = a + b \times T + c \times T^2 + d \times T^3$

$= -7.913 + 41.6 \times 10^{-2} \times 301.5 - 23.01 \times 10^{-5} \times 301.5^2 + 49.91 \times 10^{-9} \times 301.5^3 = 97.962$ kj/kmol-k = 1.689 kj/kg-k

$C_v = Cp - R = 1.689 - 0.0744 = 1.6146$ kj/kg-k

$Y = Cp/Cv = 1.6890/1.6146 = 1.0461$

$W_c = (1.0461/0.0461) \times 0.728 \times 10^5 \times 0.5012[(5.361/0.728)^{0.0461/1.0461} - 1] = 76.152$ kj/kg

Now COP = Qr/ W_c = 277.22/76.152 = 3.64

This shows that COP is improved with use of SLHX on refrigerant R-600a nearly by 5% for 10^0C sub cooling.

III. Experimental Setup

The refrigeration system which is assembled with the following components:

(a) Compressor
(b) Condenser
(c) Capillary Tubes
(d) Evaporator
(e) Shell and Tube Heat Exchanger
(f) Pressure gauges
(g) Digital pyrometers

Fig. 4

The setup uses a shell and tube heat exchanger carrying spirally coiled tube inside the shell and has the following specifications.

Fig. 5: Shell and Coiled Tube SLHX

For the shell:- Outer diameter = 35.4 mm, Inner diameter = 30 mm

For the tube:- Outer diameter = 7 mm, Inner diameter = 5 mm

Mean diameter of the helix = 20 mm

IV. Experimental Procedure and Observations

For the experiment the water tank is filled with known volume of water. Initial readings of energy meter, water temperature and pressure gauges reading are noted. First one capillary passage is kept open with by-pass of heat exchanger. Then the electric supply to the compressor is made ON for a period of 30 minutes. The final readings of energy meter, different pyrometers and pressure gauges are noted for one set of calculation. Same procedure is repeated for other capillaries in both by-pass and through pass conditions of the heat exchanger.

Compression work = Wc = (E. meter final reading – initial reading) x 3600/1800 kW

Now COP $= Q_r / W_c$

Table III: For Observed Data without Heat Exchanger in 30 Minutes Observation Period

Capillary Length (m)	Ev. Inlet Temp (^{0}c)	Ev.Outlet Temp (^{0}c)	Initial Water Temp.	Final Water Temp.	Comp. Inlet Press.	Comp. Outlet Press.	Initial E.Meter reading	Final E.Meter reading
2.4	-3.5	17.3	27.6	19.0	1.8	14.0	2.0	2.10
3.0	-4.8	11.5	19.0	11.2	1.7	13.0	2.10	2.19
3.6	-4.6	7.4	11.2	5.0	1.5	12.8	2.19	2.26

Calculations:

Let m_w = mass of water in the evaporator tank

$\Delta T_w = T_{wf} - T_{wi}$

Now refrigeration effect $Q_r = 4.186 \times m_w \times \Delta T_w / 1800$ k-Watt

Here $m_w = \rho_w \times$ water volume $= 1000 \times (30 \times 30 \times 24/10^6) = 21.6$ kg

Table IV: Calculation on without SLHX

Capillary Length	Comp. Pressure Ratio	Qr Value In KW	Wc Value In KW	COP Value
2.4	7.78	0.431	0.200	2.155
3.0	7.61	0.391	0.180	2.172
3.6	7.53	0.311	0.140	2.220

This table shows that the increase of capillary tube length causes the gradual but slow decrease of compression pressure ratio and also compression work but increase of COP in the absence of SLHX

Table V: Calculation with SLHX

Capi. Length	Comp. Pressure Ratio	Qr Value In KW	Wc Value In KW	COP Value	% Increase of COP Value
2.4	7.30	0.446	0.201	2.221	3.0
3.0	7.12	0.412	0.183	2.255	3.8
3.6	7.01	0.397	0.173	2.298	3.5

The above calculation table shows that the incorporation of SLHX results in some decrease of compression pressure ratio due to increase of inlet pressure, but slight increase of compression work due to super heating and considerable increase in refrigeration capacity for sub cooing of condensate before entering into the capillary. This results in about 3% overall increase of COP.

V. Conclusion

The overall analysis on the basis of theory and experiments verified the use of SLHX enhances the Coefficient of Performance (COP) of vapour compression refrigerator which depends on the nature of the refrigerant, design of the SLHX and the operating conditions. This article, I expect will encourage the manufacturers and readers with new ideas to develop the field of refrigeration and cryogenics.

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