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EVALUATION OF SUCTION LINE - LIQUID LINE HEAT EXCHANGE IN THE REFRIGERATION CYCLE

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ABSTRACT

The paper presents a theoretical evaluation of the performance effects resulting from the installation of a liquid line/suction line heat exchanger (llsl-hx). It examines cycle the parameters and refrigerant thermodynamic properties that determine whether the installation results in improvement of COP and volumetric capacity. The study showed that the benefit of application of the llsl-hx depends on a combination of operating conditions and fluid properties - heat capacity, latent heat, and coefficient of thermal expansion - with heat capacity being the most influential property. Fluids that perform well in the basic cycle are marginally affected by the llsl-hx, and the impact on the Coefficient of Performance and volumetric capacity may be either positive or negative. Fluids performing poorly in the basic cycle benefit from the llsl-hx installation through increase of the Coefficient of Performance and volumetric capacity.

NOMENCLATURE

β = $(1/v) \cdot (\partial v/\partial T)_p$, coefficient of thermal expansion
 COP = Coefficient of Performance
 c_p = heat capacity at constant pressure
 h_{fg} = latent heat of evaporation
 P = pressure
 q = evaporator capacity per unit mass of circulating refrigerant
 q_v = volumetric capacity (q/v at suction)
 s = specific entropy
 T = temperature
 v = specific volume
 WM = molecular weight
 w = work of compression per unit mass of circulating refrigerant
 γ = c_p/c_v , isentropic coefficient
 ϵ = $100\%(T_1' - T_1)/(T_3 - T_1)$, effectiveness of the llsl-hx

Subscripts:

c = condenser
 cr = critical
 e = evaporator
 l = liquid
 nb = normal boiling point
 r = reduced
 v = vapor or volumetric
 1-4 = key locations in a cycle, per Figures 1 and 2

Superscript ' denotes the llsl-hx cycle.

INTRODUCTION

Among many possible variations of the basic refrigeration (vapor compression) cycle, the cycle with the liquid line/suction line heat exchanger (llsl-hx) is used probably most often. As a result of employing this intra-cycle heat exchange, the high pressure refrigerant is subcooled at the expense of superheating the vapor entering the compressor. Schematics of hardware arrangement for the basic cycle and cycle with the llsl-hx are shown in Figure 1; the realized cycles are outlined on the pressure-enthalpy diagram shown in Figure 2.

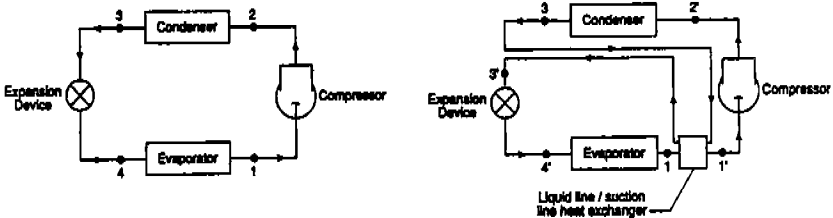


Figure 1. Schematic of hardware arrangements for the basic cycle and cycle with the liquid line/suction line heat exchanger.

The use of liquid line/suction line heat exchangers is widespread in commercial refrigeration. The heat exchangers are often employed as a means for protecting system components, by helping to ensure single-phase liquid to the expansion device and single-phase vapor to the compressor. In residential refrigerators, a capillary tube/suction line heat exchanger is used to heat the suction line above the dew-point temperature of ambient air, thus preventing condensation of the water vapor on the outside of suction line.

Employing an intra-cycle heat exchanger alters refrigerant thermodynamic states in the cycle, which may have significant, positive or negative, performance implications. For any fluid and system, a lsl-hx increases refrigerant temperature at the compressor inlet and outlet, which is a shortcoming. The Coefficient of Performance (COP) and volumetric capacity may increase for some fluid/application combinations, while for others they may decrease.

Figure 3 provides an example of the impact of the lsl-hx on COP, evaluated by theoretical calculations for five different refrigerants. The ordinate is a ratio of COP obtained for a given effectiveness of the lsl-hx (ϵ) and COP obtained for the same fluid working in the basic cycle ($\epsilon=0$). For R134a, the increase of COP was the highest, as much as 9.1% at the theoretical limit of 100% effectiveness of the heat exchanger. On the other hand, at low values of the lsl-hx effectiveness, R22 showed degradation in performance, some of which was recovered at 100% effectiveness.

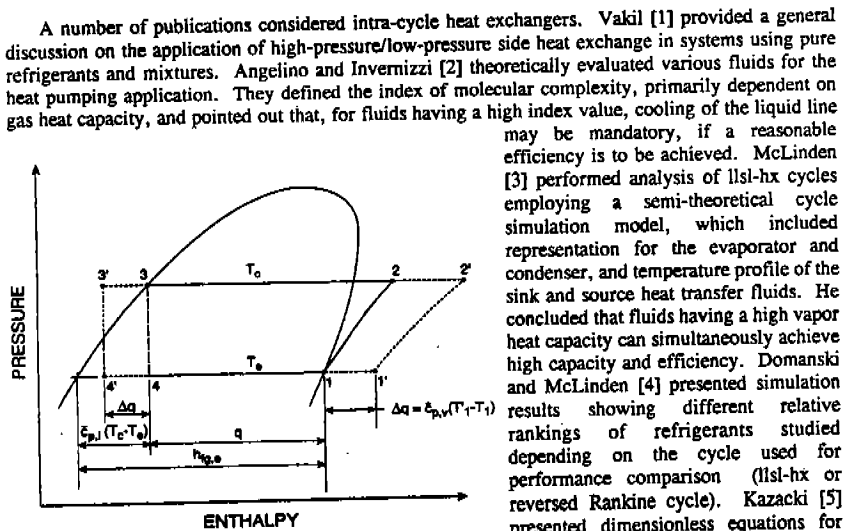


Figure 2. Key refrigerant state points in the basic cycle and lsl-hx cycle

A number of publications considered intra-cycle heat exchangers. Vakil [1] provided a general discussion on the application of high-pressure/low-pressure side heat exchange in systems using pure refrigerants and mixtures. Angelino and Invernizzi [2] theoretically evaluated various fluids for the heat pumping application. They defined the index of molecular complexity, primarily dependent on gas heat capacity, and pointed out that, for fluids having a high index value, cooling of the liquid line may be mandatory, if a reasonable efficiency is to be achieved. McLinden [3] performed analysis of lsl-hx cycles employing a semi-theoretical cycle simulation model, which included representation for the evaporator and condenser, and temperature profile of the sink and source heat transfer fluids. He concluded that fluids having a high vapor heat capacity can simultaneously achieve high capacity and efficiency. Domanski and McLinden [4] presented simulation results showing different relative rankings of refrigerants studied depending on the cycle used for performance comparison (lsl-hx or reversed Rankine cycle). Kazacki [5] presented dimensionless equations for COP and volumetric-capacity effectiveness of the internal heat

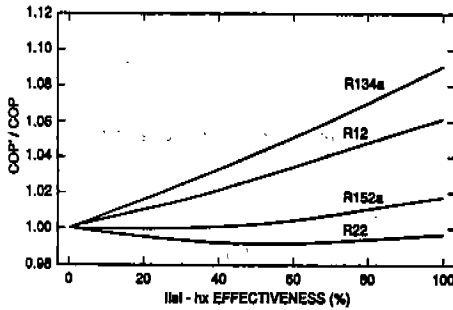


Figure 3. Change of the COP at different effectiveness of the llsl-hx at $T_c = -23^\circ\text{C}$ and $T_e = 42.5^\circ\text{C}$

exchange. A thorough discussion of liquid-subcooling and vapor-superheating effects is given in the book by Gosney [6].

This paper presents an evaluation of the liquid line/suction line heat exchange on a theoretical basis. It examines the cycle parameters and refrigerant thermodynamic properties that determine whether the installation of the llsl-hx results in improvement of COP and volumetric capacity. Twenty nine fluids were screened for their sensitivity to the liquid line/suction line heat exchange and evaluated for a few practical applications.

PERFORMANCE IN THE BASIC AND LLSL-HX CYCLE

The Coefficient of Performance

Consider the performance of a vapor compression cycle with the liquid line/suction line heat exchange, taking as a reference the performance of the same cycle without the heat exchange at the same saturation temperatures in the evaporator and the condenser. Figure 2 presents the two cycles considered. By definition, we may calculate the COP for these cycles (an apostrophe denotes the llsl-hx cycle):

$$\text{COP} = \frac{q}{w}$$

$$\text{COP}' = \frac{q + \Delta q}{w + \Delta w} = \text{COP} \frac{1 + \frac{\Delta q}{q}}{1 + \frac{\Delta w}{w}} \approx \text{COP} \left(1 + \frac{\Delta q}{q} - \frac{\Delta w}{w} \right)$$

by Taylor series, neglecting higher order terms.

The COP multiplier, $(1 + \Delta q/q - \Delta w/w)$, has to be greater than one if the cycle COP is to benefit from the llsl heat exchange. For obvious reasons, $\Delta q/q$ is always greater than 0. Also, $\Delta w/w$ is always positive, since the slope of constant entropy lines (which idealize the compression process) decreases with increasing vapor superheat on the pressure-enthalpy diagram (Figure 2). Thus, the relative rates in which $\Delta q/q$ and $\Delta w/w$ change determine the performance impact of the llsl-hx.

We can evaluate the terms involved using a few simplifying assumptions. Considering that isotherms in the liquid region are nearly vertical on the pressure-enthalpy diagram, we can express the evaporator capacity, q , in terms of fluid properties and the evaporator and condenser temperatures:

$$q = h_{f,c} - \bar{c}_{p,l} \cdot (T_c - T_e) \quad \text{where} \quad \bar{c}_{p,l} = \frac{1}{T_c - T_e} \int_{T_e}^{T_c} c_{p,l} dT$$

The change in the evaporator capacity, Δq , is equal to the amount of heat exchanged between the high-pressure liquid and low-pressure suction vapor:

$$\Delta q = \bar{c}_{p,v} \cdot (T_1' - T_1) \quad \text{where} \quad \bar{c}_{p,v} = \frac{1}{T_1' - T_1} \int_{T_1}^{T_1'} c_{p,v} dT$$

Treating refrigerant vapor as ideal gas with a constant heat capacity, we can express the work of isentropic compression by the familiar relation:

$$w = \frac{\gamma}{\gamma - 1} P_1 \cdot v_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \quad \text{and} \quad w' = \frac{\gamma}{\gamma - 1} P_1 \cdot v_1' \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

Introducing these relations and rearranging, we obtain the following equation for a fractional COP change of the lsl-hx cycle over the basic cycle:

$$\frac{\text{COP}'}{\text{COP}} = \frac{1 + \frac{\bar{c}_{p,v} (T_1' - T_1)}{h_{fg,o} - c_{p,l} (T_c - T_o)}}{1 + \frac{v_1' - v_1}{v_1}} = \frac{1 + \frac{T_1' - T_1}{h_{fg,o}/c_{p,v} - (T_c - T_o) c_{p,l}/c_{p,v}}}{1 + B_v \cdot (T_1' - T_1)}$$

For this ratio to be greater than 1, $\frac{1}{h_{fg,o}/c_{p,v} - (T_c - T_o)c_{p,l}/c_{p,v}} > B_v$,

where $B_v = \frac{v_1' - v_1}{v_1'(T_1' - T_1)}$ is an average coefficient of thermal expansion.

The derived equation allows one to make a preliminary evaluation of the COP improvement potential due to installation of the lsl-hx. Also, it provides us with the insight as to which properties influence the COP of the lsl-hx cycle. The equation indicates that the COP improvement is advanced if $h_{fg,o}/c_{p,v}$ and B are minimized, and $(T_c - T_o) \cdot c_{p,l}/c_{p,v}$ is maximized. The ratio of heat capacities of liquid and vapor exerts a stronger influence at large temperature lifts between the evaporator and condenser. The effect of all the terms is scaled with the amount of superheat gained by the suction vapor, $T_1' - T_1$, which at the theoretical limit is equal to the difference between the temperatures of the saturated refrigerant in the evaporator and condenser.

During our discussion, we have to remember that we are examining the relative performance of the lsl-hx cycle using as a reference the performance of the same fluid in the basic, reversed Rankine cycle. We may also remind ourselves that relations other than described above influence refrigerant performance in the basic refrigeration cycle.

A graphical representation of the "goodness" of thermodynamic properties for the reversed Rankine cycle (and also for the Carnot cycle) are the slopes of the liquid and vapor saturation lines on the temperature-entropy diagram. Ideally, both lines should be vertical. On the compression side, with a positive slope of the saturation line, the end-state of the compression would lay within the two-phase region - an undesired refrigerant condition for positive displacement compressors. With a negative slope, excessive compression work is required. On the expansion side, only a positive slope of the saturated liquid line is possible (Morrison, [7]). This line's gradient is related to the part of evaporator capacity that is lost due to liquid flashing.

The saturation-line slopes can be represented as proposed by Duffield and Hodgett [8]:

$$\frac{dT}{ds} = \frac{T}{c_p - \beta \cdot h_{fg} \frac{v}{v_1}}$$

where c_p , β and v are values for the phase (liquid or vapor) for which the saturation line is considered. The saturation lines would be vertical if the denominator was equal to zero. This condition can be satisfied on the vapor line but cannot for the liquid line due to a small value of β , and v_1 being much smaller than v . Since the slope of the saturated liquid line is always positive, it follows that performance of any refrigerant in the reversed Rankine cycle would improve if liquid heat capacity were smaller and the latent heat larger. We should note that the opposite property trends, larger liquid heat capacity and smaller latent heat, result in performance improvement when a lsl-hx is installed.

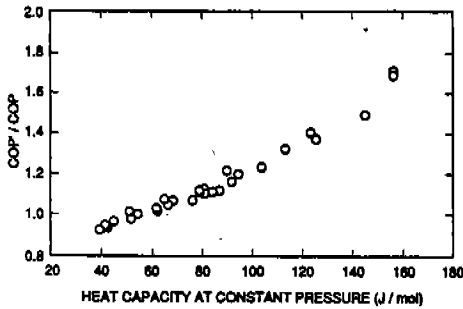


Figure 4. The relative COP of the l1sl-hx cycle (100% l1sl-hx effectiveness) at $T_{c,r}=0.82$ and $T_{e,r}=0.65$ as a function of $c_{p,v}$ of sat. vapor at $T_c=0.65$

hand, the choice of the same reduced temperatures results in larger temperatures lifts for fluids of higher critical temperatures, emphasizing the ratio of liquid and vapor heat capacities as the influencing factor for a COP change.

Based on the simulation results, we examined which of the relevant thermodynamic properties (see Table 1) have the most significant impact on the COP change of the l1sl-hx cycle over the COP of the basic refrigeration cycle. The tightest "eyeball fit" was obtained for vapor heat capacity at constant pressure at the evaporator outlet, shown in Figure 4. A similarly good fit was obtained for liquid heat capacity; the latent heat of evaporation and the coefficient of thermal expansion correlated more poorly.

The dominating impact of heat capacity on the performance of the basic refrigeration cycle can be demonstrated analytically considering the denominator in the relation for dT/ds . As pointed out by Morrison [9], at the normal boiling point, we can approximate the latent heat of evaporation by the Hildebrand variation of the Trouton's law [10]:

$$h_{fg} \approx d \cdot T_{nb} + R \cdot T_{nb} \cdot \ln T_{nb} + d_{cor}$$

where d is a constant, and d_{cor} represents a small correction dependent on the shape of molecule and its polarity. Noting that $v_v \gg v_l$ and $\beta=1/T$ for ideal gas, we obtain the relation for the vapor side:

$$c_p - \beta_v \cdot h_{fg} \frac{v_v}{v_v - v_l} \approx c_p + (d + R \cdot \ln T_{nb} + d_{cor}/T_{nb}),$$

in which variation of c_p between different fluids dominate the value of the relation since $\ln T_{nb}$ is a weak function in comparison. For the saturated liquid line, this strong influence of c_p is even more apparent since $\beta_l < \beta_v$, and $v_v \gg v_l$.

Figure 5 presents the COPs obtained for the basic cycle and for the l1sl-hx cycle at 100% effectiveness of the liquid line/suction line heat exchanger. The results are arranged in the ascending order of vapor heat capacities at constant pressure, $c_{p,v}$, of the evaluated refrigerants. Table 1 lists the refrigerants and their relevant properties in the ascending order of critical temperatures.

Figure 5 shows that the COPs of fluids working in the basic cycle are significantly different. It is interesting to notice that the fluids which perform particularly poorly in the basic cycle benefit most profoundly from switching to the l1sl-hx cycle. Refrigerants rated best for the basic cycle marginally benefitted from the l1sl-hx or, in a few cases, their performance even deteriorated. If we consider the best refrigerant results independently of the application cycle, their COPs are much more uniform.

The twenty nine fluids listed in Table 1 were screened for their performance sensitivity for the liquid line/suction line heat exchange. The performance was simulated for the basic cycle and the l1sl-hx cycle at the same reduced temperatures in the condenser and evaporator, $T_{c,r} = 0.82$ and $T_{e,r} = 0.65$, respectively, and with the assumption of isentropic compression.

The selection of the same reduced temperatures, rather than absolute temperatures, allowed examining the fluids at their best operating temperature range, while testing them against the same COP limit of the Carnot cycle, in this case equal to 3.82. On the other

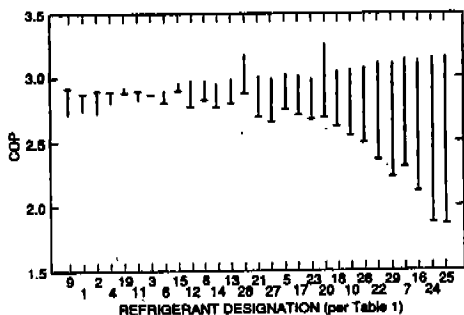


Figure 5. The COP of the basic cycle and l1sl-hx cycle for $T_{c,r}=0.65$ and $T_{c,r}=0.82$

(The horizontal short lines mark the COP for the basic cycle. The vertical lines indicate the difference between the COP for the basic cycle and l1sl-hx cycle.)

The Volumetric Capacity

Following the same approach as for the COP, we may derive the expression for the volumetric capacity for the l1sl-hx cycle, q_v' . Starting with the volumetric capacity for the basic refrigeration cycle, as a reference,

$$q_v = \frac{q}{v_1}$$

the following equation may be obtained for the volumetric capacity of the l1sl-hx cycle:

$$q_v' = q_v \frac{1 + \frac{\Delta q}{q}}{1 + \frac{\Delta v_1}{v_1}}$$

The term $\Delta v_1/v_1$ is equal to $\Delta w/w$, if ideal gas is assumed. Thus, by comparison with the expression for COP of the l1sl-hx cycle and with limitations of the assumptions made in deriving these equations, the rate of change of the volumetric capacity is equal to the rate of change of the COP.

Simplified vs. Exact Theoretical Calculations

Simplified calculations (performed with the aid of the derived equations) will carry errors resulting from simplification made during derivation of these equations. Volumetric-capacity calculations will carry the error of representing the difference of enthalpy of saturated liquid at two temperature levels by the product of the temperature difference and mean heat capacity at a constant pressure. COP calculations will be affected by the same error, as well as by the error of assuming the compression process as that of ideal gas.

These two errors are not involved if we perform calculations using the generic equations for the COP and volumetric capacity, and evaluate all the properties needed using an equation of state and other rigorous thermodynamic relations. Such an evaluation is performed by CYCLE0, a theoretical simulation program, which is based on the Carnahan-Starling-DeSantis equation of state [11].

Simulation results of CYCLE0 for the selected 29 fluids are displayed in Figure 6. in the ascending order of $c_{p,v}$ of the refrigerants. With the exception of three fluids (8, 15, and 19) which are barely sensitive to l1sl-hx, the impact of the l1sl-hx on the COP and volumetric capacity is in the same performance direction (positive or negative), the impact on the volumetric capacity being always greater than that on the COP. It was shown earlier that the simplified equations would predict an equal impact for the COP and volumetric capacity.

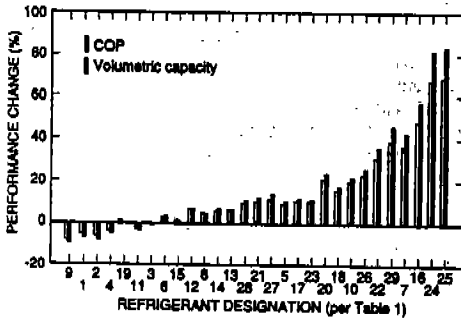


Figure 6. Change in the COP and volumetric capacity due to lsl-hx (100% effectiveness) referenced to performance of the basic cycle at $T_{c,r}=0.82$ and $T_{e,r}=0.65$

To evaluate the accuracy of the simplified equations, we compared predictions by these equations and CYCLE0. The predictions for volumetric capacity were almost identical, on average within 0.2%. The COP results were more spread. Since the COP and volumetric-capacity results calculated by the derived equations are identical, and the volumetric capacity results for both methods are very close, the spread between the COP and volumetric capacity change shown in Figure 6 is also representative of the difference between the COP predictions by the derived equations and CYCLE0. We may conclude that the simplified equations may be used with reasonable confidence for a preliminary fluid evaluation.

APPLICATION CHARTS

Among many possible graphical representations of the impact of the lsl-hx on the system COP, the representation using as coordinates a change of the COP (COP'-COP) and the temperature lift (temperature difference between the evaporator and condenser, T_c-T_e) proves to be quite simple. In this coordinate system, the lines denoting different possible evaporator temperatures, T_e , are almost straight.

Figure 7 shows charts generated for four refrigerants - R12, R134a, R152a, and R22 - assuming 100% effectiveness of the liquid-line/suction-line heat exchanger. Besides its simple form, the important feature of these charts is that they clearly separate operating conditions for which the

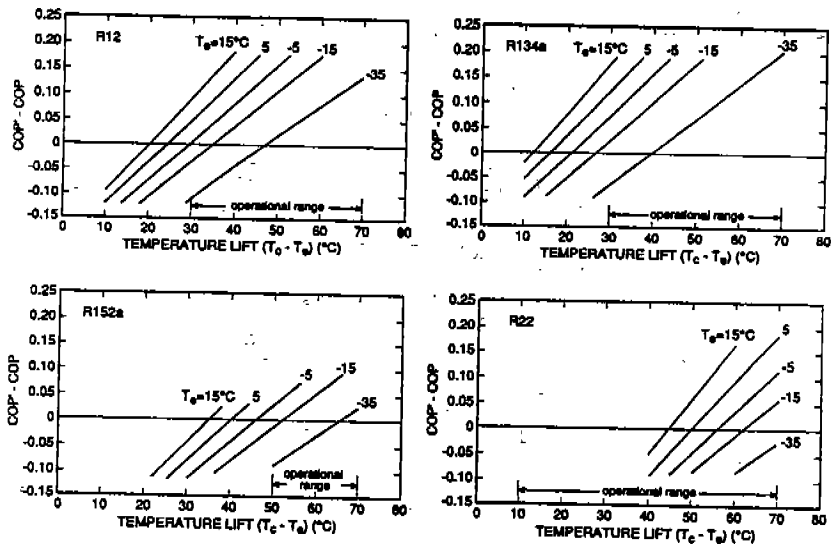


Figure 7. Change in the COP for R12, R134a, R152a, and R22 for various evaporator temperatures and temperature lifts.

implementation of the l1sl-hx can be beneficial or detrimental to the system COP. We can see that even for R12, which is commonly regarded as the refrigerant benefiting from the l1sl-hx, a loss in the COP will occur for a low evaporator temperature and low temperature lift. For example, for $T_e = -30^\circ\text{F}$ and 60°F temperature lift the loss in the COP will be approximately 0.09, while for the same lift but $T_e = 40^\circ\text{F}$, the COP will increase by 0.06. The operation ranges marked in the charts correspond to a domestic refrigerator for R152a, a domestic refrigerator and water chiller for R12 and R134a, and an air conditioner and a supermarket refrigeration unit for R22.

This impact on the COP can be explained considering changes in values of the relevant refrigerant properties utilized in the equation for COP'/COP . At a higher temperature, heat capacities are larger, while the latent capacity and coefficient of thermal expansion have smaller values. All these trends positively impact the COP for l1sl-hx cycles.

Plots generated for different refrigerants differ by the slope of the evaporator temperature lines and their positions with respect to values of the temperature lift on the abscissa. Steeper lines for the evaporator temperature indicate stronger sensitivity of COP to intra-cycle heat transfer between the suction line and liquid line.

CONCLUDING COMMENTS

The benefit of application of the intra-cycle heat exchange between the liquid line and suction line depends on the combination of the operating conditions and fluid properties - heat capacity, latent heat, and coefficient of thermal expansion - with heat capacity being the most influential property. Fluids that perform well in the basic cycle are marginally affected by the l1sl-hx, and the impact on the COP and volumetric capacity may be either positive or negative. Fluids performing poorly in the basic cycle benefit from the l1sl-hx installation by increase of the Coefficient of Performance and volumetric capacity.

This paper describes a fundamental analysis of l1sl-hx application: The results presented are based on theoretical evaluations using thermodynamic properties, with the assumption of the isentropic compression, no-pressure-drop infinite heat exchangers, and no-pressure-drop liquid line/suction line heat exchanger. In a real system, other factors such as transport properties, heat exchanger design, heat transfer fluid temperature profiles, compressor performance, etc., will also affect the system performance.

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Table 1. Selected Properties of Screened Refrigerants

Refrigerant	T_c (K)	T_n (K)	WM (g/mol)	$c_{p,l}$ (J/mol)	$c_{p,v}$ (J/mol)	$h_{f,c}$ (J/mol)	β_c (1/K)	$T_c - T_c$ (K)	P_c/P_c
1 R14	227.5	147.9	88.0	93.7	41.7	11563.2	0.007810	38.7	7.6
2 R23	299.1	194.4	70.0	100.7	42.3	16812.3	0.005708	50.8	8.9
3 R13	302.0	196.3	104.5	108.5	54.6	15296.7	0.005895	51.3	7.5
4 R170	305.4	198.5	30.1	88.2	44.9	14064.8	0.005979	51.9	6.5
5 R125	339.4	220.6	120.0	147.0	81.1	19484.4	0.005029	57.7	9.4
6 R13B1	340.2	221.1	148.9	116.9	62.0	17418.9	0.005257	57.8	7.5
7 R218	345.1	224.3	188.0	203.8	125.1	20431.6	0.004945	58.7	10.0
8 R143a	346.3	225.1	84.0	130.2	66.3	19199.0	0.004937	58.9	8.6
9 R32	351.6	228.5	52.0	94.0	39.5	19766.4	0.004797	59.8	9.0
10 R115	353.1	229.5	154.5	161.4	94.4	19642.6	0.004957	60.0	8.8
11 R22	369.3	240.0	86.5	108.9	51.7	19838.3	0.004718	62.8	8.2
12 R290	369.8	240.4	44.1	121.8	65.1	18332.6	0.004836	62.9	7.2
13 R134a	374.3	243.3	102.0	146.6	76.0	22371.8	0.004514	63.6	10.0
14 R12	384.9	250.2	120.9	124.6	68.3	19719.7	0.004607	65.4	7.6
15 R152a	386.7	251.4	66.0	124.2	62.2	21828.8	0.004401	65.7	9.0
16 RC318	388.5	252.5	200.0	231.1	145.0	24308.2	0.004359	66.0	10.9
17 R134	392.1	254.9	102.0	152.2	84.0	22807.5	0.004374	66.7	9.5
18 R124	395.6	257.2	136.5	162.2	91.7	22813.0	0.004336	67.3	9.4
19 RC270	398.3	258.9	42.1	104.3	51.1	19117.6	0.004521	67.7	6.9
20 R600a	408.1	265.3	58.1	133.2	89.7	19025.7	0.004352	69.4	7.0
21 R142b	410.3	266.7	100.5	144.8	79.4	22387.7	0.004235	69.8	8.5
22 R114	418.8	272.2	170.9	180.3	113.0	23201.2	0.004139	71.2	8.7
23 E134	420.7	273.5	118.0	164.8	87.0	26035.7	0.004011	71.5	10.6
24 R216a	453.1	294.5	220.9	241.9	156.2	27032.4	0.003766	77.0	9.9
25 R216b	453.1	294.5	220.9	242.2	156.2	26850.8	0.003779	77.0	9.8
26 R123	456.9	297.0	152.9	175.1	103.8	26313.7	0.003753	77.7	9.3
27 R11	471.2	306.3	137.4	137.8	81.0	24366.8	0.003759	80.1	7.7
28 R141b	478.0	310.7	116.9	120.2	79.1	23680.3	0.003665	81.3	7.7
29 R113	487.5	316.9	187.4	194.0	123.1	27274.6	0.003557	82.9	8.8

$$T_c = 0.65 \cdot T_{cr}$$

$c_{p,v}$, $h_{f,c}$, β_c , and P_c evaluated at saturation at T_c

$c_{p,l}$ and P_c evaluated at saturation at $T_c = 0.82 \cdot T_{cr}$