Evaluation of suction-line/liquid-line heat exchange in the refrigeration cycle*†

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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 parameters and refrigerant thermodynamic properties that determine whether the installation results in improvement of COP and volumetric capacity. The study shows 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.
(Keywords: refrigerating cycle; heat exchanger; gas; liquid; refrigerant; performance)

Evaluation de l'échange de chaleur entre la conduite d'aspiration et la conduite de liquide dans un cycle frigorifique

L'article présente une évaluation théorique de la performance après installation d'un échangeur de chaleur conduite de liquide/conduite d'aspiration (LLSL-HX). On examine les paramètres du cycle et les propriétés thermodynamiques des frigorigènes qui déterminent si cette installation entraine une amélioration du COP et de la puissance volumétrique. L'étude montre que l'avantage à utiliser cet échangeur de chaleur dépend d'un ensemble de conditions de fonctionnement et des propriétés des fluides (capacité calorifique, chaleur latente et coefficient de détente thermique), la capacité calorifique étant la propriété la plus déterminante. Les fluides qui se comportent bien dans le cycle de base sont affectés de façon marginale par l'échangeur LLSL-HX, et l'effet sur le coefficient de performance et la capacité volumétrique peut être positif ou négatif. Les fluides dont le comportement dans le cycle de base n'est guère satisfaisant tirent profit de l'installation de l'échangeur LLSL-HX, par l'augmentation du COP et de la capacité volumétrique.
(Mots clés: cycle frigorifique; échangeur de chaleur; gaz; liquide; frigorigène; performance)

Among many possible variations of the basic refrigeration (vapour compression) cycle, the cycle with the liquid-line/suction-line heat exchanger (LLSL-HX) is probably used most often. As a result of employing this intracycle heat exchange, the high-pressure refrigerant is subcooled at the expense of superheating the vapour 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*.

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 vapour 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 vapour on the outside of the 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, an LLSL-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 LLSL-HX on the COP, evaluated by theoretical calculations for four different refrigerants. The ordinate is a ratio of the COP obtained for a given effectiveness of the LLSL-HX, ε, and the COP obtained for the same fluid working in the basic cycle ($\varepsilon = 0$). 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. However, at low values of LLSL-HX

Nomenclature

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 coefficient

Greek letters

β	Coefficient of thermal expansion, $(1/v) \cdot (\partial v / \partial T)_p$
γ	Isentropic coefficient, c_p/c_v
ε	Effectiveness of the LLSL-HX $= 100\%(T'_1 - T_1)/(T_3 - T_1)$

Subscripts

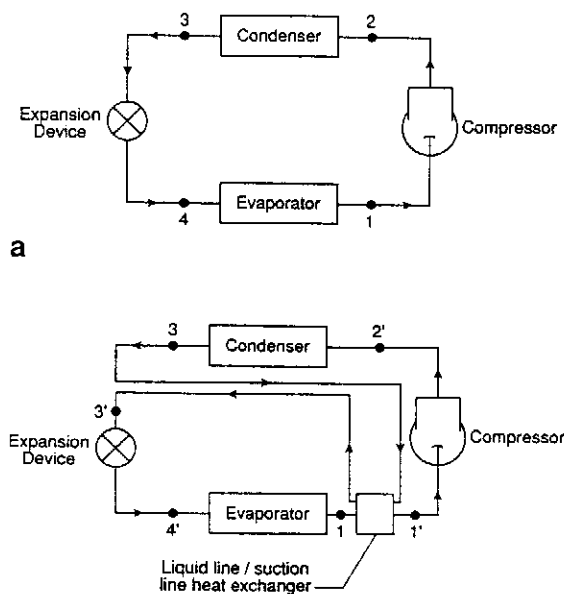|	
c	Condenser
cr	Critical
e	Evaporator
l	Liquid
nb	Normal boiling point
r	Reduced
v	Vapour, volumetric
1–4	Key locations in a cycle (*Figures 1* and *2*)

Superscript

The LLSL-HX cycle

a

b

Figure 1 Schematic of hardware arrangements for (a) the basic cycle and (b) cycle with the liquid-line/suction-line heat exchanger

Figure 1 Schéma des assemblages pour (a) le cycle de base et (b) l'échangeur de chaleur conduite de liquide/conduite d'aspiration
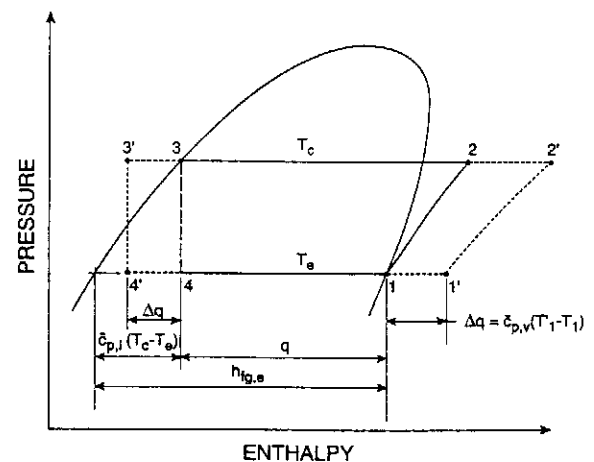

Figure 2 Key refrigerant state points in the basic cycle and LLSL-HX cycle

Figure 2 Points clés de l'état des frigorigènes dans le cycle de base et le cycle LLSL-HX

effectiveness, R22 showed degradation in performance, some of which was recovered at 100% effectiveness.

A number of publications have considered intra-cycle heat exchangers. Vakil[1] provided a general discussion of the application of high-pressure/low-pressure side heat exchange in systems using pure refrigerants and mixtures. Angelino and Invernizzi[2] evaluated theoretically various fluids for 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 LLSL-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 vapour heat capacity can simultaneously achieve high capacity and efficiency. Domanski and McLinden[4] presented simulation results showing different relative rankings of the refrigerants studied depending on the cycle used for performance comparison (LLSL-HX or reversed Rankine cycle). The performance of the LLSL-HX cycle was studied by Bykov[5] and Kazachki[6]. A thorough discussion of liquid subcooling and vapour superheating effects is given in the book by Gosney[7].
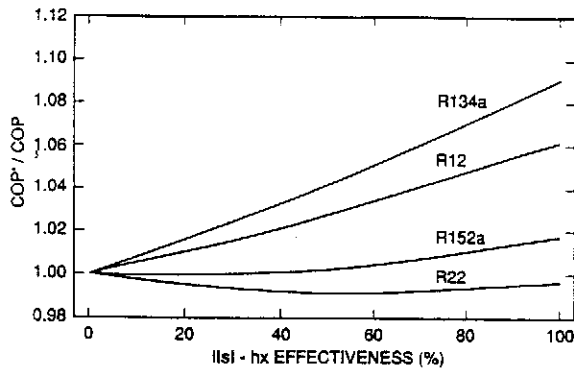
Figure 3 Change of the COP for different effectiveness of the LLSL-HX at $T_e = -23$ °C and $T_c = 42.5$ °C

Figure 3 *Changement du COP pour plusieurs efficacités de l'échangeur LLSL-HX à $T_e = 23$ deg C et $T_c = 42.5$ deg C*

This paper presents an evaluation of 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 liquid-line/suction-line heat exchange and evaluated for a few practical applications.

Performance in the basic and LLSL-HX cycle

Coefficient of performance

Consider the performance of a vapour compression cycle with liquid-line/suction-line heat exchange, taking as a reference the performance of the same cycle without 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 (a prime (') denotes the LLSL-HX cycle):

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

$$\text{COP}' = \frac{q + \Delta q}{w + \Delta w}$$

$$= \text{COP}\frac{1 + \Delta q/q}{1 + \Delta w/w}$$

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

by Taylor series, neglecting higher-order terms.

The COP multiplier, $(1 + \Delta q/q - \Delta w/w)$, has to be greater than 1 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, as the slope of constant entropy lines (which idealize the compression process) decreases with increasing vapour superheat on the pressure–enthalpy diagram (*Figure 2*). Thus the relative rates at 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:

$$q = h_{fg,e} - \bar{c}_{p,l}(T_c - T_e)$$

where

$$\bar{c}_{p,l} = \frac{1}{T_c - T_e}\int_{T_e}^{T_c} c_{p,l}\,dT$$

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

$$\Delta q = \bar{c}_{p,v}(T'_1 - T_1)$$

where

$$\bar{c}_{p,v} = \frac{1}{T'_1 - T_1}\int_{T_1}^{T'_1} c_{p,v}\,dT$$

We can approximate the work of compression of a real gas by the familiar relationship for the work of isentropic compression for an ideal gas with a constant heat capacity:

$$w = \frac{\gamma}{\gamma - 1}P_1 v_1\left[\left(\frac{P_2}{P_1}\right)^{(\gamma - 1)/\gamma} - 1\right]$$

and

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

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

$$\frac{\text{COP}'}{\text{COP}} = \frac{1 + [\bar{c}_{p,v}(T'_1 - T_1)]/[h_{fg,e} - \bar{c}_{p,l}(T_c - T_e)]}{1 + (v'_1 - v_1)/v_1}$$

$$= \frac{1 + (T'_1 - T_1)/[h_{fg,e}/\bar{c}_{p,v} - (T_c - T_e)\bar{c}_{p,l}/\bar{c}_{p,v}]}{1 + B_v(T'_1 - T_1)}$$

For this ratio to be greater than 1,

$$\frac{1}{h_{fg,e}/\bar{c}_{p,v} - (T_c - T_e)\bar{c}_{pl}/\bar{c}_{pv}} > B_v$$

where

$$B_v = \frac{v'_1 - v_1}{v'_1(T'_1 - T_1)}$$

is an average coefficient of thermal expansion.

Although the derived equation is mathematically equivalent to those given previously[5,6] we present this form of the equation because it is more suitable for gaining an insight as to which properties influence the COP of the LLSL-HX cycle. The equation indicates that the COP improvement is advanced if $h_{fg,e}/c_{p,v}$ and B are minimized, and $(T_c - T_e)c_{p,l}/c_{p,v}$ is maximized. The ratio of heat capacities of liquid and vapour 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 vapour, $T'_1 - T_1$, 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 LLSL-HX cycle using as a reference the performance of the same fluid in the basic, reversed Rankine cycle. We may also remind ourselves that relationships other than that 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) is provided by the slopes of the liquid and vapour 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 lie within the two-phase region: an undesirable 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[8]. This line's gradient is related to the part of the evaporator capacity that is lost due to liquid flashing.

The saturation line slopes can be represented as proposed by Duffield and Hodgett[9]:

$$\frac{dT}{ds} = \frac{T}{c_p - \beta h_{fg}\left(\dfrac{v}{v_v - v_1}\right)}$$

where c_p, β and v are values for the phase (liquid or vapour) 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 vapour line but cannot for the liquid line owing to a small value of β_1 and v_1 being much smaller than v_v. As the slope of the saturated liquid line is always positive, it follows that the performance of any refrigerant in the reversed Rankine cycle would improve if the 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 an LLSL-HX is installed.

The 29 fluids listed in *Table 1* were screened for their performance sensitivity for liquid-line/suction-line heat exchange. The performance was simulated for the basic cycle and the LLSL-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. For simulations we used CYCLE0, a theoretical cycle program based on the Carnahan–Starling–DeSantis equation of state[10], which performs direct cycle calculations without the assumptions made during derivation of the COP'/COP relationship.

The selection of the same reduced temperatures, rather than absolute temperatures, allowed examination of 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. The choice of the same reduced temperatures results in larger temperature lifts for fluids of higher critical temperatures, emphasizing the

Table 1 Selected properties of screened refrigerants

Tableau 1 Propriétés de certains frigorigènes

	Refrigerant	T_{cr} (K)	T_e (K)	WM (g mol^{-1})	$c_{p,l}$ (J mol^{-1})	$c_{p,v}$ (J mol^{-1})	$h_{fg,e}$ (J mol^{-1})	β_v (1/K)	$T_c - T_e$ (K)	P_c/P_e
1	R14	227.5	147.9	88.0	93.7	41.7	11 563.2	0.007 810	38.7	7.6
2	R23	299.1	194.4	70.0	100.7	42.3	16 812.3	0.005 708	50.8	8.9
3	R13	302.0	196.3	104.5	108.5	54.6	15 296.7	0.005 895	51.3	7.5
4	R170	305.4	198.5	30.1	88.2	44.9	14 064.8	0.005 979	51.9	6.5
5	R125	339.4	220.6	120.0	147.0	81.1	19 484.4	0.005 029	57.7	9.4
6	R13B1	340.2	221.1	148.9	116.9	62.0	17 418.9	0.005 257	57.8	7.5
7	R218	345.1	224.3	188.0	203.8	125.1	20 431.6	0.004 945	58.7	10.0
8	R143a	346.3	225.1	84.0	130.2	66.3	19 199.0	0.004 937	58.9	8.6
9	R32	351.6	228.5	52.0	94.0	39.5	19 766.4	0.004 797	59.8	9.0
10	R115	353.1	229.5	154.5	161.4	94.4	19 642.6	0.004 957	60.0	8.8
11	R22	369.3	240.0	86.5	108.9	51.7	19 838.3	0.004 718	62.8	8.2
12	R290	369.8	240.4	44.1	121.8	65.1	18 332.6	0.004 836	62.9	7.2
13	R134a	374.3	243.3	102.0	146.6	76.0	22 371.8	0.004 514	63.6	10.0
14	R12	384.9	250.2	120.9	124.6	68.3	19 719.7	0.004 607	65.4	7.6
15	R152a	386.7	251.4	66.0	124.2	62.2	21 828.8	0.004 401	65.7	9.0
16	RC318	388.5	252.5	200.0	231.1	145.0	24 308.2	0.004 539	66.0	10.9
17	R134	392.1	254.9	102.0	152.2	84.0	22 807.5	0.004 374	66.7	9.5
18	R124	395.6	257.2	136.5	162.2	91.7	22 813.0	0.004 336	67.3	9.4
19	RC270	398.3	258.9	42.1	104.3	51.1	19 117.6	0.004 521	67.7	6.9
20	R600a	408.1	265.3	58.1	133.2	89.7	19 025.7	0.004 352	69.4	7.0
21	R142b	410.3	266.7	100.5	144.8	79.4	22 387.7	0.004 235	69.8	8.5
22	R114	418.8	272.2	170.9	180.3	113.0	23 201.2	0.004 139	71.2	8.7
23	E134	420.7	273.5	118.0	164.8	87.0	26 035.7	0.004 011	71.5	10.6
24	R216a	453.1	294.5	220.9	241.9	156.2	27 032.4	0.003 766	77.0	9.9
25	R216b	453.1	294.5	220.9	242.2	156.2	26 850.8	0.003 779	77.0	9.8
26	R123	456.9	297.0	152.9	175.1	103.8	26 313.7	0.003 753	77.7	9.3
27	R11	471.2	306.3	137.4	137.8	81.0	24 366.8	0.003 759	80.1	7.7
28	R141b	478.0	310.7	116.9	120.2	79.1	23 680.3	0.003 665	81.3	7.7
29	R113	487.5	316.9	187.4	194.0	123.1	27 274.6	0.003 557	82.9	8.8

$T_e = 0.65 T_{cr}$

$c_{p,v}$, $h_{fg,e}$, β_v and P_e evaluated at saturation at T_e

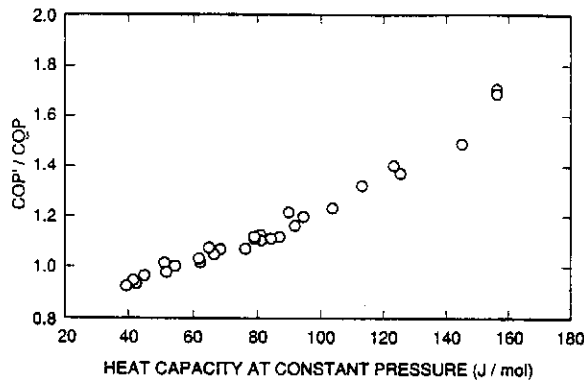$c_{p,l}$ and P_c evaluated at saturation at $T_c = 0.82 T_{cr}$

Figure 4 Relative COP of the LLSL-HX cycle (100% LLSL-HX effectiveness) at $T_{c,r}=0.82$ and $T_{e,r}=0.65$ as a function of $c_{p,v}$ of saturated vapour at $T_r=0.65$

Figure 4 COP relatif du cycle LLSL-HX (100% d'efficacité LLSL-HX) à $T_{c,r}=0,82$ et $T_{e,r}=0,65$, en fonction de c_{pv} de vapeur saturée à $T_r=0,65$

ratio of liquid and vapour 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 LLSL-HX cycle over the COP of the basic refrigeration cycle. The tightest 'eyeball fit' was obtained for vapour 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 by considering the denominator in the relation for dT/ds. As pointed out by Morrison[11], at the normal boiling point, we can approximate the latent heat of evaporation by the Hildebrand variation of Trouton's law[12]:

$$h_{fg} \approx dT_{nb} + RT_{nb} \ln T_{nb} + d_{cor}$$

where d is a constant, and d_{cor} represents a small correction dependent on the shape of the molecule and its polarity. Noting that $v_v \gg v_1$ and $\beta = 1/T$ for an ideal gas, we obtain the relationship for the vapour side:

$$c_{p,v} - \beta_v h_{fg} \frac{v_v}{v_v - v_1} \approx c_p - (d + R \ln T_{nb} + d_{cor}/T_{nb})$$

in which variation of c_p between different fluids dominates the value of the relation, as $\ln T_{nb}$ is a weak function by comparison. For the saturated liquid line, this strong influence of c_p is even more apparent, as $\beta_1 \ll \beta_v$ and $v_v \gg v_1$.

Figure 5 presents the COPs obtained for the basic cycle and for the LLSL-HX cycle at 100% effectiveness of the LLSL-HX. The results are arranged in ascending order of vapour 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 that perform particularly poorly in the basic cycle benefit most profoundly from switching to the LLSL-HX cycle. Refrigerants rated best for the basic cycle benefitted marginally from the LLSL-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.

Volumetric capacity

Following the same approach as for the COP, we may derive the expression for the volumetric capacity for the LLSL-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 LLSL-HX cycle:

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

The term $\Delta v_1/v_1$ is equal to $\Delta w/w$, if the compression work is calculated as for an ideal gas. Thus, with the ideal gas assumption, the change of volumetric capacity is expressed by the same relationship as the change of COP. In this respect, we are in contention with reference 6, where two separate relationships for the COP and volumetric capacity are presented.

Simplified vs. exact theoretical calculations

It is of interest to assess the validity of the assumptions made during derivation of the simplified relationships for COP' and q'_v. This can be done by comparing the simulation results obtained by these relations and CYCLE0. Simplified volumetric capacity calculations will carry the error of representing the difference of enthalpy of saturated liquid at two temperature levels by
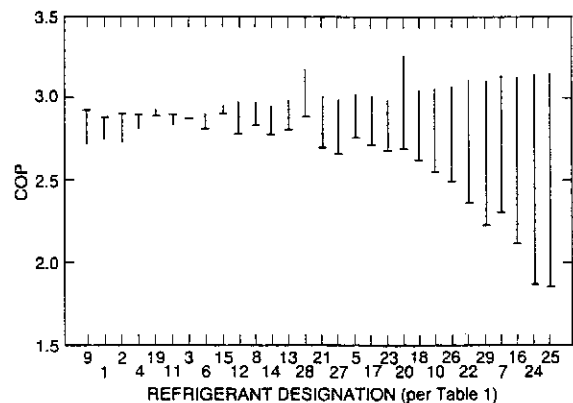

Figure 5 COP of the basic cycle and LLSL-HX cycle for $T_{e,r}=0.65$ and $T_{c,r}=0.82$. The horizontal short lines mark the COP for the basic cycle. The vertical lines indicate the difference between the COP for the basic cycle and the LLSL-HX cycle

Figure 5 COP du cycle de base et du cycle LLSL-HX pour $T_{e,r}=0,65$ et $T_{c,r}=0,82$. Les lignes horizontal courtes indiquent le COP pour le cycle de base, les lignes verticales indiquent la différence entre le COP du cycle de base et due cycle LLSL-HX
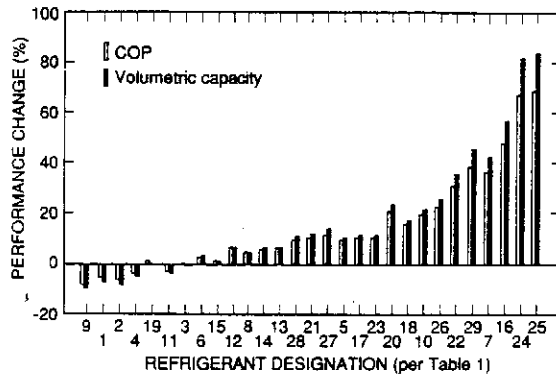
Figure 6 Change in the COP and volumetric capacity due to LLSL-HX (100% effectiveness) referenced to performance of the basic cycle at $T_{c,r} = 0.82$ and $T_{c,r} = 0.65$

Figure 6 *Modification du COP et de la puissance volumétrique due au LLSL-HX (100% d'efficacité) rapportée à la performace du cycle de base $T_{c,r} = 0.82$ et $T_{c,r} = 0.65$*

the product of the temperature difference and mean heat capacity at a constant temperature. Simplified COP calculations will be affected by the same error, as well as by the error of assuming the compression process as being that of an ideal gas. As both simplified calculations and CYCLE0 simulations use the same property algorithms, the differences in results are solely due to the simplifications stated above.

CYCLE0 simulation results, COP and volumetric capacity, are displayed in *Figure 6* for the screened refrigerants in ascending order of their molar heat capacities, $c_{p,v}$. With the exception of three fluids (8, 15 and 19), which are barely sensitive to LLSL-HX, the impact of the LLSL-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.

The predictions of volumetric capacity by CYCLE0 and the derived relationship were almost identical, on average within 0.2%. The COP results were more spread. and the derived relationship were almost identical, on average within 0.2%. The COP results were more widely spread. As 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 equation and CYCLE0. We may conclude that the simplified equations may be used with reasonable confidence for a preliminary fluid evaluation, but the accuracy of COP predictions is worse for refrigerants of large heat capacity.

Application charts

Among many possible graphical representations of the impact of the LLSL-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.
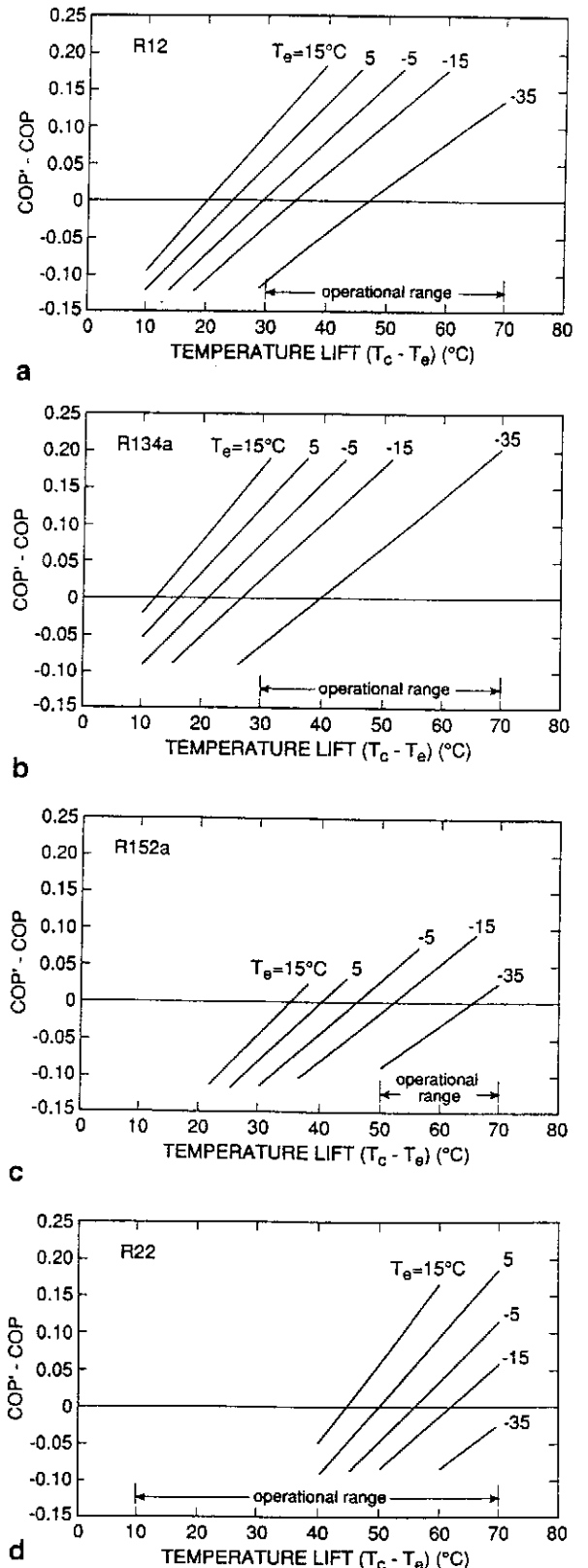

Figure 7 Change in the COP for (a) R12, (b) R134a, (c) R152a and (d) R22 for various evaporator temperatures and temperature lifts

Figure 7 *Changement du COP pour (a) R12, (b) R134a, (c) R152a et (d) R22 pour plusieurs températures de l'évaporateur et plusieurs élévations de température*

Figure 7 shows charts generated for four refrigerants – R12, R134a, R152a and R22 – assuming 100% effectiveness of the LLSL-HX. Besides their simple form, the important feature of these charts is that they clearly separate operating conditions for which the implementation of the LLSL-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 LLSL-HX, a loss in the COP will occur for a low evaporator temperature and low temperature lift. For example, for $T_e = -55\,°C$ and $40\,°C$ temperature lift the loss in the COP will be approximately 0.04, while for the same lift but $T_e = -5\,°C$, the COP will increase by 0.08. 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 conditioning and a supermarket refrigeration unit for R22.

This impact on the COP can be explained by 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 impact positively on the COP for LLSL-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.

Conclusions

The benefit of application of 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 affected marginally by the LLSL-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 LLSL-HX installation by increase of the coefficient of performance and volumetric capacity.

This paper describes a fundamental analysis of LLSL-HX application. The results presented are based on theoretical evaluations using thermodynamic properties,

with the assumption of 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 and compressor performance, will also affect the system performance.

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