EXECUTIVE SUMMARY

FINAL REPORT ARTI-21CR/611-10060-01
CO_2 COMPRESSOR-EXPANDER ANALYSIS

Background

Carbon dioxide (CO_2) is a potential substitute for HCFC refrigerants with favorable environmental properties compared to other HCFC alternatives. One of the major challenges with CO_2 in air-conditioning applications is the low energy efficiency of systems at higher heat sink temperatures . The major loss in CO_2 cycles is the throttling loss associated with the expansion process. The availability lost during the expansion process can be recovered with a work-producing expansion device, an expander. Theoretical studies have estimated improvements of the basic CO_2 cycle in the range of 40% to 60% for outdoor temperatures of 27°C to 50°C, resulting in efficiencies competitive with or better than those of current conventional systems.

Objective

The overall objective of this project is to provide the HVAC industry with elements to understand the practical performance potential of CO_2 as an alternate refrigerant in residential air-conditioning applications through the implementation of an expander – compressor system. The work analytically defines the most promising approaches from a performance and feasibility standpoint. The work accounts for the requirements inherent in the integration of the selected types of expanders into CO_2 systems, in order to provide a realistic evaluation of the system performance.

System Performance with Expander, CO_2 vs. R22

The energy efficiency of a CO_2 air-conditioning system with conventional expansion valve and without suction line heat exchanger at an outdoor air temperature of 28°C is approximately 4% to 6% lower than the energy efficiency of a comparable R22 system. At outdoor air temperatures of 50°C, the energy efficiency of the CO_2 system is approximately 43% lower. A CO_2 system with an ideal expander performs 25% to 38% more efficiently at 28°C than an R22 system with a conventional expansion valve. At 50°C, the ideal CO_2 expander system is 4% to 7% more efficient than the R22 system. If the expander efficiency is 80%, the CO_2 system performs 16% to 27% better than the R22 system at 28°C and 7% to 11% worse at 50°C. At approximately 36°C, both systems perform with a similar efficiency. With an expander of 60% efficiency, the CO_2 system performs 7% to 18% better at 28°C and 18% to 21% worse at 50°C. At approximately 31°C, the performance of both systems is equal. If the expansion valve is replaced by an ideal work-recovering expander in both systems, the CO_2 system performs 19% to 31%

more energy efficiently at outdoor air temperatures of 28°C. At 50°C, the CO_2 system performs approximately 9% to 12% worse than the R22 system. With an expander efficiency of 80%, the CO_2 system performs 10% to 21% better at 28°C and 19% to 22% worse at 50°C. If the efficiency of the expander is 60%, the energy efficiency of the CO_2 system is 4% to 14% higher than the R22 system at 28°C and 26% to 29% lower at 50°C outdoor air temperature. Table A lists the estimated coefficient of performance for the systems.

Table A: COP of CO_2 and R22 systems at ambient temperatures of 28°C and 50°C:

Expander efficiency [%]	28°C		50°C	
	CO_2	R22	CO_2	R22
100	6.9 – 7.6	5.8	2.9 - 3	3.3
80	6.4 – 7.0	5.8	2.5 – 2.6	3.2
60	5.9 – 6.5	5.7	2.2 – 2.3	3.1
0	4.8 – 5.3	5.5	1.6	2.8

Effect of Suction Line Heat Exchanger in Expander System

At most operating conditions, a suction line heat exchanger does not improve the energy efficiency of an expander system. Only if a significant fraction (more than 70%) of the work extracted from the expansion process is lost during the energy conversion does the performance benefit from an internal heat exchanger. If more than 30% of the expander work can be recovered, the suction line heat exchanger reduces the energy efficiency of the system. This is due to the reduced work output at lower expander inlet enthalpy and the increased compressor work at higher super-heat. However, it should be noted that a suction line heat exchanger can have secondary benefits, such as protection of the compressor at operation with flooded evaporator, and should not be excluded from the system without careful consideration.

Expander and Compressor Performance

Three positive displacement mechanisms have been analyzed for their potential as expansion machines in a CO_2 system:

(1) Reciprocating piston
(2) Rotary piston
(3) Scroll

Tables B and C summarize the expected performance for the devices as compressors and expanders at AHSRAE A test conditions for a range of machining tolerances (leakage gap size).

Table B: Compressor indicated, isentropic, and volumetric efficiency vs. leakage gap size

	5 µm			10 µm			15 µm		
	η_{ind}	η_{is}	η_{vol}	η_{ind}	η_{is}	η_{vol}	η_{ind}	η_{is}	η_{vol}

Recip. piston	0.93	0.77	0.88	0.91	0.76	0.84	0.85	0.72	0.79
Rotary piston	0.96	0.79	0.96	0.90	0.75	0.92	0.85	0.72	0.87
Scroll	0.87	0.73	0.88	0.64	0.56	0.65	0.39	0.36	0.38

Table C: Expander indicated and isentropic efficiency vs. leakage gap size

	5 μm		10 μm		15 μm	
	η_{ind}	η_{is}	η_{ind}	η_{is}	η_{ind}	η_{is}
Recip. piston	0.90	0.75	0.87	0.72	0.81	0.66
Rotary piston	0.94	0.79	0.84	0.69	0.78	0.63
Scroll	0.93	0.78	0.75	0.60	0.47	0.32

The performance of the scroll mechanism is most dependent on the leakage gap size. At machining tolerances above 5 μm, the scroll mechanism performs significantly worse than the other mechanisms. The performance of the scroll mechanism is dominated by the leakage across the tips of the scroll wraps. If this leakage path is eliminated, the indicated efficiency of a scroll expander, for instance, is above 90% for a flank clearance of 15 μm. Experimental results of existing scroll compressors for conventional refrigerants show a high volumetric and compression efficiency, indicating that effective techniques for leakage reduction are available. The reciprocating piston and rotary piston mechanisms perform similarly at all leakage clearances. At small leakage gap sizes, the rotary piston performs slightly better than the reciprocating piston.

The reciprocating and rotary piston expanders require externally actuated valves. With externally activated and controlled valves, current technology limits the expander frequency to values on the order of 1 Hz. The scroll mechanism is not subject to this limitation. All mechanisms require some kind of valve controls to adjust the volume ratio. If the inlet valves of an expander are not controlled, the internal volume ratio is constant. If the internal volume ratio does not match the ideal volume ratio imposed by the operating conditions, over- or under-expansion reduces the expander efficiency. Performance simulation indicates that the ideal volume ratio in a CO_2 system with optimized high-side pressure changes by less than 25% over a wide range of operating conditions. The expander performance degradation due to over- or under-expansion is below 10%, indicating that expander valve control may not be necessary.

Due to the small change of the pressure ratio over a wide range of operating conditions in a CO_2 system, the compressor and expander indicated efficiencies are approximately constant at all operating conditions. The indicated efficiencies of a reciprocating piston compressor and a reciprocating piston expander with a leakage gap size of 10 μm are approximately 91% and 87%, respectively. With an effective isentropic efficiency 15% lower than the indicated efficiency (i.e. 76% and 72% for compressor and expander, respectively), the COP of an optimized CO_2 expander system is approximately 6.6 at 28°C outdoor air temperature and 2.4 at 50°C. Compared to a conventional R22 system, the CO_2 expander system performs 20% better at the low temperature conditions and approximately 14% worse at the high-temperature conditions. At approximately 34°C, both systems operate with the same energy efficiency.

Integration Concepts

For maximum efficiency, the mechanical work generated by the expander should be directly utilized. To reduce conversion and storage losses, the expander work can supply part of the compressor shaft work, requiring the integration of the compressor and expander on a common shaft. With this design however, the optimization of the high-side pressure becomes a challenge, because the high-side pressure is no longer independent from the compressor speed (unless the expander displacement volume is variable). A number of control options have been analyzed. System simulations indicate that an expander cycle can be operated at or close to the optimum high-side pressure without variable expander displacement or additional control valves if the compressor and expander displacement volumes are designed appropriately.