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Refrigeration System Performance using Liquid-Suction Heat Exchangers

S. A. Klein, D. T. Reindl, and K. Brownell
College of Engineering
University of Wisconsin - Madison

Abstract

Heat transfer devices are provided in many refrigeration systems to exchange energy between the cool gaseous refrigerant leaving the evaporator and warm liquid refrigerant exiting the condenser. These liquid-suction or suction-line heat exchangers can, in some cases, yield improved system performance while in other cases they degrade system performance. Although previous researchers have investigated performance of liquid-suction heat exchangers, this study can be distinguished from the previous studies in three ways. First, this paper identifies a new dimensionless group to correlate performance impacts attributable to liquid-suction heat exchangers. Second, the paper extends previous analyses to include new refrigerants. Third, the analysis includes the impact of pressure drops through the liquid-suction heat exchanger on system performance. It is shown that reliance on simplified analysis techniques can lead to inaccurate conclusions regarding the impact of liquid-suction heat exchangers on refrigeration system performance. 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.

Introduction

Liquid-suction heat exchangers are commonly installed in refrigeration systems with the intent of ensuring proper system operation and increasing system performance. Specifically, ASHRAE (1998) states that liquid-suction heat exchangers are effective in:

1) increasing the system performance
2) subcooling liquid refrigerant to prevent flash gas formation at inlets to expansion devices
3) fully evaporating any residual liquid that may remain in the liquid-suction prior to reaching the compressor(s)

Figure 1 illustrates a simple direct-expansion vapor compression refrigeration system utilizing a liquid-suction heat exchanger. In this configuration, high temperature liquid leaving the heat rejection device (an evaporative condenser in this case) is subcooled prior to being throttled to the evaporator pressure by an expansion device such as a thermostatic expansion valve. The sink for subcooling the liquid is low temperature refrigerant vapor leaving the evaporator. Thus, the liquid-suction heat exchanger is an indirect liquid-to-vapor heat transfer device. The vapor-side of the heat exchanger (between the evaporator outlet and the compressor suction) is often configured to serve as an accumulator thereby further minimizing the risk of liquid refrigerant carrying-over to the compressor suction. In cases where the evaporator allows liquid carry-over, the accumulator portion of the heat exchanger will trap and, over time, vaporize the liquid carry-over by absorbing heat during the process of subcooling high-side liquid.

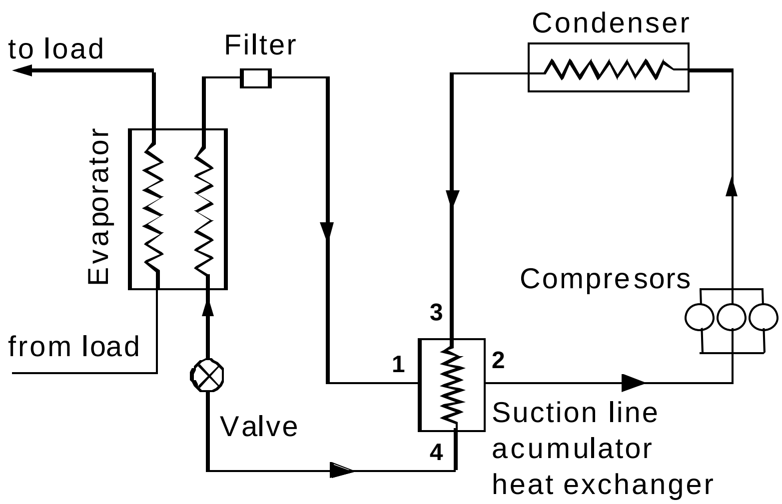

Figure 1: Schematic of typical vapor compression refrigeration system with a liquid-suction heat exchanger.

Background

Stoecker and Walukas (1981) focused on the influence of liquid-suction heat exchangers in both single temperature evaporator and dual temperature evaporator systems utilizing refrigerant mixtures. Their analysis indicated that liquid-suction heat exchangers yielded greater performance improvements when nonazeotropic mixtures were used compared with systems utilizing single component refrigerants or azeoptropic mixtures. McLinden (1990) used the principle of corresponding states to evaluate the anticipated effects of new refrigerants. He showed that the performance of a system using a liquid-suction heat exchanger increases as the ideal gas specific heat (related to the molecular complexity of the refrigerant) increases. Domanski and Didion (1993) evaluated the performance of nine alternatives to R22 including the impact of liquid-suction heat exchangers. Domanski et al. (1994) later extended the analysis by evaluating the influence of liquid-suction heat exchangers installed in vapor compression refrigeration systems considering 29 different refrigerants in a theoretical analysis. Bivens et al. (1994) evaluated a proposed mixture to substitute for R22 in air conditioners and heat pumps. Their analysis indicated a 6-7% improvement for the alternative refrigerant system when system modifications included a liquid-suction heat exchanger and counterflow system heat exchangers (evaporator and condenser). Bittle et al. (1995a) conducted an experimental evaluation of a liquid-suction heat exchanger applied in a domestic refrigerator using R152a. The authors compared the system performance with that of a traditional R12-based system. Bittle et al. (1995b) also compared the ASHRAE method for predicting capillary tube performance (including the effects of liquid-suction heat exchangers) with experimental data. Predicted capillary tube mass flow rates were within 10% of predicted values and subcooling levels were within 1.7°C (3°F) of actual measurements.

This paper analyzes the liquid-suction heat exchanger to quantify its impact on system capacity and performance (expressed in terms of a system coefficient of performance, COP). The influence of liquid-suction heat exchanger size over a range of operating conditions (evaporating and condensing) is illustrated and quantified using a number of alternative refrigerants. Refrigerants included in the present analysis are R507A, R404A, R600, R290, R134a, R407C, R410A, R12, R22, R32, and R717. This paper extends the results presented in previous studies in that it considers new refrigerants, it specifically considers the effects of the pressure drops, and it presents general relations for estimating the effect of liquid-suction heat exchangers for any refrigerant.

Heat Exchanger Effectiveness

The ability of a liquid-suction heat exchanger to transfer energy from the warm liquid to the cool vapor at steady-state conditions is dependent on the size and configuration of the heat transfer device. The liquid-suction heat exchanger performance, expressed in terms of an effectiveness, is a parameter in the analysis. The effectiveness of the liquid-suction heat exchanger is defined in equation (1):

$$e = \frac{(T_2 - T_1)}{(T_3 - T_1)} = \frac{(T_{vapor,out} - T_{vapor,in})}{(T_{liquid,in} - T_{vapor,in})} \tag{1}$$

where the numeric subscripted temperature (T) values correspond to locations depicted in Figure 1. The effectiveness is the ratio of the actual to maximum possible heat transfer rates. It is related to the surface area of the heat exchanger. A zero surface area represents a system without a liquid-suction heat exchanger whereas a system having an infinite heat exchanger area corresponds to an effectiveness of unity.

The liquid-suction heat exchanger effects the performance of a refrigeration system by influencing both the high and low pressure sides of a system. Figure 2 shows the key state points for a vapor compression cycle utilizing an idealized liquid-suction heat exchanger on a pressure-enthalpy diagram. The enthalpy of the refrigerant leaving the condenser (state 3) is decreased prior to entering the expansion device (state 4) by rejecting energy to the vapor refrigerant leaving the evaporator (state 1) prior to entering the compressor (state 2). Pressure losses are not shown. The cooling of the condensate that occurs on the high pressure side serves to increase the refrigeration capacity and reduce the likelihood of liquid refrigerant flashing prior to reaching the expansion device. On the low pressure side, the liquid-suction heat exchanger increases the temperature of the vapor entering the compressor and reduces the refrigerant pressure, both of which increase the specific volume of the refrigerant and thereby decrease the mass flow rate and capacity. A major benefit of the liquid-suction heat exchanger is that it reduces the possibility of liquid carry-over from the evaporator which could harm the compressor. Liquid carryover can be readily caused by a number of factors that may include wide fluctuations in evaporator load and poorly maintained expansion devices (especially problematic for thermostatic expansion valves used in ammonia service).

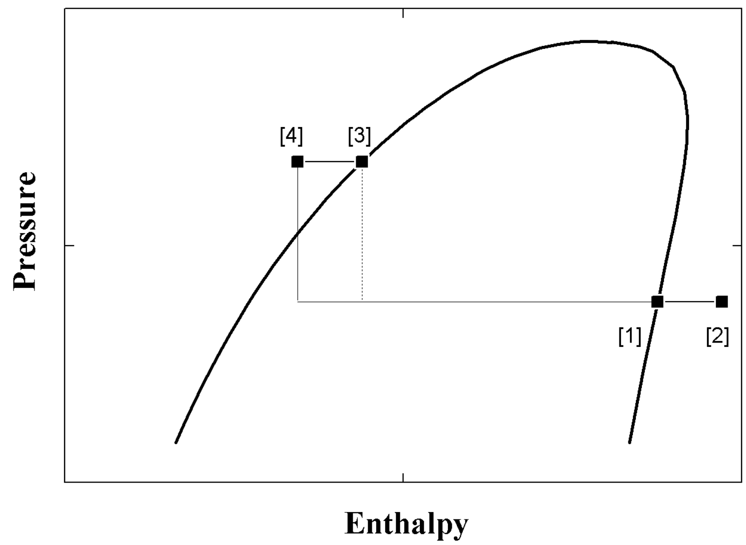

Figure 2: Pressure-Enthalpy Diagram showing effect of an idealized liquid-suction heat exchange

Heat Exchanger Effect on Capacity Neglecting Mass Flow Rate Corrections

Without a liquid-suction heat exchanger, the refrigerating effect per unit mass flow rate of circulating refrigerant is the difference in enthalpy between states 1 and 3 in Figure 2. When the heat exchanger is installed, the refrigeration effect per unit mass flow rate increases to the difference in enthalpy between states 1 and 4. If there were no other effects, the addition of a liquid-suction heat exchanger would always lead to an increase in the refrigeration capacity of a system. The extent of the capacity increase is a function of the specific refrigerant, the heat exchanger effectiveness, and the system operating conditions. The effect of a liquid-suction heat exchanger on refrigeration capacity can be quantified in terms of a relative capacity change index (RCI) as defined in equation (2):

$$RCI = \left(\frac{Capacity - Capacity_{nohx}}{Capacity_{nohx}} \right) \times 100\% \tag{2}$$

where

Capacity is the refrigeration capacity with a liquid-suction heat exchanger
$\text{Capacity}_{\text{no hx}}$ is the refrigeration capacity for a system operating at the same condensing and evaporating temperatures without a liquid-suction heat exchanger

Refrigeration cycle performance calculations were carried-out using a commercial equation solving program (Klein and Alvarado, 1998) with refrigerant property data provided by the REFPROP 6 data base (McLinden et al. 1998). The results presented here assume that refrigerant exits the evaporator as a saturated vapor at the evaporator pressure (state 1 in Figure 1) and exits the condenser as a saturated liquid at the condenser pressure (state 3). The effects of superheat at the evaporator exit and subcooling at the condenser exit were investigated and found not to have any significant effect on the relative capacity index defined in equation 2 or on the general results described in this paper. Compressor performance is quantified in terms of an isentropic efficiency. Different constant values of the isentropic efficiency between 0.5 and 1.0 were investigated. In addition, an empirical expression for the isentropic efficiency as a function of temperature and pressure ratios was investigated by Klein and Reindl (1998). However, the calculated values of relative capacity index and the general conclusions of this paper were not affected by the different methods or values used to quantify compressor performance.

When a liquid-suction heat exchanger is employed, the refrigerant entering the compressor (state 2) has been superheated by heat exchange with the liquid exiting the condenser which causes the liquid to enter the expansion device in a subcooled state (state 4). In practice, the beneficial effects of a liquid-suction heat exchanger are offset by the refrigerant pressure drops that occur in the heat exchanger. Performance estimates are first provided for no pressure losses. A method for correcting the estimates for pressure losses is provided later in the paper.

Calculated relative capacity indices are presented in Figure 3 for different refrigerants and heat exchanger effectiveness values at a fixed saturated evaporator temperature of -20°C (-4°F) and a saturated condensing temperature of 40°C (104°F). These calculations assume the refrigerant flow rate to be constant and no pressure losses through the liquid-suction heat exchanger. The effect of these assumptions is considered in following sections. The results in Figure 3 indicate the potential increase in capacity possible by subcooling the liquid refrigerant before expansion. An increase in capacity is observed for all refrigerants although there is considerable variation in the magnitude of the effect. The relative capacity increase for refrigerant R507A at a heat exchanger effectiveness of unity is 58.5% while the increase in relative capacity for R717 (ammonia) at the same conditions is only about 13%. The relation between the relative capacity index and liquid-suction heat exchanger effectiveness is nearly linear. The relative capacity index is affected by both the saturated evaporator and condensing temperatures. For example, the relative capacity indices for R507A are 84% and 38% at a condenser temperature of 40°C (104°F) and evaporator temperatures of -40°C (-40°F) and 0°C (32°F), respectively, while the relative capacity indices for R717 are 17% and 9% at these same conditions. The effect of saturated evaporator and condenser temperatures is quantified later in terms of the temperature lift defined as the difference between the saturated condensing and evaporating temperatures.

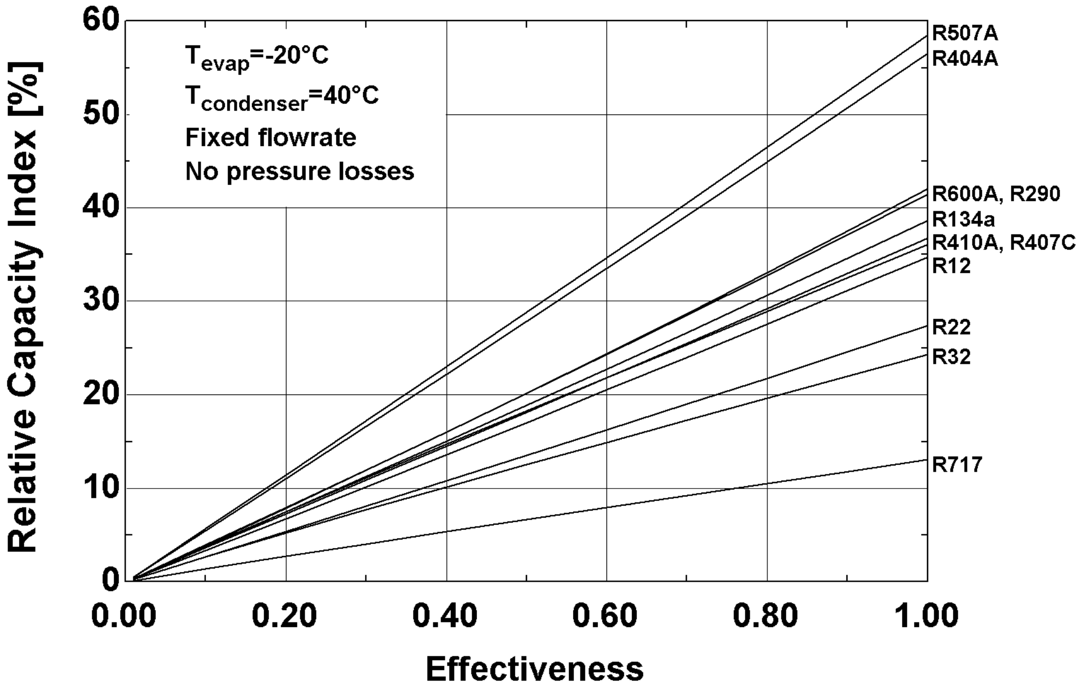

Figure 3: System capacity change as function of the liquid-suction heat exchanger effectiveness ignoring corrections for system mass flow rate changes

Heat Exchanger Effect on Capacity with Mass Flow Rate Corrections

A critical element not included in the calculated results shown in Figure 3 is the effect that superheating the compressor suction gas has on the mass flow of refrigerant delivered by the compressor. Most compressors are fixed volumetric flow devices (i.e. they operate at a fixed displacement rate); consequently, the mass flow of refrigerant the compressor delivers will be a function of the suction specific volume (Stoecker, 1988). The refrigeration capacity can be expressed in terms of the compressor displacement rate and a volumetric efficiency, refrigerant suction density, and change in enthalpy across the evaporator as indicated in equation (3):

$$\mathrm{C}apacity = CFM \;\; h_v \;\; r_1 \left(\mathrm{h}_1 - \mathrm{h}_4\right) \tag{3}$$

where

CFM	is the volumetric displacement rate of the compressor
h_v	is the compressor volumetric efficiency
r_1	is the density of refrigerant at the compressor inlet
h_1	is the specific enthalpy of refrigerant entering the compressor
h_4	is the specific enthalpy of the refrigerant entering the expansion device

The volumetric efficiency can be approximately represented in terms of the ratio of the clearance volume to the displacement volume, R, and the refrigerant specific volumes at the compressor suction and discharge, v_1 and v_2, as indicated in equation (4):

$$h_v = 1 - R\left(\frac{v_1}{v_2} - 1\right) \qquad (4)$$

As the effectiveness of the liquid-suction heat exchanger increases, the refrigerant entering the compressor at state 2 achieves a greater degree of superheat which reduces both its density and the compressor volumetric efficiency. Pressure losses on the low-pressure side of the heat exchanger result in a further reduction in refrigerant density which is considered below. 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 2); 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°C (-4°F) and a saturated condensing temperature of 40°C (104°F) is shown in Figure 4.

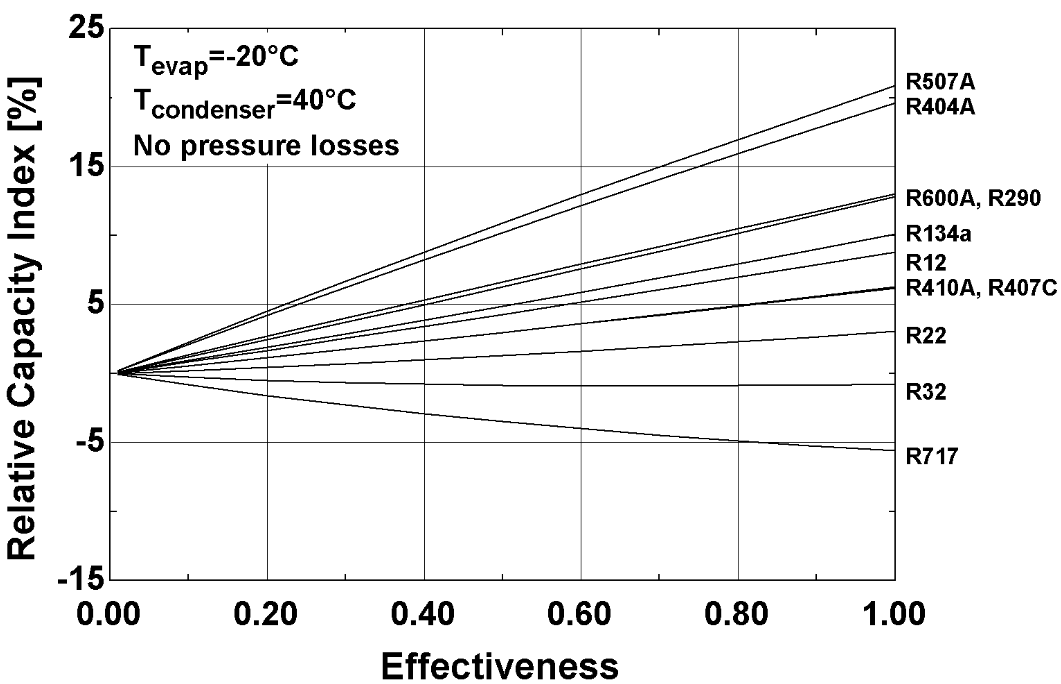

Figure 4: Relative capacity (and relative system COP) index as a function of liquid-suction heat exchanger effectiveness for various refrigerants at -20°C evaporating temperature and 40°C condensing temperature.

In addition to the influence of liquid-suction heat exchangers on system capacity, it is also important to consider their influence on the system coefficient of performance. This requires knowledge of how the refrigeration system power varies with liquid-suction heat exchanger performance. Threlkeld (1962) develops the following approximate expression for compressor work (on a per unit mass flow rate basis) assuming a polytropic compression process:

$$W = \frac{nP_2 v_2}{(n-1)}\left[\left(\frac{P_3}{P_2}\right)^{(n-1)/n} - 1\right] \tag{5}$$

where P_2 is the absolute pressure at the compressor suction, P_3 is the absolute pressure at the compressor discharge, v_2 is the refrigerant specific volume at the compressor suction and n is a polytropic index. The compressor power can be calculated knowing the refrigerant mass flow rate and the motor efficiency as given by equation (6):

$$\dot{W}_{compressor} = \dot{m}_{ref} \cdot \frac{W}{\eta_{motor}} = \frac{CFM}{v_2\,\eta_{motor}} \cdot \frac{nP_2 v_2}{(n-1)}\left[\left(\frac{P_3}{P_2}\right)^{(n-1)/n} - 1\right] \tag{6}$$

The compressor volumetric displacement rate is solely a function of motor speed and independent of the liquid subcooling and suction superheat produced by a liquid-suction heat exchanger. The compressor suction pressure is controlled (typically by loading and unloading the compressor) as is the discharge pressure (typically, by controlling the capacity of the heat rejection device). Neither the compressor suction or discharge pressure are a function of the liquid subcooling or suction superheat that results from the installation of a liquid-suction heat exchanger. The polytropic constant, *n*, is also assumed to not be a function of the level of liquid subcooling or suction superheat. As a result, the compressor power is unaffected by the operation of a liquid-suction heat exchanger, assuming the pressure drops in the heat exchanger are negligible.

Since the system COP change is directly related to the change in capacity, the percentage change in system COP is equivalent to the percentage change in system capacity, again assuming the pressure drops in the heat exchanger to be negligible. Accounting for the decrease in refrigerant mass flow rate that results from increasing the suction inlet temperature, the effect of a liquid-suction heat exchanger on COP with various refrigerants is identical to that found for capacity in Figure 4. Pressure losses in the liquid-suction heat exchanger have different effects on COP and capacity, as noted below.

Correlation of Results (neglecting pressure losses)

The results in Figure 4 indicate that a liquid-suction heat exchanger increases system capacity (and COP) for some refrigerants and decreases it for others. It is logical to question what causes the refrigerants to behave differently. An analysis and explanation of the behavior of different refrigerants is presented by Domanski and Didion (1994). Using a simple model that assumes isentropic compression and ideal gas behavior, they show that the improvement in COP resulting from the use of a liquid-suction heat exchanger should improve if $\Delta h_{vap}/C_{p,v}$ (enthalpy of vaporization at the evaporation temperature divided by isobaric specific heat of the vapor) and B are minimized and $(T_{cond}-T_{evap})(C_{p,L}/C_{p,v})$ is maximized. The parameter B is an average coefficient of thermal expansion defined as:

$$B = \frac{v_2 - v_1}{v_2 (T_2 - T_1)} \tag{7}$$

where states 1 and 2 are identified in Figure 1. Domanski and Didion note that the ratio of heat capacities of liquid and vapor exerts stronger influence with increasing temperature lifts. They tabulate the properties relating to liquid-suction heat exchanger performance for 29 refrigerants. However, some refrigerants of current interest, such as R507A, R404A, R407C, R410A, and R717, are not included in their results.

Domanski and Didion caution that relationships other than those they identified with their simple model influence refrigerant performance in the basic refrigeration cycle. They then investigate liquid-suction heat exchanger performance using a simulation model. Property data in the model are based on the Carnanhan-Starling-DeSantis equation of state that was employed in the REFPROP 4 and 5 programs (Gallagher et al., 1993). They present simulation results for 29 refrigerants; however, it is difficult to directly compare the performance of alternative refrigerants because the simulation results are presented for a reduced saturated condensing temperature of 0.82 and a reduced saturated evaporating temperature of 0.65. As a result, the simulation results for each refrigerant are at different saturated condensing and evaporating temperatures and at differing temperature lifts. Application charts are presented for four refrigerants to quantify the effect of temperature lift, but the effect of pressure losses in the liquid-suction heat exchanger is not addressed.

One objective of this paper has been to identify a general correlation of liquid-suction heat exchanger performance for different refrigerants. The parameters identified by Domanski and Didion were first investigated to determine whether simulation results could be correlated; however, a satisfactory correlation could not be established since these parameters do not include all of the refrigerant-specific influences on cycle performance (as noted by Domanski and Didion). A systematic evaluation of the dimensionless refrigerant properties revealed that the relative capacity index for a specified temperature lift correlates well with the dimensionless quantity $\Delta h_{vap}/(c_{p,L}\, T_c)$ where Δh_{vap} is the enthalpy of vaporization at the evaporator pressure, $c_{p,L}$ is the specific heat of saturated liquid refrigerant at the evaporator temperature and T_c is the critical temperature of the refrigerant. The relationship between the relative capacity index and this dimensionless quantity is shown in Figure 5 for the 11 refrigerants investigated in Figure 3 and 4 at an evaporator temperature of -20°C (-4°F) and a condensing temperature of 40°C. The line shown in the figure represents a best-fit second-order polynomial which represents the relationship with a R^2 of 0.95.

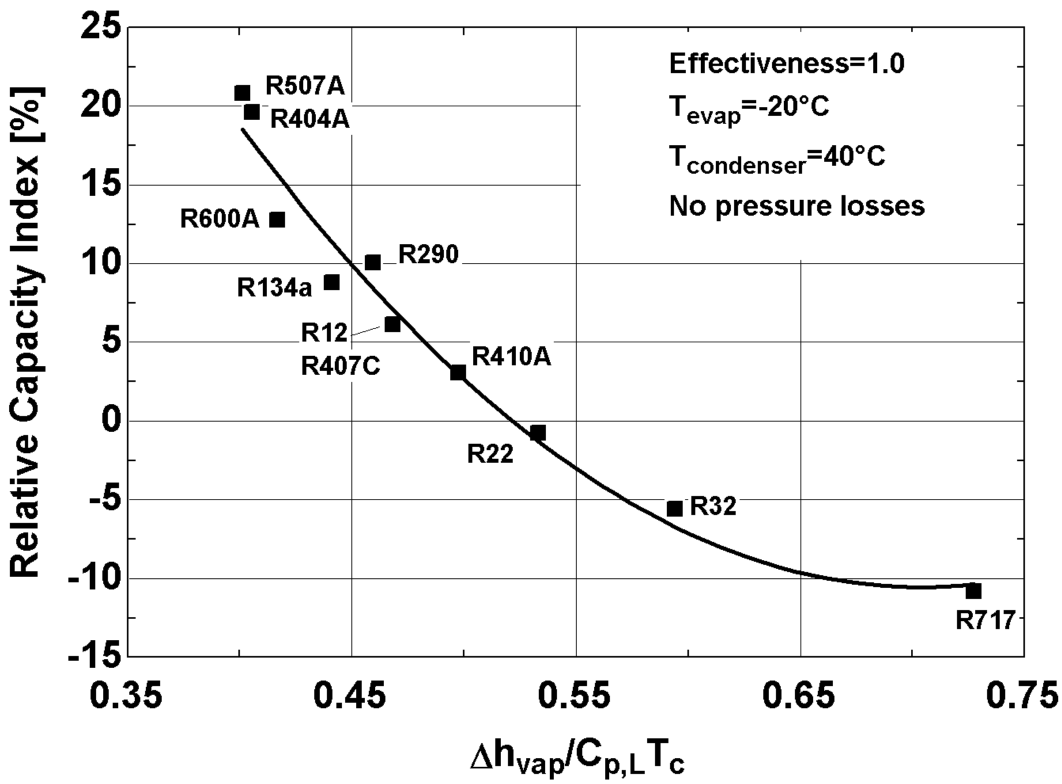

Figure 5: Relative capacity index versus $\Delta h_{vap}/(c_{p,L}\ T_c)$ at saturated evaporating and condensing temperatures of -20°C and 40°C, respectively.

The relative capacity index is also a strong function of condensing and evaporating temperatures as shown by Domanski and Didion (1994); however, it is the difference between these temperatures, the temperature lift, rather than the individual temperatures that affects the performance of a liquid-suction heat exchanger. Simulation results were obtained for a range of evaporator temperatures between -40°C and 10°C and for condensing temperature between 10°C and 60°C. These relative capacity (RCI) results are presented in Figure 6 in terms of D = $\Delta h_{vap}/(c_{p,L}\ T_c)$ and L, the temperature lift. The eleven refrigerants used in this investigation are not identified in Figure 6 to avoid clutter, although their position can be surmised from Figure 5. Linear regression was used to correlate the results in Figure 6 (for ε=1) and similar results for other values of ε. The resulting correlation is presented in equation (8):

$$\begin{aligned} RCI/e &= -3.0468 + 19.3484\ D - 19.091 D^2 + 1.2094\ L + 0.02101\ L^2 - 5.9980\, D\, L \\ &\quad - 0.02797\ D L^2 + 5.52865\ D^2 L \end{aligned} \tag{8}$$

where

D = $\Delta h_{vap}/(c_{p,L}\ T_c)$

L = $(T_{cond} - T_{evap})$

The lines shown in Figure 6 were generated using equation 8. Equation 8 fits the simulation data with a standard deviation in relative capacity of 0.34 and an R^2 of 0.95. The agreement of the fit and simulation results is better at low lifts. Relative capacity was found to be linearly dependent on the liquid-suction heat exchange effectiveness and this relationship is included in equation 8. Figure 6 shows that liquid-suction heat exchangers offer the highest capacity (and, therefore, COP) at low values of $\Delta h_{vap}/(c_{p,L}\ T_c)$ and at high temperature lifts. Equation (8) provides a general means of estimating the capacity improvement expected from a liquid-suction heat exchanger for any refrigerant and temperature lift within the range of values investigated. However, neither Figure 6 nor equation (8) account for pressure losses in the liquid-suction heat exchanger. This additional effect is considered in the following section.

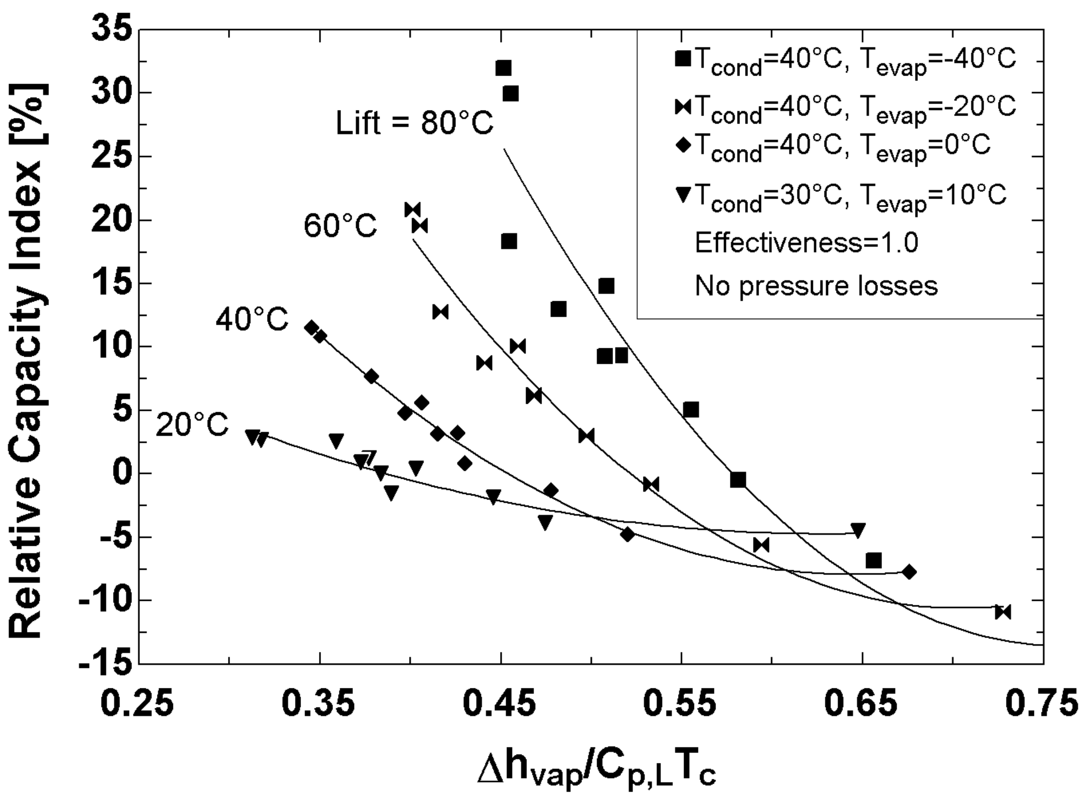

Figure 6: Relative capacity index vs $\Delta h_{vap}/(c_{p,L}\ T_c)$ for various temperature lifts for a liquid-suction heat exchanger with no pressure losses and effectiveness=1.0

Effect of Pressure Losses in the Liquid-Suction Heat Exchanger

The results presented in Figures 3-6 all assume that there are no pressure losses in the liquid-suction heat exchanger. The relative capacity index and COP will both be reduced if pressure losses occur. Pressure drops are unavoidable in heat exchangers. However the magnitudes of the pressure drops in the liquid and vapor lines can not be predicted in general since they depend on the heat exchanger design as well as the refrigerant properties.

The pressure drops in the liquid and vapor lines have different effects. A pressure drop in the liquid (high pressure) line will have much less effect on capacity and COP than a pressure drop of equal magnitude in the suction (low pressure) line. The result of the pressure loss in the liquid line is to reduce the pressure of the refrigerant upstream of the expansion device. Assuming that the pressure drop is sufficiently small such that flashing does not occur ahead of the valve, the pressure drop will have little effect on relative capacity because the liquid refrigerant is nearly incompressible and its properties are not affected by the reduction in pressure. The tendency to flash before the valve is reduced by the reduction in liquid refrigerant temperature as it passes through the heat exchanger.

A pressure loss in the vapor (low pressure) leg of the liquid-suction heat exchanger affects both capacity and COP. The pressure loss reduces the density of the refrigerant entering the compressor and thereby results in reduced refrigerant mass flow rate which in turn results in reduced capacity. In addition, more work per unit mass is required to increase the pressure to the level in the condenser and the volumetric efficiency is reduced, as indicated in equations (4) and (5). Since compressor power is unaffected by the increased superheat, the effect of the liquid-suction heat exchanger on COP is identical to relative capacity index. However, pressure loss affects capacity and the compressor power differently, so changes in COP will not necessarily be the same as changes in capacity when a liquid-suction heat exchanger with pressure losses is introduced.

Refrigeration systems having a liquid-suction heat exchanger were simulated for a range of temperature lifts, effectiveness values, and pressure losses for the eleven refrigerants identified in Figure 5. The simulation results indicate that the effect of pressure loss in the liquid-suction heat exchanger on refrigeration capacity and COP can be represented in terms of a non-dimensional pressure loss defined as the pressure loss in the low pressure leg of the liquid-suction heat exchanger divided by the absolute pressure in the evaporator. The effect of this non-dimensional pressure loss on refrigeration capacity is shown in Figure 7. The ordinate in Figure 7 is the capacity of the refrigeration system divided by the capacity that the system would have if there were no pressure losses in the low-pressure leg of the liquid-suction heat exchanger, all else being the same. Figure 7 indicates that there is a linear relationship between the reduction in capacity and the non-dimensional pressure loss. The relationship is independent of the liquid-suction heat exchanger effectiveness. At temperature lifts below 40°C, there is no discernable dependence on the refrigerant but some dependence becomes evident at higher lifts. The information in Figure 7 can be approximately represented using equation (9):

$$\frac{Capacity}{Capacity_{no\,pressure\,losses}} = 1 - \left(1.042 - 7.32\times10^{-7} L^3\right)\left(\frac{\Delta P_{HX}}{P_{evap}}\right) \tag{9}$$

The effect of pressure loss on COP is shown in Figure 8. The pressure loss results in reduced refrigerant mass flow rate which causes reduced capacity. Reducing the refrigerant mass flow rate tends to reduce compressor power. However, the increased pressure ratio resulting from the pressure loss tends to increase compressor power. The net effect on COP can be represented in terms of the non-dimensional pressure loss and temperature lift, independent of the liquid-suction heat exchanger effectiveness. There is very little effect of refrigerant for the eleven refrigerants investigated. The information in Figure 8 can be represented with equation (10):

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1-\left(2.37-0.0481L+3.01\times10^{-4}L^2\right)\left(\frac{\Delta P_{HX}}{P_{evap}}\right) \qquad (10)$$

This paper provide a means of estimating the effect of a liquid-suction heat exchanger for any refrigerant for which property data are available. Equation (8) is first used to determine the relative capacity index for a liquid-suction heat exchanger of specified effectiveness assuming that there are no pressure losses. Then, the result obtained from equation (8) is multiplied by the factor in equation (9) to account for reduced capacity resulting from pressure losses in the low pressure leg of the heat exchanger. The product of the result obtained in equation (8) and the factor in equation (10) indicates the net result of the liquid-suction heat exchanger on COP. At this point, an economic assessment can then be made to determine the overall merit of the liquid-suction heat exchanger.

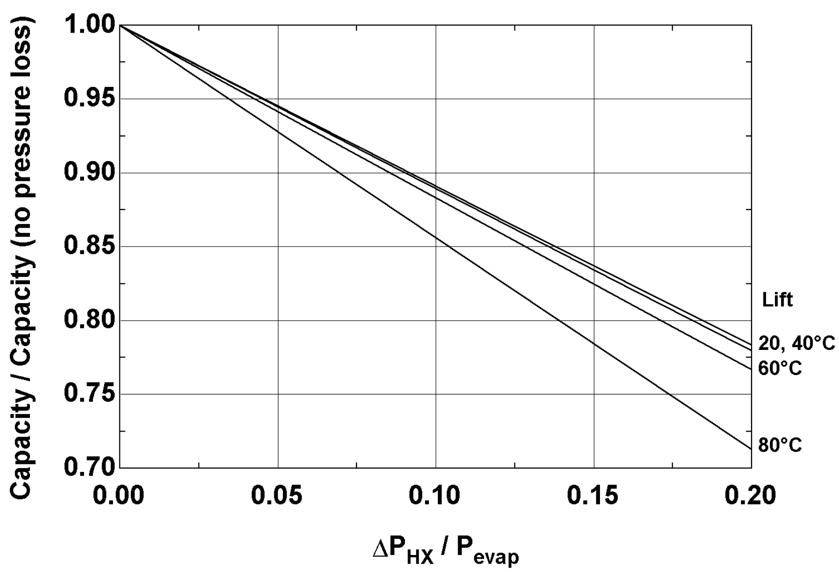

Figure 7: Correction to the relative capacity index to account for pressure loss in the low pressure leg of the liquid-suction heat exchanger.

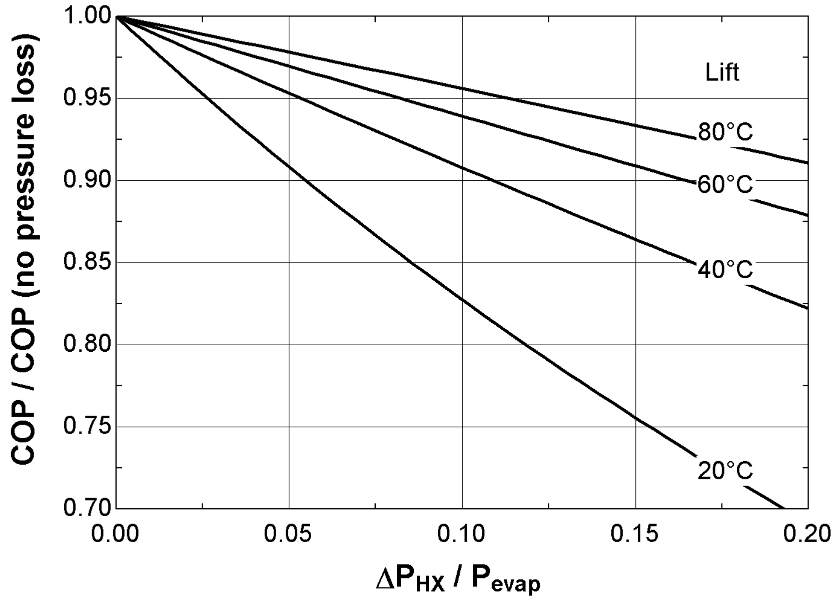

Figure 8: Correction to the COP to account for pressure loss in the low pressure leg of the liquid-suction heat exchanger.

Conclusions

By neglecting the reduction in refrigerant mass flow rate, one would conclude that liquid-suction heat exchangers lead to performance improvements for any refrigerant. Under closer evaluation, 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. Although the compressor power is only slightly affected by the change in state of the refrigerant entering the compressor, the refrigerant mass flow rate is reduced. Consequently, the advantage of liquid-suction heat exchangers depends on competing effects. Figures 4 illustrates the influence of liquid-suction heat exchangers (with no pressure losses) on the performance of a refrigeration system for a number of refrigerants accounting for changes in compressor volumetric efficiency. The effect of a liquid-suction heat exchanger (with no pressure losses) on the refrigeration capacity can be correlated in terms of the temperature lift and a dimensionless grouping equal to the enthalpy of vaporization at the evaporator temperature divided by the product of the liquid specific heat (evaluated at the evaporator temperature) and the critical temperature. The effect of pressure losses in the low pressure leg can be quantified in terms of a non-dimensional pressure difference. From this analysis, it can be concluded that liquid-suction heat exchangers are most useful at high temperature lifts and for refrigerants having a relatively small value of $\Delta h_{vap}/(c_{p,L}\ T_c)$. The potential performance advantage of a liquid-suction heat exchanger is reduced due to pressure losses in the heat exchanger. A general method of estimating the magnitude of the reduction is provided in Figure 7 and equation (9). The liquid-suction heat exchanger is detrimental to system performance in systems using R22, R32, and R717 at all temperature lifts investigated. The results obtained for R134a, R12 and R22 follow the same trends as the results of Domanski and Didion, (1994). However, the present research expands their results by examining additional refrigerants and an alternative method of correlating the performance results. Even though the liquid-suction heat exchanger has a negative impact on system performance, the system does benefit from the heat exchanger by preventing vapor in the liquid line before the expansion valve. The system designer must thus be very careful in choosing when to install a liquid-suction heat exchanger in a refrigeration system.

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