HEAT PUMPS IN COLD CLIMATES

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ABSTRACT

A simulation model for pure and mixed refrigerants is described that is suitable for conducting thermodynamic cycle evaluations of R-22 alternatives. As applied to the basic vapor compression cycle, this dual-mode program (BICYCLE) allows appropriate engineering constraints to be maintained for both design and off-design analyses of heat pumps. BICYCLE uses refrigerant property routines based on the Carnahan-Starling-DeSantis (CSD) equation-of-state and can handle multicomponent mixtures. The off-design COP and capacity predictions of the BICYCLE model for R-22 are compared to results from the more hardware-based ORNL Heat Pump Design Model to validate performance trends.

The relative performance of selected R-22 alternatives are evaluated for the design and off-design conditions that comprise the four ARI heat pump rating points. HFC mixtures considered are the R-32/R-125 azeotrope, low-glide binary and ternary zeotropes containing R-32/R-134a and R-125, and a higher glide binary mixture of R-32/R-227ea. Modeling assumptions and performance predictions are compared with those of Domanski and Didion (1993). Effects of passive and active composition shifting on low-temperature heating capacity are considered. HSPF rankings are determined for DOE Regions IV and V.

BACKGROUND

Simplified cycle evaluation models can provide an effective means of narrowing the list of possible R-22 alternatives and of evaluating the relative merits of proposed candidates. However, to be reliable indicators, these programs should predict refrigerant conditions that are closely representative of those to be expected in eventual hardware applications of each alternative.

McLinden and Radermacher (1987) reviewed methods for comparing the performance of pure and mixed refrigerants and proposed that equivalent total heat exchanger loading and external fluid temperatures provide the basis for such comparisons. Their recommended approach was most appropriate for single-design point conditions but has since been applied in questionable ways that go beyond the original premise.

In Table 1, some simplified models that are of most relevance to the present investigation are noted. Other models of related interest have been described by McLinden and Radermacher (1987) and Domanski and McLinden (1990).

TABLE 1
Summary Of More Recent Simplified Cycle Screening and Evaluation Models For Mixtures

MODEL	REFERENCE
CYCLE	Connon, 1984
CYCLE-7	McLinden and Radermacher, 1987
CYCLE-Z	Rice and Sand, 1990
CYCLE-11	Domanski and McLinden, 1990
SERCLE TERCLE	Jung and Radermacher, 1991
HAC1	Radermacher and Jung, 1991
CYCLE-11.DT	Pannock and Didion, 1991
CYCLE-10	Fischer, 1992
CYCLE-11.UA CYCLE-11.UADT	Domanski and Didion, 1993
Dew-Point-Based	Fischer and Sand, 1993

All of the above methods except for the CYCLE-11.UA model of Domanski and Didion (1993) are primarily single-design-point (i.e., "unicycle") models. While many of the earlier models listed in Table 1 (and those of Fischer, 1992 and Fischer and Sand, 1993) were applied at only one design condition, more recent uses have generally been over a range of ambients (Jung and Radermacher, 1991, 1993; Pannock and Didion, 1991; Domanski and Didion, 1993).

The models of Connon (1984) and Fischer and Sand (1993) were refrigerant-side specified (either average heat exchanger temperatures or dew points, respectively). The remaining models in Table 1 have specified air-side temperatures (inlet and often outlet) and contain some means of heat exchanger specification that allows the corresponding refrigerant-side temperatures to be determined.

Typical Rating Conditions for Heat Pumps

There are four steady-state rating point conditions for air-to-air heat pumps as specified by ARI Standard 210/240 (ARI 1989). As shown in Table 2, there are low- and high-temperature conditions for both heating and cooling modes. The high-temperature cooling condition is the design point condition where the rated nominal (cooling) capacity is determined. This is the condition at which the compressor displacement is determined. The remaining three conditions are referred to as off-design rating conditions. The low-temperature cooling point is also the test condition used for the seasonal energy efficiency rating (SEER).

TABLE 2
ARI Steady State Rating Conditions for Air-to-Air Heat Pumps

Rating Conditions	Heating Mode		Cooling Mode	
	Low-Temp. °C (°F)	High-Temp. °C (°F)	Low-Temp. °C (°F)	High-Temp. °C (°F)
Outdoor Coil	-8.33 (17)	8.33 (47)	27.8 (82)	35.0 (95)
Indoor Coil	21.1 (70)	21.1 (70)	26.7 (80)	26.7 (80)

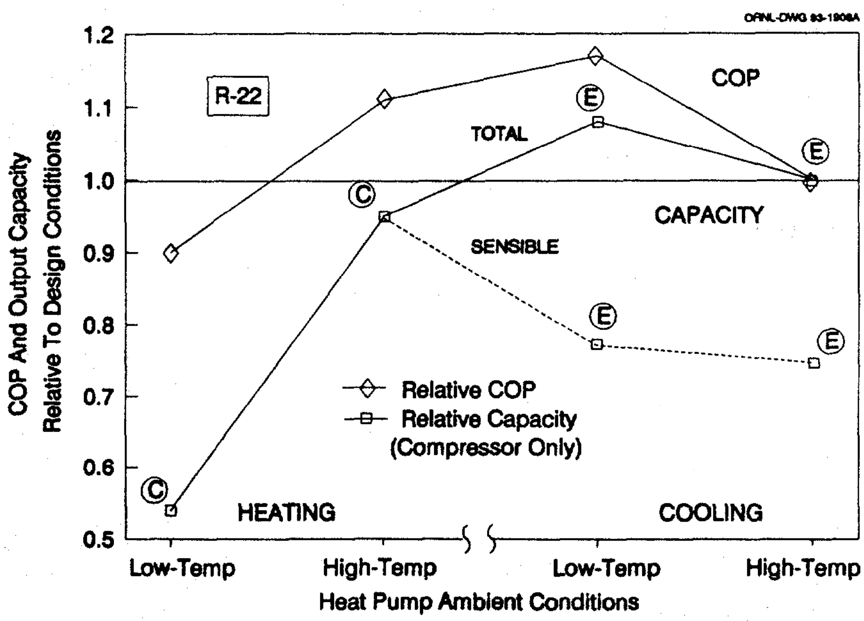
General Performance Trends of Air-To-Air Heat Pumps

The performance trends of air-to-air heat pumps versus ambient temperature are given in Figure 1 as predicted by the ORNL MODCON program (Rice 1991, 1992) for a representative configuration. The compressor-only COP and output capacity (for heating or cooling duty as appropriate) have been normalized by the cooling values at the nominal design condition of 35°C (95°F). The COP is seen to vary less than 20% in cooling and 10% in heating from the design value even though the theoretical heating COP is higher by one unit than the cooling COP at the same conditions. Output capacity is given by evaporator capacity in the cooling mode and by condenser capacity in heating mode as denoted by "E" and "C" in Figure 1.

In contrast to the COP, the total output capacity changes by a factor of two. This large capacity change is the combined result of the lower suction vapor density at the colder compressor inlet saturation temperatures in the heating mode (for compressors of constant swept volume) and the higher pressure ratio conditions on the compressor (giving lower volumetric efficiency). A large capacity change while COP remains nearly constant, implies that the compressor power also changes by a factor of two. Therefore accurate off-design COP predictions in heat pumps require good estimates of both capacity and power.

Both total and sensible cooling capacity are given in Figure 1 to illustrate that while total output capacity of the indoor coil is highest at the low-temperature cooling rating condition, the sensible output capacity, which determines the mean temperature difference (MTD) across the indoor coil, is highest at the high-temperature heating condition. This observation will later have implications with regard to proper modeling of the indoor coil in heating mode.

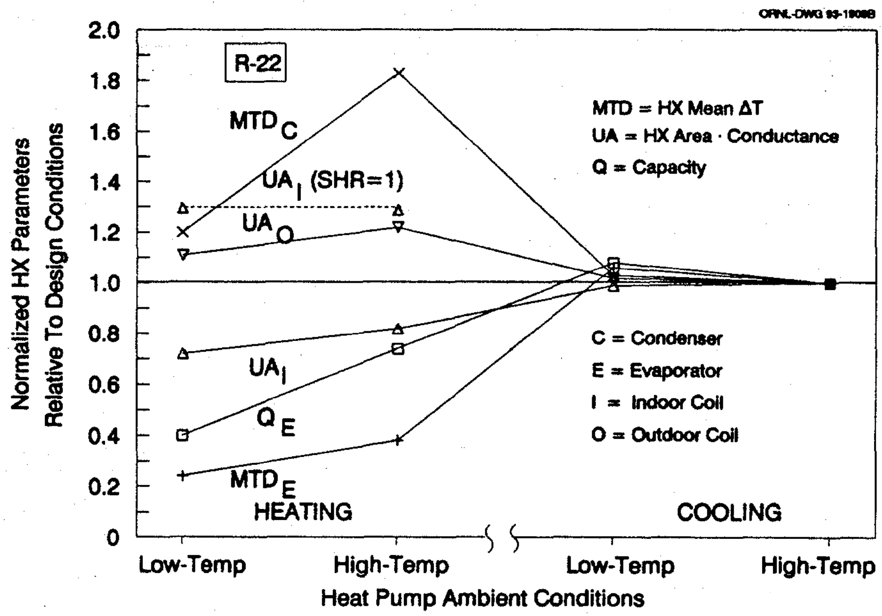
Figure 1: System Performance Trends of Air-to-Air Heat Pumps at Heat Pump Rating Conditions For R-22



Heat Exchanger (HX) Performance Trends of Air-To-Air Heat Pumps

In Figure 2, the corresponding HX performance measures are plotted for the evaporator and condenser or indoor and outdoor coil as appropriate. All the HX parameters are normalized with respect their values at the cooling design condition. While the HX parameters remain fairly constant over the cooling conditions, the switch to heating mode with differently sized HX capacities and reversed coil functions results in wide variations in condensing or evaporating performance. The UA levels of the indoor and outdoor coil are seen to remain more constant although variations of +20 to -25% are shown due to the change in refrigerant-side heat transfer coefficients from evaporating to condensing and vice-versa and from the change in refrigerant mass flow rate from cooling to heating mode.

Figure 2: Heat Exchanger Performance Trends of Air-to-Air Heat Pumps at Heat Pump Rating Conditions For R-22



In recent evaluations of R-22 alternatives, some investigators have used constant values of various evaporator and condenser performance parameters. Radermacher and Jung (1991, 1993), in an assessment for the U. S. Environmental Protection Agency, assumed constant evaporator capacity Q_E for all conditions along with constant UA levels for the indoor and outdoor coils. From Figure 2, the assumption of constant Q_E is seen to be inappropriate for heating-mode conditions—especially at the lower temperature heating conditions. Pannock (1992) assumed constant mean temperature differences (MTDs) for the condenser and evaporator in his analytical assessment of binary mixtures for heat pumps although, in his experimental apparatus, heat exchanger conditions more representative of actual heat pump conditions were maintained.

Domanski and Didion (1993) assumed constant UA levels for both heat exchangers at all ambients and further allowed the evaporator capacity to adjust at off-design conditions through the use of a constant compressor displacement and volumetric efficiency represented as a function of pressure ratio. Because of these assumptions, their representation of the *trends* of heat exchanger UA levels and capacities are the most consistent of all the simplified models reviewed with respect to Figure 2. For this reason, the analysis of Domanski and Didion is the most appropriate for comparison to the present work.

Implications of Neglecting Dehumidification Effects

When one further considers the *absolute* magnitude of the UAs used for the indoor and outdoor HXs, one of the simplifying assumptions of the Domanski and Didion approach becomes significant. Their approach, as in all the other simplified models, assumes that the heat exchanger capacities are due only to sensible heat transfer across air-to-refrigerant temperature differences. Because there is a sizeable fraction (typically 25 to 30%) of the cooling capacity that is due to dehumidification of the air stream, the implications of a sensible-heat-only (sensible-heat-ratio[SHR] = 1) assumption bear further investigation.

The effect of a sensible-heat-only assumption on cycle conditions depends on how UA levels are determined. In the case of Domanski and Didion (1993) and in the present analysis, UA levels are derived from known R-22 saturation temperatures at the design cooling condition. UA levels derived for the indoor coil under the assumption of SHR=1 will be too large by 25 to 30% for typical design SHR values of 0.70 to 0.75.[1]

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[1] If UA levels are known a priori from a separate HX model, the effects of a SHR=1 assumption will be seen as inaccurate predictions of refrigerant temperatures for the indoor coil at all conditions

This error occurs because the calculated UA level is determined from the basic HX heat transfer relationship of Q=UA•MTD. Because MTD is calculable from the specified air and R-22 temperatures, UA is directly related to the magnitude of Q. If total capacity is used for Q instead of sensible capacity, the derived UA level will be overestimated by the ratio of total to sensible capacity—the inverse of SHR. The latent portion of the heat transferred to the refrigerant does not affect the MTD although it must be accounted for in the refrigerant-side energy gain.

UA levels that are overestimated at the design cooling condition will primarily affect system performance predictions in the heating mode. Here, the oversized indoor coil, when applied to a heat rejection load that is 100% sensible, will give a condenser temperature that is too low. This error can be significant, especially at the high-temperature heating condition where, as shown in Figure 1, the indoor coil is the most heavily loaded.

In Figure 2, the net effect of the SHR=1 assumption on the indoor coil UA in the heating mode is shown by the dotted line. This error, when combined with the trend for the indoor coil UA to decrease in heating mode, could give an overestimate of the actual UA_I level in heating mode by as much as 85% (1.3/0.7). The evaporator sensible loading for heating at the higher ambients is also reduced because of dehumidification effects (SHRs of 0.9 or less). The significance of proper SHR treatment in the present context is that the UA levels affect the cycle performance of mixtures relative to pure refrigerants (Rice 1993).

APPROACH

Objective of Present Approach

The objective of this work is to provide a simplified yet accurate approach to modeling design and off-design performance of multi-component mixtures in a vapor-compression cycle. The resulting dual-mode cycle evaluation model (BICYCLE) for R-22 alternatives should be capable of reasonably matching heat pump performance trends of R-22 as predicted in more detailed hardware-based models (such as the ORNL MODCON heat pump design model, Rice 1991, 1992) by using overall HX performance parameters.

Improvements Of Present Approach

The BICYCLE evaluation model has the following improvements over previous simplified approaches:

- first-order effects of dehumidification on proper HX representation,
- constant airflow rates in combination with representative SHRs, and
- generic UA trends with ambient.

First, the influence of dehumidification on the determination of representative heat exchanger UA levels and loading is included. Second, the effect of dehumidification on the air-side temperature changes is considered. This, when combined with the use of constant air-flow rates, gives condenser air-side temperature differences (ranges) that are higher and evaporator ranges that are lower than the specified values of Domanski and Didion (1993) in the heating mode. The use of constant air-flow rates further allows the air-side glide to change appropriately at off-design conditions for R-22 alternatives with higher or lower off-design capacities. Third, also included are the +20% to -25% variations in UA levels from the changes in coil functions and conditions that result when the heat pump switches from cooling operation to heating.

We used baseline saturated R-22 conditions at the compressor inlet and exit, respectively, of 8.9°/47.8°C (48°/118°F) at the cooling design point. These conditions gave a design condenser UA about twice that for the evaporator. The 7.2°/46.1°C (45°/115°F) condition used by Domanski and Didion gave a three-to-one UA ratio when realistic sensible heat ratio effects were included—a ratio which we felt was unrealistically high. The net result in implied HX sizes was that our indoor coil was approximately 15% percent larger in cooling and 15% smaller in heating and the outdoor coil was about 5 percent smaller in either mode than that used by Domanski and Didion.

The remaining significant difference between the present analysis and that of Domanski and Didion regards the types of heat exchanger configurations and constraints that were applied.

Heat Exchanger Models and Assumptions

Fixed refrigerant pressure drops of 34.5 kPa (5 psia) and zero exit subcooling and superheat were assumed in the heat exchangers for comparability with Domanski and Didion (1993). Also, in each case, a constant total UA level across both heat exchangers was assumed. Beyond these assumptions, the heat exchanger analysis was conducted differently with regard to heat exchanger flow configuration and other imposed constraints.

Counterflow HXs were assumed in the present study to assess the full performance potential of mixtures. Domanski and Didion used a more conservative cross-flow assumption that was intended to more closely approximate existing hardware. A crossflow configuration penalizes mixture performance in heat pumps to varying degrees dependent on UA levels and number of rows as shown by Rice (1993).

Stevens (1957) has shown that four rows of countercrossflow configuration essentially provide counterflow performance. Therefore the assumption of counterflow implies that both coils have multiple rows, are circuited for countercrossflow, and that—for heat pumps—some means is provided to maintain unidirectional refrigerant flow in both heating and cooling modes. Although split system heat pumps typically have only one row outdoor coils, many package and rooftop units have multirow outdoor coils. Also, Rice (1993) has noted that it is more important with regard to heat pump system performance for the indoor coil to have four rows than for the outdoor.

A further constraint imposed by Domanski and Didion that was not adopted here was a fixed arithmetic mean refrigerant temperature in the evaporator (indoor coil) at the design cooling condition. We chose instead to allow the mean refrigerant temperature in the evaporator to seek its own thermodynamic value based on equivalent total UA levels and equal design capacity. In our analysis, the UA in each heat exchanger remains fixed whereas Domanski and Didion shifted area from the evaporator to the condenser for each gliding mixture to keep the mean evaporator temperature from rising or falling. The rationale for this additional constraint was an assumption that the same mean evaporator temperature was needed to maintain the same amount of dehumidification. However, it has been shown by Rice (1993) that an equivalent amount of sensible heat can be transferred with the same UA and the same heat transfer driving potential (same MTD) while the mean refrigerant temperature becomes more thermodynamically favorable for mixtures (e.g., higher in the case of an evaporator). Similarly, one can reason that an equivalent amount of latent heat transfer (i.e., dehumidification) can occur in an evaporator that has the same UA and mass transfer driving potential but has a higher mean refrigerant temperature that results from a more uniform HX loading (and less irreversibility) due to improved glide matching. To assume a constant mean refrigerant temperature is to negate the fundamental Lorenz effect that is the basis of improved cycle performance from better refrigerant-to-air temperature glide matching (Lorenz 1894).

The condenser superheat region was treated in a manner similar to that of Radermacher and Jung (1992) which is consistent with the approach used in the ORNL heat pump model as described in Fischer and Rice (1983). Condensation is assumed to occur at the tube wall even though the bulk refrigerant is superheated. This results in a heat transfer driving potential that is based on the difference between the refrigerant dew-point (rather than superheated refrigerant) and the air temperatures. Domanski and Didion (1993) treat the superheated region separately and assume condensation occurs only in the two-phase section of the heat exchanger (Domanski 1993).

Beyond these assumptions, the heat exchanger analyses were conducted in a generally comparable fashion in the manner described by Domanski and McLinden (1990) and Rice and Sand (1990).

Compressor Model and Assumptions

Compressor volumetric efficiency was represented in the model as a linear function of pressure ratio to provide adjustments for the different pressure ratios of various mixture alternatives to R-22. A linear regression fit was made to calorimeter data for R-22 and azeotropic and low-glide zeotropic R-32 blends in the same reciprocating compressor model (where the compressor displacement had been adjusted to give nearly the same design capacity). This assumption for volumetric efficiency is comparable to that made by Domanski and Didion (1993).

Isentropic efficiency is assumed to be the same for different refrigerants but was varied with ambient temperature according to a calorimeter performance map for R-22. The use of a constant polytropic efficiency by Domanski and Didion for all refrigerants at all ambients will likely yield less realistic absolute COPs at off-design conditions but this should have only a minor effect on relative COP comparisons.

Compressor shell heat loss was assumed to be 10% of the input compressor power requirements based on typical values observed by Miller (1988).

ANALYSIS PROCEDURE

Procedure at Design Cooling Conditions

The procedure used to compare all refrigerant mixtures on a consistent basis was to maintain constant heat exchanger loading at design conditions. This is the same general constraint adopted by McLinden and Radermacher (1987), Rice and Sand (1990) and Domanski and Didion (1993). The present implementation is most similar to that of Domanski and Didion except for the removal of their additional constraint on equal mean refrigerant temperature in the evaporator as described earlier.

For R-22, known refrigerant saturation temperatures, superheat and subcooling levels, and refrigerant pressure drops are specified along with desired air-side temperature ranges and a typical evaporator sensible heat ratio to back out the required UA levels and airflow rates for each heat exchanger and the displacement required to provide a specified design cooling capacity. These UA levels and airflow rates are then held constant and the compressor size for each R-22 alternative is determined that will provide the same design cooling capacity. A sensible heat ratio of 0.75 was assumed for all the refrigerants at the design condition.

Procedure at Off-Design Conditions

Once the design values of UAs , airflow rates, and refrigerant-specific displacements are known, they are applied at each of the off-design conditions along with the generic off-design UA corrections and appropriate sensible heat ratios. In this way, off-design capacities and air temperature changes can change appropriately for the refrigerant mixtures based on the off-design suction density characteristics of each alternative at conditions determined for fixed UA levels and air flow rates.

VALIDATION OF PRESENT APPROACH

The more hardware-based ORNL MODCON program was used to establish a baseline heat pump and performance characteristics for comparison to the BICYCLE model. MODCON provided design saturation temperatures and typical values of pressure drop, air-flow rates, and sensible heat ratios. With these values and capacity at design conditions specified, the BICYCLE code was used to determine the required UAs and the R-22 compressor size. These design parameters were held fixed for off-design calculations at the low-temperature cooling conditions and at the two heating mode conditions.

The design and off-design COPs and capacities predicted with the BICYCLE code using this analysis procedure are compared in Figures 3 and 4, respectively, with those from the hardware-based MODCON program. The effect of various

Figure 3: COP Comparisons of the BICYCLE Program Versus the ORNL Heat Pump Model For R-22

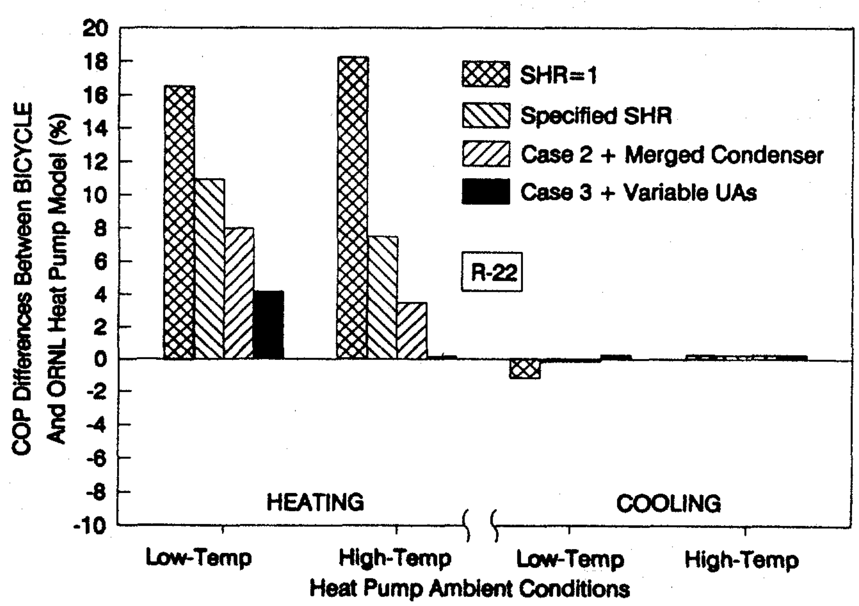
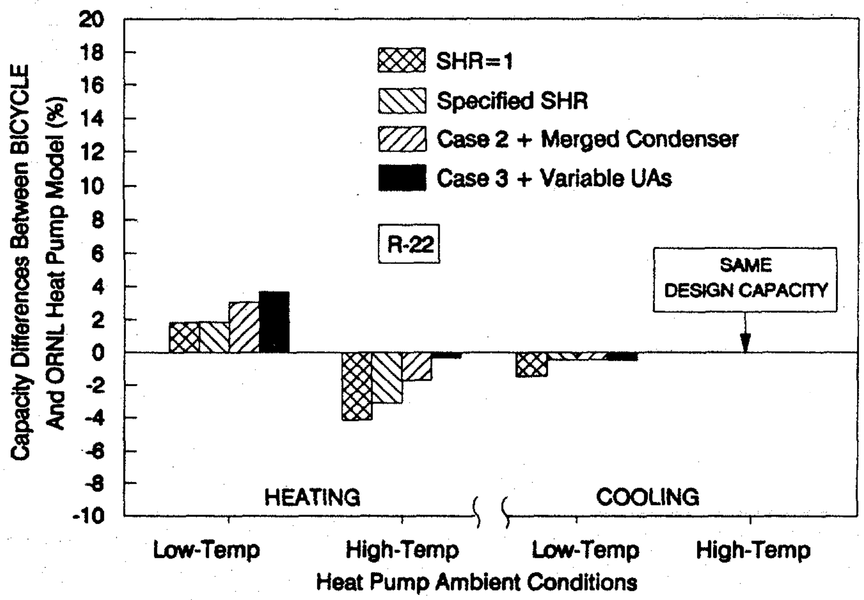


Figure 4: Capacity Comparisons of the BICYCLE Program Versus the ORNL Heat Pump Model For R-22



simplifying assumptions on the accuracy of the results are also shown in these figures by introducing the model improvements one-by-one. The first case at each ambient represents the case with SHR=1, a standard (separate) treatment of the condenser superheat region, and constant UA levels.

Errors in predicted capacity are less than ± 4% in all cases while cooling COPs are within 1.5%. Discrepancies in predicted heating COP range from a high of 17 to 19% with no modeling improvements to a low of 0.5 to 4% with all changes active. The shift in model assumption from an SHR of 1 to realistic SHR levels results in the biggest improvements of 5.5% and 11% at the low- and high-temperature heating conditions, respectively. The revised (merged) condenser superheat treatment and the use of generic UA corrections each provide an additional 3 to 4% improvement.

With all improvements included, the BICYCLE-predicted COPs, capacities, and UA levels agree closely with those from MODCON (within 1% except at low-temperature heating where COP and capacity were 3 to 4% high). BICYCLE-predicted off-design saturation temperatures were within 0.75°F of the MODCON values at all conditions. These comparisons served to confirm that BICYCLE was working properly for the tested options.

ANALYSIS RESULTS WITH R-22 MIXTURE ALTERNATIVES

With the mixture evaluation model validated for R-22, a series of cases were run to predict the design and off-design performance of four R-32-based HFC mixtures. The four mixtures and their compositions were selected from the R-22 alternatives being tested under the ARI Alternative Refrigerants Evaluation Program [AREP] (ARI 1992). These four AREP mixtures cover the range of R-22 alternatives from azeotropes through lower- and higher-glide zeotropes[2].

One azeotrope, two low-glide blends, and one high-glide mixture were chosen. They are, respectively, the R-32/R-125 [60/40 wt%] azeotrope, a low-glide binary mixture of R-32/R-134a [30/70 %wt], the low-glide ternary blend of R-32/R-125/R-134a [30/10/60 %wt], and a high -glide binary of R-32/R-227ea [35/65 wt%]. The R-32 azeotrope is known commercially as AZ-20 from Allied-Signal and the low-glide ternary is the original formulation of AC-9000 from DuPont.

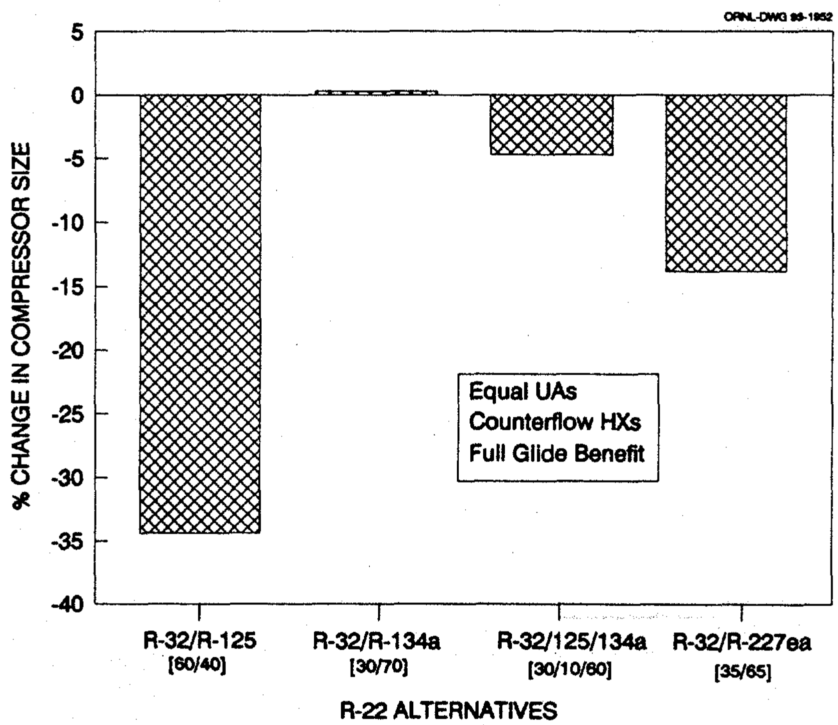
The CSD equation-of-state (EOS) as provided in REFPROP V 3.0.4 was used as the basis for calculating the thermodynamic properties of the pure and mixed refrigerants (Gallagher et al, 1992). The CSD EOS coefficients and the interaction

[2] Low-glide mixtures have a temperature change from bubble-to-dew-point of around 5 C° or less which is half as large or less than the air temperature ranges (changes) across the heat exchangers of air-to-air heat pumps at design conditions. High-glide mixtures have bubble-to-dew-point temperature differences that are close to or greater than the design air temperature ranges.

coefficients for R-32, R-125, R-134a, and R-227ea used by Domanski and Didion (1993) were also adopted for the present cases (from Domanski 1993) in order to eliminate thermodynamic properties as a potential source of difference in analysis results.

The mixture alternatives were evaluated in the BICYCLE model for the same UA levels, air-flow rates, and sensible heat ratios as for R-22. The compressor displacements relative to R-22 required to obtain the same design capacity are shown in Figure 5. AZ-20, the high-pressure azeotrope, requires a compressor 1/3 smaller than for R-22. The R-32/R-134a blend is predicted to require essentially the same size compressors as R-22 if full glide matching benefits are realized in the heat exchangers while AC-9000 would require a 5% smaller compressor. (AC-9000 was formulated to give the same capacity as R-22 in existing equipment. If full glide benefits are realized from HX redesign, this mixture should gain 5% in capacity from the resulting higher suction density and lower discharge pressure.)

Figure 5: Required Compressor Displacement Relative to R-22 for Four HFC Mixture Alternatives With the Same Design Capacity



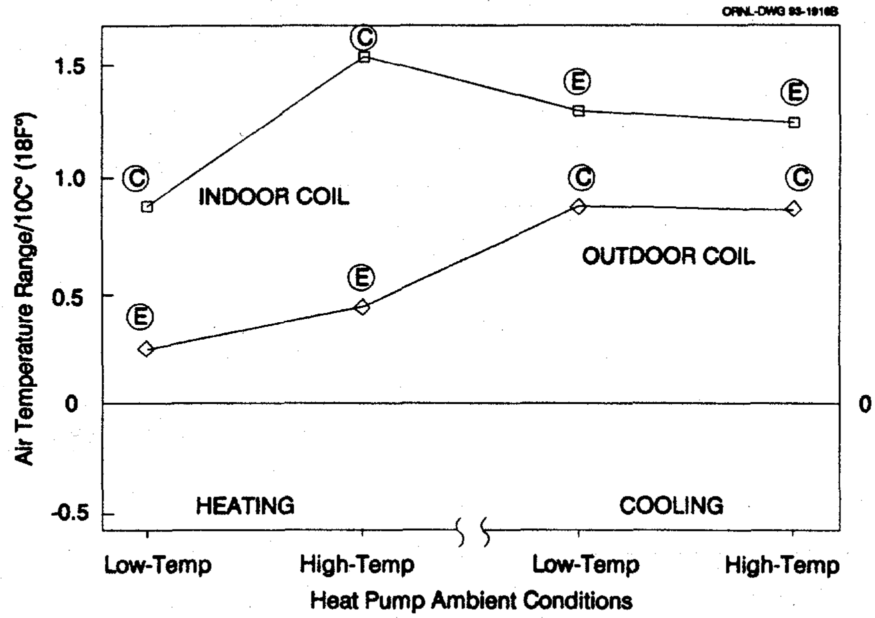
Air Temperature Ranges

The air-side temperature ranges (absolute differences between HX inlet and exit air temperatures) at the four rating conditions with R-22 are shown in Figure 6 for the indoor and outdoor coils. The function of the coils as evaporators or condensers are denoted by circled E's and C's. The ranges have been normalized by dividing by a nominal temperature difference of 10 C° (18 F°), a value which is near the average of the indoor and outdoor air-side ranges at the design cooling condition. The temperature ranges are the same for all refrigerants considered at the high-temperature cooling condition (because of the same design capacity and air flow rates) and will vary at off-design conditions directly with the relative off-design capacities.

Looking at Figure 6, one can see that the glides for the outdoor coil are considerably less than the indoor coil—especially in heating mode. The ratio of indoor to outdoor coil air-side ranges are about 1.5 to 1 in cooling and more like 3 to 1 in heating mode.

Applying the same analysis based on evaporator and condenser function, the evaporator range is 1.5 times larger than the condenser in the cooling mode but 3 times smaller than that of the condenser in the heating mode. The temperature ranges for condensers are seen to be more uniform except for a peak at the high-temperature heating condition. Evaporator air-side ranges vary by a factor of 3-or-4 to 1. Clearly, close air- to refrigerant-glide matching at all conditions would require some active means of adjusting glide for each condition as described by Radermacher (1986). The ideal average refrigerant glide to match these air-side ranges would be about 10 C° in cooling and about 6 C° in heating.

Figure 6: Air-Side Temperature Ranges at Heat Pump Rating Conditions —Indoor and Outdoor Coils With R-22



Refrigerant Glide Matching

In Figures 7 and 8, the refrigerant-side glides computed for R-22 and the four mixture alternatives are shown for the indoor and outdoor coils, respectively. The glides have been normalized by the same nominal temperature difference of 10 C° (18 F°) as for the air-side temperature ranges. Refrigerant glide is defined as positive if temperature from inlet to exit is increasing in the evaporator and decreasing in the condenser (i.e., positive if the refrigerant glide through the HX is toward the inlet air temperature).

Figure 7: Refrigerant-Side Temperature Glides for Four R-22 Alternatives—Indoor Coil

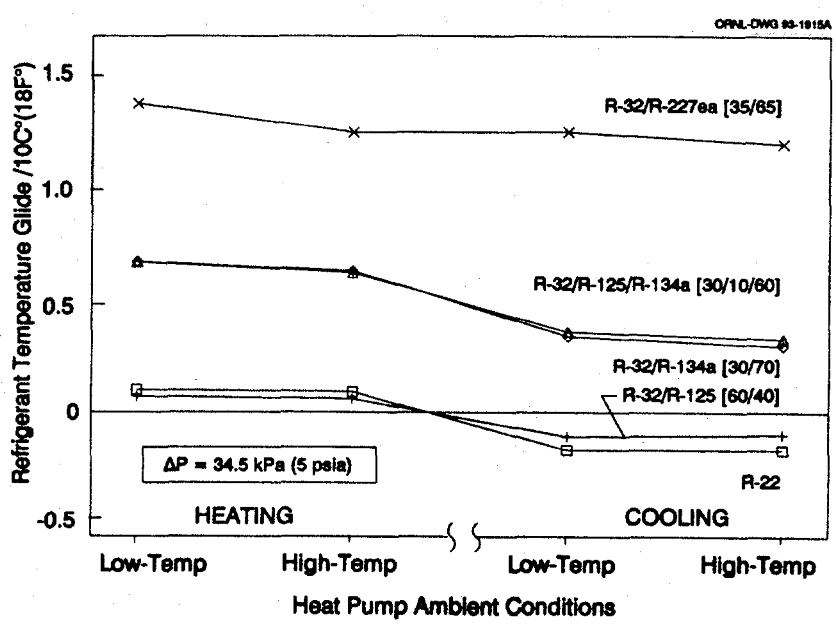
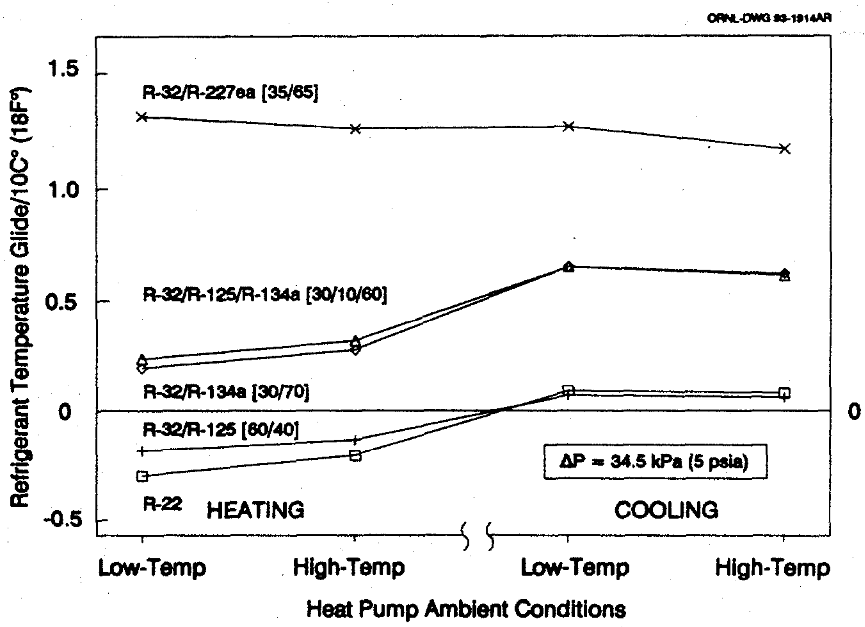


Figure 8: Refrigerant-Side Temperature Glides for Four R-22 Alternatives—Outdoor Coil



The assumed constant pressure drop of 34.5 kPa (5 psia) is seen to result in a negative glide for R-22 and a reduced glide for the evaporator conditions (indoor coil in cooling and outdoor coil in heating). The pressure drop adds about 1 C° to the glide in the condenser while reducing the evaporator glide by 1.5 to 2 C°for R-22 with larger effects for AZ-20 and the low-glide mixtures. The effect is larger for evaporators because the change of saturation temperature for a fixed pressure change is larger for typical refrigerants at lower temperatures. For R-22 and AZ-20, refrigerant-side pressure drops have the effect of making evaporator operation mildly cocurrentflow and condenser operation slightly counterflow.

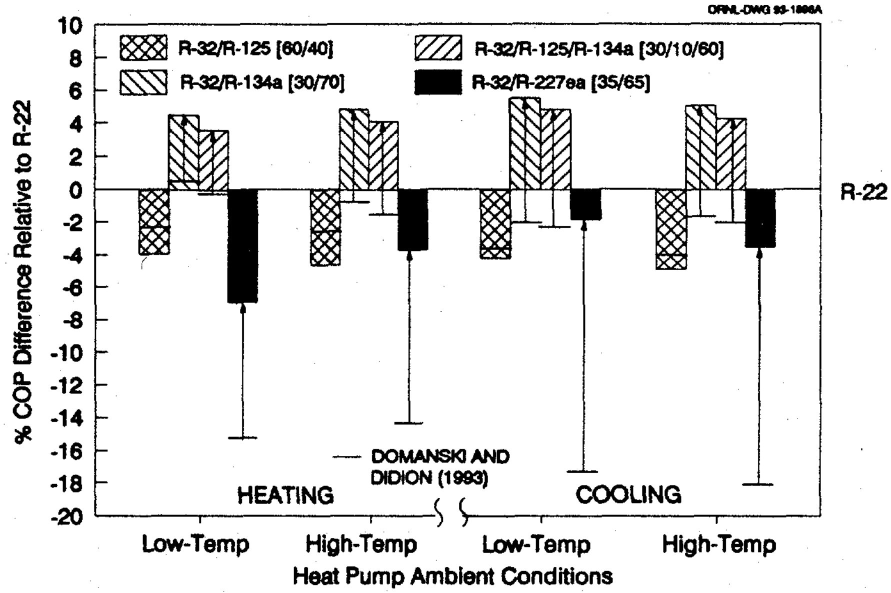
Figures 7 and 8 can be compared to Figure 6 to determine how well each mixture matches the air-side temperature ranges at the different operating conditions. The high-glide blend is seen to match the air-side range of the indoor coil almost exactly in cooling mode and quite well in heating mode. The air-side range of the outdoor coil is overglided by the high-gliding mixture by a factor of 1.5 to 1 in cooling mode and by ratios of 3-to-1 to 5-to-1 at the high-and low-temperature heating conditions. This suggests that glide matching benefits for R-32/R-227ea [35/65] should be largest in the cooling mode and less in the heating mode as overgliding penalties offset to some extent the close indoor coil glide matching.

For the low-glide blends, an surprisingly good match is seen on the outdoor coil between refrigerant glide and air-side range. The refrigerant glide is at most only 25% higher than the air temperature change. For the indoor coil, a good match also occurs at the low temperature heating condition, while elsewhere the refrigerant underglides the air by a factor of 2 in heating and 4 in cooling. However, because there is no significant overglide anywhere and because there is usually one well-matched glide of moderate size at each condition, some glide matching benefit should be realized at each condition.

COP and Capacity Comparisons of Selected Mixtures To R-22

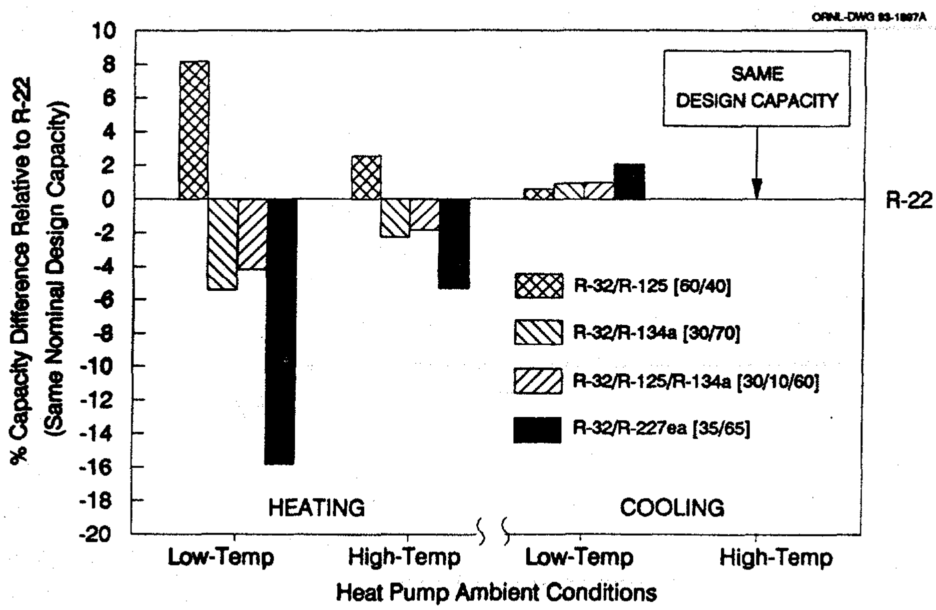
COP and output capacity for the mixtures relative to R-22 are shown in Figures 9 and 10, respectively. In Figure 9, the COPs for the lower-glide mixtures -- AC-9000 and R-32/R-134a [30/70] -- show a potential 4 to 5% gain with counterflow heat exchangers (obtainable with four rows of crosscounterflow circuiting with unidirectional refrigerant flow). These predictions are 5 to 6% more favorable than those of Domanski and Didion (1993)—as shown by the horizontal bars in Figure 9—where crossflow heat exchangers were assumed and the evaporator mean temperature was held constant. The BICYCLE-predicted cooling performance of the high-glide R-32/R-227ea mixture was about 15% higher than estimated by Domanski and Didion because this mixture benefits most from glide matching and counterflow configuration in the cooling mode. The arrows in Figure 9 highlight the performance gains, relative to Domanski and Didion, predicted from the present analysis.

Figure 9: Relative COPs Predicted by the BICYCLE Model for Four R-22 Alternatives —Counterflow Heat Exchangers



In Figure 10, the cooling and heating capacities for the R-32 blends are shown relative to R-22 at the same ambients. All mixture systems had the same capacity as R-22 at the design cooling condition for the reasons discussed earlier. The R-32 azeotrope has the most desirable off-design capacity trends, followed by the ternary and binary low-glide blends.

Figure 10: Relative Capacities Predicted by the BICYCLE Model for Four R-22 Alternatives —Counterflow Heat Exchangers



Effect of Excessive Refrigerant Glides on Heating Performance

The higher glide R-32 mix has a considerably poorer relative capacity as the heating ambient is lowered. This is because the high-glide R-32 blend pinches significantly in the outdoor evaporator, especially at the low temperature heating condition, as can be seen from a comparison of the air temperature ranges versus refrigerant glide from Figs 6 and 8 for the outdoor coil in heating mode. HX temperature pinching results in a much lower mean refrigerant temperature for the blend and, because of this, a much poorer heating capacity. This was confirmed by a comparison with thermodynamic-property-only predictions which indicate that, at the same mean refrigerant temperatures, R-32/R-227ea would have had a relative heating capacity similar to AZ-20.

The poor low-temperature heating performance predicted for the high-glide R-32 zeotrope demonstrates a disadvantage of excessive glide and presents a challenge of matching the higher glide requirements in cooling mode while minimizing evaporator penalties in the heating mode. Suction-to-liquid line intercooling in cooling mode will be investigated in the near future as a way to widen the mixture glides preferentially in cooling mode.

Effect of Passive and Active Composition Shifting

The low-glide zeotropes are predicted to have low-temperature heating capacities 4 to 6% less than R-22 while the capacity of AZ-20 is greater than R-22 by 8%. As a result, AZ-20 shows a 12 to 14% advantage over the low-glide R-32 mixes at the low-temperature heating ambient.

However, the zeotropes have a potential offsetting factor in that composition shifting in heating mode can boost the R-32 composition of the circulating refrigerant and thereby the capacity. To evaluate the possible size of this effect, scenarios of passive and active composition shifting were considered.

For the passive composition shifting, a 6 percentage point increase in the R-32 concentration was assumed based on AC-9000 data reported by Shiflett (1993). A 2 percentage point increase in the R-125 concentration was also assumed. In active composition shifting, an 18 percentage point increase in R-32 (close to the 20% used by Radermacher and Jung 1992) and a 6 percentage point increase in the R-125 were postulated. For the passive cases, concentrations of [36/64 wt%] R-32/R-134a and [36/12/52 wt%] for R-32/R-125/R-134a were used. The active compositions assumed were [48/52] and [48/16/36], respectively.

The passive composition shift occurs in a heat pump system with a suction line accumulator (receiver) that "accumulates" a liquid refrigerant level that rises as the ambient decreases in the heating mode. The vapor that is drawn off through the j-tube of an accumulator is consequently richer in the more volatile, higher capacity R-32 component. In the active composition shifting scenario, additional hardware would be required to approach a single-stage distillation (rectification) of the refrigerant mixture in the heating mode (Radermacher and Jung 1992).

The results of these assumptions on the relative capacities (and COP) of the low-glide R-32 zeotropes are shown in Figure 11 for the low temperature heating condition. For the passive-shift case, the R-32 binary gains 5.4% and the ternary gains 6.5% relative to the original compositions. In the active case, the gains are 15.9 and 19.2%, respectively. Thus on systems with an accumulator, the low-glide R-32 blends could slightly exceed the capacity of R-22. With active composition shifting of the size postulated, the low-glide blends could exceed the low temperature capacity of AZ-20 by up to 7%.

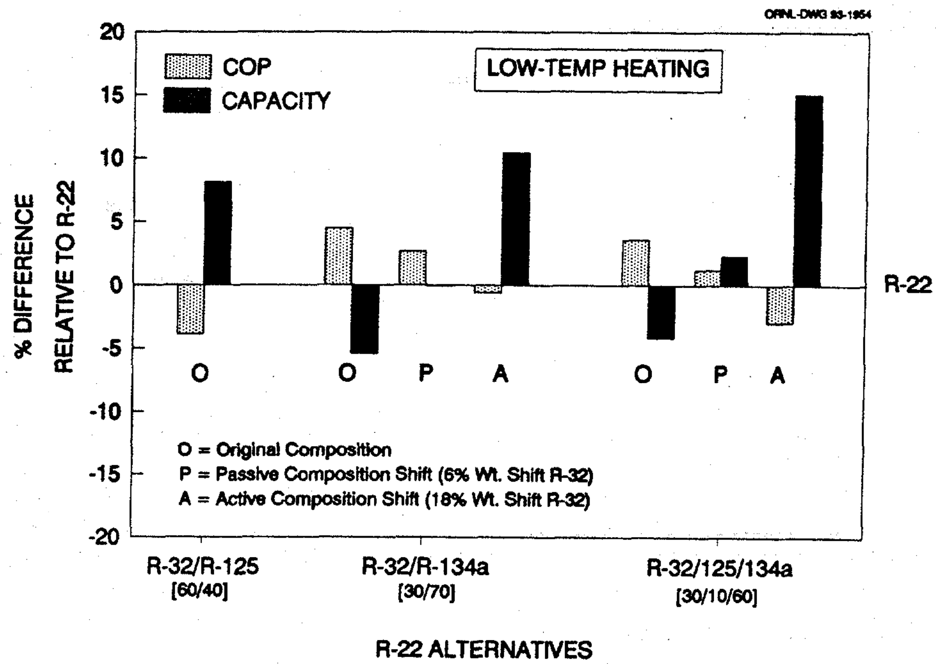
With regard to COPs, the passive composition shifting reduces the low-temperature heating COP by 1.7 to 2.3% for the binary and ternary cases, respectively, while with active shifting, the COPs drop 4.9 to 6.4%. However, the reduction of supplemental resistance heat (at a COP of 1.0) by the increase in capacity more than offsets, on a seasonal energy use basis, this slight drop in heating COP.

Seasonal Performance Comparisons For U.S. DOE Climate Regions IV and V

In Figure 12, heating seasonal performance factors (HSPFs) are shown relative to those for R-22 for all the original composition mixtures and for the composition shifting scenarios considered. U.S. DOE Climate Regions IV and V are considered so as to include the standard Region IV region (on which seasonal rating labels are based) and one region (V) more representative of heat pump applications in northern climates (CFR 1992).

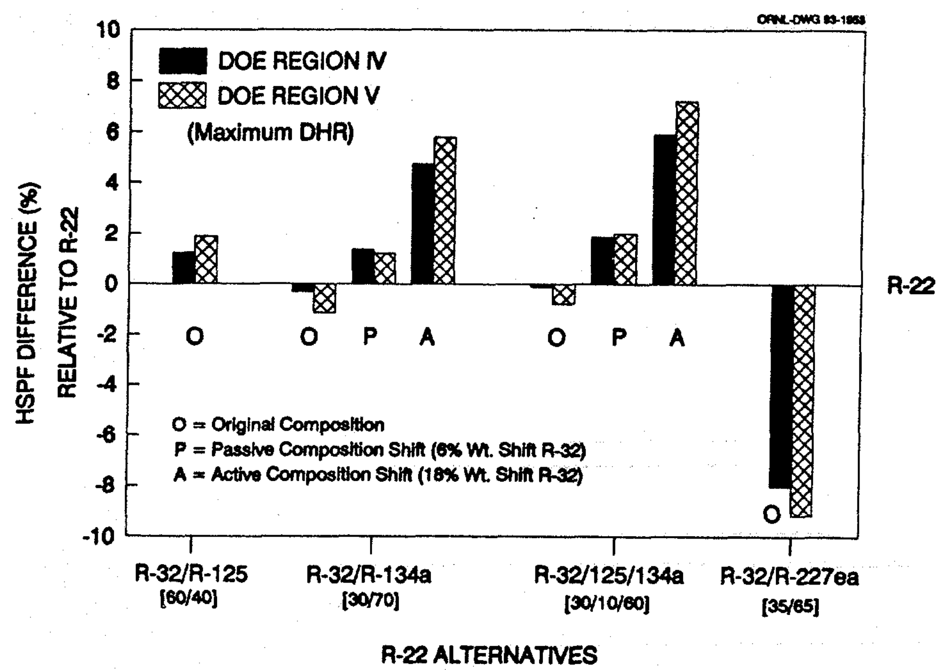
The HSPFs were calculated using the maximum design heating requirement (DHR) rather than the minimum DHR that is used in the seasonal labeling program. This was done to obtain more representative balance point temperatures (below which the heat pump capacity cannot fully meet the house load) near 0°C (32°F). Use of the minimum DHR gives balance point temperatures around -8.3°C (17°F) which can give an unrealistic estimate of the effects on HSPF of the different heating COP and capacity characteristics of the mixture cases considered (by giving more credit for COP gain and less for capacity gain than is realistic).

Figure 11: Relative Low-Temperature Heating COP and Capacity Comparisons for Four R-22 Alternatives with Passive and Active Composition Shifting



AZ-20 can be seen in Figure 12 to seasonally outperform R-22 by 1.5 to 2% in the heating mode. The azeotrope also outperforms the considered zeotropes at their original compositions by at least 1.5% to as much as 11% when compared to the high-glide case. When passive composition shifting is considered, the low-glide R-32 alternatives perform comparably to AZ-20 in DOE Regions IV and V. With active composition shifting of the amounts postulated, the low-glide zeotropes could give HSPF gains 2.5 to 4 times larger than AZ-20—for a 5% to 8% HSPF gain relative to R-22.

Figure 12: Relative HSPFs for Four R-22 Alternatives In DOE Regions IV and V with Passive and Active Composition Shifting



Comparisons of cooling energy efficiency ratios (SEERs) are not shown in a separate figure because they are directly obtainable from the COP comparisons of the different mixtures at the low-temperature cooling condition of Figure 9. There, AZ-20 is predicted to have an SEER about 4% less that R-22 while the low-glide R-32 blends could have SEERs 5 to 6% higher than R-22. The high-glide blend could perform seasonally in cooling to within 2% of conventional units.

From these seasonal comparisons, AZ-20 is seen to show a little less HSPF gain than SEER loss. The seasonal heating potential for the low-glide R-32 alternatives is seen to be less than in cooling unless active composition shifting is used. With active composition shifting, HSPF gains could be equal to those in cooling for the low-glide binary and could be 50% larger than the SEER gains for AC-9000.

CONCLUSIONS

The heat exchanger representations of previous mixture cycle models are shown to be oversimplifications—to varying degrees—of off-design HX performance in heat pumps. With more realistic HX assumptions, a simply applied UA-based evaluation model—BICYCLE—is shown to closely predict off-design COP and capacity trends in existing heat pumps. This is accomplished by the combined use of constant airflow rates, first-order sensible capacity effects, more realistic treatment of the condenser superheat region, and UA adjustments for HX function.

Initial assessment of the potential of four R-22 alternatives—assuming equal UA levels—show 5 to 15% higher cooling COP for the R-32 low- and high-glide blends, respectively, than that predicted by Domanski and Didion (1993). The low-glide binary and ternary blends are predicted to have 4 to 5% higher cooling and heating COPs and SEERs than R-22 if suitably designed cross-counterflow heat exchangers are used. These results are more encouraging than those of Domanski and Didion (1993) because all available glide matching benefits were realized for the mixtures.

Seasonal heating improvement potential for the low-glide R-32 alternatives is less than in cooling unless active composition shifting is used. HSPFs for the low-glide R-32 alternatives are equivalent to those of the R-32 azeotrope in DOE Regions IV and V if passive composition shifting is considered. The R-32 azeotrope is predicted to outperform R-22 by up to 2% in HSPF and to underperform by 4% in SEER. The high-glide R-32 blend performs only slightly poorer than R-22 in SEER but has HSPFs more than 8% poorer due to evaporator overgliding and the resulting effects of heat exchanger pinching.

RECOMMENDATIONS

Further investigation should be made of mixtures with higher-glides than AC-9000—nearer 10 C° (18 F°) in cooling—that have similarly favorable thermodynamic properties (e.g., good COP, low pressures, linear refrigerant glides) and improved heating capacity. These characteristics could possibly be combined with cycle modifications to achieve lower glides in heating and perhaps to further boost heating capacity. An analysis of most promising mixtures and cycle designs of this nature could help determine the maximum available benefits in heat pumps from glide matching with known mixtures. This information would aid manufacturers in assessing whether the available glide matching potential is sufficient to warrant the HX redesign required to take full advantage of higher-glide mixtures.

The capability of the BICYCLE model to predict the relative design and off-design pressure drops of R-22 alternatives (based on relative refrigerant flow requirements at the same design capacity) should be included in this evaluation. Routines to estimate mixture flammability should be added to the model to identify the most promising non-flammable candidates.

A remaining significant improvement in the cycle evaluation of R-22 alternatives would be to include the effects of transport properties (thermal conductivity, viscosity, and surface tension) and refrigerant-optimal HX circuit design on the relative UA levels and pressure drops of refrigerant mixtures. Models with this capability will more completely establish the R-22 alternatives with the best overall design and off-design performance potential.

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