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Evaluation of Carbon Dioxide as R-22 Substitute for Residential Air-Conditioning

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ABSTRACT

This paper compares the performance of CO_2 and R-22 in residential air-conditioning applications using semi-theoretical vapor compression and transcritical cycle models. The simulated R-22 system had a conventional component configuration, while the CO_2 system also included a liquid-line/suction-line heat exchanger. The CO_2 evaporator and gas cooler were microchannel heat exchangers originally designed for CO_2. The R-22 heat exchangers employed the same microchannel heat exchangers as CO_2 with the difference that we modified the refrigerant passages to obtain reasonable pressure drops. The study covers several heat exchanger sizes. The R-22 system had a significantly better coefficient of performance (COP) than the CO_2 system when equivalent heat exchangers were used in the CO_2 and R-22 systems, which indicates that the better transport properties and compressor isentropic efficiency of CO_2 did not compensate for the thermodynamic disadvantage of the transcritical cycle in comfort cooling applications. An entropy generation analysis showed that the CO_2 evaporator operated with fewer irreversibilities than did the R-22 evaporator. However, the CO_2 gas cooler and expansion device generated more entropy than their R-22 counterparts and were mainly responsible for the low COP of the CO_2 system.

INTRODUCTION

Due to the Montreal Protocol and subsequent regulations, the air-conditioning industry is in the process of evaluating and introducing new refrigerants as replacements for chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). Finding a suitable replacement for R-22, by far the most commonly used refrigerant for these applications, is proving to be a challenge. Hydrofluorocarbon refrigerants (HFCs) and their mixtures have been considered as the primary replacements for R-22, with the R-410A zeotropic mixture being perhaps the favorite substitute because of similar efficiencies reported for R-22 and R-410A systems. In recent years, carbon dioxide has received significant interest as a possible R-22 replacement. This interest is due to carbon dioxide's environmental attributes: CO_2 naturally exists in the atmosphere and has a lower global warming potential (GWP) than hydrofluorocarbons (HFCs) such as R-410A.

If refrigerant selection is to be made to mitigate global warming, both the refrigerant (direct) and energy (indirect) contributions of the refrigerant must be considered. The refrigerant effect is related to the ability of the refrigerant's molecule to trap the earth's infrared radiation once this molecule is released into the atmosphere. The energy effect is related to the trapping of the earth's infrared radiation by the carbon dioxide released into the atmosphere by the fossil fuel power plant providing electricity for the air conditioner. Hence, the energy impact is affected by the system efficiency. These two contributions are recognized in indexes that were developed for evaluating global warming contributions of different technologies, e.g., total equivalent warming impact (TEWI) or life cycle climate performance (LCCP). Calm (1993) showed that the energy contribution is much more significant than the refrigerant contribution for unitary air-conditioning equipment. Depending on the assumed scenario (refrigerant leak rate, operating hours, servicing practices, equipment efficiency, etc.), the energy contribution constituted between 93.2% and 99.5% of a TEWI for an 8.8 kW residential system. The study by Sand et al. (1997) also showed the dominating

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influence of the energy effect. These results emphasize the need for residential air-conditioning systems employing alternative refrigerants to have similar efficiencies as existing systems.

The importance of high efficiency becomes even more obvious when we consider the current and possible future regulatory initiatives to increase the minimum efficiency for residential equipment. Based on the debate associated with the current rule-making process (DOE 2001), we can expect the minimum seasonal energy efficiency ratio (SEER) for residential equipment to be increased from 10.0 to 12.0 or 13.0 starting in 2007.

There are relatively few studies in the open literature on the use of CO_2 in residential air-conditioning applications as compared to the number of studies on the use of CO_2 in automotive air-conditioning systems. Among theoretical studies, Bullock (1997) analyzed the performance of a CO_2 cycle by varying the temperature of CO_2 entering the gas cooler up to the critical point. He found that the CO_2 system is less efficient than the R-22 system by 30% in the cooling mode and by 25% in the heating mode. He recommended that to obtain comparable efficiency with typical unitary equipment, a CO_2 system would require the use of an expander and the use of a significantly improved compressor and heat exchangers. Robinson and Groll (1998) theoretically investigated the performance of a transcritical CO_2 cycle with and without an expansion turbine and compared it to a conventional R-22 cycle. They found that the coefficient of performance (COP) of the CO_2 cycle with an internal heat exchanger with an effectiveness of 100% is 72% to 80% of the R-22 cycle, while the COP of the CO_2 cycle with complete work recovery and without an internal heat exchanger is 87% to 100% of the R-22 cycle at evaporating temperatures above –25°C (13°F). Hwang and Radermacher (1998) theoretically compared the performance of a conventional R-22 cycle with a transcritical CO_2 cycle for water-heating and water-chilling applications. They suggested that tap water-heating is a potential application for CO_2 since its COP is approximately 10% better than that of R-22 for a wide range of ambient temperatures. In the water-heating application, CO_2 benefited from a good temperature glide match in the counterflow gas cooler.

The experimental studies found in the literature for CO_2 residential air conditioners compared the performance of a prototype CO_2 system with conventional R-22 and R-410A systems. Aarlien and Frivik (1998) compared the performance of a prototype CO_2 system with a baseline R-22 system in the cooling and heating modes. The CO_2 system consisted of microchannel heat exchangers with the same core volumes as the finned-tube heat exchangers used in the baseline system. They found that the COP of the CO_2 system in the cooling mode was from 0.5% to 14% lower than that of the R-22 system, while the COP in the heating mode was from 3% to 14% higher than that of the R-22 system. Boewe et al. (1999) investigated the influence of a liquid-line/suction-line heat exchanger (llsl-hx) on the performance of a CO_2 air conditioner. The use of the llsl-hx increased both cooling capacity and COP but also increased the compressor discharge temperature, imposing an effectiveness limit. Beaver et al. (1999a) presented experimental results for a prototype CO_2 system and a commercially available R-410A heat pump in the cooling mode. The CO_2 system utilized microchannel heat exchangers, while the R-410A system used fin-tube heat exchangers having the same core volume as the CO_2 heat exchangers. The CO_2 system yielded a comparable COP to the R-410A system at an ambient temperature of 27.8°C (82°F), while its COP was 10% lower at an ambient temperature of 35°C (95°F). The airside pressure drops in the CO_2 heat exchangers were lower than those of the R-410A system, which allowed the CO_2 system to have higher airflow rates than the R-22 system for the same fan powers. As a follow-up to this study, Richter et al. (2000) measured the performance of the CO_2 and R-410A systems in the heating mode. When matched for heating capacity at an ambient temperature of 8.3°C (47°F), the CO_2 system yielded 0% to 6% lower COP and approximately 13% greater capacity than the R-410A system at an ambient temperature of 8.3°C (47°F). At an ambient temperature of 16.7°C (62°F), the COP of the CO_2 system was 99% to 101% of the R-410A system, and the capacity was 93% to 105% of the R-410A system.

In the cited experimental studies, the cooling COP of the CO_2 system is either comparable or inferior to that of R-22 or R-410A systems. In each case, the CO_2 results were obtained for systems using high-performance microchannel heat exchangers, while the R-22 or R-410A results were measured for systems equipped with conventional finned-tube evaporators and condensers. These arrangements provided a clear performance advantage for CO_2 since microchannel heat exchangers offer superior heat transfer performance over their finned-tube counterparts. For example, Kim and Bullard (2001) compared the performance of a microchannel condenser with a conventional finned-tube condenser in a window-type R-22 air conditioner. They found that, compared to the finned-tube heat exchanger of the same capacity, the core volume and weight of the microchannel condenser were smaller by 55% and 35%, respectively.

It should be viewed as a truism that a fair evaluation of performance potentials of competing refrigerants should be carried out using equivalent equipment. In this case, using microchannel heat exchangers in each system is in order. Laboratory tests of CO_2 and R-22 systems with microchannel heat exchangers optimized for each fluid would lead to a fairer evaluation for the fluids tested. This experimental path would be the preferred method but also would be the most time consuming and expensive. As an alternative, we analyzed the performance of CO_2 and R-22 using semi-theoretical simulation models, which account for different thermodynamic and transport properties of the refrigerants and allow one to represent different refrigerant circuitry arrangements in the heat exchangers. In this study, we used the CO_2 microchannel heat exchangers for R-22. The heat exchangers had the same air-side configurations, but we redesigned the R-22 refrigerant passages to attain reasonable refrigerant pressure drops. All significant refrigerant and system-related factors are considered in the model or discussed if omitted. Consequently, we believe that this approach provides a realistic evaluation of CO_2 performance potential as an R-22 substitute.

SIMULATION MODELS

Modeling Approach

As did Brown et al. (2002), we used two related simulation models: CYCLE-11.UA and CYCLE-11.UA-CO_2. CYCLE-11.UA is a semi-theoretical model for evaluating performance of refrigerants or refrigerant mixtures in the vapor compression cycle (Domanski and McLinden 1992; Domanski et al. 1994a). The model performs simulations for user-specified temperature profiles of the heat source and heat sink. The simulated system includes the compressor, evaporator, condenser, isenthalpic expansion device, and llsl-hx. The evaporator and condenser are represented by their *UA* values, which is a product of the overall heat-transfer coefficient (*U*) and heat-transfer area (*A*). The llsl-hx is represented by its effectiveness. The user must specify either cross-flow, counterflow, or parallel-flow configuration for the evaporator and condenser, with refrigerant superheat and subcooling, where appropriate. The model employs FORTRAN subroutines from the beta version of REFPROP 7.0 to calculate refrigerant thermophysical properties.

CYCLE-11.UA-CO_2 (Brown and Domanski 2000; Brown et al. 2002) is an adaptation of CYCLE-11.UA to the carbon dioxide transcritical cycle. The model uses Span and Wagner (1996) routines to calculate thermodynamic properties, and it calculates thermal conductivity and viscosity based on Vesovic et al. (1990). Because of transcritical operation, the input to the model includes the gas cooler pressure. Alternatively, at the user's option, CYCLE-11.UA-CO_2 can optimize the gas cooler pressure to maximize the COP.

Both models employ the same convergence logic, which establishes a thermodynamic loop with respect to the heat sink and source temperatures while satisfying constraints imposed by the input data. For the evaporator, condenser, and gas cooler, the models aim to obtain agreement between the ΔT calculated using refrigerant and external heat-transfer fluid (HTF) temperatures and the ΔT calculated from the following basic heat transfer relation:

$$\dot{Q}_{hx} = \dot{m}_r \Delta i_r = UA\Delta T_{hx} \qquad (1)$$

The ΔT calculated using refrigerant and HTF temperatures is a harmonic mean of the ΔTs for the individual sections (e.g., two-phase and superheated refrigerant) weighted with fractions of heats transferred in these sections. Nonlinearities in the refrigerant temperature profiles are accounted for by dividing individual sections into smaller segments, as necessary (Domanski and McLinden 1992). Validation of CYCLE-11.UA and CYCLE-11.UA-CO_2 are presented in Domanski et al. (1994b) and Brown and Domanski (2000), respectively.

Compressor

We adopted empirical correlations for volumetric and isentropic efficiencies (Equations 2 and 3, respectively) from Brown et al. (2002).

$$\eta_v = 0.8263\left[1 - 0.09604\left(\theta^{\frac{1}{\gamma}} - 1\right)\right] \qquad (2)$$

$$\eta_s = 0.9343 - 0.044780\theta \qquad (3)$$

These expressions do not necessarily model the performance of any given compressor, but rather intend to capture the claim by several researchers (e.g., Lorentzen and Pettersen 1993; Pettersen 1994) who postulate that a lower pressure ratio inherently results in higher compressor efficiency. Following this postulate, the CO_2 compressor should have a higher efficiency than the R-22 compressor since the pressure ratio θ for a CO_2 system is on the order of 2.0 to 2.5 whereas for an R-22 system it is on the order of 2.3 to 3.1, where the ranges are due to different operating conditions. The opposing view is that the higher efficiencies measured for CO_2 compressors compared to the current-production R-22 compressors are rather a result of compressor design itself. Given this controversy, we decided to use Equations 2 and 3 to be consistent with experimental studies in which CO_2 compressors were found to be more efficient than R-22 compressors and to give CO_2 a possibly valid credit.

Refrigerant-to-Air Heat Exchangers (Evaporator, Condenser, Gas Cooler)

All three refrigerant-to-air heat exchangers had a pure cross-flow configuration. We accounted for the impact of thermophysical properties on the refrigerant heat-transfer coefficient and the pressure drop on a relative basis. The scheme started with a simulation at a reference-point operating condition for which we imposed *UA* values to obtain realistic temperature differences between the carbon dioxide and HTFs. Since the total resistance to heat transfer is composed of the resistance on the HTF side, resistance due to heat exchanger material, and resistance on the refrigerant side ($R_{hx} = R_{HTF} + R_{tube} + R_r$), the following equation holds:

$$R_{HTF} + R_{tube} = \frac{1}{UA_{hx}} - \frac{1}{h_r A_r} \qquad (4)$$

where $1/UA_{hx} = R_{hx}$ and $1/h_r A_r = R_r$. The left-hand side of the equation, $R_{HTF} + R_{tube}$, varies insignificantly with operating conditions and is not affected by the refrigerant used. We can calculate this value for the reference-point operating condition with imposed UA_{hx} and A_r and calculated value of refrigerant heat-transfer coefficient, h_r. Then, we can use this value for other operating conditions or other refrigerants to simulate UA_{hx} according to Equation 5:

$$UA_{hx} = \frac{1}{R_{HTF} + R_{tube} + \frac{1}{h_r A_r}} \quad (5)$$

We used this procedure starting with a simulation for the carbon dioxide system at a selected reference-point operating condition and applied Equation 5 to calculate *UA* values for other simulations for CO_2 and all simulations for the R-22 system. This representation of the heat exchangers ensured that the heat exchangers in the CO_2 and R-22 systems had the same air-side heat-transfer resistances, which is typically the dominant heat transfer resistance in a refrigerant-to-air heat exchanger. At the same time, this method accounted for differences in refrigerant-side heat-transfer coefficients between CO_2 and R-22 for different operating conditions.

CYCLE-11.UA and CYCLE-11.UA-CO_2 employ in-tube flow correlations to calculate refrigerant-side heat-transfer coefficients. Both models used the Kandlikar (1990) correlation to predict the evaporative heat transfer coefficient. Kandlikar developed this model for traditionally sized tubes and refrigerants. However, Pettersen et al. (2000) showed that the Kandlikar (1990) correlation provides the best prediction capability for subcritical evaporation of CO_2 in microchannel tubes and recommended using a fluid-dependent coefficient of 1.0 due to the similarities between the thermophysical properties of water and CO_2.

Cavallini et al. (2000) showed that the correlation of Shah (1979) provides as good or better predictions for the condensation heat transfer coefficient than any of the other common models. Eckels et al. (1998) also reported that the Shah correlation shows good agreement with experimental data for various fluids, including R-22. For these reasons, CYCLE-11.UA uses the correlation of Shah (1979) to calculate the condensation heat transfer coefficient. For calculating the gas cooler heat-transfer coefficient, we followed the suggestion of Pitla et al. (1998) and provided CYCLE-11.UA-CO_2 with the correlation of Krasnoshchekov et al. (1969). (See Brown et al. [2002] for the detailed equations.) For simulation purposes, the gas cooler, condenser, and evaporator are divided into 30 equal-temperature interval segments for which local heat-transfer coefficients are calculated. The mean of these values for a given heat exchanger is used as an average heat-transfer coefficient.

Pressure drop in the heat exchangers is evaluated on a relative basis using a scheme similar to the one for calculating the *UA* values (Domanski at al. 1994a). Since the gravitational and momentum components are small, the model calculates only the frictional pressure drop. To predict two-phase pressure drop (in-tube evaporation of CO_2, R-22, and condensation of R-22), the model uses the frictional term of the modified Pierre correlation (Choi et al. 1999). For the transcritical CO_2 cooling process, the model calculates the frictional pressure drop using the conventional Blausius correlation for the single-phase friction factor for turbulent in-tube flow, $f = 0.184\ Re^{-0.2}$, as given in Kakaç and Liu (1998). CYCLE-11.UA-CO_2 calculates the pressure drop for each of the 30 gas cooler segments. The total pressure drop is the sum of these individual pressure drops.

Liquid-Line/Suction-Line Heat Exchanger

The llsl-hx is of a counterflow configuration and employs a user-specified effectiveness to model its thermal performance. The pressure drops on the high-pressure and low-pressure sides of the llsl-hx are simulated in the same manner as for the gas cooler.

Expansion Device

Both cycle models use a variable opening expansion device, which realizes the isenthalpic expansion process. The R-22 simulations approximated the use of a TXV by holding constant the evaporator superheat and condenser subcooling at 5°C (9°F). For the CO_2 transcritical cycle, heat rejection is above the critical point in most cases. Thus, subcooling at the gas cooler outlet cannot be specified, and different transcritical cycles with different COPs can be established depending on the gas cooler pressure. Optimizing this pressure for maximum COP was discussed by Pettersen (1994), among others. McEnaney et al. (1999) further pointed out the practical high-side limitation for the gas cooler pressure due to temperature limitations of the compressor materials, which in their case was 140°C (284°F). Considering these issues, CYCLE-11.UA-CO_2 includes the following options for the gas cooler pressure: (1) imposed, (2) optimized for maximum COP, and (3) optimized for maximum COP overridden by the 140°C (284°F) compressor discharge temperature limit.

SIMULATED SYSTEMS AND CONDITIONS

The CO_2 system consisted of a compressor, gas cooler, llsl-hx, variable opening expansion device, and evaporator. The components and system were selected to match the system studied experimentally by Beaver et al. (1999b). The compressor had the same displacement volume. The gas cooler and evaporator had the same physical characteristics, e.g., refrigerant passage diameters, number of circuits, and air-side surface. The baseline CO_2 system had a nominal cooling capacity of 10.56 kW at a gas cooler air inlet temperature of 35°C (95°F) and an evaporator air inlet temperature of 26.7 °C (80°F) with a corresponding COP of 3.3 (EER of 11.25). The effectiveness of the llsl-hx was 0.24.

The R-22 system consisted of a compressor, condenser, variable opening expansion device, and evaporator, but it did not include an llsl-hx. Since the compressors were single speed, the R-22 compressor had an increased displacement volume to obtain the same cooling capacities as the CO_2 system at an ambient temperature of 35°C (95°F). Considering the air-side, the condenser was the same heat exchanger as the CO_2 gas cooler, and the evaporator was the same for the two refrigerants. On the refrigerant side, we redesigned the refrigerant passages to obtain reasonable refrigerant pressure drops for the R-22 heat exchangers. In redesigning the refrigerant

TABLE 1
Relevant Parameters for the Various Heat Exchangers

	N	n	L (m)	P_t (mm)	D_H (mm)	A_{HTF} (m^2)	A_r (m^2)	V_{hx} (m^3)
CO_2 evaporator	41	11	0.813	2.48	0.79	22.5	2.73	0.018
R-22 evaporator	41	6	0.813	5.74	1.45	22.5	3.45	0.018
CO_2 gas cooler	80	11	0.626	2.48	0.79	50.4	4.10	0.026
R-22 condenser	80	6	0.626	5.74	1.45	50.4	5.17	0.026

TABLE 2
Pressure Drop and Heat Transfer Performances at an Ambient Temperature of 35°C (95°F) for the Various Heat Exchangers

	ΔP (kPa)	$T_{sat,\ drop}$ (°C)	h (W m^{-2} K^{-1})	R_{HTF}/R_{tot} (%)
CO_2 evaporator	54.0	0.49	2518	84
R-22 evaporator	18.9	0.92	1087	72
CO_2 gas cooler	115.0	–	2476	83
R-22 condenser	49.8	1.09	2463	84

passages, we increased the cross-sectional areas of the individual passages but did not alter the cross-sectional profile of the flattened tubes. Figure 1 shows cross-sectional views of the CO_2 and R-22 refrigerant tubes, Table 1 lists the relevant geometrical parameters for the various heat exchangers, and Table 2 shows the corresponding pressure drop and heat transfer performances at an ambient temperature of 35°C (95°F). Since the external profile of the flattened microchannel tube was not changed, the air-side resistances were the same for the different systems; however, an increase in the refrigerant flow area caused a reduction in the R-22 heat transfer coefficients due to lowered mass fluxes. We may note that the bulk of resistance to heat transfer occurred on the air-side, and the effect of different refrigerant-side heat transfer coefficients was of secondary importance.

Three air temperatures at the condenser/gas cooler inlet, 27.8°C, 35°C, and 40.6°C (82°F, 95°F, and 105°F), and one air temperature at the evaporator inlet (26.7°C [80°F]) comprised the operating conditions. In addition to the baseline systems, we simulated systems in which both refrigerant-to-air heat exchangers had larger *UA* values relative to their baseline values. Results from these simulations estimate possible energy efficiency gains that can be achieved via heat exchanger improvements. As we increased the *UA* values, we adjusted the compressor swept volumes to maintain the 35°C (95°F) reference point cooling capacity.

SIMULATION RESULTS AND DISCUSSION

Figure 2 shows the CO_2 transcritical cycle and the R-22 subcritical cycle at an ambient temperature of 35.0°C (95.0°F). This condition represents the baseline simulation for each system. The significantly higher discharge temperature

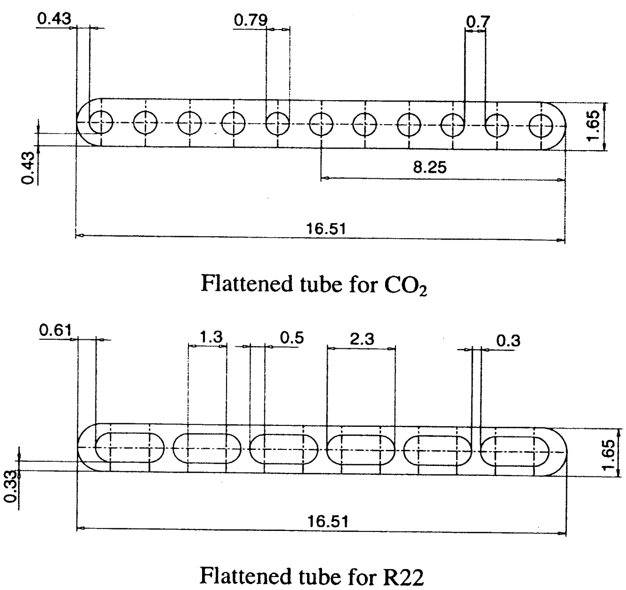

Figure 1 *Cross-sectional view of the heat exchangers (note: all dimensions in millimeter).*

and larger temperature change for CO_2 in the high-pressure heat exchanger are the most visible differences. The large CO_2 glide is the reason for significant temperature mismatch since the refrigerant-to-air heat exchangers are cross-flow. The figure also shows that the temperature approach for the CO_2 gas cooler is lower than the one in the R-22 condenser, which benefits the COP of carbon dioxide, a fact pointed out by some CO_2 researchers (e.g., Petterson 1999). However, even qualitative visual examination suggests that the benefit of a lower temperature approach cannot compensate for the penalty due to the air/CO_2 temperature mismatch.

Figure 3 shows COPs as a function of ambient temperature and heat exchanger *UA* values. At an ambient condition of 27.8°C (82.0°F) for the baseline *UA* values, the R-22 system COP is approximately 41% higher than the CO_2 system COP. The COP gap reaches 52% at 35.0°C (95.0°F) and 57% at 40.6°C (105.0°F). With increasing values of *UA*, COPs for both refrigerants increase and tend to approach maximum levels for given operating temperatures. The figure shows that the COP gaps between R-22 and CO_2 increase with larger heat exchangers, indicating that the two refrigerants have different COP limits in residential applications.

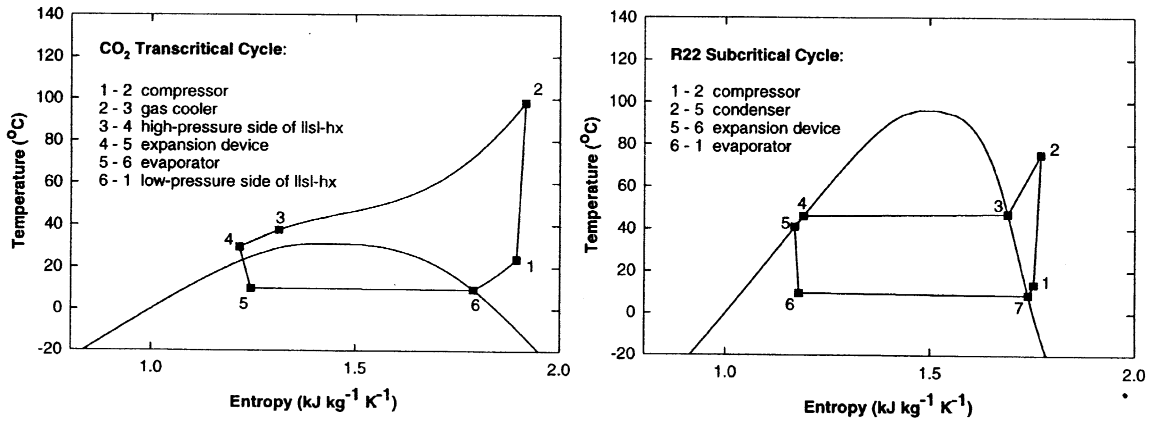

Figure 2 *CO_2 transcritical cycle and R-22 subcritical cycle at an ambient temperature of 35°C (95°F) with baseline UA values.*

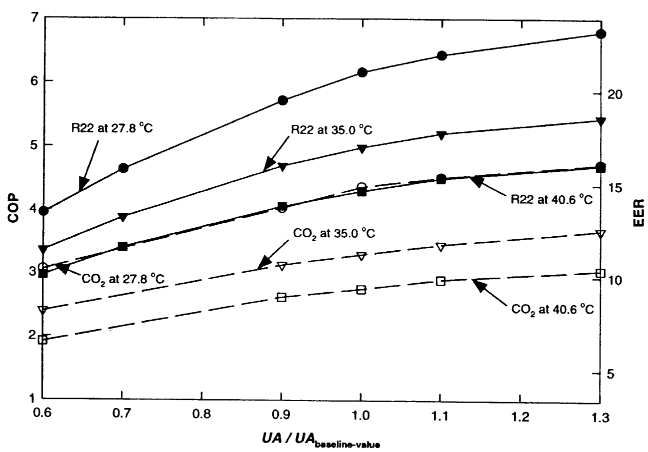

Figure 3 *COP and EER comparison between CO_2 and R-22 systems. (COPs and EERs include capacity and compressor power only.)*

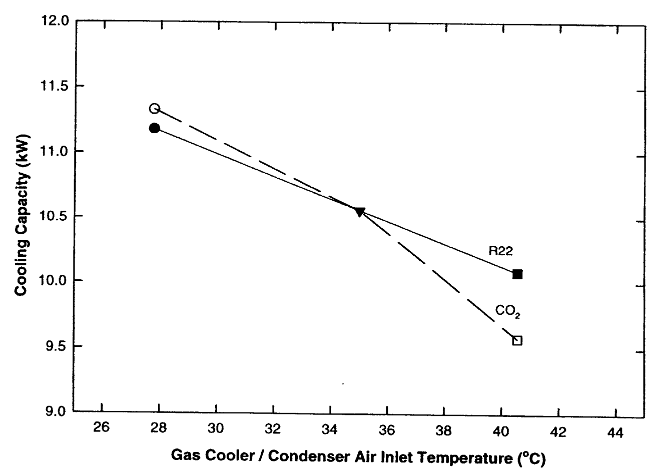

Figure 4 *Cooling capacities for CO_2 and R-22 systems with the baseline heat exchangers.*

The COP parity between R-22 and CO_2 can be obtained when the R-22 system employs heat exchangers having a significantly lower *UA* value that the CO_2 system. For example, at an ambient temperature of 27.8°C (82°F), the CO_2 COP for the baseline *UA* is matched by the R-22 system equipped with the heat exchangers having 65% of the baseline *UA*. At an ambient temperature of 35°C (95°F), the COP of the CO_2 system drops approximately 10% below that of the R-22 system. When we consider the indication by Kim and Bullard (2001) that an R-22 finned-tube condenser can be substituted in the system by a microchannel condenser having a 55% lower core volume, these simulation results explain the reason for equivalent efficiencies for CO_2 (microchannel) and R-22 or R-410A (finned tube) systems reported from some experimental studies. We may note that similarity of COPs for R-22 and R-410A in equivalent equipment has been reported by several authors and also obtained in R-410A simulations performed for this study at selective conditions. The R-410A simulation points were very close to those obtained for R-22.

Figure 4 shows cooling capacities of CO_2 and R-22 as a function of ambient temperature. The cooling capacities are identical at an ambient temperature of 35°C (95°F) due to an imposed capacity match. R-22 capacity decreases linearly with increasing ambient temperature. The CO_2 capacity decrease is steeper than that of R-22 and is particularly pronounced at high ambient temperatures. Figure 5 shows compressor power as a function of ambient temperature and heat exchanger *UA* values. At an ambient temperature of 27.8°C (82°F) for the baseline *UA* values, the CO_2 system's compressor power is approximately 38% greater than for the R-22 system.

We can obtain some interesting insight into performance of the CO_2 and R-22 systems and its components by reviewing the entropy generation compared in Figure 6. Entropy gener-

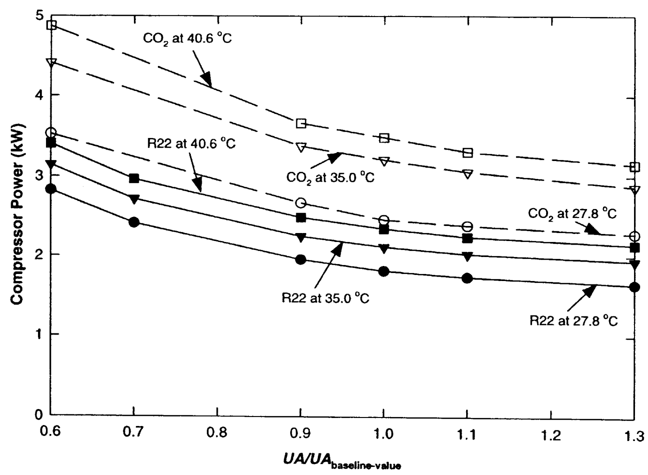

Figure 5 *Compressor power consumption comparison between CO_2 and R-22 systems.*

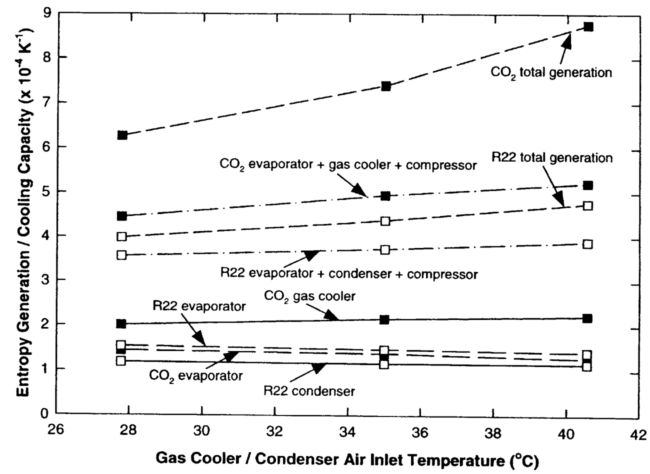

Figure 6 *Comparison of entropy generation per unit cooling capacity for CO_2 and R-22 systems.*

ation in a process or a cycle is a measure of its irreversibility. Entropy generation by the system is a summation of entropy generations by all system components and is reflective of the system's efficiency. Consequently, higher total entropy generation corresponds to a lower COP, and vice versa. For example, the correspondence between COP and entropy generation for two systems working with zeotropic mixtures is presented in Domanski et al. (1994b). In calculating the entropy generation data shown in Figure 6 we followed the method presented in this publication. The figure uses an entropy-per-capacity ordinate in recognition of the fact that an absolute scale would result in different entropy generations for two systems of the same efficiency if their capacities were different. The amount of entropy generated in the CO_2 evaporator is somewhat smaller than that in the R-22 evaporator. This can be attributed to the better transport properties of CO_2, which resulted in a lower temperature difference between the air and CO_2. On the other hand, the entropy generation in the CO_2 gas cooler is considerably greater than in the R-22 condenser. The large CO_2 temperature glide (approximately 61°C [109.8°F] for CO_2 versus 34°C for R-22 at an ambient temperature of 35.0°C [95°F] and for the baseline *UA* values) and the larger amount of heat needed to be rejected by the CO_2 gas cooler than by the R-22 condenser (6% higher at an ambient temperature of 35°C [95°F] for the baseline *UA* values) cause a significant amount of entropy generation (heat-transfer irreversibilities) in the cross-flow heat exchanger. However, the largest difference is for the processes that involve the expansion device for R-22 and the expansion device and the llsl-hx for CO_2. These entropy generations are represented in Figure 6 by the differences between the total entropy generation lines and the lines combining the compressor, evaporator, and condenser or gas cooler entropy generations. For CO_2, the profound share of entropy generation is in the expansion device; without the llsl-hx the entropy generation during the expansion process would be even greater. Hence, the large

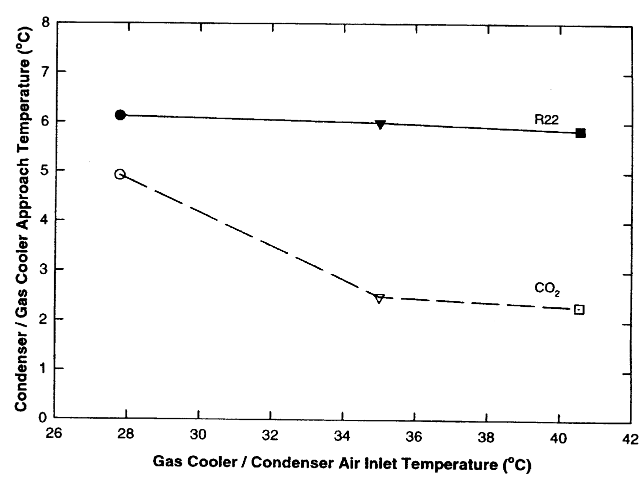

Figure 7 *Condenser/gas cooler approach temperature comparison between CO_2 and R-22 systems.*

entropy generations in the gas cooler and expansion device are responsible for the higher total entropy generation for CO_2. Although not explicitly shown in Figure 6, we note that the entropy generation in the CO_2 compressor is approximately 34% greater than in the R-22 compressor despite the fact that the CO_2 compressor has a higher isentropic efficiency due to the lower compressor pressure ratio. This is possible because the entropy analysis included the effect of the higher energy input required by CO_2 and the effect of different R-22 and CO_2 mass flow rates required to provide the same cooling capacity. Hence, isentropic efficiency does not provide complete information on the influence of the compression process on system performance when different refrigerants are compared.

Entropy generation information indicates the processes in the system that need to be improved to obtain better COP. The results attest that the CO_2 system would benefit from a work recovery device that could minimize throttling losses more effectively than the llsl-hx. Regarding the CO_2 gas cooler, large entropy generation is unavoidable to significant degree because of high CO_2 temperature glide and cross-flow heat transfer, since implementation of a counterflow CO_2-air heat exchanger may prove to be impractical, just as it proved to be impractical for zeotropic mixtures, for comfort cooling applications. However, CO_2 may provide a comfort advantage in the heating mode, and its high temperature glide can be exploited for water-heating applications.

SIMULATION TRADE-OFFS

It is proper to note the features in the simulation models that were introduced to account for peculiarities of the transcritical CO_2 cycle and to point out the limitations in the simulations that might benefit one fluid over the other. We provided a benefit to the CO_2 compressor for its lower compression ratio. Because of the CO_2 system's lower compressor pressure ratio, the CO_2 compressor has a higher isentropic efficiency as given by Equation 2. In this way, CO_2 receives the credit postulated by some researchers. For the baseline *UA* values at ambient temperatures of 27.8°C, 35°C, and 40.6°C (82°F, 95°F, and 105°F), the CO_2 compressor's isentropic efficiencies are 84.5%, 82.9%, and 82.5%, respectively. For the same conditions, the R-22 compressor's isentropic efficiencies are 82.5%, 80.9%, and 79.7%, respectively.

The simulations agree with Pettersen's observations (Pettersen 1999) that a CO_2 gas cooler can obtain a closer approach temperature than an R-134a automotive condenser for equal size heat exchangers. We would expect the same to hold true for the R-22 residential system's condenser as well, as shown in Figure 7. The approach temperature in the CO_2 gas cooler is 2.3°C to 4.9°C (4.1°F to 8.8°F), while for the R-22 condenser the range is 5.8 °C to 6.1 °C (10.4°F – 11°F). The closer approach temperature of CO_2, however, did not overcome the thermodynamic penalty associated with the large temperature glide resulting in high entropy generation, as previously discussed. In addition, we would also expect additional heat transfer irreversibilities in the CO_2 gas cooler as compared to the R-22 condenser due to longitudinal conduction effects, which we did not include in the simulations. The aluminum in the CO_2 gas cooler experiences large longitudinal temperature gradients because of the large CO_2 glide.

One drawback to CYCLE-11.UA-CO_2 is that it is not a full simulation model that can completely capture local effects in the heat exchangers. For example, as mentioned previously, the gas cooler is divided into 30 equal-temperature interval segments for which the local CO_2 heat-transfer coefficient is calculated. The mean of these values is used as an average CO_2 heat-transfer coefficient in the gas cooler. Thus, one concern might be that we do not properly capture the effects of increased heat transfer coefficients near the critical point due to the sharp increases in transport properties near the critical temperature. This, however, should be of secondary importance since the bulk (approximately 80%) of the resistance to heat transfer occurs on the airside. In addition, the rapid changes in transport properties occur in a relatively tight band of temperatures near the critical point, implying that only a small fraction of the gas cooler will experience refrigerant temperatures near this range. This is due to the large refrigerant temperature glide experienced in the gas cooler and also to the fact that we are investigating systems operating at moderate to high ambient temperatures (27.8°C, 35°C, and 40.6°C [82°F, 95°F, and 105°F]).

CYCLE-11 simulations do not consider refrigerant maldistribution in the microchannel heat exchangers that may occur due to the nonhomogeneity of the entering two-phase refrigerant. This simplification benefits R-22 over CO_2 because two-phase refrigerant maldistribution is driven by a difference in densities between the liquid and vapor, which is lower for CO_2 than for R-22. In the evaporator, for example, for a saturation temperature of 5°C (41°F), the density ratios for R-22 and CO_2 are 51.0 and 7.9, respectively. In the condenser, for a saturation temperature of 45°C (113°F), the R-22 density ratio is 14.7. In the gas cooler we expect CO_2 to be homogeneous. Ongoing research is investigating methods to minimize refrigerant maldistribution in microchannel heat exchangers. At this point, it is difficult to estimate the impact of not considering refrigerant maldistribution in the simulations.

For simplicity of presentation, when simulating systems with different heat exchanger *UA* values, we applied the same *UA* increments to the evaporators, R-22 condenser, and CO_2 gas cooler. An argument could be made that, to maintain humidity control, future COP improvements will be mainly sought by increasing the condenser and gas cooler *UA* values, while changes in evaporator *UA* values will be minimal. Considering that the CO_2 evaporator always had a higher saturation temperature than the R-22 evaporator, somewhat larger increases in the R-22 evaporator *UA* values could be implemented than for the CO_2 evaporator.

CONCLUDING REMARKS

We compared performance of CO_2 and R-22 in residential air conditioners using vapor compression and transcritical cycle simulation models. We considered a current-production configuration for the R-22 system (compressor, condenser, expansion device, and evaporator) and a CO_2 system, which was additionally equipped with the llsl-hx. In this analysis, the R-22 evaporator and condenser were derived from the CO_2 refrigerant-to-air heat exchangers by modifying the refrigerant circuits to obtain reasonable refrigerant pressure drops. We accounted for the better transport properties of CO_2 and its higher compressor isentropic efficiency due to a lower compression ratio. Also, the simulations credited CO_2 by predicting a lower approach temperature in the gas cooler as compared to the R-22 condenser.

The analysis shows CO_2 having significantly inferior COP to R-22 when these fluids are evaluated in systems equipped with comparable microchannel heat exchangers. The COP disparity depends on ambient temperature and heat exchanger *UA* values. For the baseline *UA* values, the COP for CO_2 ranged from being lower by 42% at 27.8°C (82°F) to 57% at 40.6°C (105°F). Hence, better transport properties and better compressor isentropic efficiency of CO_2 did not compensate for its thermodynamic disadvantage compared to R-22, even if an llsl-hx is used in the CO_2 cycle to reduce throttling irreversibilities. Considering that R-410A (leading R-22 alternative) and R-22 attain similar COPs in equivalent equipment, CO_2's performance can be viewed as being similarly lower when compared to R-410A.

The entropy generation calculations indicated that CO_2 has somewhat better performance than R-22 in the evaporator, but it has poorer performance in the gas cooler than R-22 in the condenser. The large CO_2 temperature glide and the larger amount of heat needed to be rejected by the CO_2 gas cooler are responsible for the higher entropy generation. The CO_2 compressor also generated more entropy than its R-22 counterpart. The highest level of irreversibilities, however, were realized in the CO_2 expansion device, and, together with the irreversibilities in the gas cooler, were chiefly responsible for the low COP. The CO_2 system would benefit from an efficient work recovery device. While being inherently disadvantaged in comfort cooling applications due to cross-flow heat exchange in the gas cooler, the CO_2 transcritical cycle lends itself to water-heating applications with counterflow heat exchange.

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NOMENCLATURE

A = area (m^2)
COP = coefficient of performance
c_p = specific heat at constant pressure (kJ kg^{-1} K^{-1})
c_v = specific heat at constant volume (kJ kg^{-1} K^{-1})
D_H = hydraulic diameter of refrigerant passage (mm)
EER = energy efficiency ratio (kBtu h^{-1} kW^{-1})
h = mean heat transfer coefficient (kW K^{-1} m^{-2})
i = specific enthalpy (kJ kg^{-1})
llsl-hx = liquid-line/suction-line heat exchanger
L = heat exchanger length (m)
$\dot{m}$ = mass flow rate (kg s^{-1})
n = number of refrigerant passages per flattened tube
N = number of parallel flattened tubes
P = pressure (kPa)
P_t = perimeter of refrigerant passage (mm)
$\dot{Q}$ = heat transfer rate (kW)
R = resistance to heat transfer (K kW^{-1})
RPM = revolutions per minute
Re = Reynolds number
SEER = seasonal energy efficiency ratio (kBtu h^{-1} kW^{-1})
s = specific entropy (kJ kg^{-1} K^{-1})
T = temperature (°C)
U = overall heat-transfer coefficient (m^2 K kW^{-1})
V = volume (m^3)

Greek Letters

γ = specific heat ratio ($c_{p/}c_v$)
η = efficiency
θ = pressure ratio

Subscripts and Superscripts

HTF = external heat-transfer fluid
hx = heat exchanger
r = refrigerant
s = isentropic
sat, drop= drop in saturation temperature between inlet and outlet
V = volumetric

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