TRANSCRITICAL CO_2 MOBILE HEAT PUMP AND A/C SYSTEM EXPERIMENTAL AND MODEL RESULTS

C.W. Bullard, J.M. Yin, P.S. Hrnjak*
Air Conditioning and Refrigeration Center (ACRC)
University of Illinois, 1206 W. Green St. Urbana, IL 61801, USA

ABSTRACT

This article presents the results of the experimental runs of a prototype of R744 (CO_2) refrigeration system operating in a both air conditioning and heat pump mode when heat rejection is done in supercritical region. The prototype system is sized for a compact car. Data presented are in the limited range of operation. Further optimization and extension of operating range is underway. Test facilities for such experiments and systems are described. The prospect of extending the ability of a mobile a/c system to a mobile heat pump operation is very promising.

INTRODUCTION

Transcritical CO_2 systems are attracting significant attention in last several years not only due their environmental impact, but also due to unexpectedly good performance. The performance of some such systems were presented by Pettersen et al. (1993, 1994, 1997a and b), University of Maryland CEEE, several companies as well as by our group in Yin et al. (1998), Boewe et al. (1999a,b), Beaver et al. (1999a, b), etc… In few earlier articles we have analyzed and compared performance of a prototype of a transcritical CO_2 system with the same volume and air-side pressure drop of heat exchangers as in a typical, of-the-shelf R134a mobile system. Results showed slightly worse performance of CO_2 system at very high ambient temperatures (above 45^{o}C) close between 35 and 45^{o}C and better performance at lower ambient temperatures.

We continue to working in the same area. There are four such systems that we are exploring at this moment. These systems are indicative of air conditioning systems used in typical compact cars and sport utility/military vehicles in the USA. Heat exchangers used in the baseline systems are typical for the respective vehicle size. The R744 systems are designed to have similar or smaller heat exchanger core volumes, face areas, and air side pressure drops. In this article we will focus to and present results for heat pump operation of the system first designed R744 system (MAC1) as shown in Table 1 and in the Figure 3.

One of the reasons to be focused to heat pump application are our modeling analysis that indicated great potential of transcritical CO_2 system operation in the heat pump mode. Favorable heat pump operation could append additional reason for considering transcritical CO_2 systems as a viable alternative to existing R134a systems.

* Author to whom correspondence should be addressed, pega@uiuc.edu

EXPERIMENTAL FACILITIES

The test facility is shown in Figure 1. Two environmental chambers have been constructed for each heat exchanger (outdoor and indoor), each containing wind tunnel with variable speed blower and different piping for the two refrigerants. Each chamber and heat exchanger can operate in both regimes: heat rejecting and absorbing. Third chamber in between is for the compressor.

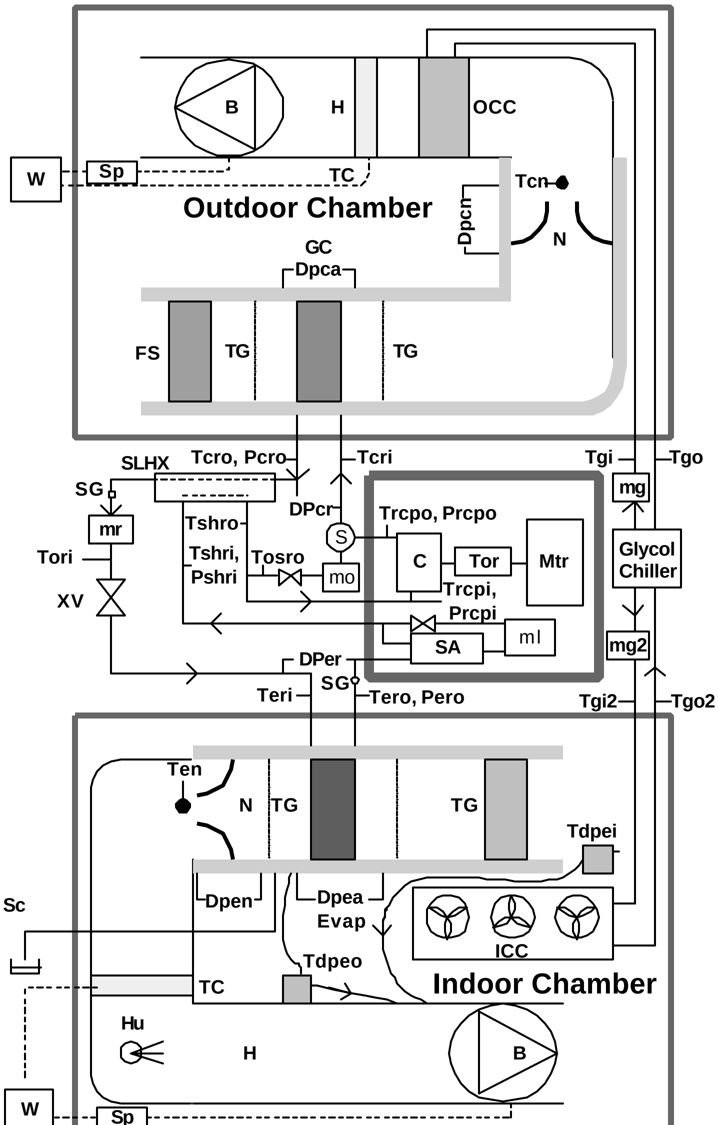

Figure 1. Test facility for R744 air conditioning system

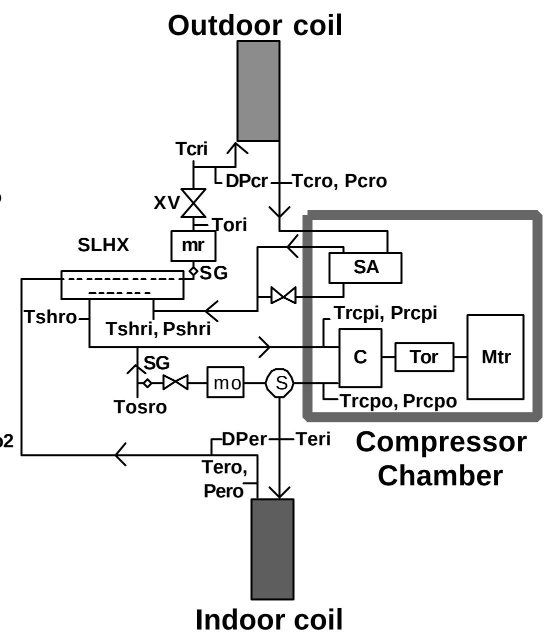

Figure 2. R744 heat pump system

B – blower, **C** – compressor, **Dp** – differential pressure, **Evap** – Evaporator-indoor coil, **FS** – flow straightener, **GC** – gas cooler/outdoor coil, **H** – heater, **Hu** – humidifier, **ICC** – indoor cooling coil, **mg & mg2** – glycol mass flow meter, **mo** – oil mass flow meter, **mr** – refrigerant mass flow meter, **Mtr** – motor, **N** – nozzle, **OCC** – outdoor cooling coil, **P** – pressure, **RH** – relative humidity, **S** – separator, **SA** – suction accumulator, **Sc** – condensate scale, **SG** – sight glass, **SLHX** – suction line (internal) heat exchanger, **Sp** – speed controller, **Th** - tachometer, **T** – thermocouple, **TC** – temperature controller, **TG** – thermocouple grid, **Tor** – torque transducer, **W** – watt transducer, **XV** – expansion valve (any type). **Indices:** **a** – air, **c** – condenser/gas cooler/outdoor coil, **cp** – compressor, **dp** – dew point, **e** – evaporator/indoor coil, **g & g2** – glycol, **i** – inlet, **n** – nozzle, **o** – outlet, **r** – refrigerant, **sh** – suction line (internal) heat exchanger

Three energy balances are obtained for each heat exchanger: air-side, refrigerant-side and room calorimetry. Three independent methods are used instead of the two required by all applicable standards, not only to facilitate determination of system capacities, but also to have two at least whenever refrigerant calorimetry is not reliable due to two-phase exit. This occurs mostly during transients, in some specific operating conditions, and with constant area expansion devices. Three independent procedures also improve our ability to troubleshoot early tests. Room calorimetry is probably the most accurate. The walls are made of 30cm thick polyurethane. There are five thermocouples on both sides of each wall, floor, and ceiling of each environmental chamber. Transmission losses are carefully calibrated so that error is within ±0.1% of capacity measured, all dry energy inputs (electric) are measured within ±0.2%. Special care is taken to ensure uniformity of the temperature and velocity profiles at the inlet to the heat exchangers, and representative reading of the exit air temperatures and humidities. Test results show agreement between the independently determined capacities to be within ±5%, primarily due to uncertainties in air-side calorimetry.

Table 1: Comparison of components for mobile system

System	Refrigerant	R134a	R744	R134a	R744
System	Name	MAC1HFC	MAC1	MAC2HFC	MAC2
Compressor:	Type	Reciprocating	Reciprocating	Reciprocating	Reciprocating
Compressor:	Displacement [cm^3]	155	20.7	164	variable
Expansion device		Orifice tube	Manual or back pressure valve	Orifice Tube	Manual or back pressure valve
Outdoor heat exchanger	Description	Wavy Al fins, round Al tubes, 21 pass, OD = 6mm	Microchannel, brazed Al tubes, 3-pass, parallel flow	Microchannel, 4-pass (9-8-5-4)	Prototype
Outdoor heat exchanger	Mass [kg]	2.0	2.3	3.5	
Outdoor heat exchanger	Face area [cm^2]	36.1 x 54.4 = 1964	36.8 x 53.0 = 1950	36.5 x 66.7 = 2434.6	35.5 x 60.7 = 2154.9
Outdoor heat exchanger	Core depth [cm]	2.2	1.65	3.175	1.905
Outdoor heat exchanger	Core volume [cm^3]	4320	3320	7730	4105
Outdoor heat exchanger	Air side surface [m^2]	7.2	5.2	8.4	6.8
Outdoor heat exchanger	Refrigerant side surface area [m^2]	0.40	0.49	--	0.71
Indoor heat exchanger	Description	Brazed Al plate (drawn cup, laminated), 4-pass, 17 plates	Microchannel, brazed, Al, 7-pass, parallel flow	Brazed Al plate, 1-pass, 18 plates	Prototype
Indoor heat exchanger	Mass [kg]	1.8	2.2	2.1	
Indoor heat exchanger	Face area [cm^2]	18.4 x 22.0 = 405	18.2 x 22.4 = 408	25.4 x 23.5 = 597	25.2 x 17.64 = 445
Indoor heat exchanger	Core depth [cm]	9.2	9.1	7.6	7.8
Indoor heat exchanger	Core volume [cm^3]	3720	3710	4537	3471
Indoor heat exchanger	Air side surface [m^2]	3.5	4.2	4.4	4.0
Indoor heat exchanger	Refrigerant side surface area [m^2]	0.55	0.66	--	0.94
Internal HX	Description	No	Al., coaxial tube, vapor in annulus, counter-flow, 1.5m	No	Prototype

Special care was taken to develop test facilities that will produce accurate data in wide operating ranges, primarily in steady-state but also in transient (mostly cycling) conditions. Coriolis mass flow meters, together with immersion thermocouples and electronic pressure transducers on one end and differential between inlet and exit of every component yield refrigerant-side capacity determinations repeatable within ±1%.

Compressor is placed in the separate chamber that simulates temperature conditions in the engine compartment. A torque meter is between the compressor and the clutch to measure power at the compressor shaft, excluding belt and clutch losses. The suction accumulator is located in the same enclosure, as it would be in the real system. Lines are as short as possible to be representative for real system even in cycling mode.

Two modes of operation (air conditioning and heat pump) are illustrated in Figures 1 (refrigerant piping as a part of the whole schematics) and Figure 2 (just refrigerant schematic in heat pump mode) for transcritical CO_2 operation only. More details about the facility could be found in Boewe et al. (1999b).

Figure 3 presents photos of all elements of the MAC1 system in the schematic of the heat pump configuration.

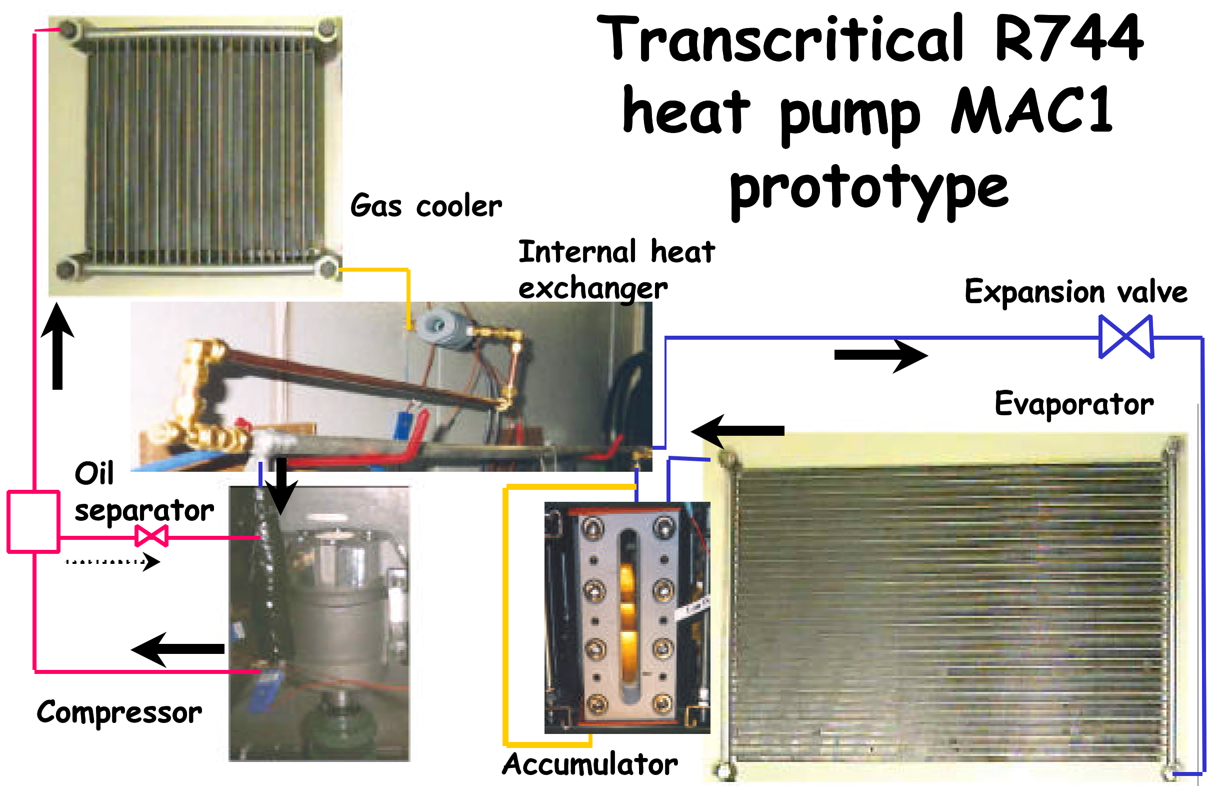

Figure 3 Elements of the first R744 heat pump system operating in transcritical mode

SOME EXPERIMENTAL RESULTS

The test matrix was designed to reflect quasi steady state conditions of vehicle heating up at moderately cold ambient weather. Ambient temperatures are varied in the range –10 to 20°C, indoor temperatures in the range –10 to 20°C, all for constant air flow rate (AFR) over the indoor coil (0.134 m^3/s), and the outdoor coil (0.434 m^3/s) and compressor speed 950 rpm representing idling condition. The evaporation temperature is established by the load/capacity equilibrium. Heat rejection pressure was varied by adjusting the expansion valve when needed. Figure 4 shows the effect of high side pressure on capacity, HPF (heating performance factor $HPF=Q_{heat}/W \sim COP+1$) and compressor work in heat pump mode. Data shown in Figure 5 are taken for the COP maximizing high side pressure or at just above critical pressures. It is clear that it is possible to get much greater capacity from the system by increasing high side pressure if needed. The effect of different operating parameters in a/c mode explored and discussed more in details in Park et al. (1999).

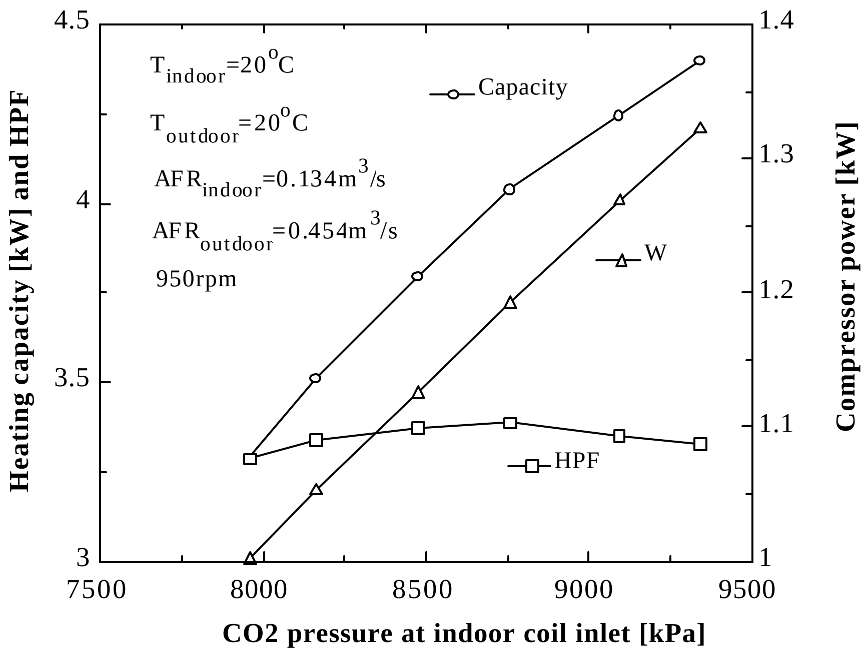

Figure 4. Effect of high side pressure on heat pump performance at a given condition

Figure 5 shows that both capacity and heating performance factor increase with the rise of ambient and indoor temperatures. However it is very important to notice both the open and closed triangles: their high location indicates that both capacity and heat performance factors are the highest when needed the most – at the start of the operation. That fact could have a crucial importance to passenger comfort because that is the operating conditions where conventional heating systems lack capacity. The fact that capacity could be further augmented by increasing operating pressure at insignificant reduction of HPF as shown in Figure 4 amplifies the benefits of heat pumping for increasing thermal comfort in the automobiles.

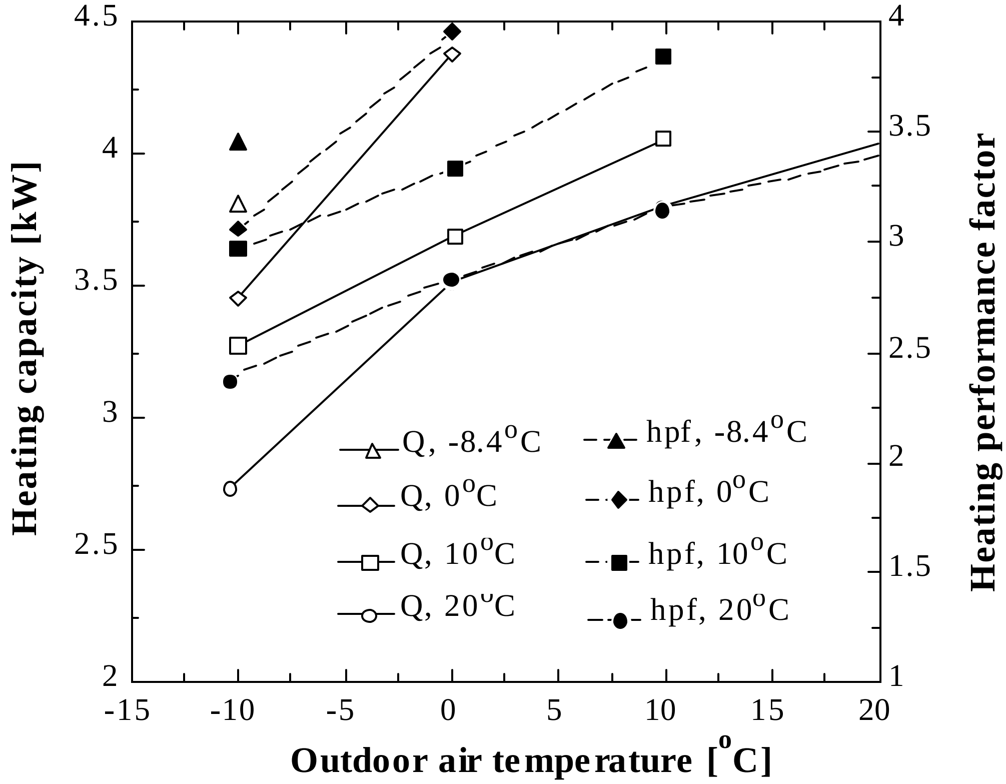

Figure 5. R744 heat pump performance (capacity and COP) at different indoor and outdoor conditions

Another important element in passenger's comfort is the temperature of air discharged to the compartment. Figure 6 indicates that even at the operating conditions shown (-10°C outdoor and -8.4°C indoor) air discharge temperature is warm and comfortable (16°C at least, at the cold startup). With increase of the high side pressure to increase capacity (as described above) air discharge temperatures rise further.

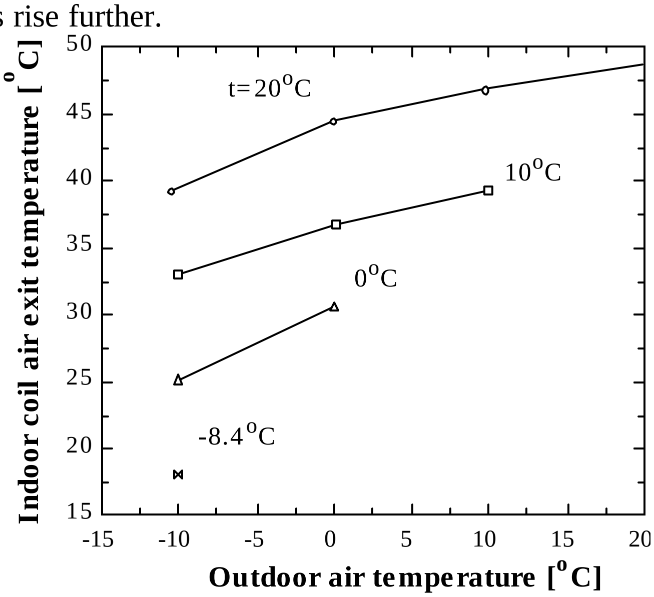

Figure 6. Air discharge temperatures as a function of ambient temperature for constant passenger compartment temperature (air inlet to the indoor coil). Even at cold starts promising potential for improving passenger's comfort

Figure 7 illustrates operation of the system showing the transcritical R744 cycle at one operating condition 10°C/10°C. There are both measured values (solid circles) and system modeling results indicated by solid lines. Triangles and squares represent the hot and cold sides of internal heat exchanger, bow tie gas cooler and solid square compressor suction determined by the model.

Besides being a good illustration it is also an indication of several possible improvements in the system performance by advancing heat exchangers and other elements. We are currently working on that issue.

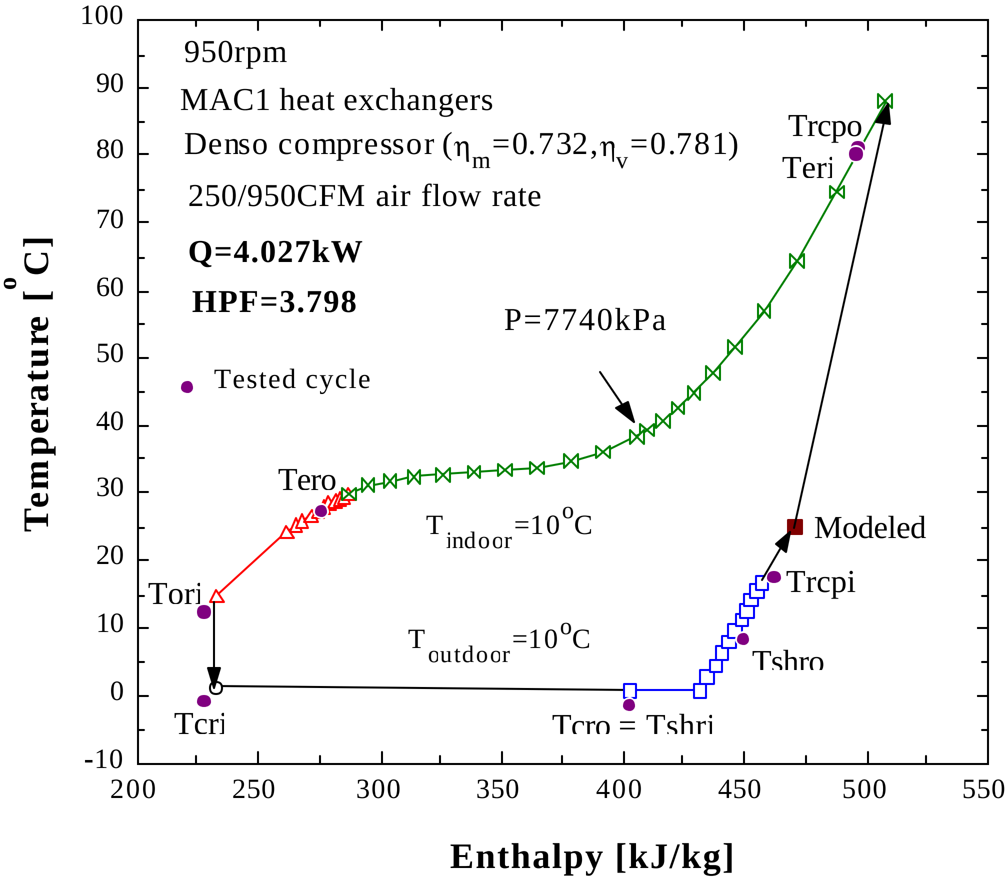

Figure 7. R744 cycle for operation at one condition (10°C/10°C)

Figure 8 depicts the compressor efficiency based on isentropic work needed to compress refrigerant flow and power on the shaft. Flow rate is measured by mass flow meter mr (Figure 1) while shaft power by the torque meter Tor. One should have in mind that some refrigerant bypass the flow meter with oil returned from the oil separator to the suction of the compressor.

All data shown are obtained with technically dry air to provide steady state operation. Air was dehumidified prior to measurements to close to 0°C dew point temperature. Final water removal is done on the glycol coils. We are aware of potential problems in the heat pump mode when operating at low evaporation temperature in humid environment. This is one of the reasons, besides desire to explore distribution issues, why we have run some experiments at higher humidity. Figure 9 presents the photo of the outdoor coil while running in frosting condition. Our group is continuing exploring this issue further and more in depth.

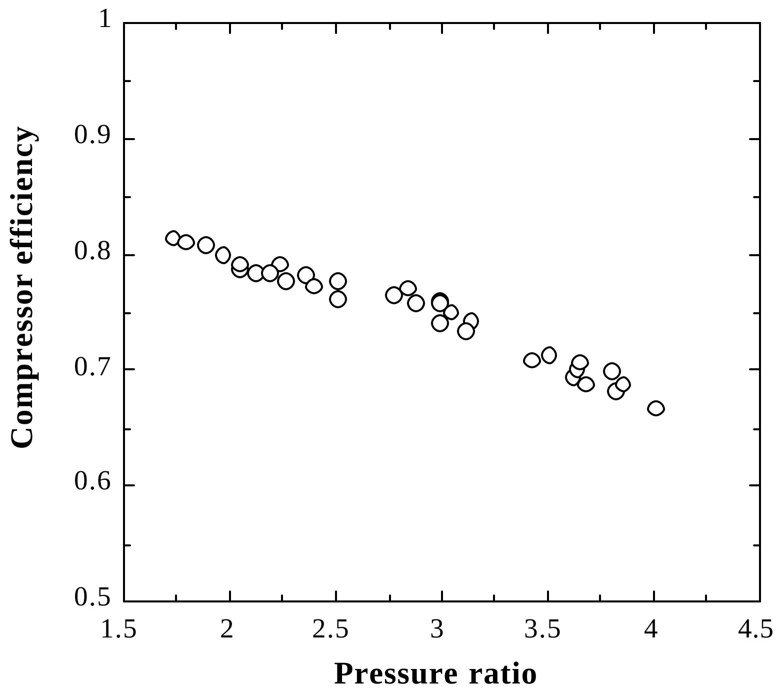

Figure 8. Compressor efficiency in heat pump mode

Figure 9. Outdoor coil frosting indicates imperfect operation (maldistribution) and problems in operation at high ambient humidity

Figure 10 presents data for the same system in the air conditioning mode. It is shown just for comparison reasons. More detailed results are described in earlier publications by the same group, Yin et al. (1998), Boewe et al. (1999a,b), Beaver et al. (1999a, b), etc.

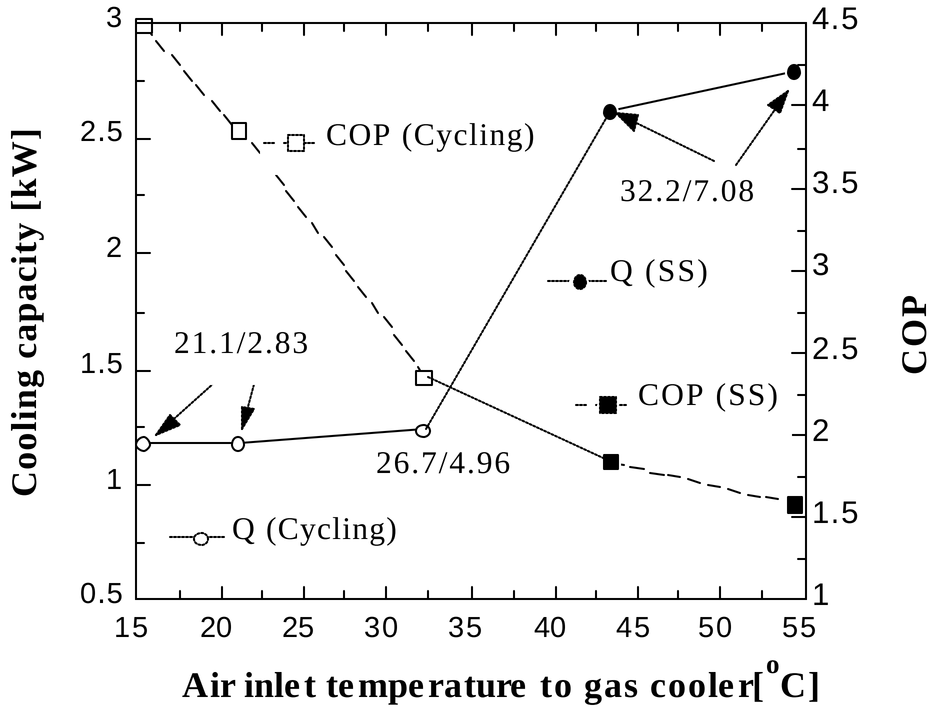

Figure 10 Performance of the same system in a/c mode. Compressor speed is 950 rpm. Non cycling data are shown as SS (steady state). First number is indoor air temperature [°C] and second is air flow rate [m³/min]. Airflow rate over the gas cooler was 22.7m³/min.

CONCLUSION

This paper presents first results in heat pump operation of an ongoing project in transcritical R744 systems for mobile air conditioning and heat pump application. It is to our best knowledge the first report about the heat pump operation of the mobile system with transcritical CO_2 in the open literature. Data show significant heating capacity at relatively low ambient temperatures. Maybe even more important are the facts that capacity is not significantly reduced at low operating temperatures and that capacity at heating up transient is actually higher. This is extremely important – we have capacity exactly when needed.

ACKNOWLEDGMENT

We are grateful to US DoE, US Army CECOM, Hydro Aluminum A.S. and Delpi Harrison for funding this research.

CONTACT

Pega Hrnjak, Adj. Associate Professor, Associate Director ACRC, University of Illinois at Urbana-Champaign, 1206 W. Green St. Urbana, IL 61801, USA, e-mail: pega@uiuc.edu.

REFERENCES

Bhatti, M.S, (1997) A critical look at R-744 and R-134a mobile air conditioning systems, SAE paper 970527, SAE Congress Proceeding, pp. 117-141

Pettersen, J., R. Aarlien, P. Neksaa, G. Skaugen and K. Aflekt, (1997a). A comparative evaluation of CO_2 and R22 residential air-conditioning systems in a Japanese climate, IIR/IEA Workshop in CO2 technology in refrigeration, heat pumps and air conditioning systems, Tronheim, Norway

Pettersen, J. and Hafner, A. (1997b). Energetischer Wirkungsgrad und TEWI von CO_2-Fahrzeug-Klimaanlagen. Seminar: Fahrzeugklimatisierung mit Naturlichen Kaetlemitteln, Karlsruhe March 8, C.F. Mueller Verlag

Pettersen, J., (1994) An efficient new automobile air-conditioning system based on CO2 vapor compression, ASHRAE Transactions, paper OR-94-5-3, pp. 657-666

Lorentzen, G. and Pettersen, J. (1993) A New, Efficient and Environmentally Benign System for Car Air-Conditioning, Int. Journal of Refrigeration, Vol. 16, No1, pp 4-12

Yin, J. M., Y. C. Park, R. P. McEnaney, D. E. Boewe, A. Beaver, C. W. Bullard, P. S. Hrnjak, (1998). Experimental and model comparison of transcritical CO_2 versus R134a and R410A system performance. IIR conference Gustav Lorentzen, Oslo, Proceedings, Preprints, pp. 331-340

Boewe, D., J. M. Yin, Y. C. Park, C. W. Bullard, P. S. Hrnjak. (1999a). The role of the suction line heat exchanger in transcritical R744 mobile a/c systems. SAE paper 1999-01-0583

McEnaney, R., Y. C. Park, J. M. Yin, C. W. Bullard, P. S. Hrnjak (1999a). Performance of the prototype of a transcritical R744 mobile a/c system. SAE paper 1999-01-0872

Park, Y. C., J. M. Yin, C. W. Bullard, P. S. Hrnjak, (1999). Experimental and model analysis of control and operating parameters of transcritical CO_2 mobile A/C system. VTMS, London, May 1999

McEnaney, R.P., P.S.Hrnjak, (2000) Control strategies for transcritical R744 systems, SAE 2000 World Congress, paper 2000-01-1272

Beaver, A., J.M.Yin, C.W. Bullard, and P.S.Hrnjak , (1999a) Experimental and model study of the heat pump/air conditioning systems based on transcritical cycle with R744, to appear in Proc. XXth Congress of Refrigeration, Sidney Australia

McEnaney, R.P., D.E.Boewe, J.M.Yin, Y.C.Park, C.W.Bullard, and P.S.Hrnjak, (1998b) Experimental comparison of mobile A/C systems when operated with transcritical CO_2 versus conventional R134a, Purdue Conference Proceedings, pp. 145-150

Beaver, A.C., J.M.Yin, C.W.Bullard, P.S. Hrnjak, (1999b) An Experimental Investigation of Transcritical Carbon Dioxide Systems for Residential Air Conditioning, ACRC report CR-18, Urbana

McEnaney, R.P., J.M.Yin, C.W.Bullard, P.S. Hrnjak, (1999c) An Investigation of Control-Related Issues In Transcritical R744 and Subcritical R134a Mobile Air Conditioning Systems, ACRC report CR-19, Urbana

Boewe, D.E., R.P.McEnaney, Y.C.Park, J.M.Yin, C.W.Bullard, P.S. Hrnjak, (1999b) Comparative Experimental Study of Subcritical R134a and Transcritical R744 Refrigeration Systems for Mobile Applications, ACRC report CR-17, Urbana, IL