Research in Transcritical R744 at ACRC, University of Illinois at Urbana-Champaign

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1989년 프레온계 냉매의 대체 가능한 물질로서 CO_2는 재 출현하게 되었고, 그 이후 관련 연구는 점차 증가하였다. 1996년 북미 지역에서 CO_2 연구는 일리노이 대학내 ACRC(Air Conditioning and Refrigeration Center)에서 최대로 시작되었다. 연구비의 90 % 이상은 산업체에 의해 지원되었고, 나머지는 미국 정부에 의해 지원되었다.

본 고는 자동차 및 가정용 에어컨 및 열펌프 시스템 및 구성부품 개선에 대한 전반적인 연구 활동에 관해 기술하였다. 또한 시스템 성능 비교 결과는 열전달, 압력강하, 사이클 변환, 냉매 분배에 대한 연구 지침을 제시하였고, 지속적인 연구 노력을 통하여 천연냉매를 이용하여 간접적인 지구온난화 가스 방출을 최소화하기 위한 시스템 효율을 증가 시키고 있다. CO_2의 고유 특징인 초월임계 CO_2 사이클, CO_2의 열역학 및 전달 물성치를 통한 효율 개선, 내부 열교환을 통한 사이클 성능 개선방안도 기술되었다.

INTRODUCTION

Carbon dioxide re-emerged as a possible replacement for fluorocarbon refrigerants in 1989 [1] and the intensity of research efforts has increased steadily since that time. The largest research effort in North America began in 1996 at the University of Illinois Air Conditioning and Refrigeration Center (ACRC); it has been conducted by five faculty, seven visiting scholars from four countries, and many MS and PhD students. More than 90 % of the funding has been provided by industry, and the remainder by the US government.

This paper provides a brief overview of that research, which has focused on automotive and residential air conditioning and heat pumping systems. Results from system-level performance comparisons have guided and set priorities for more fundamental investigations of basic processes, such as heat transfer, pressure drop, cycle alterations, and refrigerant distribution. The resulting insights have provided the foundation for a new generation of components and systems, designed using simulation models that were developed and validated during experimentation with the early prototypes. Current research efforts continue to be guided by the need to increase system operating efficiency to minimize indirect greenhouse gas emissions, while eliminating direct emissions through the use of a natural refrigerant.

Background on the unique characteristics of the transcritical carbon dioxide cycle can be found in many publications. The efficiency-enhancing advantages associated with carbon dioxide's unique thermodynamic and transport properties (e.g. low pressure ratio; slope of vapor pressure curve; boiling behavior near the critical point) were discussed in [2]. For descriptions of options for modifying the cycle through use of internal heat exchangers (e.g. multistage compression; internal heat exchange; expanders)

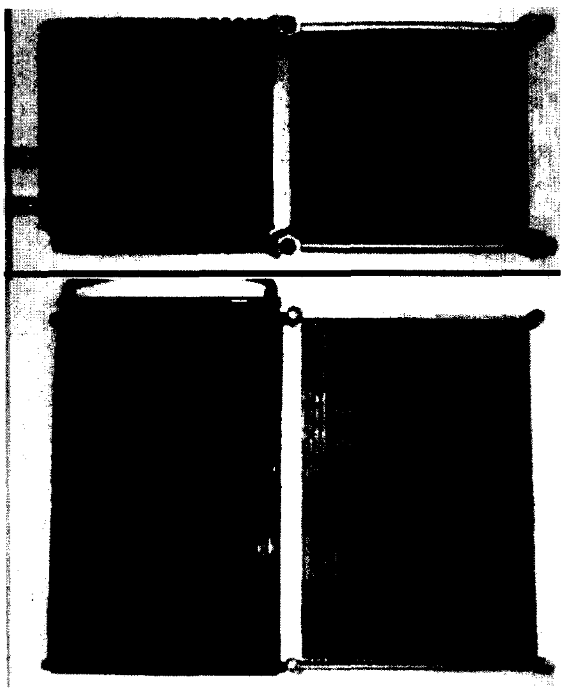
AIR CONDITIONING SYSTEMS

To provide the capability for experiments on both systems and components, separate environmental chambers were constructed to simulate a very wide range of indoor and outdoor climate conditions. Each chamber contains a wind tunnel, and has parallel sets of refrigerant piping to facilitate comparisons of CO_2 and other refrigerants. The heavily insulated walls with heat transmission measurement, plus accurate metering of refrigerant and air mass flow rates make it possible to obtain three energy balances for both the indoor and outdoor heat exchangers : air-side, refrigerant-side and room calorimetry.

Automotive cooling

The first R744 prototype system tested at ACRC was designed and built to match closely the weight, heat exchanger dimensions, face velocities and air-side pressure drops of a commercially available clutch-cycling R134a system for a Ford Escort.

Heat exchangers for the R744 systems were made of flat aluminum extruded multiport tubes to deal with the high-pressure refrigerant, making it possible to maximize air-side heat transfer area within the overall weight constraint. Photographs of the evaporators (top) and condenser/gas cooler (bottom) are



R134a R744

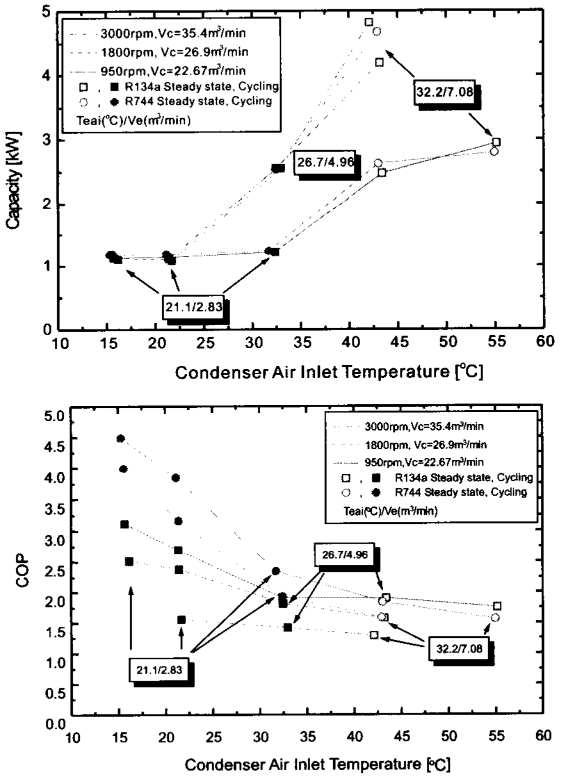
[Fig. 1] Main components of the first R744 mobile air conditioning system prototype compared to R134a baseline. Evaporators are on the top; heat rejecting heat exchangers below.

shown in **Figure 1**. The 21 cc compressor was of swashplate design, and the system could be controlled by either a needle valve or a back-pressure valve. Test matrices were defined for the purpose of developing and validating component and system simulation models, as well as supporting data-to-data comparisons of R744 and R134a at normal, seasonal and extreme operating conditions.

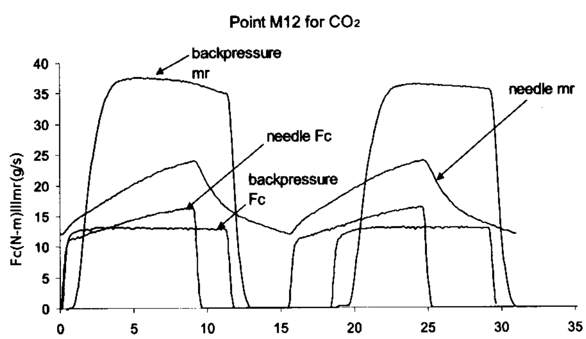
The R-744 system was sized to provide approximately equal capacity at the extreme high-temperature (54.4 ℃) idling condition. However its COP fell 10 % short of the base-line system at that point. At outdoor ambient temperatures below 40 ℃ where most a/c operation occurs, the R-744 system COP exceeded that of the baseline R-134a system up to 40 % as shown in **Figure 2**.

Experiments were conducted on both systems in both steady state and cycling modes. Cycling behavior is very sensitive to the type of expansion device used. A fixed orifice is simplest, but is unable to maintain the high-side pressure at its COP-optimizing level during the on-cycle as reflected by the large differences in refrigerant flow rate shown in **Figure 3**. It is also clear that the backpr-essure valve maintains a steady constant co-mpressor torque than the fixed orifice. Control options are discussed in[3]. Additional exp-erimental results and a more detailed inve-stigation of the COP-maximizing high side pressure strategy is presented in[4], which found a linear relationship between gas cooler exit temperature and COP maximums over a wide range of operating conditions.

Based on the results of an analysis of a a large number of experiments and some new concepts, a second-generation prototype system was designed and is serving as the

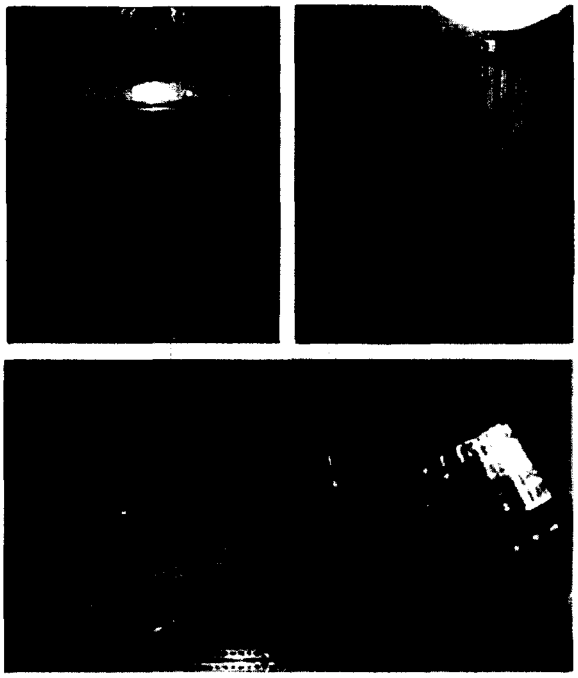


[Fig, 2] Comparison of the first R744 mobile air cond-itioning system prototype performance to to R134a baseline



[Fig, 3] Characteristic behavior in cycling focus for current research. It is equipped with a variable-displacement compressor, and heat exchangers configured to exploit the unique transport and thermodynamic properties of R744. Some of these design features and model validation results will be discussed in the Components section below.

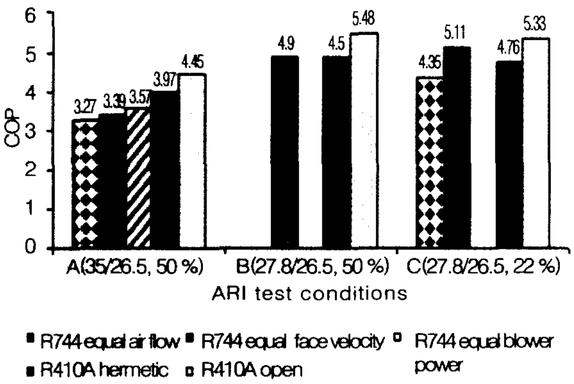
Recently due to the need to develop systems having lower greenhouse warming impacts, and based on the results of initial research on alternative refrigerants, consortium of auto manufacturers has undertaken through SAE a rigorous experimental comparison involving testing and evaluation of several competing systems in the ACRC laboratories, with extensive instrumentation under identical operating conditions. The systems are: baseline production R134a ; transcritical carbon dioxide ; advanced R134a ; and propane with secondary loop.



[Fig. 3] Heat exchagers of the first R744 residential air conditioning system prototype (microchannel) compared to R410A baseline (round tube plate fins). Outdoor units are up while evaporators are below.

Residential cooling

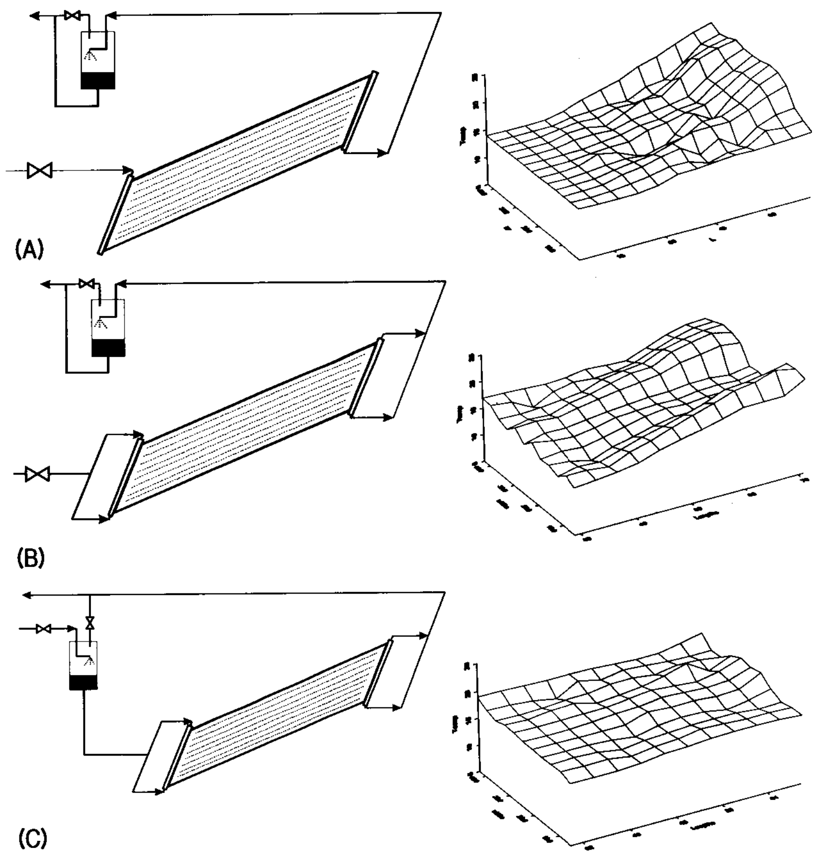
A similar set of experiments was conducted on a prototype North American-style ducted split air conditioning system. The baseline R410A system selected was the most efficient commercially available, and the R744 prototype heat exchangers were designed to match as closely as possible its overall package dimensions, as shown in Figure 4. Experiments were conducted at three standard ARI test conditions, and the results confirmed the very good performance of the baseline R410A system. The prototype R744 system achieved the same cycle COP as the baseline system at test conditions B and C (26.7/27.7 ℃). And as expected, it exhibited at 10 %



[Fig. 5] COP comparison of R744 prototype in a/c mode compared to R410A system at identical capacities (10.5 kW). Details of three air flow rate over evaporator are discussed in[7].

lower COP at the higher ambient temperature condition A (26.7/35℃), see **Figure 5**. Since microchannel heat exchangers have lower air-side pressure drop, the R744 system can benefit from greater air flow rate with no penalty in fan or blower power. This illustrates how next-generation component designs can be guided by early prototype comparisons : as alternatives to increasing air flow rate, benefits could be realized through direct fan power savings, or by increasing air-side heat transfer area until pressure drop equals that of the baseline.

The model validation effort was based on these initial experiments, and a large discrepancy was noted between simulation and data for the evaporator. A thermocouple grid was then installed downwind of evaporator, provided evidence of significant refrigerant maldistribution among the parallel microchannel tubes. **Figure 6** (b) shows how even symmetrical feeding of the header results in the middle tubes receiving excess liquid while the outer ones receive excess vapor. **Figure 6** (c) shows how this problem was eliminated through use of controlled flash gas bypass - a new concept to improve distribution and increase heat transfer coefficients while reducing pressure drop in evaporator increasing heat exchanger effec-tiveness by 17 % and increasing the evaporating temperature by 3.8 ℃ as presented in[5]. Since compact high-performance heat exchangers such as these are likely to be needed for efficient heat

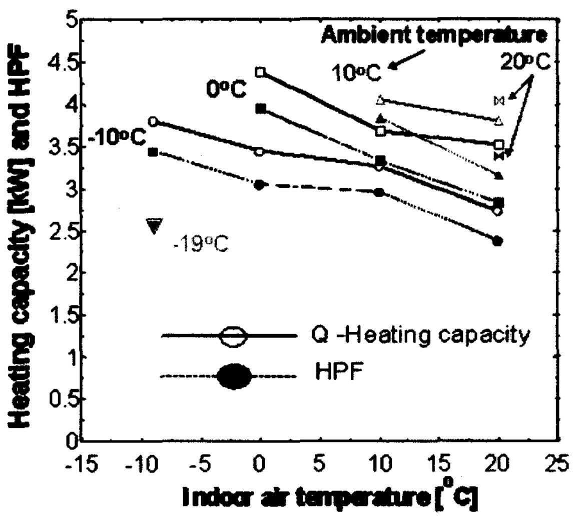


[Fig. 6] Effects of flash gas removal expressed through air exit temperature profiles shown on the right. On the left a) regular DX configuration with single side inlet, b) same as a) just with two inlets and c) flash gas removal. All three operations are completed with same evaporator capacity. In both a) and b) cases evaporation temperature was 7℃ while in c) it was 10.8 ℃ that consequently increased COP for 20 %.

pumps in the future, solv-ing the maldistribution problem has become a high priority for systems using all kinds of refrig-erants. Accordingly a substantial amount of research is now being cond-ucted to understand the behavior of developing two-phase flow in hea-ders, and on schemes for bypassing vapor to minimize degradation of evaporator performance.

HEAT PUMPING SYSTEMS

The inherent thermodynamic disadvantage of the transcritical cycle is its high heat rejection temperature. Unless there is a need for that high-temperature heat, competing cycles will have an efficiency advantage. In automotive air conditioning, R744 exploits the high refrigerant-to-air DT to make the gas cooler ultra-compact, which produces energy efficiently indirectly by enabling more aerodynamically streamlined



[Fig. 7] Automotive heat pump performance at different indoor and outdoor conditions

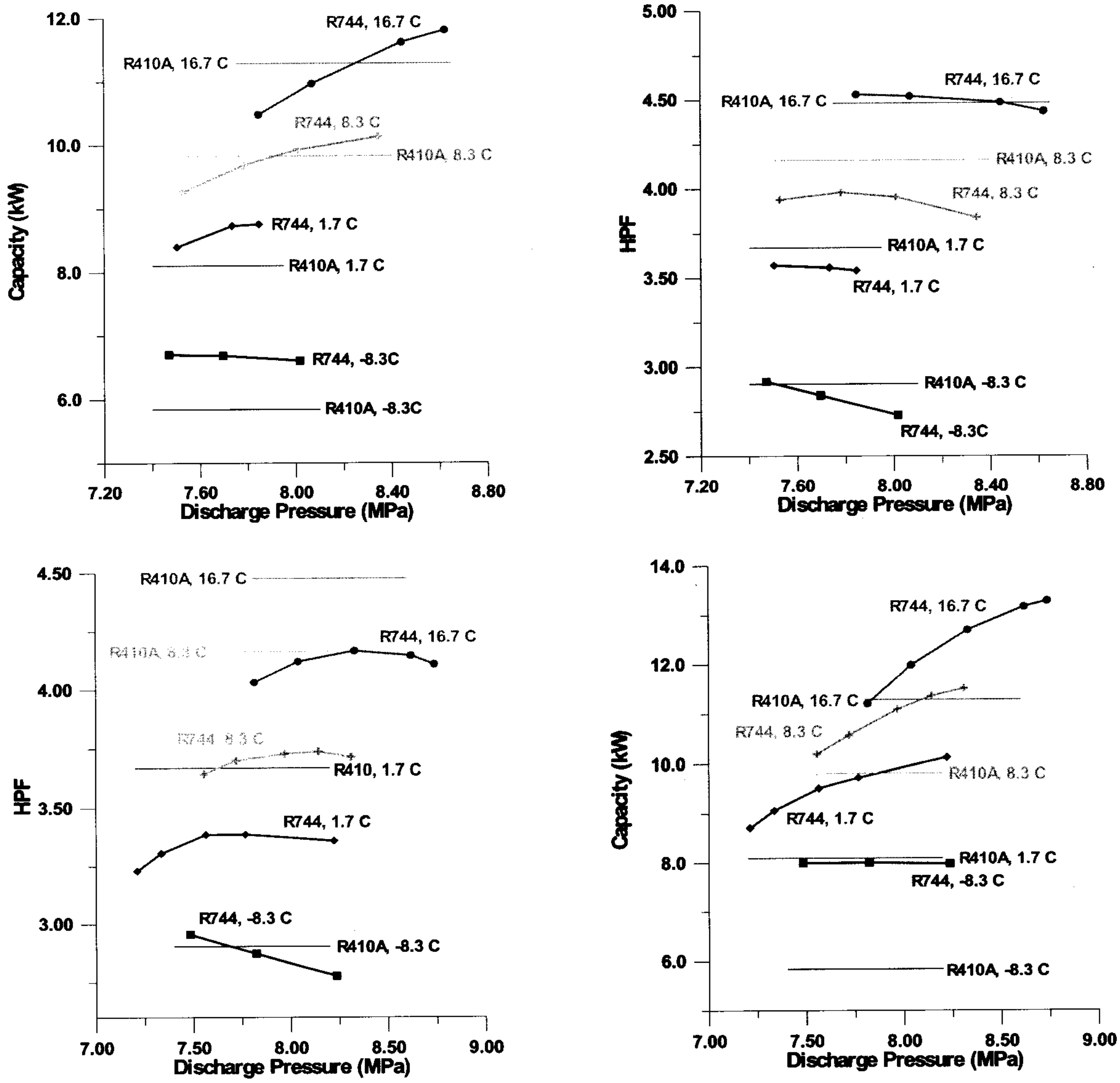
vehicle design. In heating mode, both the automotive and residential sectors offer opportunities for deriving benefit from the high-temperature heat rejected from the gas cooler. As cars become more energy-efficient, there is inadequate waste heat in the engine coolant to maintain comfort in cold weather; an R744 heat pump can deliver air instantly at high tem-perature. Today's residential heat pumps deliver large amounts of air at temperatures very near that of human skin, causing discomfort via evaporative cooling. On the other hand the transcritical cycle can deliver air at 60C, achie-ving the same level of comfort as a gas forced air furnace while quietly moving far less air.

Similarly, R744 can provide heat via a hydronic secondary loop without the energy penalty that would be incurred in a heat pump operating on a subcritical cycle.

Automotive heating

The first published results of R744 heat pump experiments were obtained by running our original auto a/c prototype system in reverse. Although its cross-counterflow indoor heat exchanger was far from ideal, the data shown in **Figure 7** illustrate the essential features of an automotive heat pump: capacity is highest at startup, when it is needed most; the capacity is at least three times higher than what could be obtained from an electric resistance or friction heater, due to the high heat pumping effi-ciency; capacity and efficiency decline slowly as the car warms up and heat becomes ava-ilable from the engine coolant. These initial results have proven quite valuable in guiding the design and development of improved components for next-generation systems. See [6] for a more detailed discussion of system performance in heating mode.

One reason heat pumps are not currently employed in automobiles is that R134a has severe disadvantages as a heat pump fluid. Regardless of the fluid, however, air-to-air heat pumps present substantial technological challenges that have not yet been addressed: the outdoor heat exchanger will accumulate frost, and perhaps ice as water is splashed on it from the road. Very little is known about frosting and condensate drainage from ultra-



[Fig. 8] Residential heat pump performance at equal a/c (bottom) and h/p (top) capacities compact microchannel heat exchangers, and what is known suggests that the difficulties may be substantial[6), 7)]. Residential heat pumps defrost by switching into air conditioning mode, but that is not a desirable option for automobiles because the there is so little air in the passenger compartment. Connecting to another heat source for defrosting may also be challenging, especially after a series of short trips during which the engine never reached normal operating temperature. On the other hand during normal operation, the heat pump needs only to provide supplemental capacity for a short period after startup, while the engine is warming up. After that, sufficient heat should be available for defrosting; the only problem is how to transfer it.

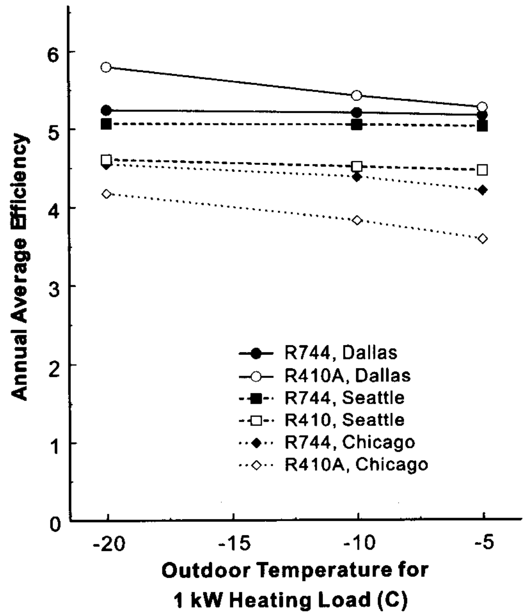
Residential heating

Similar experiments were conducted on the original prototype residential split system reversing its operation to study heating performance. Since the original baseline system was a/c-only (obtained before the first R410A heat pump system became available), the package dimensions for the heating comparisons were no longer equal: the baseline subcritical R410A system had larger heat exchangers. System performance was compared for two configurations: first when the R744 prototype semi-hermetic compressor speed was set to match heating capacity at 8.2 ℃ outdoors and 21.1 ℃ indoors ; and second when the air conditioning capacity was matched at 35 ℃ outdoors and 26.5 ℃ indoors. The R744 system had slightly lower HPF in heating mode, but its higher capacity at lower outdoor temperatures reduced the need for less efficient supplemental heat, see **Fig. 8**. The results helped establish priorities for subsequent analyses of the effects of R744's transport properties on heat exchanger (T's; and ways to optimize the geometry of the next generation of reversible heat exchangers for transcritical R744 heat pump systems.

To complement the experimental comparisons of competing system, a parallel analytical effort was undertaken to relatively simple simulation model that could analyze many of the essential differences between transcritical and subcritical systems. To keep the comparison geometry-independent, the air-side heat transfer and pressure drop characteristics were assumed to be identical, and the "ideal" assumption of infinite heat exchanger areas was retained. The analysis began with a simple comparison of ideal thermodynamic cycles which, as expected, showed R410A to be more efficient in both heating and cooling mode. Then as constraints and non-ideal considerations (e.g. finite air flow rates required by to achieve comfort and limit fan power) were introduced simultaneously for both systems, the efficiency gap narrowed. The results suggest that R744 heat pumps could be more efficient than R410A in cold climates, due to their higher capacity at cold temperatures and correspondingly less need for electric resistance backup heat. In climates dominated by cooling loads, such as the southern US, subcritical R410A systems have a clear advantage in annual average efficiency, as shown in

Figure 9. Imposition of finite-area constraints on the heat exchangers would benefit R744, but probably not enough to reverse these results.

Despite its simplicity, the analysis helped quantify the individual differences between the two systems and cycles. The pressure ratio of the R744 cycle is lower and the resulting difference in compressor work is magnified by real compressor efficiencies, it is not sufficient to close the gap. The superior thermophysical properties of R744, together with the small slope of the vapor pressure curve, allow for substantially increased heat transfer coefficients with negligible thermodynamic penalties from the associated pressure drop, enabling R744 systems to meet a given dehumidification constraint at a slightly higher evaporating temperature than R410A. At supply air temperatures greater than 40 ℃, the ideal R744 cycle is slightly more efficient than R410A and provides substantially higher capacity at colder outdoor temperatures. This advantage appears not only for air-to-air heat pumps, but also for hydronic heating systems where higher quality heat is required.



[Fig. 9] Annual average efficiency for R410A and R744 systems as function of heating load requirement for 60℃ supply air in heating - model predictions

COMPONENT IMPROVEMENTS

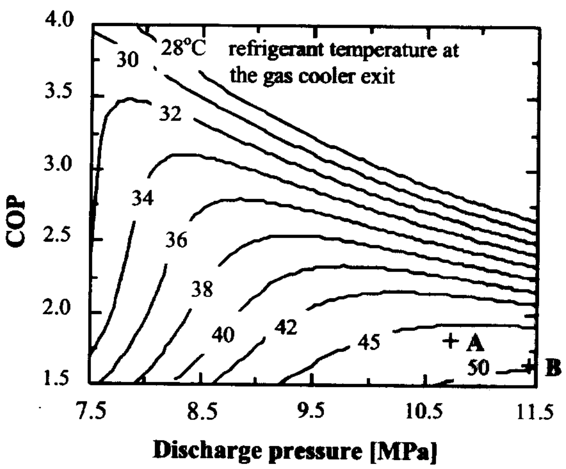
Gas coolers

The automotive industry has almost completed its transition to microchannel condensers, because of the compactness resulting from higher refrigerant-side areas, and lower air-side pressure drops which allow for higher face velocities. High face velocities not only increase air side heat transfer coefficient, but also reduce the air temperature "glide" which ultimately places a lower bound on the condensing temperature.

The transcritical carbon dioxide cycle is fundamentally different, because the outdoor air and refrigerant exit are above the critical temperature. A small change in refrigerant exit temperature can therefore produce a large change in enthalpy because specific heat becomes infinite at the critical point. This means system performance is very sensitive to gas cooler design, and there is an extra degree of freedom to select the high-side pressure that maximizes system performance. **Figure 10** shows how COP can be increased 11 % and discharge pressure reduced 5 bar by a gas cooler that cools the refrigerant exit an additional 2 ℃. Moreover the steep refrigerant temperature glide allows for ideal cycle efficiency to be achieved at finite air flow rate, in contrast to the infinite air flow required to achieve ideal efficiency in the subcritical cycle.

Using data from a diverse set of 48 operating conditions a simulation model was validated, and predicted refrigerant outlet temperature within (0.5 ℃ for most of the experimental data, given only the inlet conditions .

Additional details on the selection of heat transfer and pressure drop correlations may be found in [8), 9) and 10)]. Then the model was used to design the next-generation prototype gas cooler shown in **Figure 11**, where a multi-slab overall counterflow configuration concentrates the cool air stream on the exiting refrigerant, because the transcritical cycle is so sensitive to this exit condition[10)]. The new gas cooler design achieves approach temperature differences <2 ℃ at most operating conditions



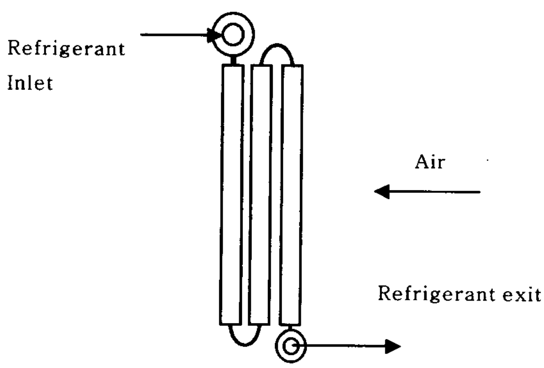
[Fig. 10] Effect of gas cooler exit temperature on transcritical cycle COP for realistic high-side pressures

because air flowing over the first slab undergoes only a small temperature change, and that $\triangle T$ is what places an upper bound on the approach temperature difference.

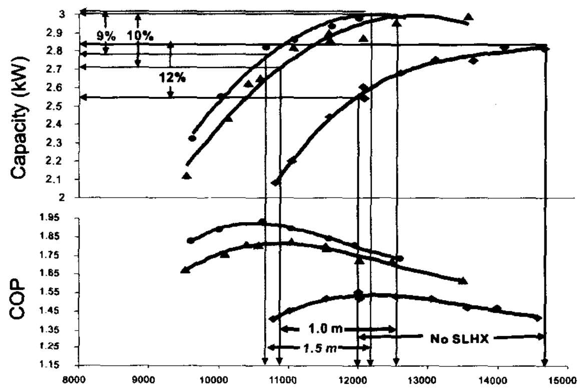
The flat tubes are vertical in this prototype, to facilitate condensate drainage and defrosting in heating mode. Finally the refrigerant flows in a single pass from the inlet to outlet, with no intermediate headers, to accommodate reversibility and facilitate refrigerant distribution in heating mode.

Internal heat exchangers

Suction-liquid heat exchange has been shown by Domanski et al. (1994) to increase ideal cycle efficiency of some refrigerants (eg R134a), and to decrease COP for others (eg R22). The effect for R410A cycle efficiency is neutral, and for R744 it is very beneficial. Through extensive experiments in our R744 prototype systems, and subsequent analyses using a validated simulation model, it has been demonstrated that internal heat exchange can increase cycle efficiency up to 25%; see Figure 12. In automotive a/c systems, internal



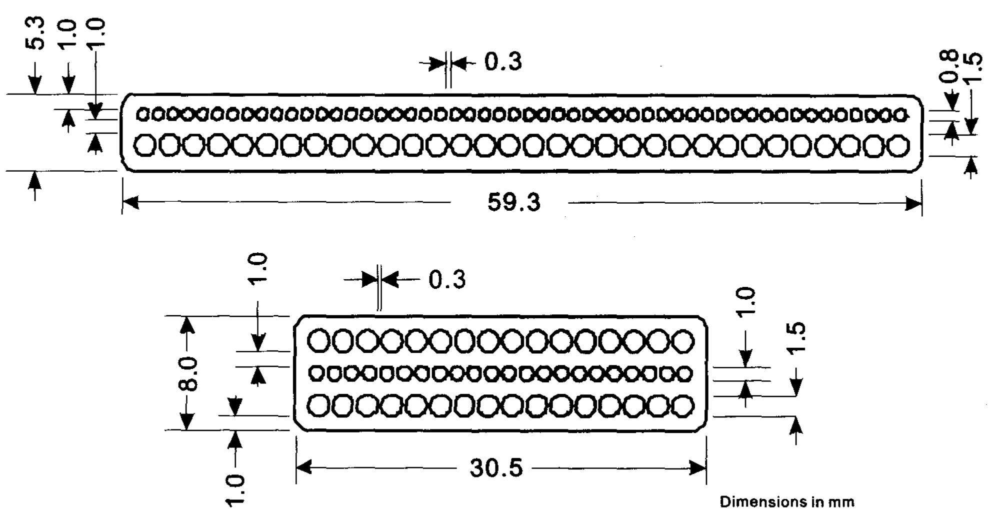
[Fig. 11] A counter flow multi-slab arrangement - side view heat exchange provides the greatest enhancement when it is needed most, while idling at high ambient temperatures. In systems without internal heat exchangers, the capacity and efficiency-maximizing discharge pressures are far apart. Internal heat exchange reduces both, and brings them closer together, creating opportunities for using less precise or simpler control systems and strategies. The analysis showed how three microchannel tubes could be stacked to provide many parallel ports to control pressure drop in the cold suction gas, while forcing the supercritical fluid through smaller ports to maximize heat transfer coefficients and areas upstream of the expansion device where larger pressure drop can be tolerated. Compared to conventional concentric tube designs, the microchannel shown in **Figure 13** reduced material requ-irements by 50 % while elimin-ating the need for long suction and liquid lines, and increasing effectiveness by 10 %.



[Fig. 12] Effect of internal heat exchanger in mobile a/c system at idling speed and 43.3 ℃ ambient

Evaporators

ins, to avoid problems with condensate rete-ntion due to bridging across louvers or fins. The same option is available for microchannel evaporators, and the issue is being revisited because of the need for both indoor and out-door heat exchangers to operate as evap-orators in reversible heat pumping cycles. However louvered fins are used in all



[Fig. 13] Illustration of new designs of internal heat exchanger automotive a/c evaporators, where condensate retention and blowoff are addressed in a variety of ways to minimize performance degradation and mold growth: eg tilting the heat exchanger about 10 degrees off the vertical, optimizing louver angle and spacing; and applying hydrophilic coatings and biocides. See Jacobi (2001) for a thorough review of air-side issues in heat exchangers with noncircular tubes.

Only a few experimental studies have been done to measure the effects of condensate retention on the performance of microchannel evaporators. The first used 30 microchannel heat exchanger samples, and found that the sensible heat transfer coefficient was impaired only slightly by wet surface at low face velocities, but was similar to that for dry a surface at higher face velocities, for fin pitch = 1.4 mm. However for those with smaller fin pitch, there was clear evidence of performance degradation by condensate bridging across the fins at all operating conditions. Based on these data, j and f-factor correlations were developed. In another investigation of the effect of inclination angle (-60 ℃$<\theta<60$ ℃), heat transfer performance for both dry and wet conditions was not influenced significantly, while the pressure drops increased consistently with the inclination angle. The heat transfer coefficients and the pressure drops for the wet conditions revealed the importance of the role of condensate drainage.

The greatest uncertainty in microchannel evaporators, however, is associated with refrigerant maldistribution in the inlet headers. The current strategy for dealing with this problem is to find ways of eliminating it, such as flash gas bypass, rather then trying to model the associated degradation. Excellent results from flash gas bypass (COP increase for 20 %) were obtained in residential CO_2 system. All simulation models of microchannel heat exchangers assume perfect distribution on the refrigerant side. Their main focus is on capturing accurately the important air-side phenomena such as the effects of condensate and inclination angle.

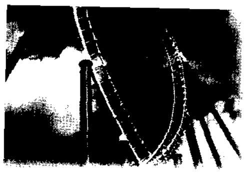
CONCLUSIONS

The paper presented condensed summary of research activities at Air Conditioning and Refrigeration Center, the University of Illinois at Urbana Champaign in transcritical CO_2 technology in last seven years. That development has been made public in 30 references cited and in numerous other reports available from the Center.

Our system results indicate better performance of R744 systems in air conditioning mode in some operating regimes (lower ambient temperatures and higher compressor speeds for mobile systems) while still not as good as R134a or R410A at higher ambient temperatures.

Better performance in heat pumping operation is spread over a wider range of operating temperatures.

Good prediction tools and understanding of relevant phenomena have been developed and are documented in publications discussed in this paper.



Newest, unpublished but completer project results will show further increase in efficiency of transcritical systems due to better control and improved components in our next system generation.

REFERENCES

1. Lorentzen, G. and Pettersen, J. A New, Efficient and Environmentally Benign System for Car Air-Conditioning, Int. Journal of Refrigeration, Vol. 16, No1, pp 4-12, 1993
2. The efficiency-enhancing advantages associated with carbon dioxide's unique thermodynamicand transport properties (e.g. low pressure ratio; slope of vapor pressure curve; boiling behavior near the critical point) were discussed
3. McEnaney, R. P., and P. S. Hrnjak, "Control Strategies for Transcritical R744 Sys-tems," SAE World Congress, Detroit, MI, SAE Paper No. 2000-01-1272, 2000
4. Park, Y. C., J. M. Yin, C. W. Bullard, and P. S. Hrnjak, "Experimental and Model Analysis of Control and Operating Param-eters of Transcritical CO_2 Mobile A/C Sys-tem," VTMS-4 Conference, London, England, 163-170, May 1999
5. Beaver, A., P. Hrnjak, J. Yin, and C. Bullard, "Effects of Distribution in Headers of Micro-channel Evaporators on Trans-critical CO_2 Heat Pump Performance," Proceedings of the International Mechanical Engineering Congress and Exposition, Orlando, FL, 55-64, Nov. 2000.
6. Giannavola, M. S., Experimental Study of System Performance Improvements in Transcritical R744 Systems for Mobile Air-Conditioning and Heat Pumping, MS Thesis, University ofIllinois, 2002
7. Ito, Masaaki, Hiroshi Ogura, Shinya Yosh-inag, Ryoichi Hoshino, Nobuhiro Waka-bayashi, Mitsuo Kudo, Hiroshi Kusumoto, "Consideration of a Parallel-Flow Heat Exchanger for a Heat Pump Application," Proc. JAR Annual Conference1996.
8. Fang, X., C. W. Bullard,. and P. S. Hrnjak, 2001, "Heat transfer and pressure drop of gas coolers," ASHRAE Transactions, 107:1, 12 pp.,
9. Fang, X., C. W. Bullard,. and P. S. Hrnjak, 2001, "Modeling and analysis of gas coolers," ASHRAE Transactions, 107:1, 9 pp.,
10. Yin, J. M., C. W. Bullard and P. S. Hrnjak, "Single-phase pressure drop measure-ments in a microchannel heat exchanger", Heat Transfer Engineering, forthcoming 2002.