CARBON DIOXIDE AS THE REPLACEMENT FOR SYNTHETIC REFRIGERANTS IN MOBILE AIR CONDITIONING

by

Dragi Lj. ANTONIJEVIĆ

Original scientific paper
UDC: 66.045.5:621.5.02/.04
BIBLID: 0354-9836, *12* (2008), 3, 55-64
DOI: 10.2298/TSCI0803055A

Based on Kyoto Protocol and the decisions of European Commission R134a refrigerant, currently dominantly used in mobile air conditioning systems, needs to be phased-out. At present automotive industry looks at carbon dioxide (CO_2; R744) as the refrigerant of the future. Apart from the environmental benefits discussed are the technical characteristics of carbon dioxide refrigeration cycle and mobile air-conditioning systems in comparison to R134a refrigerant. Analyzed are challenges emerged from the use of CO_2 as refrigerant and improvement opportunities in regards to increase of the system performance and efficiency. Particular attention is dedicated to the advantages of CO_2 utilization in prospective automotive heat pump systems.

Key words: *refrigerant, global warming potential, R134a, CO_2 (R744), transcritical cycle, air conditioning, heat pump*

Introduction

Carbon dioxide was already used as refrigerant in the beginning of the last century. In 1940s it retreated in front of the fluorocarbon chemicals that could produce good cooling cycle characteristics by much lower pressure levels in the system. In early 1990s, with increased awareness of environmental harmfulness of fluorocarbons and progress in high strength aluminum alloys development, an interest for using natural refrigerants, in particular CO_2 due to its favorable thermodynamic and transport properties [1-6], reapeared [7-9].

After the phase-out of CFC (chloro-fluoro-carbon) based refrigerants R11, R12, R13, R113, R114 , R115, and HCFC (hydro-chloro-fluoro-carbon) based R22, due to the Montreal Protocol, the search towards finding an environmentally friendly refrigerant led to the development of HFC (hydro-fluoro-carbon) based refrigerants. The most common and since year 1992 in mobile air conditioning systems almost exclusively used HFC refrigerant is R134a.

The Kyoto Protocol included fluorinated compounds, perfluorocarbons and hydrofluorocarbons in the list of greenhouse gases and hence these gases were subjected to the emission reduction requirements. Additionally, in year 2006 European Commission decided that from year 2011 maximal cumulative global warming potential (GWP) of refrigerant used in mobile air conditioning must not exceed 150 for all new types of vehicles launched onto the European Union market. Consequently, since GWP of HFC refrigerants is far above this level, and GWP of R134a as the refrigerant approved by all automotive air conditioning manufacturers worldwide is about 1300, its use in modern car air-conditioning systems needs to be phased out.

Environmental aspects

The increased environmental relevance of the mobile air conditioning is apparent in regard to the fact that during the last years mobile air conditioning has experienced its renaissance – the participation of new passenger cars with air conditioning systems sold in European Union in year 1990 was under 10% whilst in year 2008 it is expected to be over 90%.

Since successful Montreal Protocol ban of CFC and HCFC refrigerants, the main environmental concern in regard to processes and technologies using refrigerants has become the influence on global warming. Total European fluorinated gas emissions in year 1990 amounted 650 million tones of CO_2 equivalent, which was about 12% of total annual emissions. In year 2010 fluorinated gas emissions are forecast to reach 1020 million tonnes and one of the most contributing sectors would be mobile air conditioning with 22 million tones of CO_2 equivalent [10-14].

Total mobile air conditioning emissions consist of direct and indirect emissions. Direct emissions during the system lifetime are due to refrigerant handling, fugitive emissions from valves and fittings, circuit openings for repair or service reasons, ruptures of safety valves or components of the system, and end-of-life disposal. Average lifetime leakage of R134a are estimated in the range from 1.6 to 2.2 tones of CO_2 equivalent per car. It is important to note that refrigerant leakage from mobile air conditioning systems is relatively higher than from the stationary systems, primarily on behalf of significant vibration, utilization of the flexible hoses and direct compressor drive. Indirect emissions are the consequence of the increased fuel consumption in air conditioned vehicles. Depending on size and type of the vehicle, ambient and driving conditions, *etc*, air conditionning system might increase the fuel consumption of the vehicle up to 20%.

In the last years industry has been trying to find an adequate substitute for R134a refrigerant. Numerous possibilities have been examined: from improving R134a system in the direction of better sealing, end-of-life, *etc*, to inventig novel syntetic refrigerants with favourable environmental, thermophysical, and thermodynamical properties. The most seriously observed refrigerant alternatives and their relevant environmental properties – GWP, ozone depletion potential (ODP), acute toxicity exposure level (ATEL), and time weighted average amount of a substance man can be exposed to over an eight-hour day (TWA) – are shown in tab. 1.

It is obvious that CO_2 refrigerant (R744) has the most favorable environmental parameters among all the alternatives. It is non-flammable and non-toxic, and has GWP of only 1, *vs.* 1300 by R134a. Also important is the fact that it is not necessary to capture CO_2 refrigerant during the air conditioning (AC) system refilling or reparation and at the end-of-life, which simplifies handling and allows certain savings.

Table 1. Environmental properties of some refrigerants

Refrigerant	R134a	R152a	R290	Blend H	R744
GWP* [–]	1300	140	11	<150	1
ODP* [–]	0	0	0	0	0
ATEL [ppm]	50000	50000	50000	tbd	40000
TWA [ppm]	1000	1000	1000	tbd	5000
Maximal operating pressure [MPa]	2.5	2.5	2.5	2.5	14

*References: GWP = 1 for R744, ODP = 1 for R12; tbd – to be determined

CO_2 refrigeration cycle and air conditioning system features

The property that decisively defines behavior of CO_2 in a refrigeration cycle is its low critical temperature of 31.1 °C, by critical pressure of 7.38 MPa. On that account CO_2 vapor compression cycle in air conditioning and heat pump ambient heating operation works usually partly above the critical point and at much higher pressures than R134a cycle operated by the same temperature levels (fig. 1). Consequently, the process of heat rejection is not any more predominately condensation, but supercritical vapor cooling with significant gaseous refrigerant temperature decrease inside "gas cooler" heat exchanger. The low pressure side of the cycle remains subcritical evaporation.

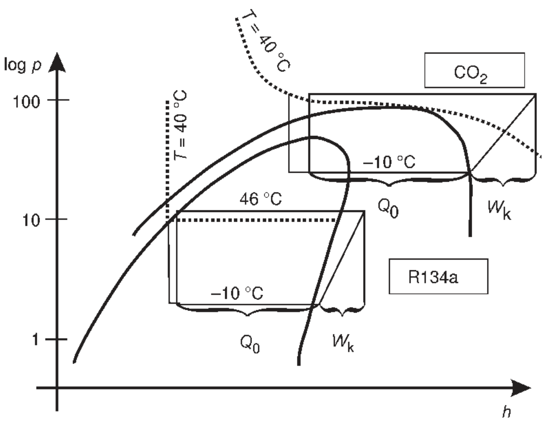

Figure 1. Typical cooling cycles of R134a and CO_2 in pressure-enthalpy diagram

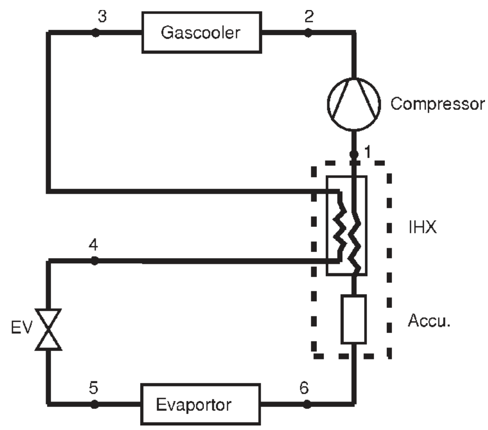

Figure 2. Layout of CO_2 air conditioning system with internal heat exchanger

For high overall air conditioning system efficiency it is vital to enter evaporator with wet vapor with highest possible potential to extract heat from the cooled air. Nevertheless, R744 air conditioning system shows certain efficiency drop by higher ambient temperatures, dictated mainly by the shape of isotherms in usual gascooler exit temperature range prohibiting effective convective heat transfer [15, 16]. Therefore it is necessary to introduce another component – internal heat exchanger (IHX in fig. 2), whose principal role is to cool high pressure refrigerant after exiting gas cooler (from point 3 to 4 in fig. 3) on behalf of heating low pressure refrigerant before entering compressor (from point 6 to 1) [16-18].

The internal heat exchanger is often designed as extruded aluminum counter current flow ("coaxial tube" heat exchanger, and might be a separate component, than taking over a part of the refrigerant transport role in the system [19], or embedded in the refrigerant accumulator for improved package reasons [20].

For given ambient temperature R744 air conditioning system capacity increases with discharge pressure increase, but the coefficient of performance (*COP*) is a function of both gascooler exit temperature and discharge pressure and it shows a clear optimum (fig. 4). Since overall cooling performance and *COP* of transcritical R744 cycle are very sensitive to gas cooler exit temperature, design of that component is of primary importance. The refrigerant temperature glide in the gascooler, enables achieving of an optimal cycle efficiency at finite air flow rate, in contrast to the infinite air flow required in the subcritical cycle air conditioning and should be utilized by countercurrent flow configuration [15].

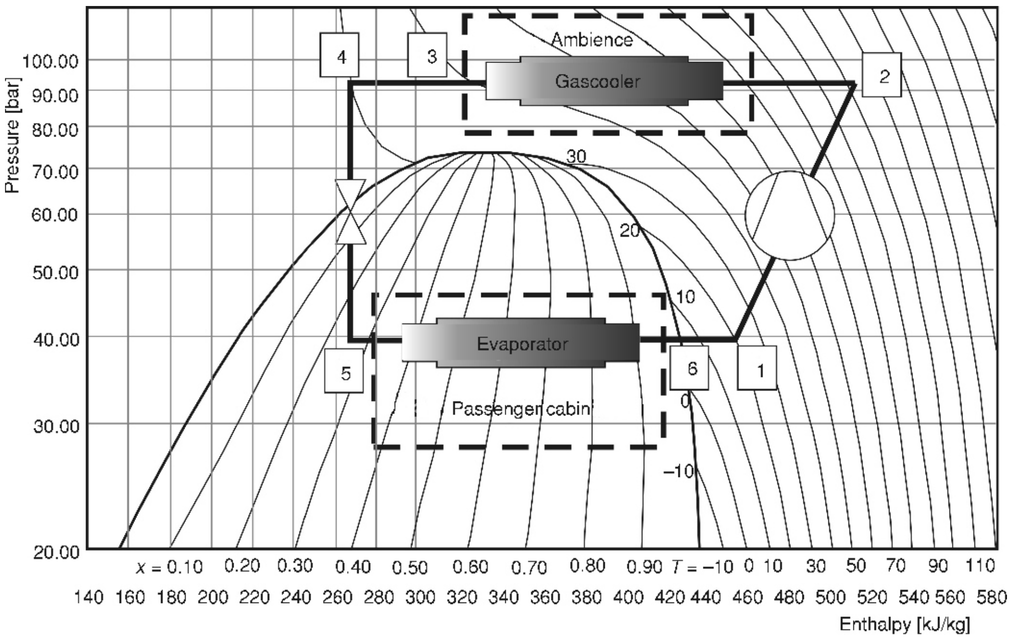

Figure 3. Pressure-enthalpy diagram of the system with internal heat exchanger

In order to manage highly efficient heat exchange in gas cooler, multiport tube and fin technology is used. Refrigerant tubes, produced by aluminum extrusion, have rows of micro channels with diameter of about 0.5 mm optimized to enhance heat transfer and provide enough strength to withstand high operating pressures in the system. Such CO_2 refrigerant tubes have thinner air side profile than R134a system tubes (fig. 5), offering thereby a possibility for larger heat transfer surface at the air side where the predominant amount of heat transfer losses of the heat exchanger is generated in most of the operating regimes. Thin microport tube in combina-

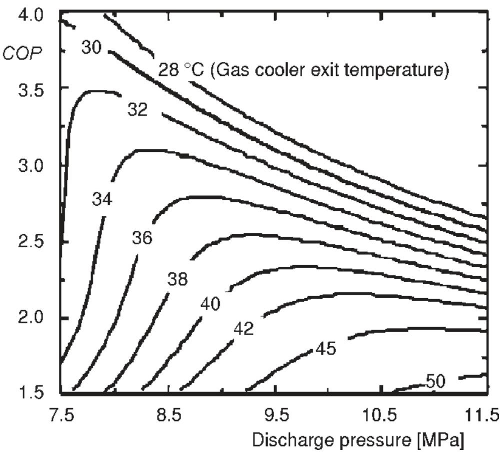

Figure 4. R744 cycle *COP* for different discharge pressures and gas cooler exit temperatures (by evaporation temperature of 3.9 °C and IHX efficiency of 0.8) [15]

Figure 5. Air side segment of microport tube / louver fin R744 heat exchanger

tion with multilouvered fins of optimized geometry, pitch and height, tends to be the optimal solution for both gascooler and evaporator of the CO_2 system [21-23]. Additionally, for highly efficient gascooler it is important to utilize optimized refrigerant distribution and circuitry and to prohibit internal heat losses by conduction [24].

In general, R744 heat exchangers offer more capacity than the R134a components of the same size and possess potential for further improvement in terms of enhancement of air side heat transfer through optimization of fin geometry for critical operating conditions, developing advanced refrigerant circuitry and becoming more package friendly (gascooler); as well as improving two phase refrigerant distribution to achieve full potential of high boiling heat transfer rates inside the micro channels, and elaborating air-side condensate retention/drainage management (evaporator).

In comparison to R134a air conditioning system *COP* of R744 system decreases faster with increase of ambient temperature and might even drop under *COP* of R134a by very high ambient temperatures. This is especially the case if the engine, and consequently compressor, work in low rpm regime [17, 25, 26]. Nevertheless, when tested back to back with R134a system, for the same working and boundary conditions, in the same vehicle and for the same components sizes, CO_2 air conditioning system shows clear supremacy in performance.

Figure 6 shows the results of performance tests of R134a and R744 air conditioning systems presented by company Visteon [27], performed for small, 1000 cm^3 engine vehicle, at ambient temperature of 43 °C and humidity of 40%, for two different driving speeds and for idling operation. It is obvious that, even by such harsh ambient conditions, achieved air temperature levels at evaporator discharge and in the passenger compartment, are better when CO_2 air conditioning system is used. For example, after 10 minutes of pull-down, average temperature at evaporator exit acheived with CO_2 AC system was 6 °C lower than the temperature provided by R134a system.

Seeing that working pressure levels in R744 system are much higher than in R134a system, usually in the range 3-11 MPa, all the components need to be designed in the novel way

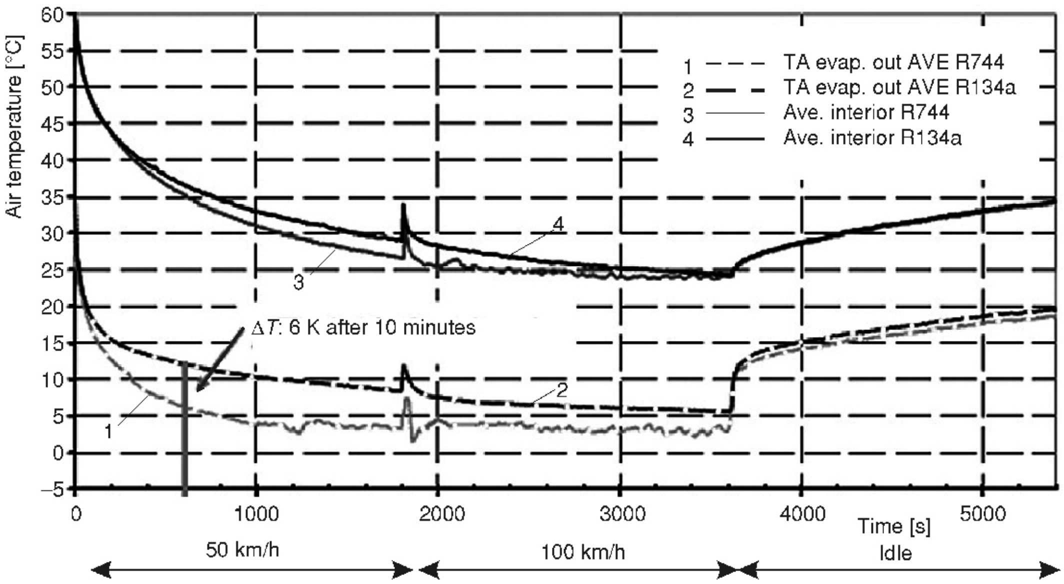

Figure 6. Back-to-back performance test results of R134a and CO_2 AC systems

to withstand and benefit from the higher pressure. Material strength as well as the wall thickness at CO_2-air boundaries must be increased compared to R134a components. Nevertheless, volume of the components and piping is smaller and therefore the stored explosion energy is not much higher than in R134a system. Utilization of extruded micro port tubes in heat exchangers is also beneficial for the components structural strength.

At present level of R744 air conditioning system development, the fuel consumption for the given system performance is already lower than with R134a system [27]. R744 compressor has lower operating pressure ratios and consequently better efficiency then R134a compressor. The size and the weight of the component are smaller. For automotive air conditioning applications single stage swash or scroll compressor technology are predominantly used. Further developments in automotive R744 AC system performance and efficiency are closely connected with system architecture and sustainable cost structure and would be probably, in parallel to the further optimizing of compressor design and operation, transcritical cycle control strategies, lubricants and refrigerant charge, examined in the area of add work recovery by use of expanders and ejectors [26, 28, 29].

Heat pump heating possibility

Commonly, passenger compartment air is heated by waste engine heat in a coolant-to-air heat exchanger. With recent developments in high efficiency engines with very low heat rejection rates, the waste heat available is not sufficient for achievement of fast heat-up and comfortable passenger compartment temperature. In current production vehicles the required comfort level is accomplished by using supplementary heating devices, such as “PTC Heater” (electrical air heater), “Glow Plug Heater” (electrical coolant heater) or “Fuel Fired Coolant Heater”, adding the system cost and overall fuel consumption.

Due to its thermophysical characteristics CO_2 is much more convenient for utilization in a heat pump heating system than R134a. The high compressor discharge temperatures and relatively high density even by very low temperatures provide ample refrigerant mass flow through the system and the efficient operation. The heating capacity is several times higher than by the electrical supplementary heating devices and highest at the start of operation when needed most. The high passenger compartment inlet air temperatures are achieved almost instantly. During the operation heat pump capacity and efficiency decrease slowly due to reduced compressor isentropic efficiency at higher temperature lift, as the passenger compartment warms up and heat becomes available from the engine coolant.

The automotive heat pump system uses principally the components of the existing air conditioning system together with some additional valves and lines. In the simplest lay-out, the evaporator of AC system is used as the gas cooler and auxiliary heater of the heat pump. The issue with such system architecture appears when the evaporator of AC system, that remains wet from earlier AC operation, is switched to the heating mode. The hot and humid air stream is then send onto cold windshield surface – fogging it instantly (“flash fogging”), which might have safety implications. Therefore modern automotive heat pump concepts usually include secondary – additional heating only – gascooler (fig. 7). That heat exchanger is much smaler in dimensions than standard AC evaporator and might be designed as a separate component or combined, for cost and package reasons, with the glycol heater core in a “three fluid heat exchanger” [30, 31].

As a secondary heat source for an automotive heat pump it is possible to use exhaust gas, engine coolant or ambient air. The amount of heat available in the exhaust gas depends on the engine load. In city traffic conditions, when substantial supplementary heating is required, exhaust heat is at its low and limits the heating performance of the heat pump.

Utilizing the heat of the engine coolant as a secondary heat source has the advantage that coolant temperature increases rapidly during the vehicle warm up, providing highly efficient heat pump operation. The problem is that the engine coolant can not be easily used in the heater core for simultaneous heating of the passenger compartment.

Using of ambient air as the heat source has the best thermodynamic justification since the heat is extracted from the external source – surrounding air. There is no necessity for additional heat source heat exchanger, since gascooler, working then in evaporation mode, is used as air-to-refrigerant heat exchanger. Additionally, the conventional heater core can be used in the usual way for passenger compartment heating once the comfort level is reached.

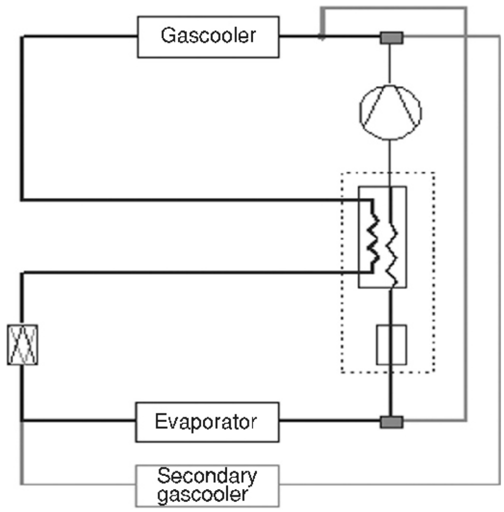

Figure 7. Automotive heat pump with air as heat source and secondary gascooler

Potential disadvantage of ambient air used as the heat source is frost formation in the air-to-refrigerant heat exchanger under certain ambient and working conditions. However, in normal operation heat pump should only provide supplemental heating for a short period after start-up while the engine and coolant are warming up. In the experiments with air-to-air heat pumps built in test vehicles, no serious issues were found in regards to the system functionality or unacceptable performance degradation [32-34]. The investigation work on optimal defrosting strategies and the other technical details of the system with air as heat source, like influence of ice formation on the mechanical characteristics and durability of air-to-refrigerant heat exchanger, is moving ahead and shortly after the start of serial production of first mobile CO_2 air conditioning systems the heat pump functionality should be expected in the passenger vehicles.

System cost

Introduction of R744 air conditioning system into mass produced vehicles requires substantial and extensive development of a completely new air conditioning system with new components, new circuiting, and new controls. Higher operating pressures of R744 than R134a system impose increased components wall thickness. Nevertheless, the volume of the components decreases. The fact that CO_2 system is less sensitive to refrigerant pressure drop than R134a system, allows further weight and package improvements, especially in piping and heat exchanger manifolds, as well as optimization of strength/weight ratio of microport tubes used in heat exchangers. Additionally, developments in manufacturing technologies have already reduced diameter of microports contributing thus to decrease of the heat exchangers weight and cost.

In the moment CO_2 system is likely to have higher production cost than R134a system but, as every new technology, it tends to decrease the costs over the time (fig. 8). If the weight of the aluminum components of R774 system drop further due to the progress in design and manufacturing, and the expenses for sensors, actuators, and controls together with the costs of mass production keep their decreasing trend, it is to believe that in 2-3 years the production cost of R744 system will reach the cost of R134a system.

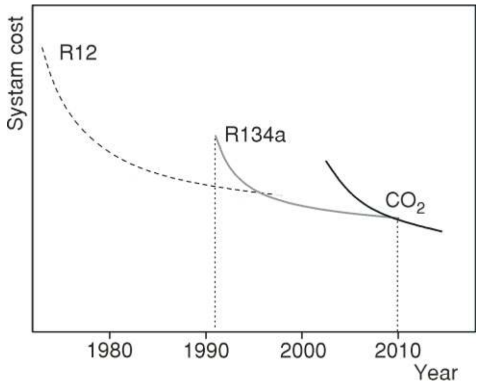

Figure 8. Automotive air conditioning system cost over the years

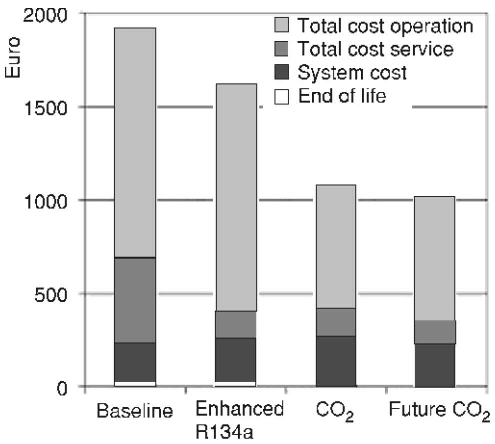

Figure 9. Total automotive air conditioning system cost during the lifetime

In case of the total lifetime costs assessment for R134a air conditioning system, the expenses involved in improvements to reduce leakage during manufacturing and system operation, as well as the expenses for the disposal of the system need to be taken into account (fig. 9). Presently the sealing of R134a system is not ideal and the refrigerant leakage through rubber hoses, fittings, *etc.* need to be compensated, adding to the total service costs. CO_2 system has better sealing, the refrigerant leakage is less harmful and the fluid itself has a low cost and no tax burden worldwide. With CO_2 as the working fluid the refrigerant recycling is not needed, so there is also no end-of-life costs associated.

In case of R744 system with heat pump functionality, further trade offs and potential cost savings in the vehicle supplemental heating system are expected.

Conclusions

Use of carbon dioxide as refrigerant in mobile and stationary air conditioning offers a sustainable solution compatible with future environmental limits on refrigerants. Since CO_2 is a natural substance nearly innoxious for environment there are no environmental concerns about its leakage during air conditioning systems production, operation, and service. There is also no necessity to capture and recycle the refrigerant at the end-of-life.

On the subject of thermodynamics and technical aspects, CO_2 automotive air conditioning system has reduced pull-down time, better efficiency and offers potential for lower fuel consumption compared to the existing R134a system. The components of the state-of-the-art CO_2 system have already smaller dimensions than R134a components and are expected to be further improved in size, efficiency, and cost. Thermodynamic and transport properties of CO_2 and the characteristics of its transcritical cycle are particularly suitable for heat pump heating operation. Integrating the heat pump function into the automotive climate control system is relatively simple, yet radically improves comfort and eliminates need for supplemental heating devices even in coldest climates.

Acknowledgment

The author would like to thank Climate Control Systems Development, Visteon Corporation (Vorentwicklung Klimatisierung, Visteon Deutschland GmbH) for the years of successful cooperation.

References

[1] Vesovic, V., *et al.*, The Transport Properties of Carbon Dioxide, *Journal of Physical and Chemical Reference Data, 19* (1990), 3, pp. 763-808

[2] Choi, K. I., *et al.*, Boiling Heat Transfer of R-22, R-134a and CO_2 in Horizontal Smooth Minichannels, *International Journal of Refrigeration, 30* (2007), 8, pp. 1336-1346

[3] Pettersen, J., Rieberer, R., Leister, A., Heat Transfer and Pressure Drop Characteristics of Supercritical Carbon Dioxide in Microchannel Tubes under Cooling, *Proceedings,* 4th IIR Gustav Lorenzen Conference on Natural Working Fluids, West Lafayette, Ind., USA, 2000, pp. 99-106

[4] Pitla, S., *et al*., Convective Heat Transfer from in-Tube Flow of Turbulent Supercritical Carbon Dioxide, Part 1: Numerical Analysis, *International Journal HVAC&R Research, 7* (2001), 4, pp. 345-366

[5] Pitla, S., Groll, E., Ramadhyani, S., Convective Heat Transfer from in-Tube Flow of Turbulent Supercritical Carbon Dioxide, Part 2: Experimental Data and Numerical Predictions, *International Journal HVAC&R Research, 7* (2001), 4, pp. 367-382

[6] Liao, S., Zhao, T., Measurements of Heat Transfer Coefficients from Supercritical Carbon Dioxide Flowing in Horizontal Mini/Meso Channels, *Journal of Heat Transfer, 124* (2002), 3, pp. 413-420

[7] Lorentzen, G., Pettersen, J., New Possibilities for Non-CFC Refrigeration, *Proceedings,* IIR International Symposium on Refrigeration, Energy and Environment, Trondheim, Norway, 1992, pp. 147-163

[8] Lorentzen, G., Revival of Carbon Dioxide as a Refrigerant, *International Journal of Refrigeration, 1*7 (1993), 5, pp. 292-301

[9] Pettersen, J., Skaugen, G., Operation of Transcritical CO_2 Vapour Compression Systems in Vehicle Air Conditioning, *Proceedings,* International Conference on New Applications of Natural Working Fluids in Refrigeration and Air Conditioning, Hanover, Germany, 1994, pp. 495-505

[10] Billiard, F., The New European Regulation on Substances that Deplete the Ozone Layer Has Come into Force: What's New?, *International Journal of Refrigeration, 24* (2001), 3, pp. 205-207

[11] Memory, S. *et al.*, Vehicular CO_2 AC System, *Proceedings on CD Rom*, 5th Vehicle Thermal Management Systems Conference, Nashwille, Tenn., USA, 2001

[12] Schwartz, W., R134a Emissions from Passenger Car Air Conditioning Systems, *Proceedings on CD Rom*, VDA Alternate Refrigerant Winter Meeting, Saalfelden, Austria, 2002

[13] Pearson, A., Carbon Dioxide New Uses for an Old Refrigerant, *International Journal of Refrigeration, 28* (2005), 8, pp. 1140-1148

[14] Neksa, P., Wolf, F., R744 the Global Solution, Advantages and Possibilities, *Proceedings on CD Rom*, VDA Alternate Refrigerant Winter Meeting, Saalfelden, Austria, 2007

[15] Kim, M. H., Pettersen, J., Bullard, C., Fundamental Process and System Design Issues in CO_2 Vapour Compression Systems, *Progress in Energy and Combustion Science, 30* (2004), 2, pp. 119-174

[16] Boewe, D., *et al*., Contribution of Internal Heat Exchanger to Transcritical R744 Cycle Performance, *International Journal HVAC&R Research, 7* (2001), 2, pp. 155-168

[17] Hrnjak, P., Design and Performance of Improved R744 System Based On 2002 Technology, *Proceedings on CD Rom*, SAE Automotive Alternate Refrigerant Systems Symposium, Scottsdale, Ariz., USA, 2003

[18] Chen, Y., Gu, J., The Optimum High Pressure for CO_2 Transcritical Refrigeration Systems with Internal Heat Exchangers, *International Journal of Refrigeration, 28* (2005), 8, pp. 1238-1249

[19] Antonijević, D., Froehling, J., Multichannel Heat Exchanger and Connection Unit (in Germany), Deutsches Patent- und Markenamt, DE10303595B4, 2005

[20] Dickson, T., Whittle, W., Stobbart, M., Internal Heat Exchanger Accumulator, US patent US6463757, 2004

[21] Petersen, J., *et al*., Development of Compact Heat Exchangers for CO_2 Air-Conditioning Systems, *International Journal of Refrigeration, 21* (1998), 3, pp. 180-193

[22] Antonijević, D., Hoffmann, H., Multichanel Flat Tube for Heat Exchanger (in German), Deutsches Patent und- Markenamt, DE102005052683.7, 2005

[23] Antonijević, D., Fin for Heat Exchanger with Flat and Parallel Refrigerant Tubes (in German), Deutsches Patent- und Markenamt, DE10360240B4, 2005

[24] Antonijević, D., Multiple Flow Heat Exchanger (in German), Deutsches Patent- und Markenamt, DE102006017434A1 (WO2007/014560), 2007

[25] Brown, J., Yana-Motta, S. F., Domanski, P. A., Comparitive Analysis of an Automotive Air Conditioning Systems Operating with CO_2 and R134a, *International Journal of Refrigerationm, 25* (2002), 1, pp. 19-32

[26] Hrnjak, P., Tehnological and Theoretical Opportunities for Further Improvement of Efficiency and Performance of the Refrigerant Candidates, *Proceedings on CD Rom*, VDA Alternate Refrigerant Winter Meeting, Saalfelden, Austria, 2007
[27] Wiesholek, F., Heckt, R., Improved Efficiency for Small Cars with R744, *Proceedings on CD Rom*, VDA Alternate Refrigerant Winter Meeting, Saalfelden, Austria, 2007
[28] Nickl, J., *et al*., Integration of a Three-Stage Expander into a CO_2 Refrigeration System, *International Journal of Refrigeration, 28* (2005), 8, pp. 1219-1224
[29] Elbel, S., Hrnjak, P., Experimental Validation of a Prototype Ejector Designed to Reduce Throttling Losses Encountered in Transcritical R744 System Operation, *International Journal of Refrigeration, 31* (2008), 3, pp. 411-422
[30] Heckt, R., Antonijević, D., Heating Heat Exchanger (in German), Deutsches Patent- und Markenamt, DE10313234A1, 2004
[31] Heckt, R., Antonijević, D., Heat Exchanger Assembly, US Patent Application Publication, US20040231825A1, 2004
[32] Antonijević, D., Heckt, R., Heat Pump Supplementary Heating for Motor Vehicles, *Journal of Automobile Engineering, 218* (2004), 10, pp. 1111-1117
[33] Heckt, R., CO_2 Heat Pump Optimized for Fuel Economy, *Proceedings on CD Rom*, VDA Alternate Refrigerant Winter Meeting, Saalfelden, Austria, 2004
[34] Tamura, T., Yakumaru, Y., Nishiwaki, F., Experimental Study on Automotive Cooling and Heating Air Conditioning System Using CO_2 as a Refrigerant, *International Journal of Refrigeration, 28* (2005), 8, pp. 1302-1307

Author´s address:

D. Lj. Antonijević
Singidunum University, Faculty for Applied Ecology
12a, Bulevar Mihaila Pupina, Belgrade, Serbia

E-mail: dantonijevic@singidunum.ac.yu

Paper submitted: March 4, 2008
Paper revised: July 7, 2008
Paper accepted: August 31, 2008