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A SIMPLIFIED CYCLE SIMULATION MODEL

FOR THE PERFORMANCE RATING OF REFRIGERANTS AND REFRIGERANT MIXTURES

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ABSTRACT

A simulation program, CYCLE11, which is useful for the preliminary evaluation of the performance of refrigerants and refrigerant mixtures in the vapor compression cycle is described. The program simulates a theoretical vapor-compression cycle and departures from the theoretical cycle as occur in a heat pump and in a refrigerator. The cycles are prescribed in terms of the temperatures of the external heat transfer fluids with the heat exchangers generalized by an average effective temperature difference. The isenthalpic expansion process is assumed. The program includes a rudimentary model of a compressor and a representation of the suction line and liquid line heat exchange. Refrigerant thermodynamic properties are calculated using the Carnahan-Starling-DeSantes equation of state. Refrigerant transport properties are not included in the simulations.

The program can generate merit ratings of refrigerants for which limited measurement data are available. An example of simulation results stresses the need for careful application of simplified models and consideration for the involved assumptions.

(Keywords: vapor compression cycle, refrigerants, modeling)

NOMENCLATURE

A	heat transfer area	Subscripts:
COP	coefficient of performance	a ambient air
C_p	heat capacity at constant pressure	c condenser
C_v	heat capacity at constant volume	cr critical
HTF	heat transfer fluid	ev evaporator
h	enthalpy	hx heat exchanger
L	length	p polytropic
n	polytropic index	s isentropic
P	pressure	v volumetric
Q	heat transfer rate	1-11 key locations in the system
R	universal gas constant	per Figure 1b or con-
T	temperature	secutive sections in a heat
U	overall heat transfer coefficient	exchanger
V	molar volume	
W	work	
γ	isentropic index	
η	efficiency	
ρ	density	

INTRODUCTION

The implication of CFC refrigerants in the destruction of stratospheric ozone and internationally agreed limitations on CFC use have spurred intensive efforts towards the identification of suitable replacement refrigerants. The selection of a refrigerant for a vapor compression machine is a long and elaborate process. A potential replacement must satisfy a number of qualifying (or gate) properties as well as possess favorable thermodynamic and transport properties (Threlkeld[1], McLinden and Didion[2]). The thermodynamic properties are most important in determining cycle performance, and they receive primary attention during the selection process, particularly in the beginning phases.

A number of methods may be used to predict the performance of the vapor compression cycle. They may be simplistically divided into the following categories:

(1) Carnot cycle analysis
(2) simple methods based on fundamental observations and principles
(3) theoretical and semi-theoretical cycle analysis
(4) detailed equipment simulation programs
(5) laboratory test of the vapor compression machine

Category (1), the Carnot Cycle, represents the limit for the Coefficient of Performance (COP) of a refrigeration cycle operating between two fixed (constant) temperatures. This cycle assumes reversible compression and expansion processes, and isothermal heat withdrawal and rejection (thus requiring infinite heat exchangers and an infinite heat sink and heat source). The coefficient of performance of the Carnot cycle is independent of fluid properties and is thus not suited for refrigerant screening studies except as a reference for the ultimate performance limit.

Category (2) consists of methods that could be used for general screening of refrigerants. They require only limited property data; typically values of key properties at some specified reference temperature. Although these methods are expressed in terms of simple rules or equations, they may be based on significant theoretical insight. The methodology proposed by Alefeld[3] is in this category. Starting from the First and Second Law of Thermodynamics, he derived a simple equation for COP. The Carnot efficiency of the cycle is modified by terms involving the isentropic efficiency of the compressor, fluid properties, and the evaporator temperature. For the HCFC fluids the fluid property term requires only the ratio of the liquid heat capacity to the latent heat of vaporization.

Angelino and Invernizzi[4] presented the thermodynamic merits of refrigerants in terms of a cycle quality defined as the ratio of actual COP to the COP of the Carnot cycle. Their study showed that the main parameters affecting the cycle quality are the complexity of the fluid molecular structure, reduced temperature at which evaporation is performed, and the fractional temperature lift, $\Delta T/T_e$. McLinden[5] reached a similar conclusion in a study evaluating refrigerants in the vapor compression cycle using reduced properties. He showed that the cycle COP is a function of the reduced temperature of the condenser, temperature lift, and the ideal gas heat capacity (C_p at the limit of zero pressure). It should be noted that the molecular structure, indicated by Angelino and Invernizzi[4], and the ideal gas heat capacity used by McLinden[5], are related since each chemical bond in the refrigerant molecule provides a certain contribution to the specific heat of the molecule.

The evaluation methods in Category (3) model the performance of refrigerants in a specified cycle and require a complete set of refrigerant thermodynamic properties. This level of evaluation ranges from a theoretical cycle composed of idealized thermodynamic processes, to a practical cycle which would include effects such as refrigerant subcooling at the condenser, refrigerant superheat at the evaporator, pressure drops in heat exchangers, and a representation of the temperature difference between fluids exchanging heat.

The fourth category of evaluation methods consists of detailed computer models (for example, Domanski and Didion[6], Fisher and Rice[7]). Complete thermodynamic and transport property data are needed as well as a detailed description of the modeled equipment. These models can provide system performance information very close to those which would be obtained from a system test. Results from these simulations can be used to compare the performance of different refrigerants in identical hardware or to evaluate the impact of hardware modifications on system performance.

The final verification of the refrigerant performance is an actual test in equipment (Category (5)). It is the ultimate and most costly evaluation. As such, refrigerant tests in equipment are usually performed in the advanced stages of refrigerant selection although, because detailed property data are not needed, it may be desirable to run such tests at an early stage.

The relative merits of these five categories of evaluation methods will depend on the goals of the evaluation. For the purposes of screening among a set of closely related fluids or mixtures and ranking their relative performance in order to select one or a few candidates for detailed study, the theoretical and semi-theoretical cycle models in Category (3) are the most appropriate. The Carnot

cycle analysis is clearly of no use for this type of screening. The methods of Category (2) would be useful for cutting the list of candidate working fluids from the many thousands of known chemical compounds to a few dozen of the most promising fluids. The general methods of Category (2) would not, however, be able to discern small differences among closely related fluids.

In progressing from the Category (3)-type models to the detailed equipment models (Category 4) and finally to equipment tests (Category 5), the results are influenced more and more by the detailed specification of the equipment and the degree to which the equipment design is optimized for a particular working fluid (typically one of the long used refrigerants). Because of this, laboratory studies, particularly those testing many different fluids in unmodified equipment, can give misleading results for new or unusual refrigerants.

Transport properties are not involved in Category (3) cycle simulations and this is their major shortcoming. It is important to consider how this shortcoming affects the performance predictions. In a sensitivity study using a very detailed heat pump model, Domanski and Didion[6] assessed the impact of fluid properties on heat pump performance. The study revealed that a change of liquid thermal conductivity of 50 percent changed the COP by only five percent, and the same magnitude change in liquid viscosity altered the COP by less than four percent. Similar changes in the transport properties of the vapor resulted in COP changes of less than 0.4 percent. (Of course, this relates to overall system performance; transport properties are considerably more important in component design, particulary heat exchangers.)

This paper describes a Category (3)-type model and illustrates how it might be used to select a working fluid for a domestic refrigerator. Emphasized in this discussion are the effects of different assumptions and levels of modeling detail on the ranking of different mixtures.
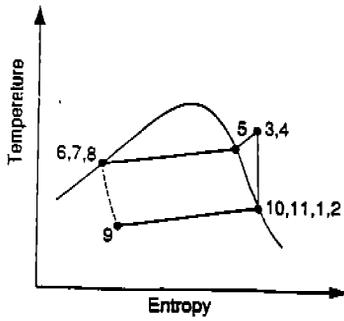
DESCRIPTION OF THE MODEL

The model described here is termed CYCLE11. The name refers to both the number of modeled refrigerant state points and its evolution from the earlier CYCLE7 model developed by McLinden[9]. The simulation options available in CYCLE11 are presented in Figure 1 in the form of thermodynamic cycles which can be simulated. The basic cycle (Figure 1a), consists of an isentropic compression, isobaric heat transfer in both heat exchangers, and an irreversible, adiabatic expansion. The most elaborate cycle modeled by CYCLE11 (Figure 1b) includes liquid line/suction line heat exchange and a rudimentary model of a hermetic compressor; this cycle is intended to approximate a household refrigerator. As an intermediate option the cycle realized by an air-conditioner or a heat pump may be simulated by specifying the appropriate refrigerant superheat at the compressor inlet and subcooling at the condenser outlet, by including the option for the hermetic compressor, and by excluding the liquid line/suction line heat exchanger.

The simulated cycle is outlined by eleven states corresponding to key locations in a real system. These states, as shown in Figure 1b, correspond to:

1 - suction line outlet, inlet to the shell of the hermetic compressor
2 - refrigerant state in the cylinder before the compression process
3 - refrigerant state in the cylinder after the compression process
4 - compressor shell outlet, condenser inlet
5 - saturated vapor refrigerant state in the condenser
6 - saturated liquid refrigerant state in the condenser
7 - condenser exit, liquid line inlet
8 - liquid line exit, inlet to the adiabatic expansion device
9 - expansion device outlet, evaporator inlet
10 - saturated vapor refrigerant state in the evaporator
11 - evaporator outlet, suction line inlet

At input, refrigerant or refrigerant mixture is selected along with the inlet and outlet temperatures of external heat transfer fluids (HTFs) and other operating conditions (shown latter in the paper). According to the solution logic shown in Figure 2, CYCLE11 iterates temperatures in the heat exchangers so the resulting average effective temperature difference agrees with the specified value within a prescribed tolerance. The output of the model includes refrigerant thermodynamic properties at the key cycle points, and refrigerating and heating COPs and capacities.

a) theoretical cycle

1-4 - compressor
4-7 - condenser
7-8 - liquid line
8-9 - expansion device
9-11- evaporator
11-1- vapor line
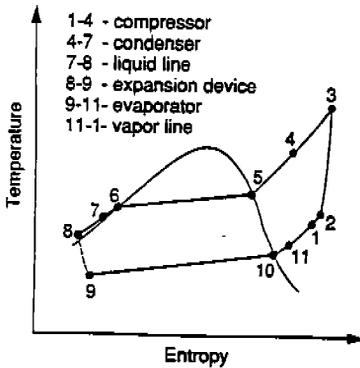
b) refrigerator cycle

Figure 1   Thermodynamic cycles simulated
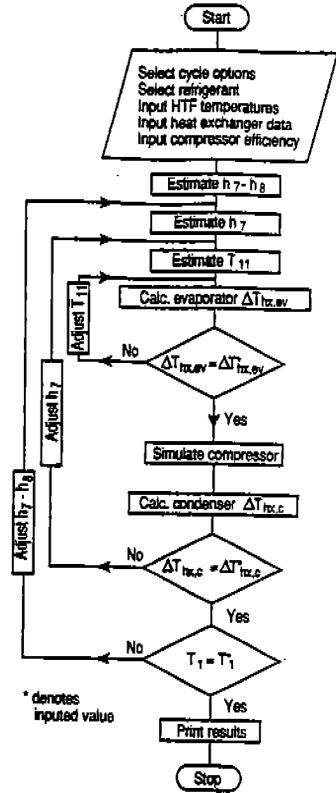   by CYCLE11

Figure 2   Solution logic for the
           refrigeration cycle

Compressor model

The CYCLE11 model has three options for simulating the compression process. They are: (1) isentropic process, (2) polytropic process, and (3) either of the these processes with the inclusion of volumetric efficiency and a representation of the heat transfer to the suction gas and from the discharge gas which occurs in a hermetic compressor. The last option provides a representation of the significant departures from theoretical compression that occur in a hermetic compressor. Not accounting for these deviations would also affect the conditions of the refrigerant entering the condenser and would thus unduly affect the overall simulation results.

Complicated heat transfer mechanisms within the compressor shell are modeled by simplifying relations. An increase in the refrigerant temperature of the low pressure refrigerant between the shell inlet and the cylinder inlet is a result of heat transfer from the electric motor windings and high pressure, high temperature refrigerant. The calculation of this temperature increase assumes that the temperature difference between the high and low pressure refrigerant is the only heat transfer driving force:

$$T_2 - T_1 = C_{1.2} \frac{(T_4 - T_1) \cdot \rho_2^{0.8}}{C_{p1,2}^{0.66}}$$

(1)

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The coefficient $C_{1,2}$ is an empirical heat conductance factor; the value of $C_{1,2}$ was adjusted to match the refrigerant temperature change experienced in an actual refrigerator operating with R12 at the test conditions prescribed by the U.S. Department of Energy[10]. The symbol $C_{p1,2}$ denotes mean heat capacity at constant pressure between states 1 and 2.

The temperature decrease of the high pressure refrigerant after the compression process is due to heat losses to the suction refrigerant and ambient air.

$$T_4 - T_3 = - C_{3,4} \frac{(T_3 - 0.5 \cdot (T_1 + T_A)) \cdot \rho_2^{0.8}}{C_{p3,4}^{0.66}} \tag{2}$$

The coefficient $C_{3,4}$ is similar to $C_{1,2}$. Equations (1) and (2) were obtained using the stated heat balances and considering that forced convection heat transfer is related to the Prandtl number (hence also heat capacity) raised to the 0.33 power (Colburn[11]), and to the Reynolds number (hence the refrigerant mass flux) raised to the 0.8 power. The density of vapor at the compressor cylinder inlet (point 2) is representative, on the relative bases, of the refrigerant mass flow through a compressor of a constant intake volume.

A polytropic analysis, rather than an isentropic analysis, is used in the simulation of the compression process. The isentropic efficiency varies with pressure ratio while polytropic efficiency better describes compressor performance at various operating conditions (Wilson[12], Schultz[13]). Using the polytropic efficiency provides more validity to a comparative evaluation of refrigerants since they may operate in a given cycle at quite different compression ratios.

The refrigerant state after compression is calculated using the polytropic efficiency (an input datum). For a known (or guessed) refrigerant discharge pressure, the refrigerant enthalpy after compression, h_3, is calculated by the relations:

$$h_3 = h_2 + \frac{W_p}{\eta_p} \tag{3}$$

$$W_p = \frac{n}{n-1} P_2 \cdot V_2 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right] \tag{4}$$

where the polytropic index, n, is defined by:

$$\frac{n-1}{n} = \frac{(\gamma-1)/\eta_p}{\gamma} \tag{5}$$

It should be noted that the isentropic index, γ, if evaluated for a real gas as the ratio of heat capacities at constant pressure and constant volume ($\gamma = C_p/C_v$) does not satisfy the equation for the isentropic work derived for ideal gas:

$$(h_3 - h_2)_s = \frac{\gamma}{\gamma-1} (P_3 V_3 - P_2 V_2)_s \tag{6}$$

where the subscript s refers to an isentropic process. Therefore, for a consistent representation of the compression process, the isentropic index, γ, is evaluated based on equation (6). For R12 at the operating pressures of a domestic refrigerator, the value of γ calculated by equations (6) is lower by approximately 14 percent than the value obtained from the heat capacity ratio.

This lower value can be verified theoretically, as shown by Morrison[14]. Starting with a partial differential relation for pressure change of gas,

$$dP = \left(\frac{\partial P}{\partial V} \right)_T dV + \left(\frac{\partial P}{\partial T} \right)_V dT \tag{7}$$

and rearranging with the Maxwell relation, $(\partial T/\partial V)_S = -(\partial P/\partial S)_V$, and the Gibbs function, a general equation for the isentropic process is obtained:

$$\left(\frac{\partial P}{\partial V}\right)_s = \frac{C_p}{C_v}\left(\frac{\partial P}{\partial V}\right)_T \tag{8}$$

Combining equation (8) with the perfect gas equation of state, $PV = RT$, results in the isentropic process equation of the ideal gas:

$$\left(\frac{\partial P}{\partial V}\right)_s = -\frac{P}{V}\frac{C_p}{C_v}, \qquad \text{or} \qquad P \cdot V^7 = const \tag{9}$$

Combining equation (8) with the virial equation of state, $PV = RT(1+B/V)$ results in the relation:

$$\left(\frac{\partial P}{\partial V}\right)_s = -\frac{P}{V}\frac{C_p}{C_v}\left[1 + \frac{B \cdot R \cdot T}{P \cdot V^2}\right] \tag{10}$$

Equations (9) and (10) are identical except for the last term of equation (10) which may be viewed as a modifying term for the heat capacity ratio. Since the second virial coefficient, B, is negative for all fluids for temperatures below approximately $2.75 \cdot T_{cr}$, it may be concluded that the isentropic index for all refrigeration applications is smaller than the ratio of heat capacity at constant pressure to the heat capacity at constant volume. (The virial equation of state is used here for clarity of presentation; the same could be proved with other real gas equations of state but with greater effort).

Volumetric capacity and efficiency.
    Volumetric capacity is defined here as the heating or refrigerating capacity per unit mass of the circulating refrigerant divided by the specific volume of the refrigerant vapor in the cylinder before the compression process. In a comparative analysis of different refrigerants, volumetric capacity is indicative of the relative capacities in the same compressor system. CYCLE11 provides volumetric capacity for an ideal compressor with zero clearance volume, no valve losses, and no high/low side gas leakage (i.e. a volumetric efficiency equal to 1).

    The model also provides a value of the volumetric efficiency, and respective value of volumetric capacity calculated for a typical clearance volume and leakage rate observed in reciprocating compressors. The relation used to calculate the volumetric efficiency has the form:

$$\eta_v = 0.96 \cdot \{1 - 0.04[(P_3/P_2)^{1/n} - 1]\} \tag{11}$$

where 0.96 is an experience factor for leakage (Hirsh[15]), and 0.04 is a factor accounting for the clearance volume.

Heat exchanger model
    Counter flow heat exchange is assumed in both the evaporator and condenser. The performance of the heat exchangers is specified in terms of an average effective temperature difference and refrigerant pressure drop. In addition, heat exchanger input data include inlet and outlet temperatures of heat transfer fluids, and refrigerant subcooling and superheat for the condenser and the evaporator, respectively.

    The heat exchanger average effective temperature difference, ΔT_{hx}, is defined by the equation:

$$\Delta T_{hx} = \frac{Q_{hx}}{UA_{hx}} \tag{12}$$

It is calculated considering individual heat exchanger sections with different flow regimes. For example, in a condenser three regimes may exist; superheated vapor, two-phase fluid, and subcooled liquid. Assuming the same overall heat transfer coefficient, U, in each section of the heat exchanger, and considering that $Q_{hx} = \sum Q_i$, and that $Q_i = U \cdot A_i \cdot \Delta T_i$ (where A_i and ΔT_i are the heat transfer area and temperature difference of an individual section), an equation is obtained in which

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the heat exchanger average effective temperature difference is the arithmetic mean of the average effective temperature differences in the individual sections weighted with the fractional heat transfer areas used by these sections.

$$\Delta T_{hx} = \frac{A_1}{A_{hx}} \Delta T_1 + \frac{A_2}{A_{hx}} \Delta T_2 + \frac{A_3}{A_{hx}} \Delta T_3 = \frac{\sum A_i \cdot \Delta T_i}{A_{hx}} \qquad (13)$$

Noting that $A_i = Q_i / U \cdot \Delta T_i$, an alternative relation is obtained in which the heat exchanger average effective temperature difference is expressed as a harmonic mean weighted with the fraction of heat transferred in individual sections of the heat exchanger:

$$\frac{1}{\Delta T_{hx}} = \frac{Q_1}{Q_{hx} \cdot \Delta T_1} + \frac{Q_2}{Q_{hx} \cdot \Delta T_2} + \frac{Q_3}{Q_{hx} \cdot \Delta T_3} = \frac{1}{Q_{hx}} \sum \frac{Q_i}{\Delta T_i} \qquad (14)$$

For single component refrigerants, an effective average temperature difference for each section is calculated as a log mean temperature difference using fluid temperatures at the ends of each section. Evaluation of the average effective temperature based only on the temperatures at each end could be highly inaccurate if the phase change exhibits a non-linear temperature profile. Thus for zeotropic mixtures, CYCLE11 splits the two-phase portion of the heat exchanger into a number of sub-sections, computes the log mean temperature for each of them, and then evaluates the heat exchanger average effective temperature difference consistently with equation (14). Mixtures of fluids having a greater difference in the pure component boiling points usually exhibit more non-linearity. As shown in Figure 3, non-linearity of the mixture R22/R123 is substantial while the temperature profile of R22/R142b is almost linear.
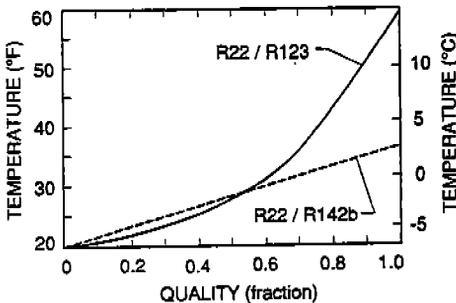

Figure 3
Temperature glide of R22/R123 and R22/R142b (50% weight fraction) for a bubble point temperature of 20°F (-6.7°C)

Refrigerant pressure drop, if specified as an input for a heat exchanger, is distributed by CYCLE11 between flow regimes in linear proportion to the heat transfer area occupied by each of the regimes. The fractional heat transfer area for a particular flow regime is given by the following relation:

$$\frac{A_i}{A_{hx}} = \frac{Q_i}{Q_{hx}} \frac{\Delta T_{hx}}{\Delta T_i} \qquad (15)$$

The two assumptions used in the derivation of the above equations (U = const and dP/dL = const regardless of refrigerant flow regime) have varying degrees of validity, depending on the application. For example, for condensers in household refrigerators these assumptions approximate reality well since most of the heat transfer resistance is on the air-side, and refrigerant pressure drop is negligible. On the other hand, for the forced convection evaporators used in air-conditioners, these assumptions are not rigorous.

Liquid line - suction line heat exchange

In a household refrigerator a significant amount of heat is transferred between the expansion device (capillary tube) and the suction line. The heat transfer between the capillary tube and the suction line is represented in CYCLE11 as the heat transfer between the liquid line and the suction line.

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The design of the suction line heat exchanger in a refrigerator ensures that the refrigerant temperature at the inlet to the compressor shell is close to the ambient temperature. CYCLE11 iterates the amount of heat exchanged between the suction line and the liquid line until the specified temperature at the compressor inlet is reached.

Representation of refrigerant properties

The refrigerant thermodynamic properties required in the cycle calculations are supplied as an independent set of property subroutines which are linked with the code for the cycle model. This structure readily permits the addition of fluids and the revision of properties.

CYCLE11 presently employs a set of property routines developed by Morrison and McLinden[16] based on the Carnahan-Starling-DeSantis (CSD) equation of state (DeSantis et al.[17]). The equation has the following form:

$$\frac{PV}{RT} = \frac{1 + y + y^2 - y^3}{(1 - y)^3} - \frac{a}{RT(V + b)} \qquad \text{where } y = \frac{b}{4V} \qquad (16)$$

The values of parameters a and b are strong functions of fluid and temperature. Morrison and McLinden showed that the CSD equation of state accurately represent both the vapor and liquid phases for HCFC refrigerants and their mixtures. Properties of 22 refrigerants and mixing coefficients for 20 mixtures are represented in CYCLE11 based on reference[18].

EXAMPLE OF SIMULATION RESULTS

To demonstrate the application of CYCLE11, several simulations were performed. A household refrigerator was simulated with inputs approximating the operating conditions specified in the US Department of Energy rating test[10]:

- temperature of HTF entering/leaving evaporator, 0°F/-7°F (-17.8°C/-21.7°C)
- temperature of HTF entering/leaving condenser, 90°F/100°F (32.2°C/37.8°C)
- average effective temperature difference for the evaporator, 11.2°F (6.2°C)
- average effective temperature difference for the condenser, 21.6°F (12.0°C)
- saturated liquid and vapor leaving the condenser and evaporator, respectively
- refrigerant temperature at the compressor shell inlet, 90°F (32.2°C)

Simulations were performed for a cycle with the liquid line/suction line heat exchanger and the hermetic compressor, as depicted in Figure 1b. A polytropic efficiency of 0.85 was used. Results for four fluids are presented in Table 1 relative to the performance of R12.

Table 1. Simulation Results for a Household Refrigerator Application
(the results are referenced to performance of R12)

Refrigerant	Weight composition	Volumetric capacity ratio v.cap/v.cap$_{R12}$		COP ratio COP/COP$_{R12}$	Condenser pressure P/P$_{R12}$
		η_v=1	calc η_v		
R134a	100%	0.968	0.865	1.052	1.05
R134	100%	0.775	0.671	1.088	0.84
R22/R142b	57% R22	1.032	1.000	1.059	1.01
R22/R152a	26% R22	1.032	1.000	1.067	1.01

Two values of volumetric capacity are provided for each refrigerant or mixture. The first value (column denoted by η_v=1) was calculated assuming no volumetric losses during the compression process. The second value incorporates the impact of the volumetric efficiency of a reciprocating compressor. The compositions of the two tested mixtures were selected to obtain the same volumetric capacity as R12. Volumetric capacities for R134a and R134 are significantly lower than that of R12. Including the volumetric efficiency decreased the volumetric capacity of R134 by as much as 10.4 percent. In terms of COP, all tested fluids

slightly outperformed R12. Condenser pressure was not excessive, being at most five percent higher than R12. R134 had the lowest pressures, with the evaporator pressure below atmospheric pressure.

It is interesting to note that different relative COP rankings are obtained if the comparison is performed using a simpler process. Two other cycles were simulated: the theoretical cycle of Figure 1a modified by inclusion of the polytropic process, and this cycle further enhanced with the liquid line/suction line heat exchange. The compression was characterized with a polytropic efficiency of 0.85 without representation of the hermetic compressor heat transfer.

As shown in Figure 4, the COPs of R134 and R134a are lower than the COP of R12 for the theoretical cycle, although they were higher than R12 for the refrigerator cycle. The difference in relative COP between these two cycles is 8.6% for R134a and 9.8% for R134. For the mixtures, the COPs are within 1.7% of R12 COP for the cycle with the liquid line/suction line heat exchanger (but without the hermetic compressor). For both the theoretical cycle and the refrigerator cycle, the mixtures have better COP than R12 by approximately six percent. This trend is not consistent with that for R134a and R134 which have COPs, relative to R12, which increase in progressing from the theoretical to the refrigerator cycle. The volumetric capacities for $\eta_v=1$ are shown in Figure 5.
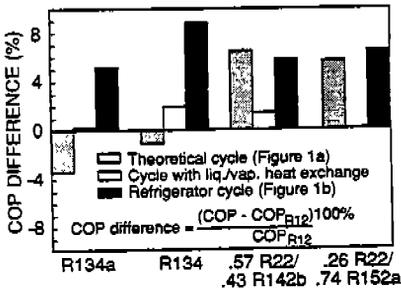
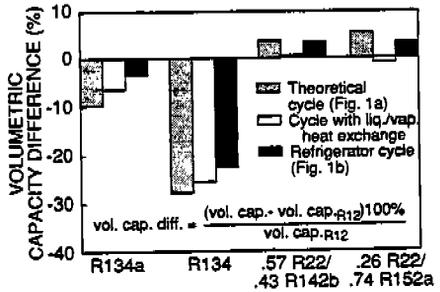Figure 4 Coefficient of Performane referenced to COP of R12

Figure 5 Volumetric capacity referenced to volumetric capacity of R12

Another set of simulations was performed to assess the error in COP prediction resulting from neglecting the temperature/enthalpy non-linearity of zeotropic mixtures undergoing a phase change. The results, presented in Figure 6, were obtained for the theoretical cycle and the following inputs:

- temperature of HTF entering/leaving evaporator 80°F/55°F (26.7°C/12.8°C)
- temperature of HTF entering/leaving condenser 82°F/100°F (27.8°C/37.8°C)
- average effective temperature difference for evaporator and condenser 12.5°F (6.9°C)

Figure 6 shows that, for most compositions, the model neglecting the non-linearity overpredicts COP. The error is as high as 8.9 percent at a composition of 0.2 weight fraction R22. At high compositions of R22, on the other hand, neglecting the non-linearities resulted in an underprediction of COP. This reversal in the
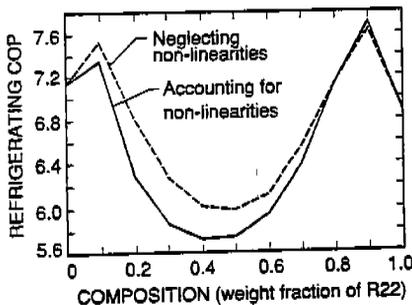
Figure 6
Prediction of refrigerating COP neglecting and accounting for the temperature/enthalpy non-linearity during phase change

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effect on COP is caused by the character of the non-linearity changing from convex at low R22 compositions to concave at high R22 compositions. These differences will vary with the mixture and conditions simulated, and cannot be generalized.

CONCLUSIONS

A model was presented to provide performance ratings of refrigerants and refrigerant mixtures operating in vapor compression cycles. The relative rating of refrigerants will vary with the application and the type of cycle simulated. Great care must be exercised in drawing broad conclusions from such ratings. For meaningful performance ratings, the simulation model should represent all the significant departures from the theoretical cycle experienced in a particular application, and these ratings should be applied only to the application simulated.

ACKNOWLEDGEMENTS

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