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Impact of Refrigerant Property Uncertainties on Prediction of Vapor Compression Cycle Performance

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NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

Executive Summary

Considerable reliance is placed on computer simulation models for evaluation of potential performance enhancement concepts. As the potential of new working fluids is evaluated, the impact of uncertain refrigerant properties needs to be better understood to properly interpret these computer simulation results. This paper presents a study of sensitivity of performance prediction to refrigerant properties for a residential, split heat pump operating in the cooling mode. The NBS steady-state heat pump model, HPSIM, was used in this study. The individual influence of the following parametric uncertainties was examined:

- liquid thermal conductivity
- vapor thermal conductivity
- liquid viscosity
- vapor viscosity
- liquid specific volume
- vapor specific volume
- liquid heat capacity
- vapor heat capacity at constant volume and heat capacity at constant pressure
- evaporation heat transfer coefficient
- condensation heat transfer coefficient
- evaporation pressure drop
- condensation pressure drop

The influence of a given parameter on the performance prediction was found by executing the program, HPSIM, with the altered parameter value. Several runs were executed for each tested parameter to cover the uncertainty range within which the value of this parameter would be expected to be known. Comparison of the heat pump capacity and power input with results of a run using an unchanged value of the parameters described the sensitivity of the cycle to each parameter. The effects on evaporator and condenser pressures and refrigerant mass flow rate are also given in the report.

Discrepancy between heat pump laboratory test results and computer model predictions may stem from uncertainties in refrigerant property predictions and inadequate hardware modeling algorithms. The impact of inaccurate property values can be evaluated using the findings of this study, the impact of simplifications in modeling algorithms have to be determined individually for each computer model.

Results of this sensitivity study are system dependent (e.g. somewhat different results would be obtained for a system with different relative sizes of heat exchangers), however, no change of the relative importance of the investigated parameters should be observed.

Best knowledge of all refrigerant parameters is essential for accurate performance predictions since the impact of individual parameter uncertainties may superimpose. The study indicated those parameters, which at the present state of the art, have the greatest impact on the uncertainty of performance prediction of the heat pump operating in the cooling mode. These parameters are: liquid transport properties, evaporative heat transfer coefficient and vapor density.

The effect of an individual parameter variation within the tested uncertainty limits was found to be as high as 7.5% for capacity (for liquid thermal conductivity) and 6% for power (for vapor specific volume). Calculations, which combined effects of individual parameter variations for a system charged with refrigerant 22, showed that uncertainty of capacity and COP predictions may be as high as 12.9% and 10.9%, respectively, if the involved refrigerant parameters are known with an error equal to the maximum deviation of the considered property correlations, and if the errors superimpose. Since properties of refrigerant 22 are among the best known, the uncertainty of capacity and COP predictions for other refrigerants and mixtures may be expected to be greater.

ABSTRACT

This paper presents a sensitivity study of a vapor compression cycle in the form of a heat pump operating in the cooling mode. The study was performed with the aid of a detailed heat pump computer model; simulation runs were made for different parametric values and the capacity and power input were compared with results of a run using an unchanged value of the parameters. The effects on evaporator and condenser pressures, and refrigerant mass flow rate are given. The independent variables (parameters) include thermodynamic and transport properties, as well as the refrigerant flow heat transfer and pressure drop coefficients. When considering the state-of-the-art limits of the individual parameter uncertainties, those which had the most effect on system performance were liquid transport properties, evaporative heat transfer coefficient and vapor density.

Key Words: Air conditioner, heat pump, sensitivity study, thermodynamic properties, transport properties, vapor compression cycle

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1. INTRODUCTION

Over the past decade, there has been considerable attention devoted to the performance of refrigeration systems in both simulation model studies and laboratory measurement studies. The accuracy of these studies tacitly depends on knowledge of the various refrigerant thermodynamic and transport properties as well as other flow parameters. The uncertainty with which the property values are known is primarily a function of the state-of-the-art accuracy of the particular property measurement technique. In the case of refrigerant mixtures, there exists very little measured data, and it is therefore necessary to employ mixing rules along with component data. This can increase the uncertainty from 10 to 50%. For example, vapor density can be evaluated quite well and usually is known within 2 percent uncertainty [1]; on the other hand, the specific volume of a liquid mixture may have an uncertainty of as much as 15 percent if density is evaluated by a mixing rule in the neighborhood of critical temperature of the lower boiling component [2]. In the case of evaporative heat transfer coefficients, the actual mixture value has been shown to be as much as 40% lower than that predicted by the ideal mixture weighting factor method [3].

The different refrigerant property and flow parameters affect the system performance with varying degrees of sensitivity. It is therefore necessary to establish the system sensitivity to each property value if the most effective information is to be obtained. Of course, accurate knowledge of refrigerant properties does not change the way the real system operates. However, the qualitative knowledge of the impact made by properties on performance on a one by one basis is desirable because it can establish the limits of uncertainty associated with simulation of refrigerant systems.

This, in turn, will indicate the areas most in need of further research on prediction and measurement methodologies for refrigerants and refrigerant mixtures.

The sensitivity analysis reported here was performed by simulating the performance of a 2.5 ton split, residential, air-to-air heat pump charged with Refrigerant 22. The heat pump was simulated in the cooling mode at DOE test A conditions [4]. Simulation of the system without altering any parameter was performed first. Then, simulation runs were performed with a specific parameter changed by an assigned multiplication factor. The change of system performance reflected system sensitivity to the altered parameter.

All simulation runs were performed with imposed refrigerant superheat of 10°F at the compressor can inlet, and imposed refrigerant subcooling of 10°F at the expansion device inlet. These restrictions are related to the common laboratory practice of selecting optimum refrigerant charge and expansion device size for a given system. These restrictions also ensure that when simulating the system with altered parameters, the observed change in performance indeed results from the altered parameter with optimized cycle by the same criteria, and not from a change in operating conditions.

The range over which a selected parameter was altered depended basically upon the uncertainty band with which this parameter is generally known and upon a need for presenting results of this study in some uniform fashion. The uncertainty with which a given parameter is known depends on the methodology by which the parameter is determined. If parameter determination is based on laboratory measurement, the difference between values of this parameter reported by two reputable sources may be considered as the uncertainty of this parameter. If this parameter is determined by a theoretical correlation based on fundamental knowledge of the molecule, the difference between a predicted value and a laboratory measurement is considered the uncertainty.

An uncertainty band usually increases significantly if refrigerant mixtures instead of single refrigerants are considered. This is a result of the need for some type of mixing rule to determine the desired property based on properties known for single refrigerants. Proper assessment of possible uncertainties is complicated in some cases by the lack of reliable data and by the large discrepancies between existing data sources for the majority of compounds.

2. HEAT PUMP SIMULATION MODEL USED

A computer model of a heat pump, HPSIM [5], used in this study, is a 'first principles' model, which was developed with emphasis on modeling phenomena taking place in the system on a local basis. The structure of HPSIM is modular with simulation of each major component handled by an independent subroutine. The model consists of 41 subprograms for heat pump component simulation, and heat transfer, fluid mechanics, and fluid property calculation. The program totals approximately 5000 Fortran statements.

For use in this sensitivity study the model, HPSIM, was modified in two areas:

- convergence tolerances were tightened throughout the program to obtain final convergence of the thermodynamic cycle within an enthalpy value 0.15 Btu per pound of circulating refrigerant.
- the logic of the program was changed to allow, in addition to existing features, the imposition of a preset value of subcooling at the expansion device inlet for any given cycle.

Heat pump components considered in HPSIM are shown in Figure 1. A thermodynamic cycle simulated by the model is presented along with the logic of the model in Figure 2.

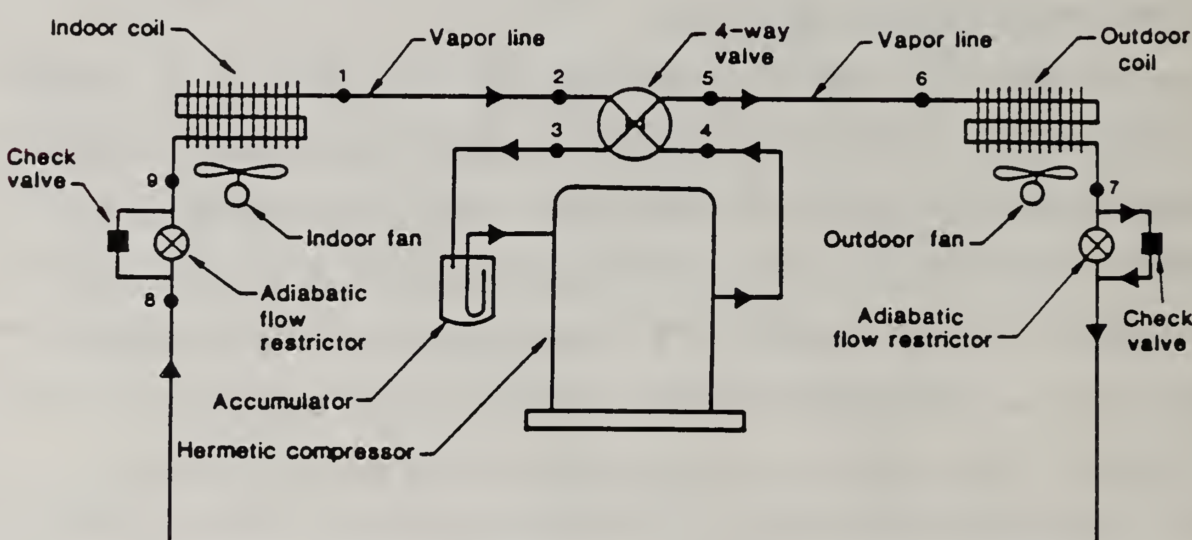
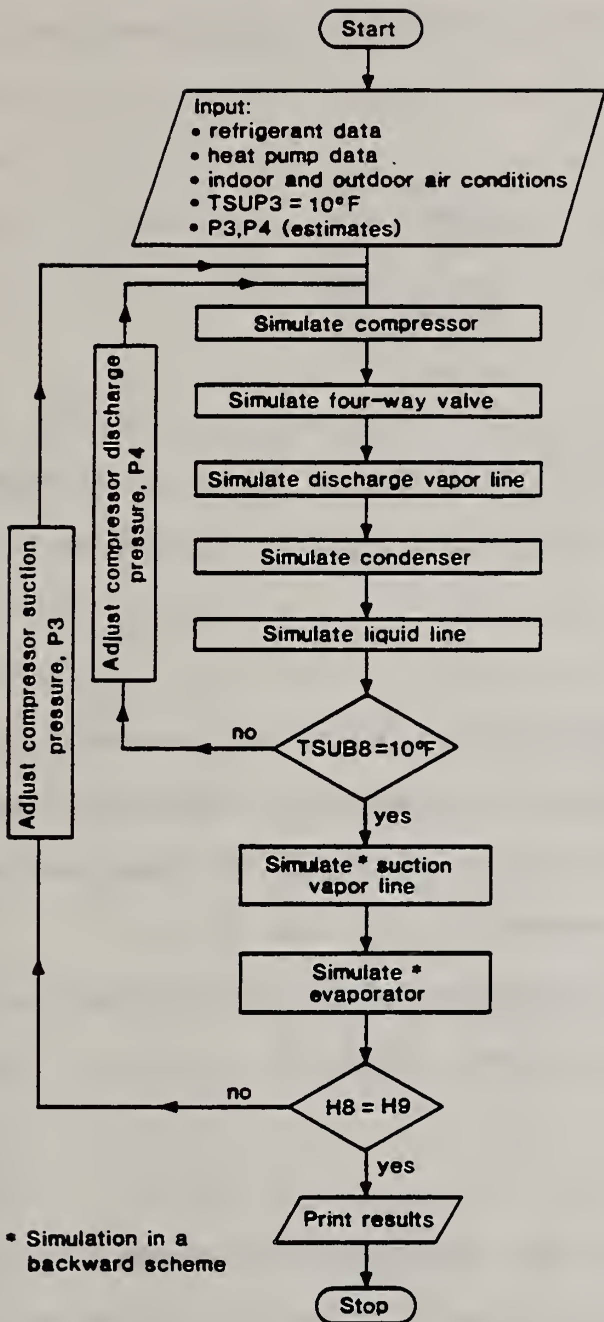


Figure 1. Schematic of a heat pump simulated by the heat pump model, HPSIM.



Symbols:

H8 - refrigerant enthalpy at the expansion device inlet

H9 - refrigerant enthalpy at the evaporator inlet

P3 - refrigerant pressure at the compressor can inlet

P4 - refrigerant pressure at the compressor can outlet

TSUP3 - refrigerant vapor superheat at the compressor can inlet

TSUB8 - refrigerant subcooling at the expansion device inlet

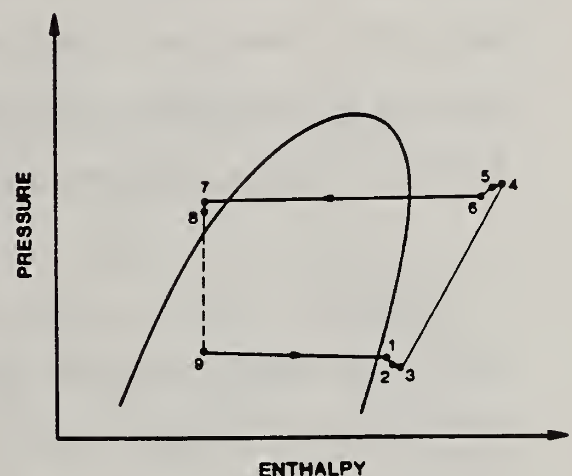


Figure 2. Logic of the heat pump model used in the sensitivity study.

The most important correlations for this sensitivity study are those associated with evaluating the refrigerant thermodynamic state, heat transfer and pressure drop. The correlations used were:

thermodynamic properties of vapor	- Downing [6]
single-phase heat transfer coefficient	- Colburn [7]
evaporation heat transfer coefficient	- Pierre [8]
condensation heat transfer coefficient	- Traviss et al. [9]
single-phase pressure drop	- Fanning [10]
evaporation pressure drop	- Pierre [11]
condensation pressure drop	- Lockhart - Martinelli [12]

Equations for calculation of thermodynamic properties of vapor are presented for reference in Appendix A. Other above listed correlations are shown in Appendix B.

Other properties which impact on performance was studied here included: density, heat capacity, thermal conductivity and absolute viscosity of liquid, and thermal conductivity and absolute viscosity of vapor. The first four of these were correlated as function of temperature only, last two as function of temperature and pressure. These correlations are described in Appendix A of the report on the heat pump model, HPSIM [5].

3. RESULTS

Results presented here refer strictly to the refrigerant side of the given thermodynamic cycle, i.e., capacity change refers to capacity delivered by the modeled coil not including the heat added by the indoor fan. By the same principle, power input change refers to compressor power only and does not include power of fans or controls.

Results are presented in the form of figures for those parameters which, when altered within the applied range changed system capacity or energy input by more than 0.5 percent. The figures show changes of capacity (%), energy input (%), refrigerant mass flow rate (%), condenser inlet pressure (psi) and evaporator outlet pressure (psi) as a function of change in a value of a specific property.

3.1 Sensitivity to Liquid Viscosity Change

Liquid viscosity data can be obtained in a number of ways. Liquid viscosity obtained by direct laboratory measurement or by a very limited extrapolation from measured data are most reliable with a generally stated uncertainty of less than 5% [1]. The best theoretical correlations provide data of an uncertainty of 15% [13]. Methods for predicting viscosity of liquid mixtures are remarkably poor (uncertainties up to 23% [13]) even when the viscosities of the pure components are accurately known.

Simulation results are shown in Figure 3. Simulation of liquid viscosity at a higher value has a detrimental effect on refrigerant pressure drop and the inside tube heat transfer coefficient, for both two-phase and single-phase flow. As a result suction pressure at the compressor decreases, discharge pressure increases, and mass flow rate of refrigerant is decreased, which is followed by a decrease of system capacity by 4% at 50% increase of vicosity. Compressor power also decreases but more slowly than capacity, making the system less efficient. A lower value of liquid viscosity is associated with similar but opposite trends.

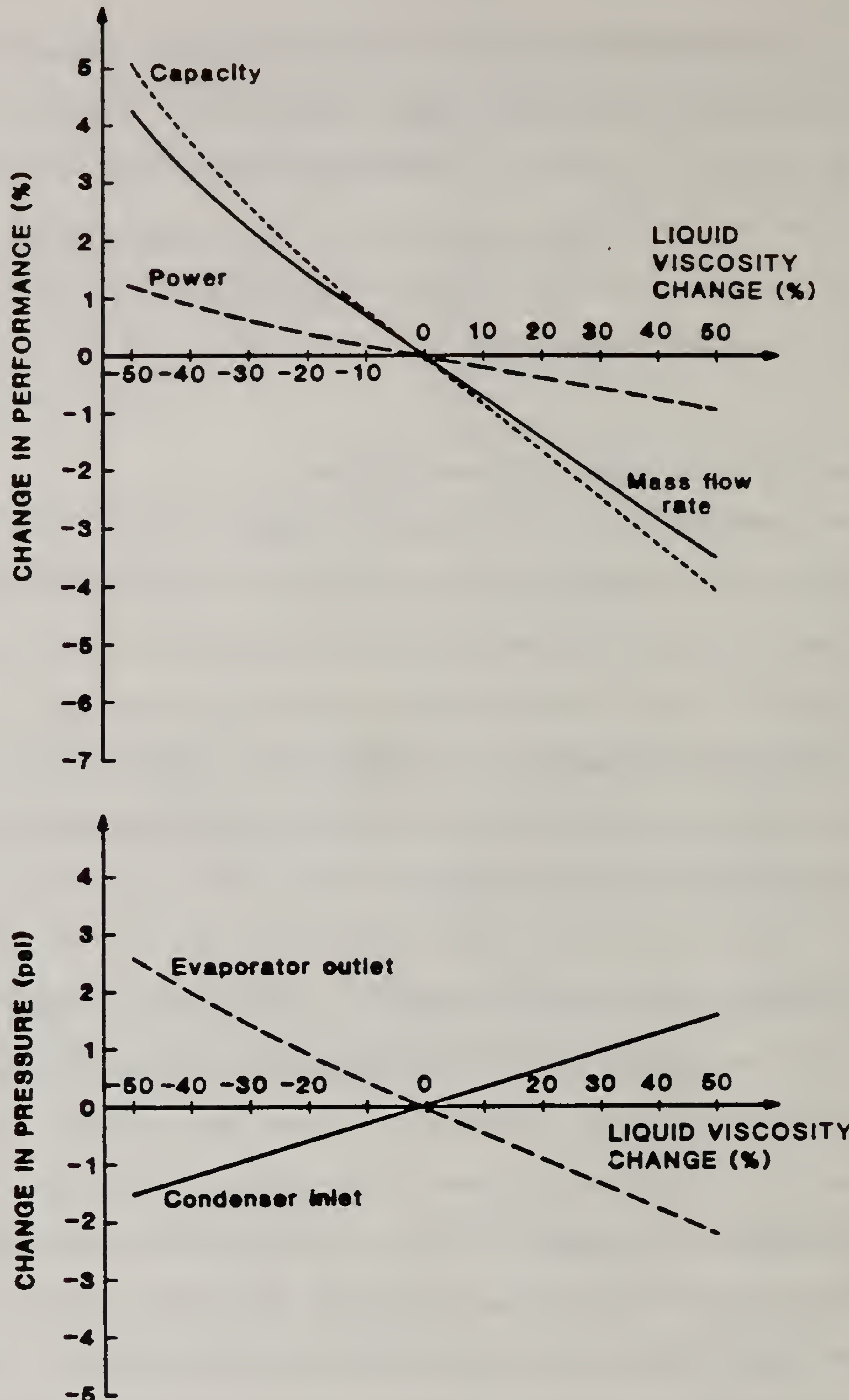


Figure 3. Sensitivity of performance of a heat pump to a change in liquid viscosity.

3.2 Sensitivity to Vapor Viscosity Change

Viscosity of a pure gas is usually a well know property with uncertainty less than 2% [13]. On the other hand, viscosity of a gas mixture (if determined through the use of pure component data and mixing rules rather than measurements on the mixture, of which there are few) may have an uncertainty of up to 17% [13]. Fortunately, however, simulation runs showed a very weak sensitivity (maximum .2%) of capacity and energy input to vapor viscosity which was varied up to $\pm$ 50%.

3.3 Sensitivity to Liquid Thermal Conductivity Change

Thermal conductivity of liquid, if assessed from experimental data, is known with up to 10% uncertainty [13]. Theoretical prediction methods for this property value can have an uncertainty of up to 30%. Prediction methods for mixtures also have 30% uncertainties if 'correct' viscosities of components are known.

Simulation results are presented in Figure 4. Change of liquid refrigerant thermal conductivity does not affect refrigerant pressure drop in the system. The observed change in level of evaporator and condenser pressures is a result of change of the inside tube heat transfer coefficient, both single-phase and two-phase, which increases with increase of liquid thermal conductivity. The consequence of the improved heat transfer is that the internal saturation temperature (and thus pressure) equilibrium levels the simulation converges to are closer to the external source and sink values. This increase of evaporator pressure and decrease of condenser pressure allows the compressor to pump more refrigerant consequently causing increased system capacity.

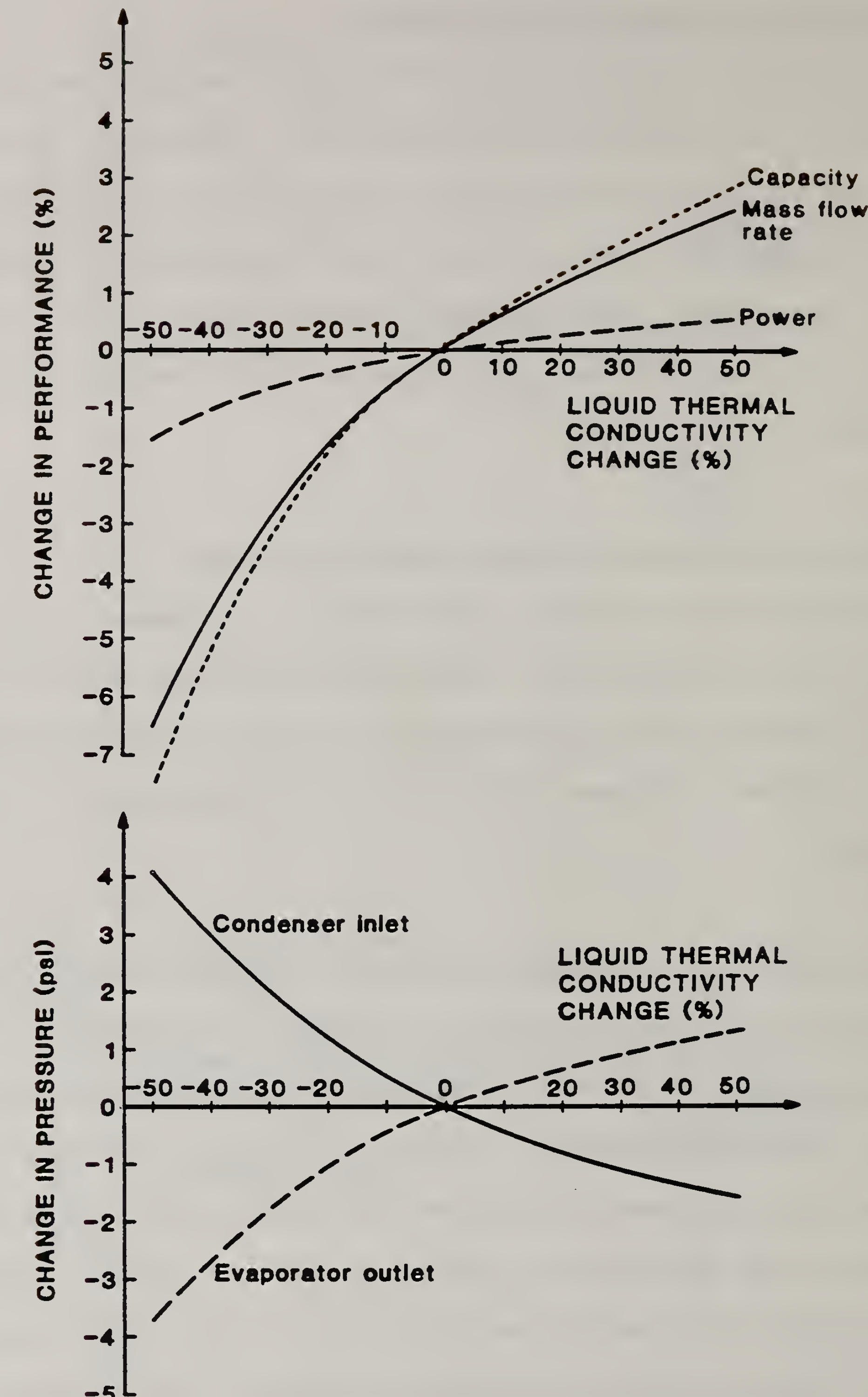


Figure 4. Sensitivity of performance of a heat pump to a change in liquid thermal conductivity.

Compressor power follows the mass flow rate pattern but not as strongly as capacity, which results in more efficient operation of the system.

It is interesting to note that sensitivity of performance is not the same over the range ±50% at which liquid conductivity was altered. Greater impact (degradation) is exhibited for decreased conductivity than the improvement of performance for increased conductivity (-7.6% versus 2.8%). This behavior can be explained by the fact that the liquid thermal conductivity profoundly affects the inside tube heat transfer resistance. Increase of liquid conductivity decreases the inside tube resistance with decreasing impact on overall heat transfer resistance, making the air-side resistance an even more dominant factor, while decreasing the liquid conductivity increases the inside tube resistance which becomes more significant and influential part of the overall heat transfer resistance of the heat exchanger.

3.4 Sensitivity to Vapor Thermal Conductivity Change

Vapor thermal conductivity is usually known with an uncertainty smaller than 10% though the error can be as high as 28% [13]. Thermal conductivity of mixtures evaluated by a mixing correlation may carry an uncertainty of similar magnitude even if accurate values for components thermal conductivities are know.

Simulation runs performed with vapor thermal conductivity altered from -50% to +50% showed very weak sensitivity of performance of the system to this property. Capacity varied from -0.5% to 0.4% respectively, while compressor power varied less than 0.1%.

3.5 Sensitivity to Liquid Specific Volume Change

Specific volume of liquid is usually known with uncertainty smaller than 1%. For liquid mixtures, the straight mixing rule (Amagat's law) is usually satisfactory and does not add more than 5% in uncertainty at low reduced temperatures. A method based on the principle of corresponding states provides specific volume for some liquid mixtures with uncertainty up to 10% [1]. None of these methods should be used at reduced temperature exceeding 0.9 for the more volatile component where prediction uncertainty may be as high as 100% [2]; an equation of state capable of handling both the liquid and vapor should be used instead [14].

In the heat pump simulation program used in this study, liquid specific volume at saturation is calculated by an independent spline which is a function of temperature only. The value obtained from such a spline was altered within ±10% in this study. It has to be mentioned that liquid specific volume has impact on the value of the latent heat and the width of the two-phase region (see equation (A4)) affecting the amount of heat per one pound of refrigerant that can be pumped by the cycle. Simulation results are presented in Figure 5. An increase in liquid specific volume decreases latent heat, increases single-phase and two-phase pressure drop, and increases the condensation heat transfer coefficient. The evaporation heat transfer coefficient evaluated by Pierre's correlation [8], used in this study, is insensitive to liquid specific volume, though other type correlations based on Lockhart-Martinelli parameter would show a decrease of the heat transfer coefficient with increased liquid specific volume. The first two factors are dominant, with the latent heat being most influential; system capacity decreases inspite of

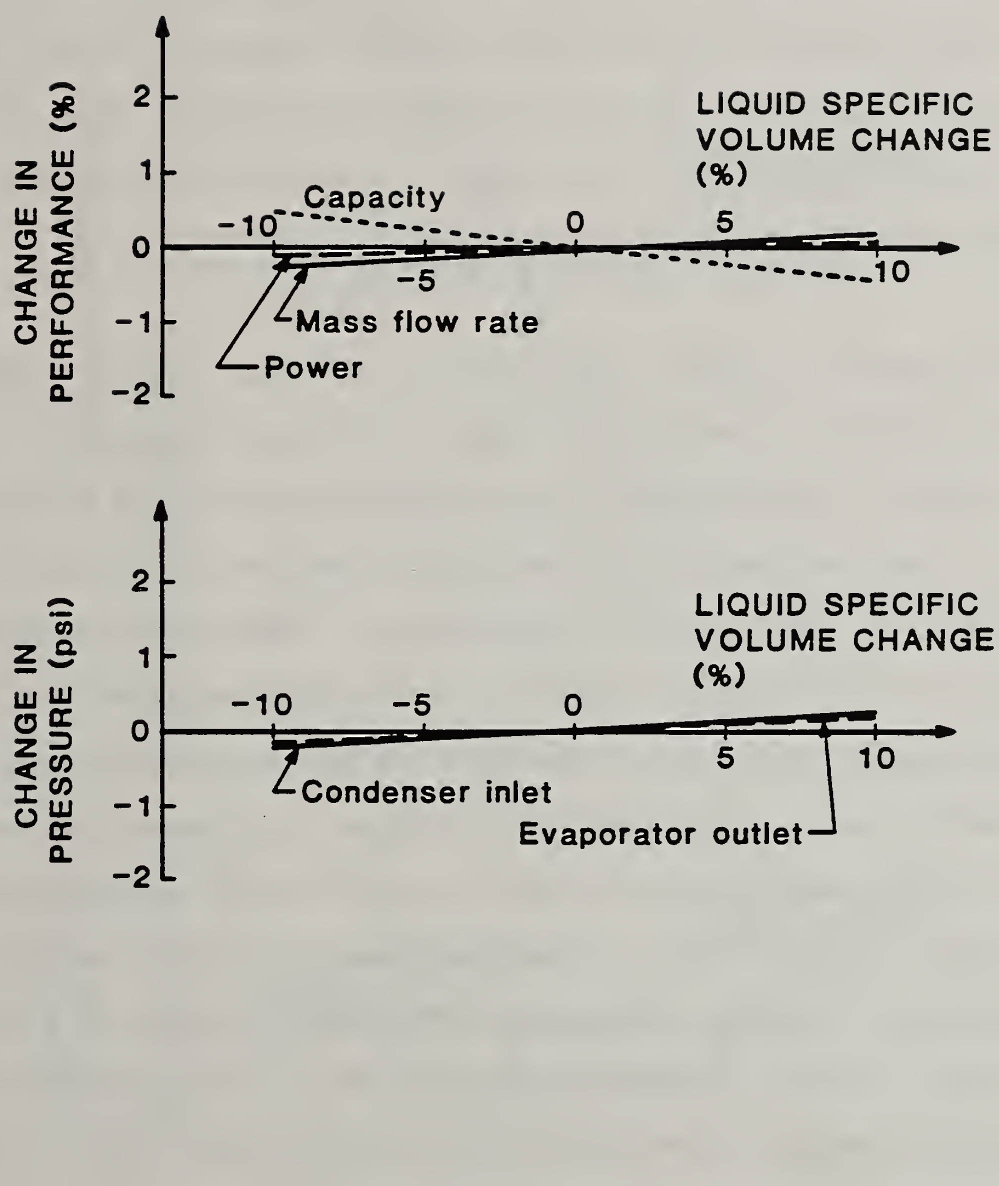


Figure 5. Sensitivity of performance of a heat pump to a change in liquid specific volume.

increased refrigerant mass flow rate. Results for decreased liquid specific volume are symmetrical but with opposite signs than those found for increased specific volume. However, the overall performance sensitivity to this property is minimal.

3.6 Sensitivity to Vapor Specific Volume Change

Vapor specific volume was altered in this study within the range ±10% which covers uncertainties with which this property is generally known. This 10% range may be exceeded in the case of mixture vapor volumes where an unanticipated nonideality in the mixture may alter the vapor pressure substantially; otherwise the state-of-the-art is better than 2%.

Change of specific volume of vapor affects the latent heat value similarly as specific volume of liquid but in opposite direction (see equation (A4)), i.e., the increase of vapor specific volume increases the width of the two-phase region. Vapor heat capacity, enthalpy and entropy are also affected (equations (A3), (A5) and (A6), respectively). Other effects associated with increase of vapor specific volume are: refrigerant mass flow rate decrease, pressure drop of single-phase vapor flow and two-phase flow increase, and condensation heat transfer coefficient increase. The changes of condensation heat transfer coefficient and pressure drop have minor and opposite affects (Figure 6). Increase of the latent heat appears to be the dominating factor, overcoming the impact of reduction of refrigerant mass flow rate (8.2%) resulting in slightly increased capacity (0.4%). System performance improves substantially at higher specific volume since the slight capacity increase is accomapnied by decreased energy consumption.

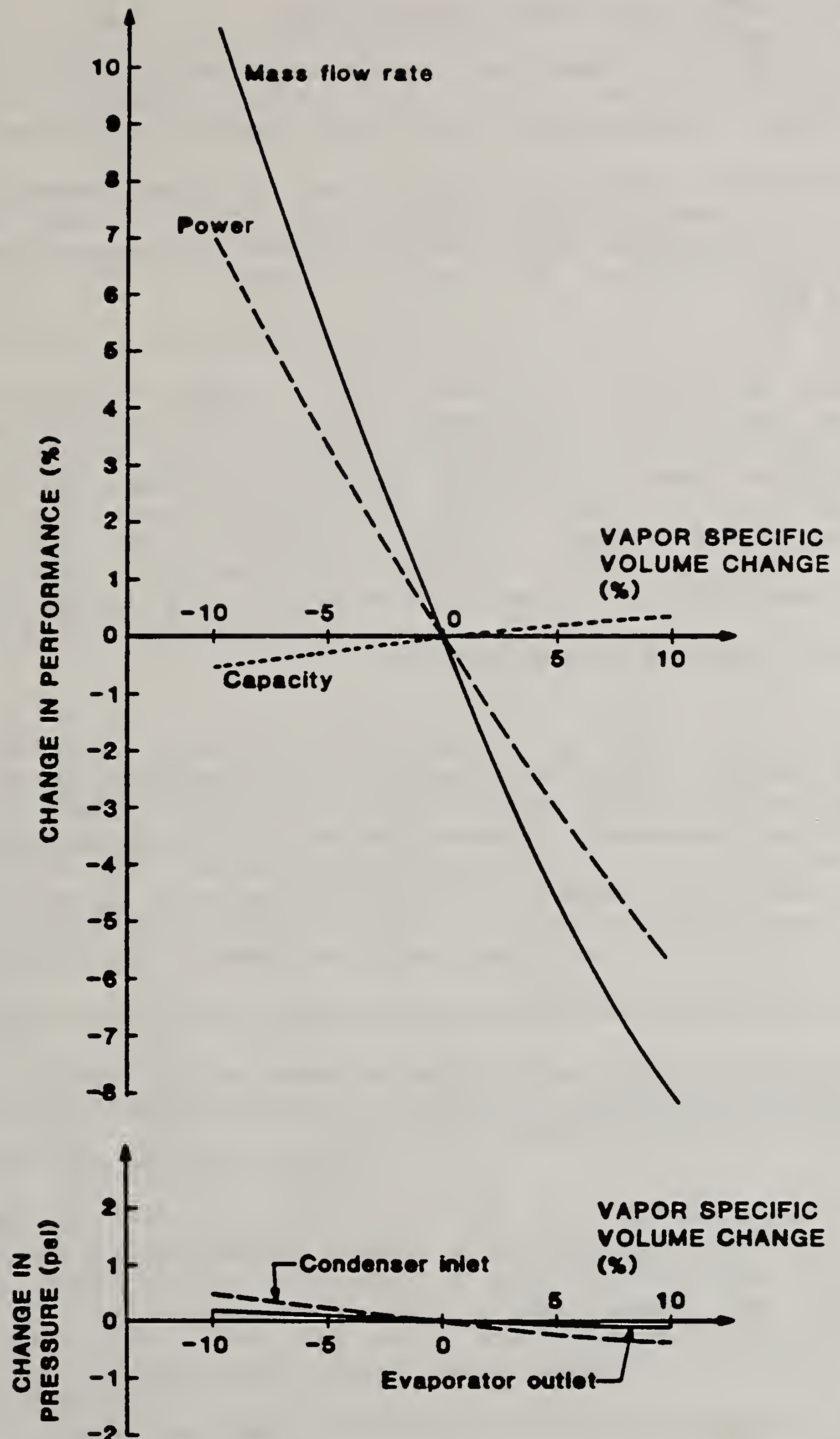


Figure 6. Sensitivity of performance of a heat pump to a change in vapor specific volume.

3.7 Sensitivity to Liquid Heat Capacity Change

Prediction methods for liquid heat capacity are usually accurate within 15% below the normal boiling point, but at higher reduced temperatures the uncertainty may be greater [1]. Specific heat of liquid mixtures has an additional 5% uncertainty due to the mixing rules for component heat capacities [13].

Liquid heat capacity as a thermodynamic property, is related to other properties through the slopes of saturated liquid and saturated vapor lines outlining the two-phase region. In assessing sensitivity of system performance on liquid specific heat it is important to realize that the outcome will depend on the origin of the uncertainty.

Liquid heat capacity may be determined by direct measurement or by vapor phase measurement and theoretical relations outlining the two-phase region. Simulation results for these two cases will differ since in the first case only uncertainty of a value of the liquid heat capacity will affect the accuracy of performance prediction, while in the latter case additionally to the liquid heat capacity, uncertainties of vapor phase measurements and theoretical relations are possibly involved.

In simulation runs, reported here, a value of liquid specific heat was altered exclusively. (This creates some theoretical inconsistencies since $\partial h/\partial T$ for saturated liquid was not changed.) As a result, the scope of sensitivity analysis is limited to the effect of altered liquid heat capacity through change of the inside tube heat transfer coefficients in the heat exchangers. Consideration of uncertainty of liquid heat capacity that would originate from inaccurate outline of the two-phase region would require modifying of the thermodynamic properties algorithm which was not practical for this study.

The results, shown in Figure 7, can be explained as follows:
a decrease in liquid heat capacity decreases both single-phase and two-phase condensation heat transfer coefficient (pressure drop is not affected) forcing condenser pressure to increase. This increase is reinforced by increased evaporator pressure, 'pulled up' by the condenser, and increased mass flow rate caused by increased density of the suction vapor. Capacity decreases, in spite of greater refrigerant mass flow rate, because refrigerant enthalpy change in the evaporator decreased.

Opposite performance trends are associated with an increase of liquid heat capacity with some difference in pressure change. This difference can be explained by the fact that condenser pressure decrease is limited by refrigerant saturation temperature which has to be above the outdoor temperature.

3.8 Sensitivity to Change of Vapor Heat Capacity at Constant Pressure and Vapor Heat Capacity at Constant Volume

Sensitivity on vapor heat capacity at constant pressure and vapor heat capacity at constant volume was tested by altering both these properties at the same time by ±10%. This range covers the likely uncertainties associated with these properties.

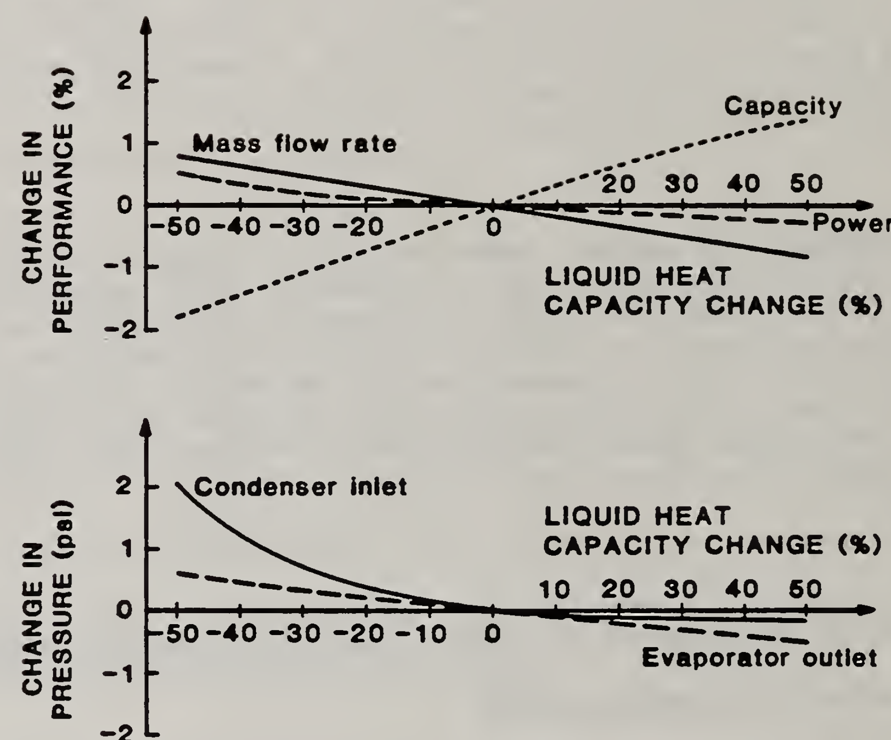


Figure 7. Sensitivity of performance of a heat pump on a change in the heat capacity of liquid.

Simulation runs showed insignificant impact on system performance resulting from the vapor heat capacities changes. Heat pump capacity and power values remained within 0.1% of the original level for 10% variation in heat capacities.

3.9 Sensitivity to Change of Evaporative Heat Transfer Coefficient

The evaporative heat transfer coefficient is one of the most difficult parameters to predict and thus has the widest variation in uncertainty. It is not uncommon that different correlations are in disagreement by as much as 50%. Aside from evaluating the impact of the uncertainty of the evaporative heat transfer coefficient for pure components in smooth tubes, analysis of tubes with enhanced internal surfaces and refrigerants comprised of non-azeotropic mixtures should be also considered. For this later case, validated mixing rules are virtually unknown, however it is known that simple weighting of pure component data can result in a 40% over prediction from measured data [3].

The effect of changed evaporative heat transfer coefficient is not equally strong for its degradation and enhancement (Figure 8). The impact is stronger with degraded coefficient because in this case the inside tube heat transfer resistance contributes more significantly to efficiency of the heat exchange.

An increase of evaporative heat transfer coefficient allows for smaller temperature difference between the ambient air and refrigerant in the evaporator causing evaporator pressure to increase. Subsequently, the suction vapor density increases enabling the compressor to pump more refrigerant. Since heat transfer resistance on the condenser side is not changed, the

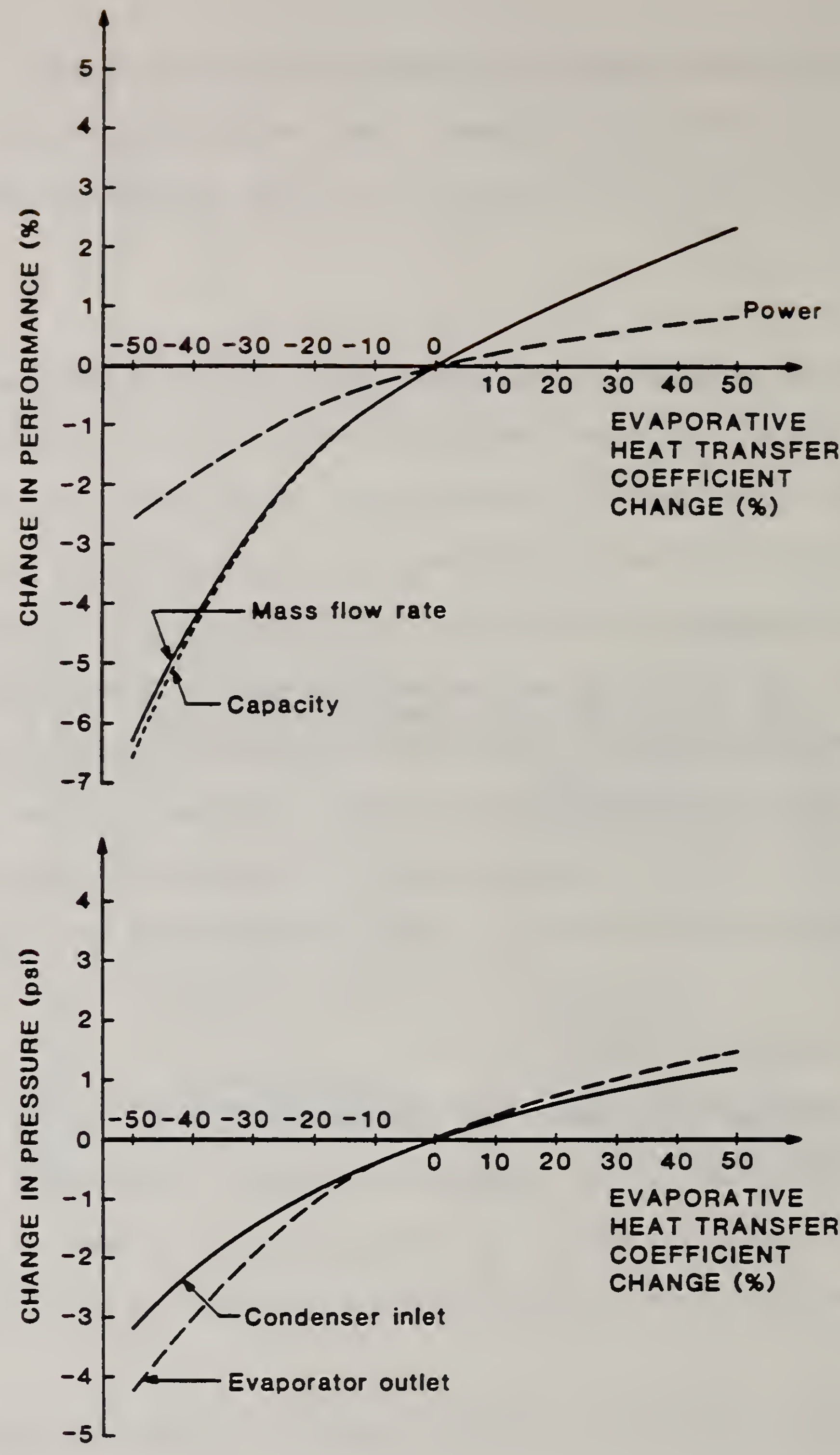


Figure 8. Sensitivity of performance of a heat pump on a change in the evaporative heat transfer coefficient.

increased refrigerant mass flow rate requires higher condenser temperature (pressure) in order to condense the refrigerant at this higher mass flow rate. Capacity of the system increases along with refrigerant mass flow rate (both curves coincide).

3.10 Sensitivity to Change of Condensation Heat Transfer Coefficient

Uncertainties in determination of the condensation heat transfer coefficient are similar to those of the evaporation heat transfer coefficient. The condensation heat transfer coefficient may also be enhanced by roughed inner surface or be degraded if a mixture is used. Therefore, just as with the evaporative coefficient, the selection of the appropriate value is application as well as property dependent.

Simulation results are shown in Figure 9. A change of the condensation heat transfer coefficient makes greater impact when the coefficient is degraded than when it is improved. Decrease in the condensation heat transfer coefficient causes a rise in condenser pressure in order to condense flowing refrigerant. Increased condenser pressure is associated with only a weak evaporator pressure increase, thus the refrigerant mass flow rates remain almost unchanged. Stronger changes (decreases) in capacity with decreased heat transfer coefficient is a result of a decreased enthalpy change in the evaporator resulting from higher condenser pressure and higher liquid enthalpy at the expansion device inlet with enthalpy at the evaporator outlet almost unchanged. Compressor power change corresponds to changes in the condenser pressure which changed compressor compression ratio.

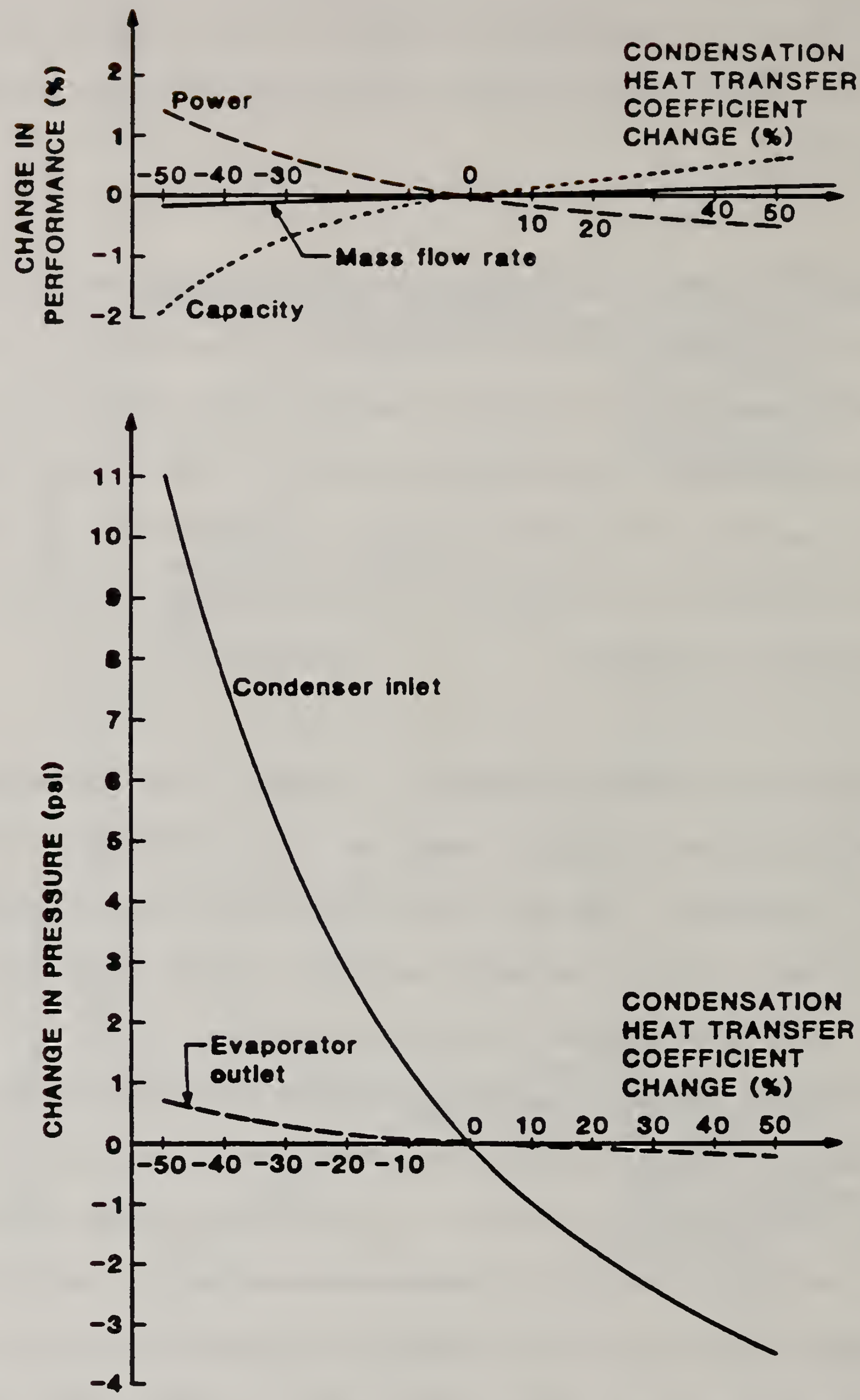


Figure 9. Sensitivity of performance of a heat pump on a change in the condensation heat transfer coefficient.

3.11 Sensitivity to Change of Evaporation Pressure Drop

The evaporative pressure drop was altered in the range from -50% to +100%, to cover possible uncertainty for pressure drop evaluation as well as cases of increased pressure drop in tubes with roughed inner surface for enhanced heat transfer coefficient. Simulation results are quite linear (Figure 10). Increased pressure drop decreases the compressor suction pressure and suction vapor density. Refrigerant mass flow rate decreases and pressure in the condenser also decreases since a smaller temperature difference between the ambient air and the condenser is required to condense refrigerant flowing at the lower rate. Refrigerant mass flow rate decrease results in capacity decrease (both curves coincide).

3.12 Sensitivity to Change of Condensation Pressure Drop

Condensation pressure drop was altered within the same range as evaporative pressure drop: -50%, +100%. Simulation results showed small sensitivity to altered two-phase pressure drop in the condenser. Capacity and power were changed within -0.1 and +0.1% range.

3.13 Sensitivity to Change of More than One Parameter

Performance sensitivity to more than one parameter's variation can be evaluated by reading the figures for the individual parameters involved and multiplying the individual performance changes. For example, if a roughened inside tube surface enhanced the inside tube heat transfer coefficient by 40% at the expense of increased pressure drop of 40%, the resultant change of system capacity, if such tubes were employed in the evaporator, would be

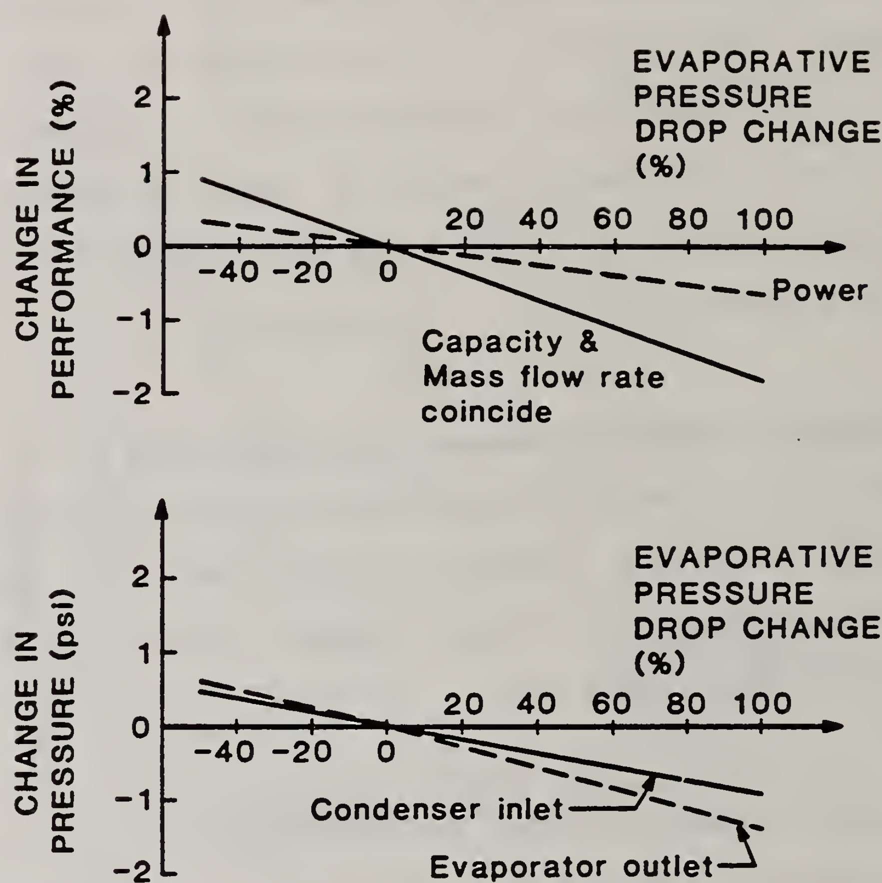


Figure 10. Sensitivity of performance of a heat pump on a change in the evaporative pressure drop.

$1.020 \cdot 0.993 = 1.013$ where:

1.020 corresponds to 2% capacity increase due to the increased inside tube heat transfer coefficient (read from Figure 8),

0.993 corresponds to 0.7% capacity decrease due to increased evaporative pressure drop (read from Figure 10).

This superimposing of effects can be done since system sensitivity was evaluated based on comparable thermodynamic cycles optimized by the same criteria by imposing 10°F vapor superheat at the compressor can inlet and 10°F liquid subcooling at the expansion device inlet. This procedure was also verified by results of simulation runs in which more than one parameter was varied.

3.14 Combined Effect of Refrigerant Properties Uncertainties on Prediction of System Performance

Accuracy of performance prediction may suffer for two reasons: inadequate hardware modeling algorithms, and inaccurate fluid property prediction. The performance prediction uncertainty band due to inadequacy of hardware modeling algorithms is, by definition, specific for each particular computer model. This uncertainty band can be found by comparing simulation and laboratory test results for different model heat pumps. Differences of disagreement between predicted and tested performance for different heat pump models would indicate the uncertainty band due to the inadequacy of simulation model algorithms.

This study was performed to provide a methodology for evaluation of sensitivity of performance prediction on uncertainties of refrigerant properties. This methodology depends on combining effects of individual parameters on system performance. Though results were generated simulating a heat pump charged with refrigerant 22, the methodology is applicable to any refrigerant since it was formulated on a relative basis. Sensitivity of performance prediction is difficult to determine rigorously because the errors with which fluid properties values are used in the simulation process are unknown. In addition, errors in fluid property values may have a cancelling effect in affecting the performance prediction, or a superimposing effect causing the simulation results to be far off.

The uncertainty band due to inaccurate refrigerant property values will be different for different refrigerants since not all refrigerants have been equally investigated and their properties equally well known. In order to provide an indication of the uncertainty in performance prediction which may result from uncertainty in fluid properties, a case for refrigerant 22 is presented below.

Previous sections and figures allow identification of the parameters which have the most significant impact on performance predictions and should be considered. These parameters are: liquid thermal conductivity, liquid viscosity, vapor specific volume, evaporation and condensation heat transfer coefficients, and evaporation pressure drop. It is assumed here that all property uncertainties taken into account have a superimposing effect on system capacity. Two scenarios are considered: one when the error in performance prediction is equal to the mean deviation for a given property correlation, and the second when the maximum deviation is taken into account. For R-22 there are a few correlations that were checked against a broad data bank and mean and maximum deviations were evaluated. These correlations for the following parameters are:

	Mean Deviation (%)	Max Deviation (%)	Source
liquid thermal conductivity	2.1	8.5	[15]
liquid viscosity	2.5	5.9	[15]
evaporation heat transfer coefficient	25.0	58.0	[16]

For the condensation heat transfer coefficient mean and maximum deviations were assumed to be the same as for the evaporation heat transfer coefficient. Deviations for vapor specific volume were assumed to have values of 0.5% (max) and 0.25% (min).

Reading system performance change for each property from the figures and combining the individual effects yielded an uncertainty of 3.7% for capacity, and 3.1% for COP prediction in the minimum deviation case, and respective uncertainties of 12.9% and 10.9% for the maximum deviation case. The most influential single parameter appeared to be the evaporative heat transfer coefficient which contributed approximately 60% to the calculated uncertainties for capacity and COP predictions.

As far as other refrigerants and their mixtures are concerned, it is expected that performance prediction uncertainty will be greater. An important conclusion that can be drawn for all refrigerants from the R22 example is that an error in the value of the evaporation heat transfer coefficient has a significant weight and evaluation of this parameter alone may provide a rough indication of the performance prediction uncertainty for single component refrigerants. For refrigerant mixtures a significant impact on performance prediction may come from errors in values of liquid transport properties and liquid specific volume. Influence of these properties may differ significantly from one mixture to another depending on the experimental data base and correlations used.

4. CONCLUSIONS

Results of this study provide information on the effect uncertainties of individual parameters have on system performance prediction. This information should be helpful in evaluation of uncertainties in performance prediction for innovative refrigerants and their mixtures for which complete property information is usually not available. The results indicate liquid transport properties, evaporative heat transfer coefficient and vapor density as those properties whose uncertainties have the most significant impact.

The effect of an individual parameter variation within the tested uncertainty limits was found to be as high as 7.5% for capacity (for liquid thermal conductivity) and 6% for power (for vapor specific volume). In reality the uncertainty of system performance prediction may be much higher since the value of each parameter is known with some uncertainty and they all influence the predicted level of performance. An example analysis for refrigerant 22 showed uncertainty of capacity and COP predictions can be as high as 12.9% and 10.9% if involved refrigerant parameters are known with an error equal to the maximum deviations of the considered property correlations. Since properties of refrigerant 22 are among the best known, the uncertainty of capacity and COP predictions for other refrigerants may be expected to be greater.

Discrepancy between heat pump laboratory test results and computer model predictions stem from uncertainties in refrigerant property prediction and inadequate hardware modeling algorithms. Inaccurate refrigerant properties values should have a similar impact on performance predictions obtained by any computer model. This impact can be evaluated using findings contained in this report. The impact of inadequate modeling algorithms is specific for each computer model and has to be evaluated using the particular model individually.

Results of this sensitivity study are system dependent (e.g. somewhat different results would be obtained for a system with different relative sizes of heat exchangers), however, no change of the relative importance of the investigated parameters should be observed.

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APPENDIX A. EVALUATION OF THE THERMODYNAMIC PROPERTIES OF VAPOR

Equations used for evaluation of the thermodynamic properties were presented by Downing [6]. These equations are given below with appropriate constants for Refrigerant 22.

Saturated Vapor Pressure

$$\log_{10} P = A + \frac{B}{T} + C \log_{10} T + DT + E\left(\frac{F - T}{T}\right)\log_{10}(F - T) \qquad \text{(A1)}$$

where:

P = pressure (psia)

T = vapor temperature (R)

A = 29.357545

B = −3845.1932

C = −7.8610312

D = 2.1909390E−3

E = 0.44574670

F = 686.1

Equation of State

$$P = \frac{RT}{V - b} + \frac{A2 + B2T + C2\exp(-KT/TC)}{(V - b)^2} + \frac{A3 + B3T + C3\exp(-KT/TC)}{(V - b)^3} \quad (A2)$$

$$+ \frac{A4 + B4T}{(V - b)^4} + \frac{A5 + B5T + C5\exp(-KT/TC)}{(V - b)^5} + \frac{A6 + B6T}{\exp(ALPHA \cdot V)}$$

P = pressure (psia)

T = temperature (R)

TC = 664.50, critical temperature (R)

V = specific volume (ft**3/lb)

A2 = -4.353547

A3 = -0.017464

A4 = 2.310142E-3

A5 = -3.724044E-5

A6 = 1.363387E+8

ALPHA = 548.2

B2 = 2.407252E-3

B3 = 7.62789E-5

B4 = -3.605723E-6

B5 = 5.355465E-8

B6 = -1.672612E+5

b = 0.002

C2 = -44.06686

C3 = 1.483763

C5 = -1.845051E-4

C6 = -1.672612E+5

K = 4.2

R = 0.124098

Heat Capacity of Vapor at Constant Volume

$$C_V = a + bT + cT^2 + \frac{f}{T^2} - \frac{JK^2T\,\exp(-KT/TC)}{TC^2}\left[\frac{C2}{V - b} + \frac{C3}{2(V - b)^2} + \frac{C5}{4(V - b)^4}\right] \tag{A3}$$

where:

C_V = heat capacity of vapor at constant volume (Btu/lb·F)

a = 0.02812836

b = 2.255408E-4

c = -6.509607E-8

f = 257.341

J = 0.185053

Other symbols as defined for the equation of state.

Latent Heat of Vaporization

$$i_{fg} = JT\left(V_g - V_f\right)\left[P(\ln 10)\left(\frac{-B}{T^2} + \frac{C}{T(\ln 10)}\right)\right. \qquad (A4)$$

$$\left. + D - E\left(\frac{\log_{10} e}{T} + \frac{F \log_{10}(F-T)}{T^2}\right)\right]$$

where:

i_{fg} = latent heat of vaporization (Btu/lb)

J = 0.185053

V_g = specific volume of saturated vapor (ft**3/lb)

V_f = specific volume of saturated liquid (ft**3/lb)

Other symbols are as described for the vapor pressure equation.

Enthalpy of the Vapor

$$i = aT + \frac{bT^2}{2} + \frac{cT^3}{3} - \frac{f}{T} + JPV + J\left[\frac{A2}{V - b} + \frac{A3}{2(V - b)^2}\right. \qquad (A5)$$

$$\left. + \frac{A4}{3(V - b)^3} + \frac{A5}{4(V - b)^4} + \frac{A6}{ALPHA}/\exp(ALPHA \cdot V)\right]$$

$$+ J\exp(-KT/TC)\left(1 + \frac{KT}{TC}\right)\left[\frac{C2}{V - b} + \frac{C3}{2(V - b)^2}\right.$$

$$\left. + \frac{C5}{4(V - b)^4}\right] + 62.4009$$

where:

i = enthalpy of vapor (Btu/lb)

symbols A2, A3, A4, A5, A6, C2, C3, K, P, T, TC, V,

ALPHA are as described for the equation of state

symbols J, a, b, c, f are as described for the heat capacity equation

<u>Entropy of the Vapor</u>

$$S = a(\ln 10)\log T + bT + \frac{cT^2}{2} - \frac{f}{2T^2} + JR\ (\ln 20)\log(V - b) \tag{A6}$$

$$- J\left[\frac{B2}{V - b} + \frac{B3}{2(V - b)^2} + \frac{B4}{3(v - b)^3} + \frac{B5}{4(V - b)^4}\right.$$

$$\left. + \frac{B6}{ALPHA}\ /\exp(ALPHA \cdot V)\right]$$

$$+ \frac{J\ K\exp(-KT/TC)}{TC}\left[\frac{C2}{V - b} + \frac{C3}{2(V - b)^2} + \frac{C5}{4(V - b)^4}\right] - 0.0453335$$

where S = entropy of the vapor (Btu/(lb·F))

symbols B2, B3, B4, B5, B6, C2, C3, C4, C5, K, T, TC, V, ALPHA are as described for the equation of state.

symbols J, a, b, c, f are as described for the heat capacity equation.

APPENDIX B. EVALUATION OF THE INSIDE TUBE HEAT TRANSFER COEFFICIENT AND PRESSURE DROP OF REFRIGERANT

The following are inside tube heat transfer and pressure drop correlations used in the model, HPSIM. List of symbols is given at the end of this appendix.

<u>Single-Phase Heat Transfer Coefficient</u> [7]

$$h = 0.023 \frac{k}{D} \cdot Re^{0.8} \cdot Pr^{0.333} \tag{B1}$$

<u>Evaporative Heat Transfer Coefficient</u> [8]

$$h = 0.0009 \frac{k_L}{D} \cdot Re \left(J \cdot \Delta x \cdot i_{fg}/L\right)^{0.5} \tag{B2}$$

<u>Condensation Heat Transfer Coefficient</u> [9]

$$h = \frac{k_L}{D}^{0.9} \cdot Re_L \cdot Pr \cdot F1^{\beta}/F2 \tag{B3}$$

where: $\beta = 1.00$ for $F1 \leq 1$

$\beta = 1.15$ for $F1 > 1$

$$F1 = 0.15 \left(X_{tt}^{-1} + 2.85\, X_{tt}^{0.524}\right)$$

$$F2 = 0.707 \cdot Pr_L \cdot Re_L^{0.5} \quad \text{for } Re_L < 50$$

$$F2 = 5 \cdot Pr_L + 5 \cdot \ln(1 + Pr_L(0.09636 \cdot Re_L^{0.585} - 1)) \quad \text{for } 50 < Re_L < 1125$$

$$F2 = 5 \cdot Pr_L + 5 \cdot \ln(1 + Pr_L) + 2.5 \cdot \ln(0.00313 \cdot Re_L^{0.812})$$

for $Re_L < 1125$ (63)

Single-Phase Pressure Drop [10]

$$\Delta P = 2 \cdot f \cdot G^2 \cdot L/(D \cdot \rho) \tag{B4}$$

where $f = 0.046 \cdot Re^{-0.2}$

Two-Phase Pressure Drop With Evaporation [11]

$$\Delta P = (f \cdot L/D + \Delta x/x_m) G^2 \cdot V_m$$

$$\text{where: } f = 0.0185 \cdot Re(L/(J \cdot i_{fg} \cdot \Delta x))^{0.25} \tag{B5}$$

Two-Phase Pressure Drop with Condensation [12]

$$\Delta P = \Delta P_L \cdot \Phi \tag{B6}$$

where: $\Delta P_L = 2f((1 - x)G)^2 \cdot L/(D \cdot \rho_L)$

$f_L = 0.046 \cdot Re_L^{-0.2}$

$\Phi = 10.0^{(A0 + A1 \cdot B + A2 \cdot B^2 + A3 \cdot B^3 + A4 \cdot B^4)}$

Φ = correlated based on [12]

A0 = 1.4

A1 = 0.87917

A2 = 0.14062

$A3 = 0.0010417$

$A4 = -0.00078125$

$B = \text{Log}_{10} X_{tt}$

List of Symbols

C_p = specific heat at constant pressure

D = inner tube diameter

f = Fanning friction factor

$G = \frac{4m}{\pi D^2}$, refrigerant mass flux

h = forced convection heat transfer coefficient

i_{fg} = latent heat of evaporation/condensation

J = mechanical equivalent of heat

k = thermal conductivity

L = tube length

m = refrigerant mass flow rate

P = pressure

$Pr = \frac{\mu Cp}{k}$, Prandtl number

$Re = \frac{GD}{\mu}$, Reynolds number

x = local quality, x_m refers to mean quality

V = specific volume

$V_m = V_L + x_m(V_V - V_L)$, two-phase mean specific volume

Φ = Lockhart-Martinelli correction factor for two-phase pressure drop

$X_{tt} = \left(\frac{1 - x}{x}\right)^{0.9} \left(\frac{V_L}{V_V}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$, two-phase flow Lockhart-Martinelli parameter for turbulent both liquid and vapor

ρ = density

μ = absolute viscosity

Subscripts refer to:

L = liquid

V = vapor

m = mean value

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☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)*

This paper presents a sensitivity study of a vapor compression cycle in the form of a heat pump operating in the cooling mode. The study was performed with the aid of a detailed heat pump computer model; simulation runs were made for different parametric values and the capacity and power input were compared with results of a run using an unchanged value of the parameters. The effects on evaporator and condenser pressures, and refrigerant mass flow rate are given. The independent variables (parameters) include thermodynamic and transport properties, as well as the refrigerant flow heat transfer and pressure drop coefficients. When considering the state-of-the-art limits of the individual parameter uncertainties, those which had the most effect on system performance were liquid transport properties, evaporative heat transfer coefficient and vapor density.

12. KEY WORDS *(Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)*

air conditioner, heat pump, sensitivity study, thermodynamic properties, transport properties, vapor compression cycle

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