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PERFORMANCE EVALUATION OF WATER CHILLER USING LSHX UNDER VARYING LOAD AND SUCTION TEMPERATURE

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Abstract

This paper considers the influence of heat exchangers to the efficiency of water chillier. A tube and tube type heat exchanger is used to compare the coefficients of performance of the vapor compression system with and without liquid suction heat exchanger. To change the operating suction temperature (evaporator temperature) the application of three different capillary tubes is to be done. The approximate temperature range will be +50C to -50C, in three steps. And performance of liquid suction heat exchanger (LSHX) will be evaluated for present water chiller. Again the evaporator will be loaded at different masse of water and with this change in load on evaporator the performance of LSHX is studied. With these varying conditions the performance of LSHX is to be evaluated with use of change in coefficient of performance, operating condenser and evaporator pressure and temperatures. The expected results are raise in COP with 6 to 9%, increase in cooling capacity.

I. INTRODUCTION

''Heat exchanger'' is a common device, in which heat interchange between a refrigerant in a suction line and a liquid refrigerant in a capillary tube occurs. It is used in most household systems in order to ensure the necessary superheat at the compressor's inlet section, while promoting certain sub cooling before the capillary tube.

The performance of Rl34a water chiller systems has been continuously improved over the last years. However, the potential benefits associated with the implementation of a liquid suction heat exchanger (LSHX) - also termed internal heat exchanger - have not been paid much attention according to the published literature. A LSHX transfers energy from the refrigerant leaving the condenser to the suction gas, resulting in a lower inlet enthalpy at the evaporator, providing a higher cooling capacity. The competing effects with respect to performance are a larger enthalpy change across the compressor and a lower mass flow rate, both effects caused by the lower suction density. An Rl34a prototype system was operated with and without a LSHX, the performance comparison is presented in this paper.

The most common in household refrigerators is the tube-in-tube heat exchanger with a capillary tube placed concentrically inside the suction tube. Another widespread design is the heat exchanger with a capillary tube welded to the outer surface of the suction tube. Heat exchangers of both types have shortcomings a heat exchanger with concentric tubing has at least two additional tube junctions and is less efficient. A heat exchanger with a capillary welded to the outer surface of the suction tube is higher in price. Such heat exchangers prevail in freezers. Usually they are made by specialized manufacturers as one piece together with evaporator.

This paper presents a relatively new heat exchanger, which is similar to the tube and tube type one from a thermodynamic point of view, but has advantages in some cases. The copper fins are mounted on inner tube to increase heat exchange area and to maintain equal-space between two tubes along length of heat exchanger.

II. LITERATURE REVIEW

S.A. Klein, D.T. Reindl, K. Brownell have studied Refrigeration system performance using liquid-suction heat exchangers in 2000 and found that liquid-suction heat exchangers increase the temperature and reduce the pressure of the refrigerant entering the compressor causing a decrease in the refrigerant density and compressor volumetric efficiency.[1]

Consequently, the refrigerant flow rate decreases with increasing effectiveness of the liquid-suction heat exchanger. The presence of a liquid-suction heat exchanger produces opposing effects on refrigeration capacity. The refrigerating effect per unit mass flow rate increases due to an increasing enthalpy difference across the evaporator (as seen in Figure 1; however, the mass flow rate itself decreases due to the effects of decreasing suction density resulting from increased temperature and reduced pressure at state 2 when pressure losses in the heat exchanger are considered. The net effect of the liquid-suction heat exchanger on the relative capacity index for eleven refrigerants at a saturated evaporator temperature of -20^{0}C and a saturated condensing temperature of 40^{0}C is shown in Figure 1.

From detailed analyses, it can be concluded that liquid-suction heat exchangers that have a minimal pressure loss on the low pressure side are useful for systems using R507A, R134a, R12, R404A, R290, R407C, R600, and R410A. The liquid-suction heat exchanger is detrimental to system performance in systems using R22, R32, and R717 [1].

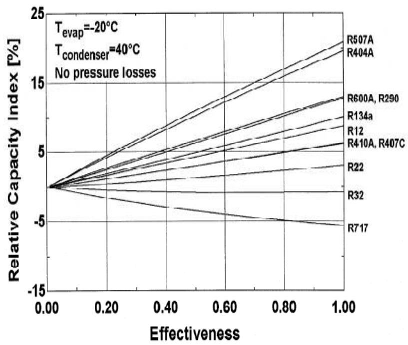


Figure 1: Effectiveness of LSHX vs. Relative capacity index for different refrigerants.[1]

M. Preissner, B. Cutler, R. Radermacher, C. A. Zhang have studied Suction Line Heat exchanger for R134a Automotive Air-

Conditioning System in 2000 and found that the COP and the capacity increased on the order of 5 to 10 % with a suction line heat exchanger with 60 % effectiveness.[2]

They also found that the performance decrease associated with an additional pressure drop at the low pressure side, calculations were carried out with a program based on fitted cycle parameters from the experimental data set, the penalty of the pressure drop on the low pressure side of the suction line heat exchanger is shown in Figure 2. A 5 bar pressure drop reduces capacity and COP by about 10%. [2]

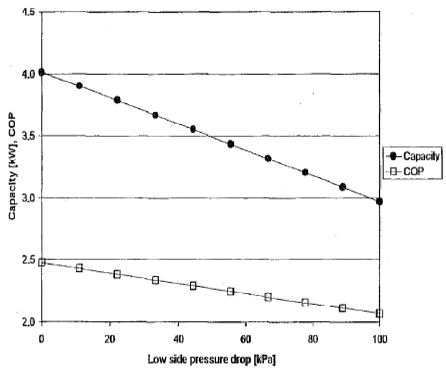


Figure 2: Performance penalty of pressure drop at the suction side. [2]

V.Dagilis, L.Vaitkus, A.Balcius has studied Liquid–gas heat exchanger for household refrigerator in 2004. They performed experiment with three type of LSHX, mostly common in household refrigerators are the tube-in-tube heat exchanger with a capillary tube placed concentrically inside the suction tube. Another widespread design is the heat exchanger with a capillary tube welded to the outer surface of the suction tube. [3]

And found that the heat exchanger with concentric tubing is much more sensitive to lower than optimal inner diameter of the suction tube. For a refrigerating system with this heat exchanger, the inner diameter, lower than the optimal by 30%, will cause a decrease of COP by 7%, which is unacceptable. All types of heat exchangers are less sensitive to higher than optimal inner diameter. Therefore, if a tube with an optimal diameter is not available, it is preferable to use a tube with a higher diameter.[3]

R. Mastrullo, A.W. Mauro, S. Tino , G.P. Vanoli has studied possible advantage of adopting a suction/liquid heat exchanger in refrigerating system in 2007 and found that its use can improve or decrease the system performance depending on the operating conditions.[4]

In Figure 3A and 3B, the COP'/ COP ratio for different two refrigerants (R-717 and R-134a) is presented. It's possible to see that the addition of the LSHX is not convenient for the R-717, for the all considered operating conditions; on the other side for the R-134a it is always advantageous [4].

Erik Bjork, Bjorn Palm has studied Performance of a domestic refrigerator under influence of varied expansion device capacity, refrigerant charge and ambient temperature in 2006. The method outlined in the introduction on how to determine the capillary tube length and quantity of charge can be recommended as it takes different thermal loads and thermal masses (food in cabinet) into consideration.

They also studied the influence of LSHX and found that Suction line heat exchangers (LSHX) are typically used in domestic refrigeration.

They increase the total efficiency and prevent external condensation on the suction line between the LSHX and the compressor. The overall influence from LSHX is that it increases the capillary tube capacity, lowers the evaporator inlet quality and increases the suction line temperature (at compressor inlet).

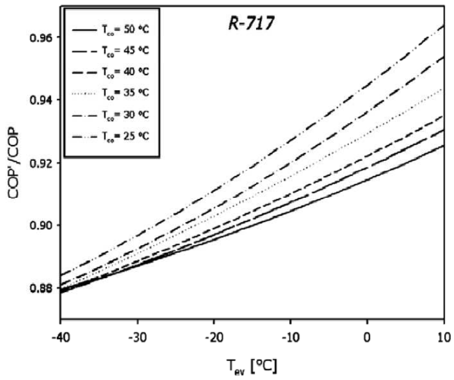


Figure 3A: The COP'/COP ratio for R-717, for various condensing temperature and evaporating temperature [4]

As a result the optimum capillary tube length needs to be longer and the charge possibly a little higher (the LSHX allows the dry-out point closer to the evaporator outlet.)[5]

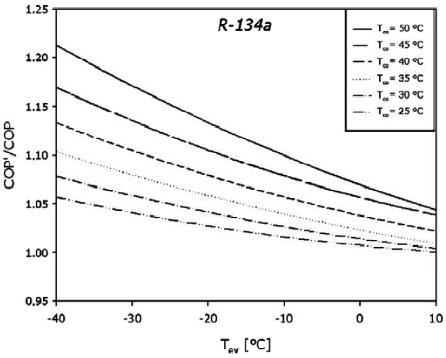


Figure 3B: The COP'/COP ratio for R-134a, for various condensing temperature and evaporating temperature [4]

R. A. Peixoto and C. W. Bullard has studied that application of capillary tube in the liquid suction heat exchanger will give better results with R134a rather than R22, due to its lower specific volume. They also found that the capillary tube LSHX have lower results than the lateral LSHX, due to decreases of flow area in suction tube. They also found that the length of LSHX with capillary tube is much long than lateral LSHX. The compressor volumetric efficiency is higher for R134a than R22 as the density of R134a is more than R22, also the mass flow rate required for R134a is less than R22 because of its higher latent heat than R22 at same saturation evaporator temperature [6]

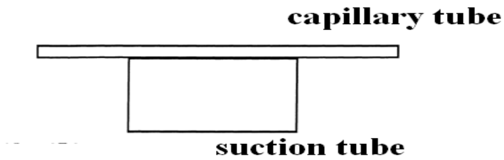


Figure 4A: lateral capillary tube LSHX [6]

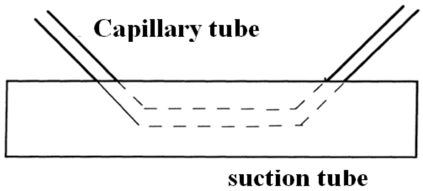


Figure 4B: concentric capillary tube LSHX [6]

III. SYSTEM DESCRIPTION

3.1 Compressor selection:

Cooling capacity calculations:

Let suppose, we want to cool 10 kg of water in ½ hr from 30^0C to 5^0C,

Therefore cooling load will be,

Qe=m*Cp*ΔT/1800………… (kW)

=10*4.2*25/1800…….. (kW)

=1050/1800…….……… (kW)

=0.5833……………….. (kW)

=0.6…………….……… (kW)

But, 1 kW cooling= 3418.80 BTU/hr cooling.

Therefore, 0.6kW cooling=2051 BTU/hr.

Therefore, selection of compressor from catalogue of "Emerson compressors".

Model-KCE444HAG

Refrigerant R134a.

Cooling capacity 1720 BTU/hr

Input power 450W

3.2 Condenser selection

Heat rejection through condenser is addition of heat supplied through evaporator and work done in compressor.

Qc= Qe + Wc

=0.6 + 0.45 kW

=1.05 kW

Therefore, selecting a condenser of 1.5 kW which is suitable for system.

Thermal model

The model is based on the mass, momentum and Energy conservation equations applied to both the refrigerant and air streams. The model predictions were compared with experimental data taken at several operating and geometric conditions.

The thermal sub-model was divided into two domains namely, air and refrigerant streams. The thermal resistances due to heat conduction through the tube and fin walls were neglected due to the relatively high thermal conductivity of the walls in comparison to the external convection heat transfer.

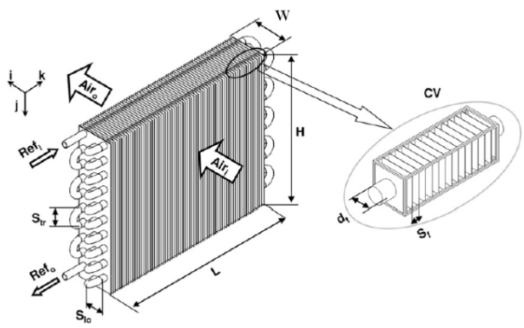


Figure 5: Schematic representation of the heat exchanger discretization. [7]

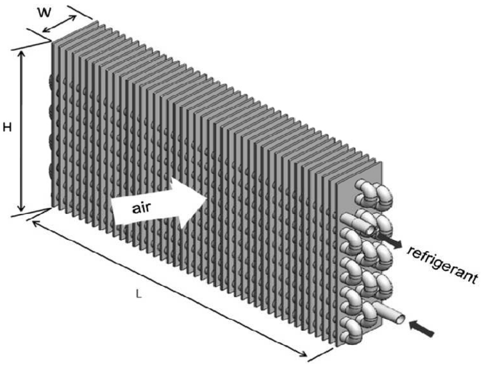


Figure 6: Tube-fin type condenser [7]

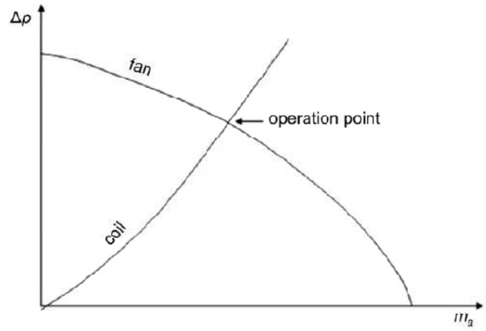


Figure 7: Schematic representation of the fan-coil hydrodynamic interaction [7]

Therefore, the air and refrigerant streams were modeled based on the following energy balances applied to the control volume illustrated in Figure 5.

$$m_r(h_{i,r} - h_{o,r}) + Q_{cv} = 0 \quad ($$

$$m_{a,cv}c_{p,a}(t_{i,a} - t_{o,a}) - Q_{cv} = 0 \quad ($$

where m_r and m_a,c_v are the refrigerant and the air mass flow rates through the control volume [kg/s], respectively, *h* is the refrigerant specific enthalpy [J/kg], and the indices *i* and *o* refer to the inlet and outlet ports of the control volume, respectively. The heat transfer rate Q_{cv} was calculated from the concept of heat exchanger effectiveness, as follows:

$$Q_{cv} = \pm\varepsilon C_{\min}(t_{i,h} - t_{i,c}) \quad (3)$$

where "+" should read "_" when the refrigerant transfers heat to the air stream (condensers and gas coolers) and "+" when the refrigerant receives heat from the air stream (evaporators). C_{min} = min ($m_r c_{p,r}$ OR m_a, $c_{p,a}$) is the lowest thermal capacity [W/K] of the streams, and $t_{i,h}$ and $t_{i,c}$ are the temperatures of the hot and cold streams at the entrance ports, respectively. The control volume effectiveness (ε) for a mixed, cross-flow, single-pass heat exchanger with single phase refrigerant was calculated.

$$\varepsilon = 1 - \exp\left(NTU^{0.2} C_r^{-1}\left(\exp\left(-C_r NTU^{0.78}\right) - 1\right)\right)$$

(4a)

Where $C_r = C_{min}/C_{max}$, and $NTU = UA/C_{min}$ is the number of transfer units. In the two-phase flow regions, the control volume effectiveness was calculated as follows:

$$\varepsilon = 1 - \exp(-NTU) \quad (4b)$$

The thermal conductance UA was obtained from

$$UA^{-1} = (\alpha_r A_r)^{-1} + \left(\alpha_a\left(A_t + \eta_f A_f\right)\right)^{-1}$$

(5)

where η_f is the fin efficiency calculated from the procedure introduced by Schmidt. Eqs. (1) to (5) were solved following the refrigerant coil arrangement (see Figure 5) assuming that each control volume behaves as an individual heat exchanger. [7]

Thus, although the air and refrigerant streams were modeled as one-dimensional, the three dimensional nature of the coil circuit was retained by the model. The heat transfer coefficients required by the model were obtained from empirical correlations. The air-side heat transfer coefficients were selected based on a study performed using a wind-tunnel facility, whereas the single and two-phase refrigerant side heat transfer coefficients were obtained from Gnielinski and Bassi and Bansal correlations, respectively. The air and refrigerant thermodynamic and thermo physical properties were calculated through the REFPROP7 software linked to the EES platform. [7]

Experimental apparatus

Experiments were carried out with a series of tube-fin heat exchanger samples particularly designed for light commercial refrigeration applications. The tests were performed using a closed-loop wind-tunnel calorimeter facility specially constructed for testing tube-fin heat exchangers according to the ANSI/ASHRAE 33 standard. The wind-tunnel comprised of a variable-speed radial fan to control the air flow rate. Compact fan-supplied tube-fin heat exchangers for light commercial refrigeration applications, i.e., with heat duties ranging from 0.5 to 2.0 kW are listed in table 1.

Table 1: Geometric characteristics of the heat exchangers. [7]

Sample	1	2	3	4
Ff (mm)	2.53	3.88	2.86	3.04
Str (mm)	25.6	25.5	25.5	25.6
Slo (mm)	21.67	20.33	20.33	22.5
dt (mm)	9.5	9.5	9.5	9.5
δf (mm)	0.14	0.14	0.14	0.14
L (mm)	304	304	304	304
H (mm)	256	153	153	256
W(mm)	65	61	61	45

The experimental apparatus was designed for air flow rates ranging from 170 and 2000 m^3/h, refrigerant mass flow rates up to 250 kg/h, and refrigerant pressures up to 20 bars.

Specifications of selected condenser-

Copper tube length (l) =8 m,

Copper tube diameter (d_t) =9.5 mm.

Fin pitch (F_f) =2.48 mm.

Material of fin is aluminum.

Condenser height (H) = 250 mm

Condenser length (L) = 270 mm and

Condenser width (W) = 50mm.

3.3 Design of heat exchanger:

We have a set of reading from ordinary water chiller, which is having same specifications as our selected compressor (Table 2).

A practical model of the heat exchanger was formed by taking into consideration the following assumptions:

1. Refrigerant vapor in the suction tube is superheated.
2. Heat exchange with environment is neglected.
3. Heat flow is considered as steady state.
4. A refrigerant in the capillary tube is in liquid state.

Now, from readings of table no. 2 it is clear that we can reduce the condenser outlet temperature around 35^0C and the heat can be rejected to vapour coming from evaporator. Therefore allowable temperature drop is around 25^{0}C.

Table 2: Readings set for ordinary water chiller.

Time	10:37	11:09	11:29
Suction Pressure P1 (bar)	2.6	2.2	2
Discharge pressure P2 (bar)	15.6	14.2	13.3
Suction temperature T1 (^{0}C)	17.1	12	2.8
Discharge temperature T2 (^{0}C)	55.3	68.5	69.3
Condenser outlet temp. T3 (^{0}C)	46.3	45.7	43.1
Capillary outlet temp. T3 (^{0}C)	9.5	4.9	1.5
Water temperature T5 (^{0}C)	25.2	12	10
Supply Voltage V (Volts)	215	214	213
Supply current I (Amp)	2.8	2.63	2.4

Therefore, heat exchanged

heat exchanged $Q= m*Cp_l*\Delta T$ kW

Heat exchanged Q= 0.1 kW

Now, heat transferred in LSHX is,

$$Q= h*A*\Delta T_{LM}$$

$$100=450*A*17$$

$$A=0.013 \quad m^2$$

As our LSHX is a tube and fin type of heat exchanger.

Therefore, total heat exchanging area equal to area of tube and area of fin. The fin is a helical wound of a copper strip having 4mm height.

Therefore, total heat exchanging area is

$$A= 2*\Pi*r*l + [2*n*((2\Pi r)^2+ (l/n)^2))^{1/2}*h] \text{ (in } m^2)$$

Where, radius of inner pipe r = 0.00325 m

Number of turns of copper fin ‘n’

Length of heat exchanger l in meter

Height of fin, h = 4mm

We have performed iterations to get optimized number turns of fin and length of heat exchanger.

Therefore, selecting a heat exchanger which is having 10 turns of fin over 44 cm tube length from table no. 3.

Table 3: Relation between 'n' and 'l'.

Number of turns 'n'	Length 'l' (m)	n/l
10	0.44	23
15	0.42	36
20	0.40	50
30	0.35	86
40	0.29	138

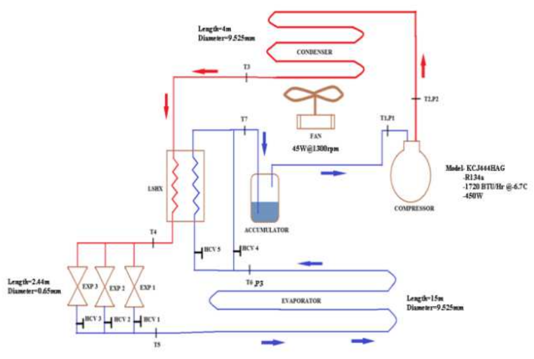
3.3 Proposed experimental setup:

The LSHX was tested in the Rl34a water chiller test facility. The evaporator is placed in a water tank of variable capacity.

The temperature of water is recorded with use of thermocouple. With the use of quantity of water used and the temperature drop reading, refrigerating effect can be calculated.

Condenser is exposed to open atmosphere and it is forced cooled. Two thermocouples and to pressure gauge are placed across the condenser, to get the reading of temperature and pressure drop across the condenser.

The schematic setup of the R134a loop is shown in Figure 8. The compressor is placed in the same climate chamber as the condenser, and consequently, is exposed to the condenser air inlet temperature. A digital voltmeter and ammeter are joined to compressor electrical supply lines, to get power supplied to compressor.



PROPOSED EXPERIMENTAL SETUP

Figure 8: Proposed experimental setup

The refrigerant leaving the compressor passes the condenser and can then be routed through the liquid suction heat exchanger. The suction line heat exchanger transfers heat to the suction gas, providing more cooling capacity and improving the cycle COP.

We have three capillaries placed, with different capacities. After passing the expansion device (manually controlled), the refrigerant evaporates in the evaporator and passes through the LSHX and enters the accumulator. Liquid refrigerant and oil are stored in this device a small amount of the oil-refrigerant mixture passes through a bleed hole at the bottom inside of the accumulator to the main refrigerant line to ensure proper compressor lubrication. The refrigerant loop is equipped with in-stream thermocouples and pressure gauges at the inlet and outlet of the main components.

IV. SYSTEM ANLYSIS

4.1 Analysis of COP:

From above observation table we can determine the COP of system for a particular load and suction temperature with and without LSHX.

Let's assume,

COP= coefficient of performance without LSHX

COP'= coefficient of performance with LSHX

Therefore we can have graphs of

A] COP' vs. suction temperature at different loads. (With LSHX)

B] COP vs. suction temperature at different loads. (Without LSHX)

C] COP' vs. load on evaporator at different suction temperature. (With LSHX)

D] COP vs. load on evaporator at different suction temperature. (Without LSHX)

With this other properties like compressor power, refrigerating capacity, pressure drop across LSHX can be studied.

V. CONCLUSION

From the available references, it is seen that there are chances of improvement in the COP of the system with the use of the LSHX system.

The effectiveness of LSHX system can be improved by proper selection and design of the LSHX.

The effect of the LSHX system on the performance of the water chiller system can be studied by variation in the operating conditions of the system.

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