FUNDAMENTAL ASPECTS OF THE APPLICATION OF CARBON DIOXIDE IN COMFORT COOLING

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ABSTRACT

This paper presents entropy generation analyses for the evaporator, compressor, gas cooler, expansion device and liquid-line suction-line heat exchanger for a transcritical carbon dioxide cycle for automotive and residential air-conditioning systems, and presents entropy generation analyses for the evaporator, compressor, condenser, and expansion device for a subcritical R134a automotive air-conditioning system and a subcritical R22 residential air-conditioning system. The analyses show that the CO_2 automotive air-conditioning system generates 36 % more entropy than the R134a automotive air-conditioning system, and that the CO_2 residential air-conditioning system generates 63 % more entropy than the R22 residential air-conditioning system. The biggest contributors to the lower CO_2 performance are irreversibilities associated with the heat rejection process and the expansion related processes.

INTRODUCTION

Over the last two decades, the refrigeration and air-conditioning industry has been undergoing major changes due to an increasing awareness of the impact that the use of refrigeration and air-conditioning equipment can have on the environment. The major environmental concerns have been ozone depletion and global warming. The Montreal Protocol and its amendments have largely settled the ozone depletion problem. To date, most practitioners have chosen HFC refrigerants as replacements for CFC and HCFC refrigerants (Dupont, 2001). However, the continued use of HFC refrigerants is questioned because of their global warming potentials.

With this increasing scrutiny on the use of HFC refrigerants, many researchers are investigating so-called natural refrigerants, e.g., hydrocarbons, air, water, ammonia, and carbon dioxide, with carbon dioxide receiving substantial focus beginning in the early 1990's (e.g., Lorentzen and Pettersen, 1992). Since that time, several research groups have studied carbon dioxide in various applications, including, mobile air conditioning, heat pumping, residential and commercial air conditioning, and commercial refrigeration.

Given the intense interest in the transcritical CO_2 cycle for air-conditioning applications, this paper attempts to contribute to the understanding of how the transcritical CO_2 cycle compares to typical subcritical halocarbon cycles. To this end, we use an entropy generation methodology to compare the amounts of entropy generation in each of the primary system components, and to demonstrate the possibilities for improvements to the transcritical CO_2 cycle.

1 SYSTEM ENTROPY GENERATION

The purpose of an air-conditioning system is to transfer heat from a low-temperature source to a high-temperature sink while using the least amount of work, i.e. to maximize the Coefficient of Performance (COP) for a given cooling capacity at given source and sink temperatures. We can restate this goal in terms of entropy using the Gouy-Stodola Theorem. That is, the purpose of an air-conditioning system is to transfer entropy from a low-temperature source to a high-temperature sink while generating the least amount of entropy, or stated in another way, the goal is to generate the least amount of entropy for a given cooling capacity.

To begin the analysis, consider the generic air-conditioning system shown in Fig. 1 operating in a steady-state cycle. The energy balance is:

$$\dot{W} = \dot{Q}_H - \dot{Q}_L \tag{1}$$

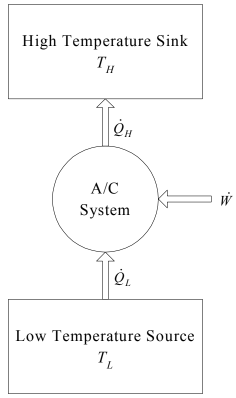

Figure 1: Generic air-conditioning system

and the entropy balance is:

$$0 = -\frac{\dot{Q}_H}{T_H} + \frac{\dot{Q}_L}{T_L} + \dot{S}_{gen} \tag{2}$$

where $\dot{S}_{gen}$ denotes the entropy generation. $\dot{S}_{gen}$ is equal to zero if the system operates as a Carnot cycle, and $\dot{S}_{gen}$ is greater than zero if the system deviates from the Carnot behavior. The more irreversible the cycle (i.e., the larger the entropy generation), the further the system will be from the ideal Carnot behavior.

Combining the energy and entropy balances yields:

$$0 = -\dot{W} + \dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) + T_H\dot{S}_{gen} \tag{3}$$

If the system operates as a Carnot cycle, then $\dot{S}_{gen} = 0$ and eq. (3) reduces to an expression for the Carnot cycle work, $\dot{W}_{Carnot}$:

$$\dot{W}_{Carnot} = \dot{Q}_L\left(\frac{T_H}{T_L} - 1\right) \tag{4}$$

Substituting eq. (4) into eq. (3) yields:

$$\dot{W} = \dot{W}_{Carnot} + T_H\dot{S}_{gen} \tag{5}$$

If we use the Gouy-Stodola Theorem $\left(\dot{W}_{lost} = T_H\dot{S}_{gen}\right)$, we can rewrite eq. (5) as:

$$\dot{W} = \dot{W}_{Carnot} + \dot{W}_{lost} \tag{6}$$

Equations (5) and (6) demonstrate that the actual work is simply the sum of the Carnot work and the lost work potential, $\dot{W}_{lost}$, due to entropy generation. In order to minimize the actual work (approach the Carnot work), one must minimize the entropy generation. Note that we derived eqs. (5) and (6) without considering any details of the system. However, in practice one must design a particular machine so we can rewrite eqs. (5) and (6) as:

$$\dot{W} = \dot{W}_{Carnot} + T_H \sum_{components}\dot{S}_{gen} = \dot{W}_{Carnot} + \sum_{components}\dot{W}_{lost} \tag{7}$$

Since our stated goal for the air-conditioning system is to maximize COP for a fixed cooling capacity, we can rewrite eq. (7) in terms of COP by dividing through by Q_L and rearranging to obtain:

$$COP = \frac{COP_{Carnot}}{1 + \frac{COP_{Carnot} T_H \sum\limits_{components} \dot{S}_{gen}}{\dot{Q}_L}} = \frac{COP_{Carnot}}{1 + \frac{COP_{Carnot} \sum\limits_{components} \dot{W}_{lost}}{\dot{Q}_L}} \tag{8}$$

The advantage of writing the COP in terms of entropy generation (or lost work) is that for fixed application temperatures (T_L and T_H) and fixed cooling capacity, we recognize that we must minimize $\sum\limits_{components} \dot{S}_{gen}$ (or $\sum\limits_{components} \dot{W}_{lost}$) in order to maximize COP. Thus, if we consider and compare different refrigerants we may compare the entropy generation (or lost work) for the component(s) that accomplishes each of the four basic tasks of a vapor compression refrigeration cycle, i.e., compression, heat rejection, expansion, and heat addition.

To examine the thermodynamic merits of the CO_2 transcritical cycle operating at moderate to high ambient temperatures, we present two examples: automotive air-conditioning systems operating with CO_2 and R134a, and residential air-conditioning systems operating with CO_2 and R22. Brown *et al*. (2002a) and Brown *et al*. (2002b) describe the automotive air-conditioning systems and residential air-conditioning systems, respectively, in detail and provide simulation results for the various systems. In both cases they simulated the various systems on an equivalent basis. They maintained equal cooling capacities between the CO_2 and halocarbon systems, and for the halocarbon systems employed modified versions of the newer generation micro-channel heat exchangers used in the CO_2 systems. Here, we briefly describe their systems and then use some of their simulation results in our analyses.

2 SIMULATED SYSTEMS

2.1 CO_2 Automotive Air-Conditioning System

Brown *et al*. (2002a) simulated two automotive systems: a CO_2 air conditioner and an R134a air conditioner. The CO_2 system consisted of a compressor, gas cooler, liquid line/suction line heat exchanger (llsl-hx), variable opening expansion device, and evaporator. The components and system were selected to match the system studied experimentally by McEnaney *et al*. (1999). 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 CO_2 system had a nominal cooling capacity of 3.32 kW at a gas cooler inlet temperature of 43.3 °C, an evaporator air inlet temperature of 26.7 °C, and a compressor speed of 1000 RPM.

2.2 CO_2 Residential Air-Conditioning System

Brown *et al*. (2002b) simulated two residential air-conditioning systems: a CO_2 air conditioner and an R22 air conditioner. 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*. (1999). 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 35.0 °C, an evaporator air inlet temperature of 26.7 °C, and a compressor speed of 1319 RPM.

2.3 Conventional R134a and R22 Systems

The R134a automotive system and R22 residential system were based on two respective CO_2 air conditioners. Both halocarbon systems consisted of a compressor, condenser, variable opening expansion device, and evaporator, but did not include a llsl-hx since commercially available systems do not normally include a one. The air-side of the R134a and R22 condensers and evaporators were the same as their CO_2 counterparts. On the refrigerant side, the refrigerant passages were redesigned to obtain typical refrigerant pressure drops for R134a and R22. In both cases, rearranging the circuitry reduced the pressure drop penalty at the expense of a lower refrigerant-side heat-transfer coefficient. These changes led to somewhat larger refrigerant side areas for the heat exchangers. For example, for the R22 system the refrigerant side areas for the evaporator and condenser/gas cooler were 26 % greater than those of the CO_2 system. The R134a compressor had an increased displacement volume to obtain the same cooling capacity as the CO_2 system at 1000 RPM and an ambient temperature of 43.3 °C. The R22 compressor had an increased displacement volume to obtain the same cooling capacity as the CO_2 system at 1319 RPM and an ambient temperature of 35.0 °C.

In what follows, we present analyses for each primary component for CO_2 and R134a operating in automotive air-conditioning applications, and for CO_2 and R22 operating in residential air-conditioning applications.

3 EVAPORATOR ENTROPY GENERATION

For a refrigerant-to-air heat exchanger operating at steady-state, the entropy balance is:

$$\dot{S}_{gen} = \dot{m}_{air}\left(s_{out} - s_{in}\right)_{air} + \dot{m}_{ref}\left(s_{out} - s_{in}\right)_{ref} \tag{9}$$

where $\dot{m}_{air}$ and $\dot{m}_{ref}$ are the mass flow rates of air and refrigerant, respectively. In the analyses of Brown *et al*. (2002a) and Brown *et al*. (2002b), the compared systems had the same cooling capacity and operated with the same air mass flow rates and inlet air temperatures. Given these constraints, the entropy generation on the air-side is the same for the CO_2 and R134a evaporators for the automotive air conditioners, and for the CO_2 and R22 evaporators for the residential air conditioners. Therefore, we only need to compare the $\dot{m}_{ref}\left(s_{out} - s_{in}\right)_{ref}$ terms to understand which refrigerant will generate more entropy in the evaporator.

To this end, we can express the refrigerant entropy change in terms of other thermodynamic properties by considering the thermodynamic relation for a simple compressible substance:

$$T\mathrm{d}s = \mathrm{d}h - v\mathrm{d}P \tag{10}$$

where h and v are the specific enthalpy and volume, respectively, and P is the pressure. Therefore, the rate of entropy change for the refrigerant stream is simply:

$$\dot{m}_{ref}\left(s_{out} - s_{in}\right)_{ref} = \dot{m}_{ref}\int_{in}^{out}\frac{\mathrm{d}h}{T} - \dot{m}_{ref}\int_{in}^{out}\frac{v}{T}\mathrm{d}P \tag{11}$$

In both examples, the halocarbon refrigerants operated with slightly lower evaporation temperatures than did CO_2. Thus, the first term on the right-hand side of eq. (11) is greater for the halocarbon refrigerants since the cooling capacity is the same for both refrigerants in each application. The second term is also larger for the halocarbon refrigerants since they have considerably greater refrigerant volumetric flow rates than does CO_2 (roughly eight times greater for the automotive example and roughly five times greater for the residential air-conditioning example). The saturation temperatures are not significantly different between CO_2 and the halocarbon refrigerant in each application. The pressure drops for CO_2 are larger than those for the halocarbon refrigerants, however the larger CO_2 pressure drops do not override the dominating influence of the greater halocarbon volumetric flow rates in eq. (11). Note: both terms on the right-hand side of eq. (11) are positive quantities. Therefore, the halocarbon refrigerants will generate greater amounts of entropy than CO_2 in the evaporators for these applications.

For the automotive air-conditioning example of Brown *et al*. (2002a), the entropy generation per unit cooling capacity for the CO_2 evaporator is $1.41 \cdot 10^{-4}$ K^{-1} versus $1.72 \cdot 10^{-4}$ K^{-1} for the R134a evaporator, and the corresponding value of lost work is 0.148 kW for the CO_2 evaporator and 0.180 kW for the R134a evaporator. Therefore, the R134a evaporator generates 22 % more entropy than does the CO_2 evaporator.

For the residential air-conditioning example of Brown *et al*. (2002b), the entropy generation per unit cooling capacity for the CO_2 evaporator is $1.38 \cdot 10^{-4}$ K^{-1} versus $1.47 \cdot 10^{-4}$ K^{-1} for the R22 evaporator, and the corresponding value of lost work is 0.448 kW for the CO_2 evaporator and 0.478 kW for the R22 evaporator. Therefore, the R22 evaporator generates 7 % more entropy than does the CO_2 evaporator. Hence, when operating at a fixed cooling capacity with fixed application temperatures, a CO_2 evaporator will generate somewhat less entropy than will an evaporator operating with a halocarbon refrigerant.

4 COMPRESSOR ENTROPY GENERATION

Several researchers postulate that the isentropic efficiency of a compressor is a function of its compression ratio with isentropic efficiency decreasing with increasing compression ratio. This implies that CO_2 compressors inherently have higher isentropic efficiencies than halocarbon compressors because they operate at a lower compression ratio. However, isentropic efficiency is not a complete measure of the compressor irreversibilities as employed in a system. In fact, even if the isentropic efficiency of a compressor operating with one refrigerant is greater than another compressor operating with a different refrigerant, it may still generate more entropy. To understand why this is possible consider an open, adiabatic compressor operating at steady-state. The entropy balance is:

$$\dot{S}_{gen} = \dot{m}_{ref}\left(s_{out} - s_{in}\right) \tag{12}$$

For an isentropic process the thermodynamic relation given in eq. (10) reduces to:

$$\left(\mathrm{d}h\right)_s = \left(v\mathrm{d}P\right)_s \tag{13}$$

We further recognize that the instantaneous isentropic efficiency is:

$$\eta_s = \frac{\left(\mathrm{d}h\right)_s}{\mathrm{d}h} \tag{14}$$

Combining eqs. (10), (13), and (14) with eq. (12) and then integrating, the entropy generation is:

$$\dot{S}_{gen} = \dot{m}_{ref}\left[\int_{in}^{out}\left(\frac{v}{\eta_s T}\mathrm{d}P\right)_s - \int_{in}^{out}\frac{v}{T}\mathrm{d}P\right] \tag{15}$$

One observation regarding eq. (15) is that as the isentropic efficiency numerically approaches 1.0 then the entropy generation decreases since the two integrals approach one another. However, it is also true that isentropic efficiency is only one variable in eq. (15) along with mass flow rate, temperature, pressure, and specific volume (i.e., refrigerant and cycle-related parameters). In fact, the combination of these variables, as expressed in eq. (15), determines the amount of entropy generation. Consequently, higher isentropic efficiency is not equivalent to lower entropy generation in the compressor when two different refrigerants are considered.

To illustrate this point, consider the examples of Brown *et al*. (2002a) and Brown *et al*. (2002b). For the automotive air-conditioning example, the isentropic efficiencies were 80.6 % for the CO_2 compressor and 72.3 % for the R134a compressor at an ambient temperature of 43.3 °C, based on the relation that Brown *et al.* (2002a) adopted for isentropic efficiency as a function of pressure ratio. The significantly better isentropic efficiency of the CO_2 compressor is because the CO_2 compressor operated with a compression ratio of 2.8 versus 4.7 for the R134a compressor. The entropy generation per unit cooling capacity for the CO_2 compressor is $2.24 \cdot 10^{-4}$ K^{-1} versus $2.82 \cdot 10^{-4}$ K^{-1} for the R134a compressor, and the corresponding value of lost work is 0.235 kW for the CO_2 compressor and 0.296 kW for the R134a compressor.

For the residential air-conditioning example, Brown *et al*. (2002b) give the isentropic efficiencies as 82.9 % for the CO_2 compressor and 80.9 % for the R22 compressor at an ambient temperature of 35 °C. Again, the slightly better isentropic efficiency of the CO_2 compressor is because the CO_2 compressor operated with a compression ratio of 2.4 versus 2.8 for the R22 compressor. Despite the better isentropic efficiency for the CO_2 compressor, its entropy generation per unit cooling capacity is $1.41 \cdot 10^{-4}$ K^{-1} versus $1.11 \cdot 10^{-4}$ K^{-1} for the R22 compressor, and the corresponding value of lost work is 0.459 kW for the CO_2 compressor and 0.361 kW for the R22 compressor.

The above examples show that, when comparing different refrigerants, better isentropic efficiency does not necessarily lead to lower entropy generation. In fact, for the automotive air-conditioning example the R134a compressor generates 26 % more entropy than does the CO_2 compressor; whereas, for the residential air-conditioning example the CO_2 compressor generates 27 % more entropy than does the R22 compressor.

A note should be made that the authors could not locate in the open literature any explanation for the reported better isentropic efficiency at low compression ratios, neither based on theory nor laboratory measurements in a well-controlled experiment. Since comparisons of isentropic efficiencies, by necessity, are always presented using different compressors, a question remains to what extent the better efficiency of the CO_2 compressor is due to CO_2 properties alone, and whether a different compressor design contributes to the reported efficiency superiority.

5 CONDENSER / GAS COOLER ENTROPY GENERATION

Equation (9) gives the entropy generation for the condenser/gas cooler. Combining eqs. (9) and (10) and neglecting the pressure drop term on the air-side (it contributes equally for the CO_2 and halocarbon systems), we can write:

$$\dot{S}_{gen} = \dot{m}_{air}\int_{in}^{out}\left(\frac{dh}{T}\right)_{air} + \dot{m}_{ref}\int_{in}^{out}\left(\frac{dh}{T}\right)_{ref} - \dot{m}_{ref}\int_{in}^{out}\left(\frac{v}{T}dP\right)_{ref} \tag{16}$$

which can then be rewritten as:

$$\dot{S}_{gen} = \int_{in}^{out} \delta\dot{Q}_{cond\,/\,gc}\left(\frac{1}{T_{air}} - \frac{1}{T_{ref}}\right) - \dot{m}_{ref}\int_{in}^{out} \frac{v}{T}\,\mathrm{d}P \tag{17}$$

where the $\dot{m}dh$ terms have been replaced by $\delta\dot{Q}_{cond\,/\,gc}$. The first term on the right-hand side of eq. (17) is due to heat transfer irreversibility and the second-term is due to refrigerant flow irreversibility, where the heat transfer irreversibility dominates. The entropy generation for a CO_2 gas cooler will be larger than for a halocarbon condenser since the CO_2 heat rejection rate is larger and the temperature mismatch between CO_2 and air is greater due to the CO_2 temperature glide. Even though the effective temperature difference in a CO_2 gas cooler can be the same as in a halocarbon condenser and the approach temperature in a gas cooler can be smaller than that in a condenser, the entropy generation will be larger in a gas cooler because of the large temperature mismatch over a significant fraction of the gas cooler.

For the automotive air-conditioning example of Brown *et al*. (2002a), the entropy generation per unit cooling capacity for the CO_2 gas cooler is $3.65 \cdot 10^{-4}$ K^{-1} versus $1.89 \cdot 10^{-4}$ K^{-1} for the R134a condenser, or expressed in terms of lost work the value for the CO_2 gas cooler is 0.383 kW and for the R134a condenser is 0.198 kW.

For the residential air-conditioning example of Brown *et al*. (2002b), the entropy generation per unit cooling capacity for the CO_2 gas cooler is $1.89 \cdot 10^{-4}$ K^{-1} versus $1.16 \cdot 10^{-4}$ K^{-1} for the R22 condenser, or expressed in terms of lost work the value for the CO_2 gas cooler is 0.615 kW and for the R22 condenser is 0.377 kW.

6 ENTROPY GENERATION FOR EXPANSION DEVICE AND LLSL-HX

6.1 Expansion Device

For an adiabatic expansion device operating at steady-state with a single inlet and a single outlet, the entropy balance is:

$$\dot{S}_{gen} = \dot{m}_{ref}\left(s_{out} - s_{in}\right) \tag{18}$$

Thus, similar to the compressor, the entropy generation for the expansion process is a function of the mass flow rate and the thermodynamic states at the inlet and outlet.

For the automotive air-conditioning example, the entropy generation per unit cooling capacity for the CO_2 expansion device is $2.52 \cdot 10^{-4}$ K^{-1} versus $1.70 \cdot 10^{-4}$ K^{-1} for the R134a expansion device, and the corresponding value of lost work is 0.264 kW for the CO_2 expansion device and 0.178 kW for the R134a expansion device. Therefore, the CO_2 expansion device generates 48 % more entropy than does the R134a expansion device.

For the residential air-conditioning example, the entropy generation per unit cooling capacity for the CO_2 expansion device is $1.82 \cdot 10^{-4}$ K^{-1} versus $0.65 \cdot 10^{-4}$ K^{-1} for the R22 expansion device, and the corresponding value of lost work is 0.591 kW for the CO_2 expansion device and 0.211 kW for the R22 expansion device. Therefore, the CO_2 expansion device generates 280 % more entropy than does the R22 expansion device.

Recall that the automotive and residential CO_2 systems were equipped with a llsl-hx; hence, the entropy generations in the CO_2 expansion valves were affected not only by different refrigerant properties and operating conditions but also by different effectiveness values for the internal heat exchangers. The entropy generation values presented above for the CO_2 valves are much higher than for the R134a and R22 valves, but they would be even greater if not for the use of a llsl-hx, which moves the CO_2 throttling process to a lower inlet enthalpy point. Without doing so, the inlet quality of the CO_2 evaporator would be quite large. Because of the proximity to the critical point and the small slope of the saturated liquid line, the throttling irreversibilities for CO_2 are large. For this reason, minimizing throttling losses in a CO_2 system is necessary, and a llsl-hx appears to be the most practical option for this purpose.

6.2 LLSL-HX

The entropy generations presented for the CO_2 expansion valves were calculated for CO_2 systems equipped with a llsl-hx. Although heat transfer in a llsl-hx results in entropy generation, the combined entropy generation in a CO_2 llsl-hx and expansion valve is lower than it would be if only an expansion valve were used.

For an adiabatic llsl-hx operating at steady-state, the entropy balance is:

$$\dot{S}_{gen} = \left[\dot{m}_{ref}\left(s_{out} - s_{in}\right)\right]_{high-pressure\ side} + \left[\dot{m}_{ref}\left(s_{out} - s_{in}\right)\right]_{low-pressure\ side} \tag{19}$$

For the automotive air-conditioning example, the entropy generation per unit cooling capacity for the CO_2 llsl-hx is $1.20 \cdot 10^{-4}$ K^{-1} and for the residential air-conditioning application is $0.64 \cdot 10^{-4}$ K^{-1}, and the corresponding values of the lost work are 0.115 kW for the automotive air conditioner and 0.208 kW for the residential air conditioner.

6.3 Total: Expansion Device and LLSL-HX

The total entropy generation is the sum of the entropy generated by the expansion device and by the llsl-hx. For the automotive air conditioner, the total entropy generation per unit cooling capacity for the expansion side for the CO_2 system is thus $3.72 \cdot 10^{-4}$ K^{-1} and for the R134a system is $1.70 \cdot 10^{-4}$ K^{-1}, or expressed in terms of lost work the values are 0.379 kW and 0.162 kW, respectively. For the residential air conditioner, the total entropy generation per unit cooling capacity for the expansion side for the CO_2 system is $2.46 \cdot 10^{-4}$ K^{-1} and for the R22 system is $0.65 \cdot 10^{-4}$ K^{-1}, and the corresponding values of lost work are 0.799 kW and 0.211 kW, respectively. These values translate to 219 % more entropy generation for the CO_2 expansion processes as compared to the R134a expansion device for the automotive air conditioner and to 378 % more entropy generation for the CO_2 expansion processes as compared to the R22 expansion device for the residential air conditioner.

7 OVERALL SYSTEM RESULTS AND DISCUSSION

Figure 2 summarizes the entropy generation analyses for both systems. For the automotive air-conditioning system, we calculated $\dot{W}_{lost}$ for source and sink temperatures of 7 °C for the low-temperature source and 43.3 °C for the high-temperature sink. With a cooling load of 3.32 kW, the resulting COP_{Carnot} is 7.7, and $\dot{W}_{Carnot}$ is 0.43 kW. The CO_2 system generates 36 % more entropy than does the R134a system. This translates into a COP for the R134a system that is 24 % greater than is the COP of the CO_2 system. The inferior CO_2 performance is primarily due to the heat rejection and expansion processes, which account for 67 % of the total entropy generation for the CO_2 system, whereas for the R134a system these processes account for only 44 % of the total. In terms of absolute values, the entropy generation per unit cooling capacity for these two processes is $7.37 \cdot 10^{-4}$ K^{-1} for the CO_2 system, and is only $3.59 \cdot 10^{-4}$ K^{-1} for the R134a system. That is, the entropy generation per unit cooling capacity for these processes is 205 % greater for the CO_2 automotive air conditioner as compared to its R134a counterpart.

For the residential air-conditioning system, we calculated $\dot{W}_{lost}$ for source and sink temperatures of 9 °C for the low-temperature source and 35.0 °C for the high-temperature sink. With a cooling load of 10.56 kW, the resulting COP_{Carnot} is 10.9 and $\dot{W}_{Carnot}$ is 0.97 kW. The CO_2 system generates 63 % more entropy than does the R22 system. This difference translates into a COP for the R22 system that is 37 % greater than is the COP of the CO_2 system. As for the automotive example, the inferior CO_2 performance is primarily due to the heat rejection and expansion processes, which account for 61 % of the total entropy generation for the CO_2 system, whereas for the R22 system these processes account for only 41 % of the total. In terms of absolute values, the entropy generation per unit cooling capacity for these two processes is $4.15 \cdot 10^{-4}$ K^{-1} for the CO_2 system, and is only $1.81 \cdot 10^{-4}$ K^{-1} for the R22 system. That is, the entropy generation per unit cooling capacity for these processes is 229 % greater for the CO_2 residential air conditioner as compared to its R22 counterpart.

The entropy generation analyses demonstrate that the two biggest contributors to the poorer CO_2 performance are the heat rejection process and the expansion process. The major contributor to the entropy generation in the CO_2 gas cooler is the large temperature difference between CO_2 and air over a significant fraction of the gas cooler. This temperature difference would need to be significantly reduced in order to lower the amount of entropy generation. One way to accomplish this would be to achieve glide matching between CO_2 and the external heat transfer fluid. As Fig. 2 shows, the potential benefit from using a counter-flow gas cooler, if practically feasible, is much greater for the automotive application because of the higher irreversibilities due to a greater temperature mismatch between the CO_2 and air. However, a previous study with zeotropic mixtures showed that it is rather difficult to achieve good glide matching in air-to-refrigerant heat exchangers (Marques and Domanski, 1998).

The primary reason for the large amount of entropy generation associated with the expansion process is the low critical temperature of CO_2. A llsl-hx provides inexpensive partial relief from the throttling losses. A work recovery device would be a more direct way of addressing the problem (Robinson and Groll, 1998). However, design of a cost-effective work recovery device poses significant challenges, and requires considerable research effort.

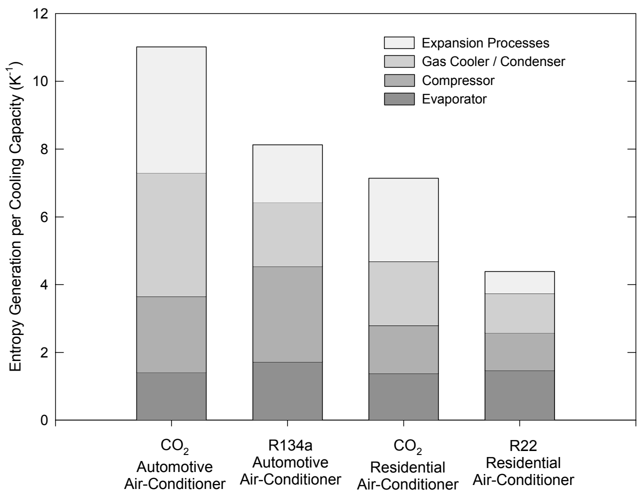

Figure 2: Entropy generation results for automotive and residential air-conditioning applications

CONCLUSIONS

In this paper, we present entropy generation analyses for the evaporator, compressor, gas cooler, expansion device and liquid-line suction-line heat exchanger for a transcritical carbon dioxide cycle for automotive and residential air-conditioning systems, and present entropy generation analyses for the evaporator, compressor, condenser, and expansion device for a subcritical R134a automotive air-conditioning system and a subcritical R22 residential air-conditioning system. The analyses show that the CO_2 automotive air-conditioning system generates 36 % more entropy than the R134a automotive air-conditioning system, and that the CO_2 residential air-conditioning system generates 63 % more entropy than the R22 residential air-conditioning system. The biggest contributors to the lower CO_2 performance are irreversibilities associated with the heat rejection process and the expansion related processes.

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