ANNEX 1

Analysis of scientific literature

Ref.	Authors	Title	Source
[McE98]	R.P McEnaney, D.E Boewe, J.M Yin, Y.C Park, C.W Bullard, P.S Hrnjak	Experimental comparison of mobile A/C systems when operated with transcritical CO_2 versus conventional R-134a	1998 International Refrigeration Conference at Purdue
[ACR02]	ACRC	SAE alternate refrigerant Cooperative Research Project	SAE cooperative research program. November 2002
[BRO02]	Brown J.S., Yana-Motta S., F., Domanski P.	Comparative analysis of an automotive air conditioning systems operating with CO_2 and R-134a	International Journal of Refrigeration. 25 (2002) 19-32
[GEN98]	H. Gentner, A Földi	Passenger car air conditioning using carbon dioxide as refrigerant	IIF-IIR Section B and E Oslo, Norway. 1998
[PRE00]	M. Preissner, B. Cutler, S. Singanamalla, Y. Hwang, R. Radermacher	Comparison of automotive air-conditioning systems operating with CO_2 and R-134a	IIF-IIR Commission B1, B2, E1 and E2, Purdue University. July 2000
[BHA97]	M.S Bhatti	A critical look at R-744 and R-134a mobile air conditioning systems	SAE International Congress. February1997

A1.1 - Experimental comparison of mobile A/C systems when operated with transcritical CO_2 versus conventional R-134a

Title	Experimental comparison of mobile A/C systems when operated with transcritical CO2 versus conventional R-134a
Authors	R.P McEnaney, D.E Boewe, J.M Yin, Y.C Park, C.W Bullard, P.S Hrnjak
Subject	Comparison of AC systems with CO2 and R-134a
Document	Paper
Compagny	Air Conditioning and Refrigeration Center , University of Illinois
Source	1998 International Refrigeration Conference at Purdue
Date	1998
Number of pages	145-150

The subject of this paper is the experimental comparison of two mobile AC systems. The first one is a standard R-134a system, whereas the second is a CO_2 prototype. Both are mounted on a Ford Escort. The comparison is based on two systems that have the same geometrical size in terms of volume and heat exchange surfaces for the condenser and the evaporator. For both systems, the compressors have a fixed swept volume so the regulation is performed by clutching and de-clutching thee compressor during the test.

Table 1 shows the technical data of all the components used for both systems.

Tableau 1 – Technical data of the two tested systems

System	Refrigerant	*R134a*	*R744*
	Type	Ford Escort	Prototype
Compressor:	Type	Reciprocating	Reciprocating
	Displacement	155 cm^3	20.7 cm^3
Expansion device		Orifice tube	Manual valve/back pressure valve
Condenser / Gas cooler	Description	Wavy Al fins, round Al tubes, 21 pass, OD = 6mm	Microchannel, brazed Al tubes, 3-pass, parallel flow
	Mass	2.0 kg	2.3 kg
	Face area	36.1 x 54.4 = 1964 cm^2	36.8 x 53.0 = 1950 cm^2
	Core depth	2.2 cm	1.65 cm
	Core volume	4320 cm^3	3320cm^3
	Air side surface	7.2 m^2	5.2 m^2
	Refrigerant side surface area	0.40 m^2	0.49 m^2
Evaporator	Description	Brazed Al plate (drawn cup, laminated), 4-pass, 17 plates	Microchannel, brazed, Al, 7-pass, parallel flow
	Mass	1.8 kg	2.2 kg
	Face area	18.4 x 22.0 = 405 cm^2	18.2 x 22.4 = 408 cm^2
	Core depth	9.2 cm	9.1 cm
	Core volume	3720 cm^3	3710 cm^3
	Air side surface	3.5 m^2	4.2 m^2
	Refrigerant side surface area	0.55 m^2	0.66 m^2
Suction line heat exchanger	Description		Two aluminum coaxial tubes, suction vapor in annulus, counter-flow
	Mass		0.7kg
	Length		1.5m

Tableau 2 – Test matrix of of operating conditions used during the work.

Test	I30	M26	I28	I23	M23	I13	M12	I6	M3	I19	M10	I17	M8	I11	M5
Condenser inlet air temperature, [°C]	15.5		21.1	32.2	21.1	43.3	32.2	54.4	43.3	32.2		43.3			
Passenger compartment air temperature, [°C]	21.1					26.7		32.2		26.6		21.1			26.6
Evaporator air flow rate, [m^3/min]	2.832					4.956		7.08							
Relative humidity, [%]	40														

(1) I : Idle condition: compressor speed = 950 rpm, condenser air flow rate = 22.67 m^3/min
(2) M : Intermediate speed: compressor speed = 1800 rpm, condenser air flow rate = 26.90 m^3/min
(3) H : Highway condition: compressor speed = 3000 rpm, condenser air flow rate = 35.40 m^3/min

The experimental data obtained are presented Table 3. Many values measured can be read including the cooling capacity, the COP, the pressure ratio, the evaporating temperature, evaporator air out temperature and the volumetric and isentropic efficiencies.

Tableau 3 – Tests results of R-134a and R-744 system

Test	Refri-gerant	Q [kW]	COP [-]	Cond. [g/s]	Tevap [°C]	Teao [°C]	Isentr. eff. [-]	Volum. eff. [-]	Pd/Ps [-]	Ontime fraction
M3	R134a	4.194	1.58	0.631	4.8	14.9	0.59	0.56	7.1	1.0
	R744	**4.672**	**1.59**	**0.666**	**8.3**	**12.8**	**0.76**	**0.79**	**2.7**	**1.0**
M12	R134a	2.548	1.82	0.318	1.5	9.7				0.64
	R744	**2.532**	**1.93**	**0.253**	**4.5**	**8.4**				**0.57**
M23	R134a	1.107	2.38	0.079	2.7	6.8				0.24
	R744	**1.149**	**3.17**	**0.071**	**2.2**	**5.8**				**0.16**
M26	R134a	1.104	2.51	0.077	2.7	7.2				0.09
	R744	**1.187**	**4.00**	**0.087**	**2.5**	**5.9**				**0.15**
I6	R134a	2.939	1.75	0.307	11.5	18.1	0.58	0.54	5.7	1.0
	R744	**2.795**	**1.57**	**0.229**	**13.7**	**17.5**	**0.70**	**0.72**	**2.6**	**1.0**
I13	R134a	2.475	1.90	0.280	3.1	9.4	0.60	0.59	5.5	1.0
	R744	**2.624**	**1.84**	**0.299**	**4.8**	**8.6**	**0.79**	**0.79**	**2.7**	**1.0**
I23	R134a	1.219	1.91	0.114	1.1	6.3				0.58
	R744	**1.237**	**2.35**	**0.092**	**0.9**	**5.4**				**0.50**
I28	R134a	1.141	2.70	0.092	2.1	7.0				0.45
	R744	**1.184**	**3.85**	**0.085**	**2.2**	**5.8**				**0.30**
I30	R134a	1.132	3.12	0.082	1.9	6.3				0.41
	R744	**1.182**	**4.49**	**0.079**	**1.9**	**5.7**				**0.29**
I19	R134a	3.142	2.58	0.346	3.4	11.1	0.66	0.67	4.5	1.0
	R744	**3.405**	**2.76**	**0.034**	**5.3**	**9.2**	**0.83**	**0.84**	**2.4**	**1.0**
M10	R134a	3.754	1.68	0.473	-1.0	9.4	0.56	0.54	6.8	1.0
	R744	**4.587**	**1.86**	**0.570**	**0.8**	**5.3**	**0.80**	**0.84**	**2.7**	**1.0**
I17	R134a	2.208	1.75	0.149	1.7	8.7	0.60	0.58	5.6	1.0
	R744	**2.290**	**1.76**	**0.090**	**4.1**	**7.5**	**0.79**	**0.79**	**2.5**	**1.0**
M8	R134a	2.817	1.27	0.261	-2.5	6.5	0.54	0.50	7.8	1.0
	R744	**3.466**	**1.31**	**0.310**	**-1.9**	**2.9**	**0.76**	**0.77**	**3.1**	**1.0**
I11	R134a	2.773	2.03	0.273	5.4	12.3	0.62	0.60	5.2	1.0
	R744	**2.780**	**2.01**	**0.214**	**7.5**	**11.2**	**0.78**	**0.79**	**2.5**	**1.0**
M5	R134a	3.480	1.44	0.401	0.8	9.4	0.56	0.52	7.5	1.0
	R744	**4.722**	**1.47**	**0.502**	**2.1**	**6.5**	**0.75**	**0.77**	**3.1**	**1.0**

Analysis of results

At high rotation speed, the CO_2 system presents a higher cooling capacity but the COP is lower than the R-134a system.

On the other hand, when the compressor rotation speed is low (idle or urban conditions), the COP of the CO_2 system is higher than the one measured for the R-134a system. But the cooling capacity is lower.

When referred to the tests presented, it seems that the combination COP / cooling capacity is better for the CO_2 system for the majority of the tests points. Nevertheless it must be noticed that for low rotation speed, the CO_2 system presents a lower cooling capacity. Moreover at high ambient temperatures (ex N = 950 RPM and T_{air} = 55°C), the COP of the R-134a system is better.

Conclusions

The comparison performed by the ACRC team shows that the CO_2 mobile AC system presents a higher cooling capacity for most conditions.

The CO_2 system has been designed to have the same cooling capacity as the R-134a system at an ambient temperature of 54°C and rotation speed equal to 950 RPM (worst case working conditions). For these conditions, the COP of the CO_2 system is about 10% lower than the one measured for the R-134a system.

For ambient temperatures lower than 40°C (which represents the majority of the climatic conditions that can be encountered), the CO_2 system presents better energy performances in terms of cooling capacity and COP.

A1.2 - SAE alternate refrigerant Cooperative Research Project

Title	SAE alternate refrigerant Cooperative Research Project
Authors	ACRC
Subject	Cooperative project
Document type	Research contract
Company	Air Conditioning and Refrigeration Center , University of Illinois
Source	SAE cooperative research program
Date	November 2002
Number of pages	17

Main goal

The main goal of this work is to provide a directly comparative engineering evaluation of the existing R-134a systems and other refrigerant technologies, specially CO_2 systems.

Three different systems have been tested under the same working conditions:
- Baseline R-134a system,
- Enhanced R-134a system,
- R-744 Prototype system.

The geometrical characteristics of the components used for each system are listed in Table 1. All systems are fitted with variable swash plate compressors (but it is not said if they are internally or externally controlled).

Table 1 : system comparison

Component	Feature	Baseline R134a	R744	Enhanced R134a
	Core Effective Face [mm]	W768xH400xD20	W689xH464xD16	W679xH465xD16
	Face area [mm^2]	307,200	319,696	315,735
Condenser	Core volume [mm^3]	6,144,000	5,115,136	5,051,760
	Aspect Ratio	0.52	0.67	0.68
	Mass [kg]	2.8	2.4	2.9
	Core Effective Face [mm]	W280xH216xD73	W305xH210xD58	W294xH216xD50
	Face area [mm^2]	60,480	64,050	63,504
Evaporator	Core volume [mm^3]	4,415,040	3,714,900	3,175,200
	Aspect Ratio	0.77	0.70	0.73
	Mass [kg]	2.0	2.7	1.6
	Type	Variable Swash -plate	Variable Swash -plate	Variable Swash -plate
Compressor	Displacement [cc.]	~165	~33	~165
	Mass(including pulley) [kg.]	6.4	7.0	6.4
SLHX	Length [mm.]		2100	
	Mass [kg.]	N/A	1.1	N/A
Receiver/Drier	Volume [cc.]			
	Mass [kg.]	0.4	1.4	IRD, see condenser
Oil Separator	Volume [cc.]			
	Mass [kg.]		0.2	
Total Mass	[kg]	11.6	14.8	10.9

The weight of systems is as follows :
- Baseline R-134a system : 11.6 kg,
- Enhanced R-134a system : 10.9 kg,
- R-744 Prototype system : 14.8 kg.

Mass includes only heat exchangers, compressor and fluid storage. It does not include refrigerant lines, fittings, valves and systems controls.

All the tests are preformed for steady state conditions. The aim was to evaluate the influence of various parameters :

- compressor rotation speed,
- ambient air temperature,
- air mass flow at the heat exchangers.

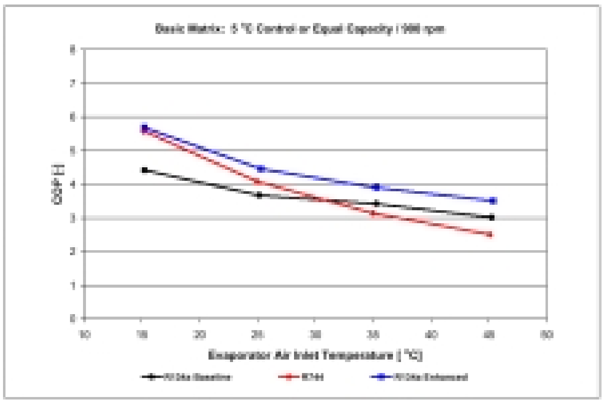
on the cooling capacity of the systems and the energy performances:

The first series of tests has been performed for the following conditions; evaporator outlet air temperature is fixed at 5°C or all systems work at equal capacity

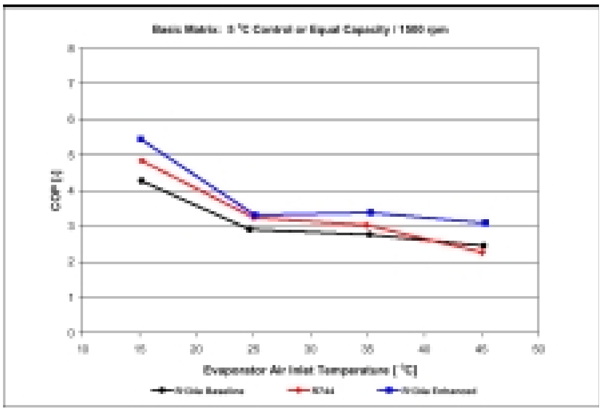
Basic Test Point Matrix					
Evaporator Air			Cond/Gas Cooler Air		Comp
T (°C)	RH (%)	Flow (l/s)	T (°C)	Flow (l/s)	rpm
45	27	109 130	45	425 850 1320	900 1500 2500
35	45	109 130	35	425 850 1320	900 1500 2500
25	65	109 130	25	425 850 1320	900 1500 2500
15	80	109 130	15	425 850 1320	900 1500 2500

The COP is a function of the ambient air temperature. The results are presented for three rotation speeds:

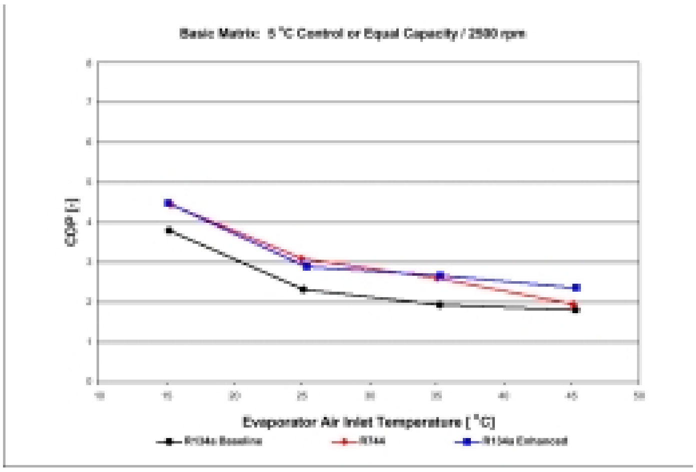
- 900 RPM,
- 1500 RPM,
- 2500 RPM



900RPM



1500RPM



2500RPM

Concerning the results presented in the three figures, it appears that whatever the tests conditions the best performances are obtained with the enhanced R-134a system.

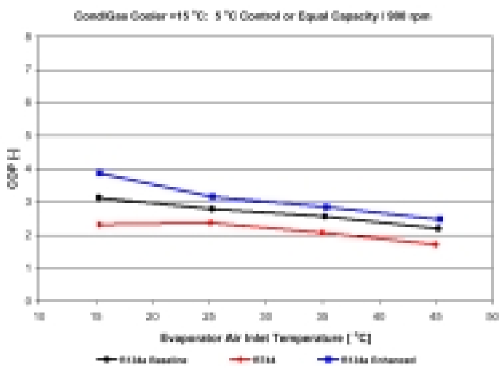
It shall be noticed that the higher the rotation speed, the lower the performances whatever the tested system. The same behavior is observed when the ambient air temperature increases.

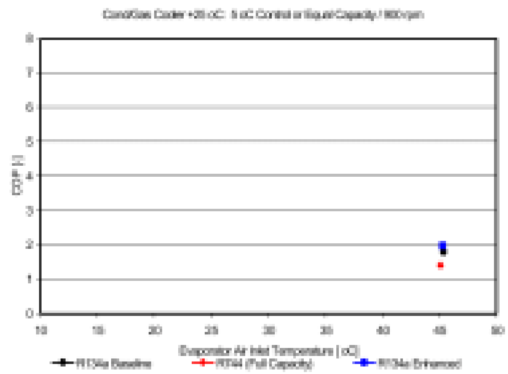
Other tests performed during the work

The main objective of this series of tests is the analysis of the influence of ambient air temperature at the evaporator and the condenser.

Condenser/Gas Cooler Air Temperature Matrix					
Evaporator Air			Cond/Gas Cooler Air		Comp
T (°C)	RH (%)	Flow (l/s)	T (°C)	Flow (l/s)	rpm
45	27	109	60 70	425	900
35	45	109	50	425	900
25	65	109	40	425	900
15	80	109	30	425	900

Tests have been carried out at only 900 RPM. It shall be noticed that during the urban traffic, the engine rotation speed can vary from 900 RPM to 2000 RPM depending on the way the driver drives the car.

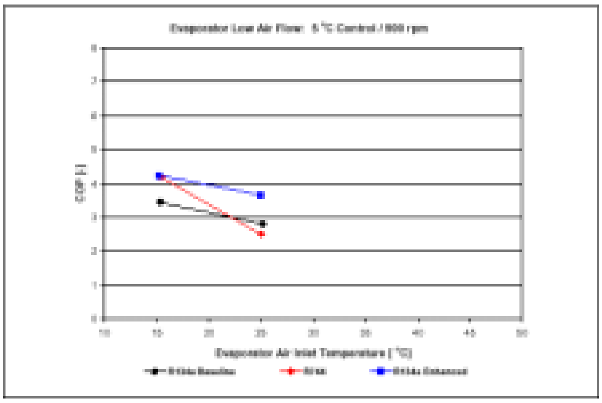


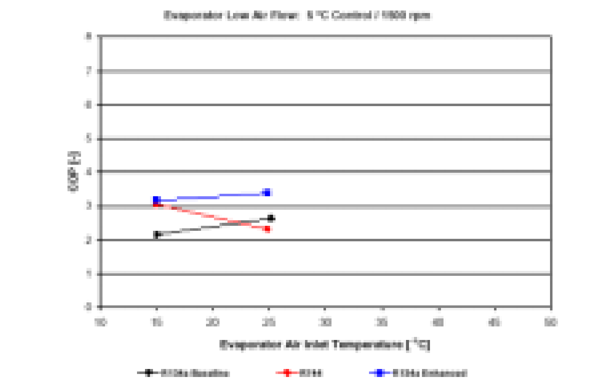


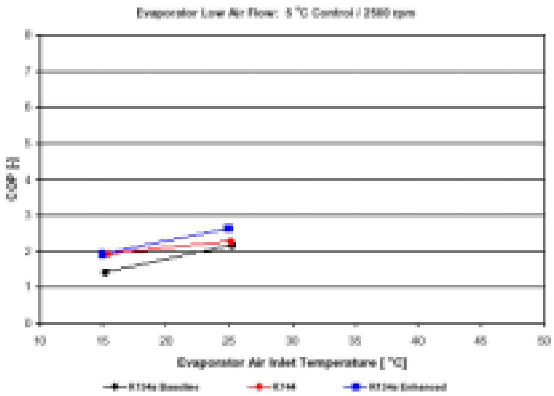
It is obvious that when the condenser / gas cooler inlet air temperature increases, the energy performances of the CO_2 system are lower than the ones measured for both R-134a systems (baseline and improved). When the condenser / gas cooler inlet air temperature is equal to the ambient air temperature +15°C, the COP of the CO_2 system is nearly 50% lower than the R-134a enhanced system at Tair = 15°C, and 35% lower at Tair = 45°C.

The purpose of the last series of tests is to simulate low blower operation in moderate ambient temperature conditions. Evaporator low air flow was run at 25°C and 15°C conditions. The table below shows the tests matrix.

Evaporator Low Air Flow Matrix					
Evaporator Air			Cond/Gas Cooler Air		Comp
T (°C)	RH (%)	Flow (l/s)	T (°C)	Flow (l/s)	rpm
25	65	28	25	425	900
				850	1500
				1320	2500
15	80	28	15	425	900
				850	1500
				1320	2500







The results show that the higher the rotation speed, the lower the energy performances whatever the system tested.

When the three systems are compared, it appears that at 15°C, the enhanced R-134a and the CO_2 systems present the same energy performances (higher that the R-134a baseline system). But at 25°C, the CO_2 system performances fall below the performances measured on the two R-134a systems. The CO_2 system COP is between 35% (at 900 RPM) and 20% (at 2500 RPM) lower than the COP measured for the enhanced R-134a system COP.

A1.3 Comparative analysis of an automotive air conditioning systems operating with CO_2 and R-134a

Title	Comparative analysis of an automotive air conditioning systems operating with CO_2 and R-134a
Authors	J. S. Brown[1], S. F Yana-Motta[2], P. Domanski[3]
subject	Theoretical comparison of AC systems with CO_2 and R-134a
document	Paper
Company	1. Catholic University, Washington, DC, USA 2. Honeywell International Inc, Buffalo, NY, USA 3. National Institute of Standards and Technology, Gaithersburg, MD, USA
Source	International journal of refrigeration 25 (2002)
Date	2002
Number of pages	19-32

This paper evaluates the performance advantages of R-744 and R-134a automotive air conditioning systems using semi-theoretical cycle models.

1. Introduction

The authors explain why, according to the Montreal Protocol and consequent regulation, the air-conditioning industry is evaluating and introducing new refrigerant (with low GWP) as replacements to CFCs and HCFCs.

Due to the high warming potential of the R-134a (GWP = 1300), the automotive industry is investigating fluids with low GWP with a particular attention to R-744.

This paper presents the energy performances of two systems using R-134a and R-744. It is important to focus on this point because a refrigerant's environmental impact on the climate change is determined not only by its direct effect (GWP), but also by the CO_2 released upon burning fossil needed to power the air-conditioning system (indirect effect).

2. Literature review

The authors present first an interesting literature review. In the early 1990's, Lorentzen and Pettersen [1] initiated a renewed interest in CO_2, Petersen, Aarline, Kruse et al [2],[3] carried out an intense research effort. Lorentzen and Pettersen developed and tested a R-744 prototype. They showed that it has comparable performance to an R-12 system.

The authors mention the Race project and Gentner work [4], McEnaney et al [5] and Preissner [6] et al paper, which are presented in detail in the present study.

J.S Brown mention the work performed by Bullock [7]. Bullock theoretically analyzes performance of R-744 as refrigerant in a vapor compression cycle with the condenser temperature reaching the critical point. His results showed that the R-744 system is less efficient than the R-22 system by 30% in the cooling mode and by 25% in the heating mode. He concludes that to obtain mid-level efficiency of typical unitary equipment, the R-744 system would require an efficient expander and significantly improved compressor and heat exchangers.

Hwang and Radermacher [8] theoretically compared the performance of R-22 and R-744 and they found that water-heating is a promising application since the performance of the R-744 system is 10% higher than R-22 across a wide range of ambient temperatures.

Boewe et al [9] investigated the effect of the internal heat exchanger on the COP and the capacity of a R-744 system. The internal heat exchanger increases both capacity and COP. For idle conditions, the COP is increased by 26% and the cooling capacity by 10%. In all the idle conditions, the optimal pressure resulted in discharge temperatures below 140°C. For tests performed for driving conditions, the optimum pressure was not reached because it would result in compressor discharge temperatures above 140°C.

Bhatti [10] explored several possibilities to improve the COP of the R-134a system. His strategies included an increase in compressor efficiency, increase in condenser effectiveness, decrease evaporator air-side pressure drop by improved condensat removal and a decrease in oil circulation.

Zietlow and al [11] performed an experimental study to improve the COP of and the cooling capacity. With a high-efficiency scroll compressor, a TXV expansion valve and a micro-channel condenser with a receiver and a sub-cooling section, the authors show that the compressor power consumption was reduced by 28% whereas the COP increased by 24%.

From this brief literature review, J. S. Brown observes that the theoretical COP of an R-744 system is lower than the theoretical COP of an R-134a system, whereas some experimental studies demonstrated that an R-744 system could provide the performance level of a current-production R-134a system. The studies that reported equivalent performance were based on tests of R-744 and R-134 systems which were not equivalent component-wise. The use of the internal heat exchanger is the most obvious difference between the two systems. The use of different-technology heat exchangers has also a major impact on the performances.

3. Simulation model

The model used during this study is a semi empirical model. The simulated system includes the compressor, the condenser, the evaporator, the expansion device and the internal heat exchanger (for R-744 only).

The compressor is assumed to be an open-type (for both systems). The volumetric efficiency is modeled using the following expression (for both systems).

$$\eta_v = 0.8263.\left[1 - 0.09604.\left(\theta^{\frac{1}{\gamma}} - 1\right)\right]$$

The correlation was obtained by fitting the data of McEnaney et al and Park et al for R-744 compressor and R-134a compressor respectively.

The isentropic efficiency was fitted using the curve presented by Rieberer and Halozan for an R-744 compressor based on experimental data of four authors.

$$\eta_{is} = 0.9343 - 0.4478.\theta$$

Several researchers postulated that a lower pressure ratio results in higher compressor efficiency. Following this postulate, the R-744 should have higher isentropic efficiencies. The pressure ratio for an R-744 system is on the order of 3, whereas for an R-134a system, it is on the order of 5 to 7. The opposing view is that the higher efficiencies measured for R-744 compressors compared to the current-production are rather a result of compressor design itself. Given this controversy, it was decided to use for both R-744 and R-134a compressors the equation above to be consistent with experimental studies in which R-744 compressors were found to be more efficient than R-134a compressors.

The heat exchangers chosen for both systems are made with micro-channel tube to have a fair comparison.

The internal hat exchanger was just used for the R-744 system.

Both cycle model assume that the expansion device process is isenthalpic. The R-134a simulation approximated the use of a TXV with constant superheat and sub-cooling. For R-744 transcritical cycle, the following options were chosen:

- Gas cooler pressure imposed,
- Gas cooler pressure optimized for maximum COP,
- Gas cooler pressure optimized for maximum COP overridden by the 140°C compressor discharge temperature limit.
-

4. Simulations

Three air temperatures were chosen at the gas cooler/condenser, for the tests:

- 32.2°C,
- 43.3°C
- 48.9°C.

and one temperature at the evaporator inlet : 26.7°C.

Two compressor rotation speeds were chosen:

- 1000 RPM (for low speed),
- 3000 RPM (for high speed).

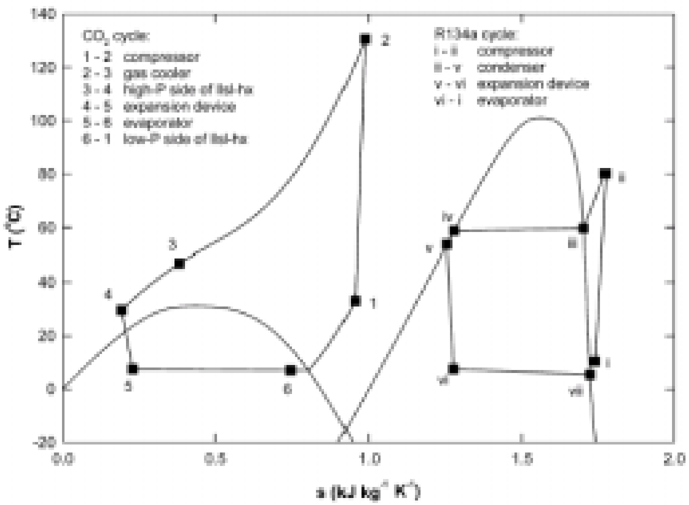


Fig. 1. CO_2 transcritical cycle and the R134a subcritical cycle at an ambient temperature of 43.3°C and 1000 compressor RPM.

Figure 1 shows the R-744 transcritical and the R-134a sub-critical cycle obtained at an ambient temperature of 43.3°C for 1000 RPM.

The significantly higher discharge temperature and larger temperature for R-744 than for R-134a in the high-pressure side are the most visible differences. The large R-744 glide is the reason for the significant temperature mismatch because the refrigerant-to-air heat exchangers are cross-flow. The figure also shows that the temperature approach for the R-744 gas cooler is lower than for the R-134a condenser, which benefits the R-744. However even qualitative visual examination suggests that the benefit of a lower temperature approach cannot compensate for the penalty due to the air/R-744 temperature mismatch.

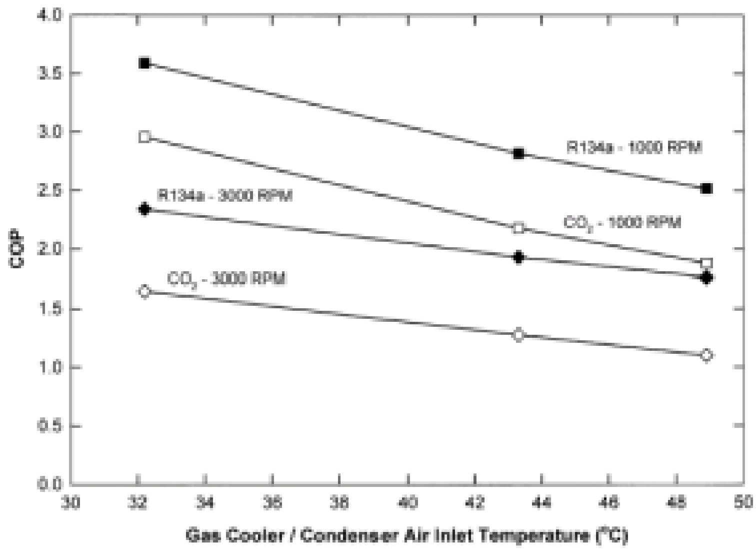


Fig. 2. COP comparison between CO_2 and R134a air conditioning systems.

Figure 2 shows that at 1000 RPM and with an ambient air temperature of 32.2°C, the R-134a system COP is 21% higher than the R-744 system COP. The COP gap increases to 29% at 43.3°C, and to 34% at 48.9°C. At 3000 RPM, the R-134a COP is 42% higher than that of the R-744 system. The gap increases to 51% at 43.3°C and to 60% at 48.9°C.

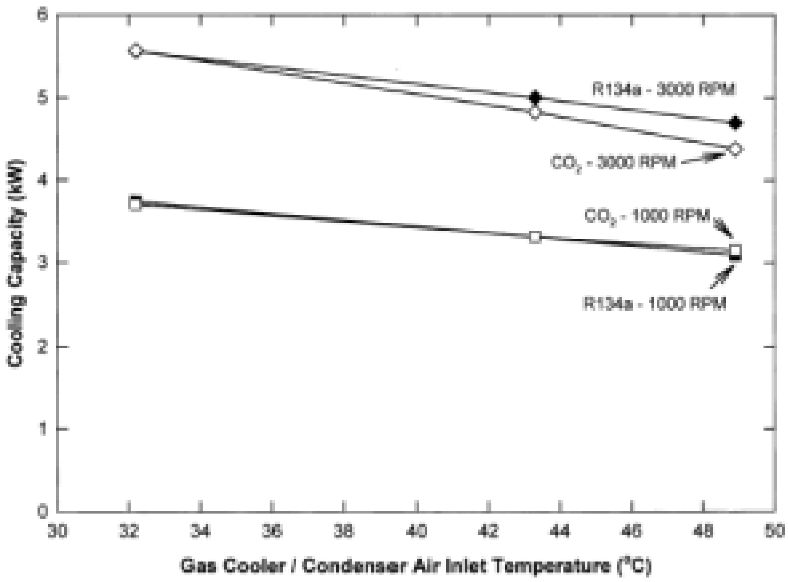


Fig. 3. Cooling capacity comparison between CO_2 and R134a air conditioning systems.

Figure 3 shows the cooling capacity as a function of ambient temperature and rotation speed. For both systems, the cooling capacity is nearly the same. But the R-134a is a bit higher depending on the testing conditions. The higher the ambient temperature, the higher the difference between the two systems.

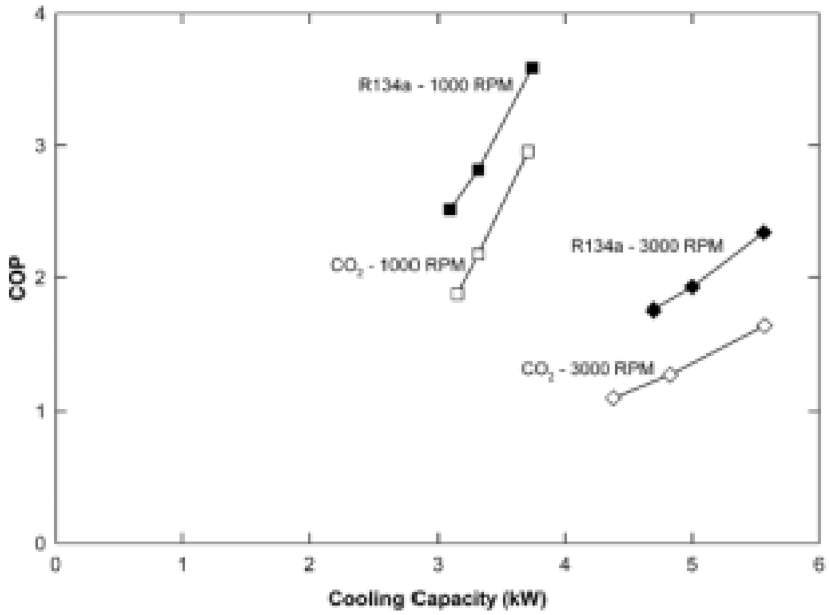


Fig. 4. COP comparison versus cooling capacity between CO_2 and R134a air conditioning systems.

Figure 4 shows the interdependence between the cooling capacity and the COP for both systems.

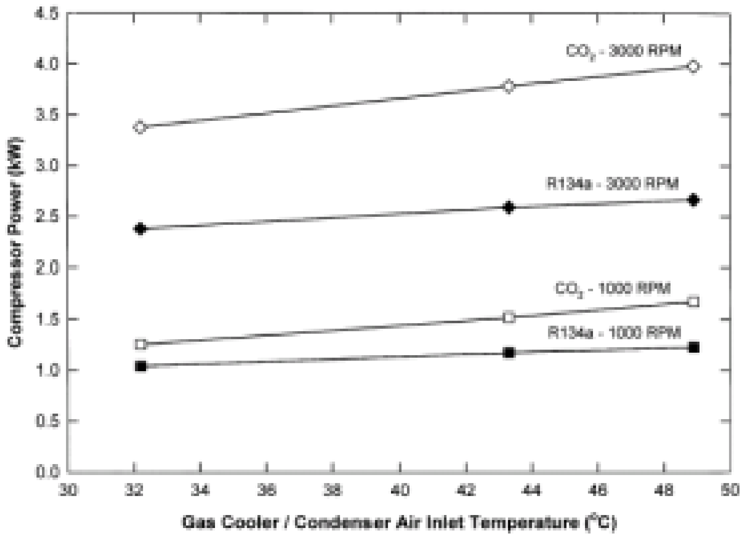


Fig. 5. Compressor power consumption comparison between CO_2 and R134a air conditioning systems.

Figure 5 shows compressor input power as a function of the ambient air temperature and the compressor rotation speed. It is obvious that the mechanical power absorbed by the R-744 system is much higher than the energy consumed by the R-134a system.

At 1000 RPM and 32.2°C, the R-744 system power is 20% higher than the power absorbed by the R-134a system. The gap increases to 29% at 43.3°C and to 36% at 48.9°C. At 3000 RPM and 32.2°C, the R-744 system power is 42% higher than the power absorbed by the R-134a system. The gap increases to 46% at 43.3°C and to 49% at 48.9°C.

5. Simulation trade-offs

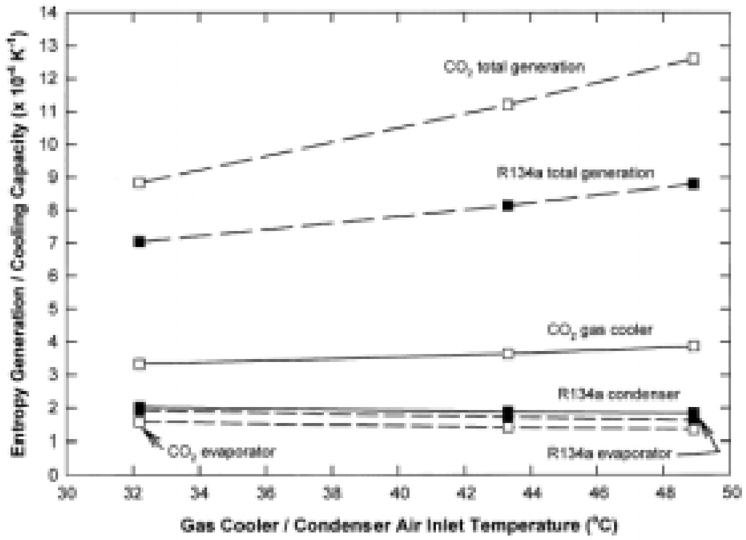


Fig. 6. Comparison of entropy generation per unit cooling capacity for CO_2 and R134a air conditioning systems at 1000 compressor RPM.

An insight into R-744 and R-134a systems' irreversibility can be gained, reviewing the entropy generation information shown Figure 6 for the refrigerant-to-air heat exchangers and for both systems. The figure uses an entropy-per-capacity ordinate in recognition of the fact that an absolute scale would result in different entropy generations for two systems of the same capacity, if their capacities were different.

The entropy generated at the evaporator are similar for both systems. R-744 produces a smaller amount of entropy. This is due to the better transport properties of the R-744. On the other hand, at the gas cooler, the entropy generation is much higher for the R-744 system. The large R-744 temperature glide (around 80°C compared to 25°C for the R-134a) and the larger amount of heat to be rejected (3.7-11.7% at 1000RPM and 12.7-13.4% at 3000 RPM) causes a significant amount of entropy generation (heat transfer irreversibilities). The large amount of entropy generation for the R-744 in the gas cooler is responsible for the higher total generation unavoidable in a cross-flow gas cooler heat exchangers for high-glide fluids.

Hence the cross-flow gas cooler will negatively affect the performance of the transcritical R-744 cycle while application of a counter-flow gas cooler may offer COP advantage if glide matching with external heat-transfer fluid is obtained.

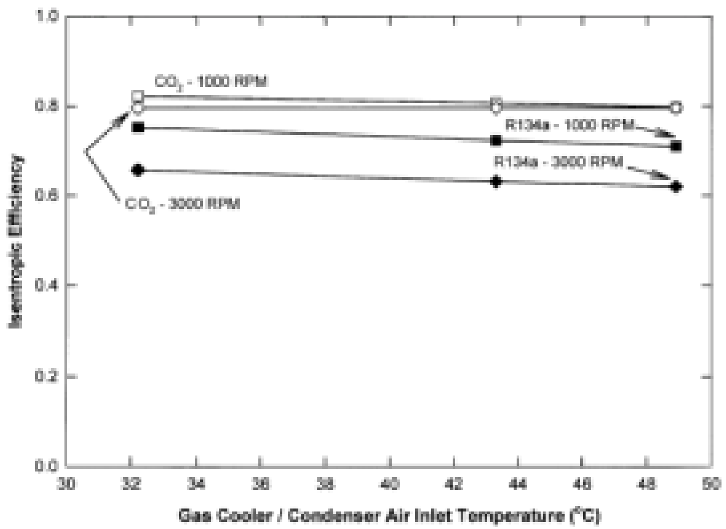


Fig. 7. Compressor isentropic efficiency comparison between CO_2 and R134a air conditioning systems.

Because of the lower pressure ratio, the R-744 compressor has higher isentropic efficiency as it can be seen Figure 7, and thus whatever the testing conditions. The higher the rotation speed, the lower the isentropic efficiency of the R-134a compressor, whereas the isentropic efficiency is nearly constant for the R-744 compressor. This is undoubtedly an advantage for the R-744 system.

The approach temperature in the R-744 gas cooler is between 3.0 and 5.8°C whereas for the R-134a condenser the range is between 8.5 and 11.2°C. The closer approach temperature however, did not overcome the thermodynamic penalty associated with the large temperature glide resulting in high entropy generation.

The R-134a system would benefit from the use of an internal heat exchanger.

References

[1] Lorentzen, G Petersen J. , New possibilities for non-CFC refrigeration. In proceeding of the IIR International Symposium on Refrigeration, Energy and Environment, Trondheim, Norway,1992 p 147-163

[2] Petersen J, Aarlien R., Progress in CO2 vapor compression systems. Thermal science and engineering 1998, 25-34

[3] Kruse H, Heidelck R, Suss J, The application of CO2 as refrigerant. Bulletin of the International Institute of refrigeration 2000, 3-18.

[4] Gentner

[5] McEnaney

[6] Preissner

[7] Bullok CE,. Theoretical evaluation of carbon dioxide in sub-critical and transcritical cycles. In proceedings of ASHRAE/ NIST Refrigerants Conference, Gaithersburg, Maryland, 1997, p20-26

[8] Hwang Y, Radermacher R. Theoretical evaluation of carbon dioxide refrigeration cycle. HVAC and Research 1998 4(3):245-63

[9] Boewe, D, Yin J, Park YC, Bullard CW, Hrnjak PS. The role of suction line heat exchanger in trans-critical R-744 mobile A/C systems. SAE international Congress and Exposition, Detroit, Michigan, Paper 1999-01-0583, 1999

[10] Bhatti MS. Enhancement of R-134a automotive air conditioning system. SAE international Congress and Exposition, Detroit, Michigan, Paper 1999-01-0870, 1999

[11] Zietlow D, Sharma A, Meyer J. An experimental study of power reduction opportunities for automotive air conditioning systems. In proceedings of the 20th International Congress of Refrigeration, Sydney; Australia, 1999

A1.4 - Passenger car air conditioning using carbon dioxide as refrigerant

Title	Passenger car air conditioning using carbon dioxide as refrigerant
Authors	H. Gentner, A Földi
Subject	CO_2 AC systems evaluation
Document type	Paper
Company	BMW AG, Müchen
Source	IIF-IIR Section B and E Oslo, Norway
Date	1998
Number of pages	15

Objectives

In the years 1994 to 1997, European car manufacturers and suppliers carried out the R&D project “RACE. The main objective of this project was the development and the investigation of passenger mobile air-conditioning system using CO_2 as refrigerant.
The main steps of this project were:

- the design of the system,
- the elaboration of control strategies,
- the preparation of prototypes and tests in wind tunnels,
- the environmental and technical evaluation compared to the state-of-the art R-134a systems.

CO_2 cycle description

The field of operating conditions for MAC systems exceeds the low critical temperature of CO_2 (31,1°C) so the cycle is transcritical. The condenser is replaced by a gas cooler. It shall be noticed that in such cycles, the high pressure (that can reach 150 bar) has a major influence on the energy performances of the system in terms of cooling capacity and COP.

The high volumetric cooling capacity of CO_2 allows to reduce the refrigerant mass flow rate and thus allows to reduce the compressor swept volume (the ratio between R-134a and CO_2 is around 5) and the internal diameter of heat exchangers tubes, avoiding a significant increase of wall thickness to resist to the high pressures reached by the refrigerant.

The chosen heat exchangers are built using the serpentine technology. The suction line heat exchanger is a compact extruded tube.

The expansion device is a back pressure valve but a solenoid can be used to control the refrigerant mass flow rate depending on how the system works.

The CO_2 compressor developed during this project is a swash plate compressor with variable displacement. The stroke volume of this compressor is 26 cm^3. A special development has been made for the shaft seal which has to work against a pressure difference of 50 bar. Between 800 and 8000 RPM, the volumetric efficiency is nearly constant whatever the rotation speed of the compressor.

Experimental investigation of the car prototype

The reference car used during the RACE project is the BMW 5 series. This car allowed the comparison of the series R-134a system and the new CO_2 prototype system.
The CO_2 system is composed by:

- A tube fin evaporator,
- A spherical type receiver,
- A counter flow internal heat exchanger,

- A variable displacement reciprocating compressor with a stroke volume of 26 cm3,
- A flat tube serpentine gas cooler,
- Solenoid expansion valve,
- Sensors for ambient temperature, ventilation temperature, high side and suction pressure for capacity and high side pressure control.

The system has been charged with 800g of CO_2.
The car was tested in wind tunnel for the conditions showed Table 1.

Tableau 1 – Time course and boundary conditions

Phase	Precondi-tioning	Passive Heating	32 km/h, 3. Gear	64 km/h, 4. Gear	Idle
Time [min]	480	90	30	30	80
Temperature [°C]	40	40	40	40	40
Rel. Humidity [%]	25	25	25	25	25
Sol. Radiation [W/m²]	0	950	950	950	950
Engine Speed [rpm]	0	0	1570	2290	830
Velocity of Wind [km/h]	12	0	32	64	7
Inside Moistening [g/h]	0	0	0	0	0
General Settings			Fan: max. Temperature: min. Air Recirculation: 100 % Air Distribution: 100 % Ventilation		

Figure 1 shows the results obtained during the tests
Figure 1 indicates that at idle conditions, the evaporator outlet air temperature increases above 10°C. Reasons are:

- A decrease in the engine speed at 830 RPM causes an insufficient refrigerant mass flow rate. A slight improvement of the displacement or the volumetric efficiency seems to be necessary.

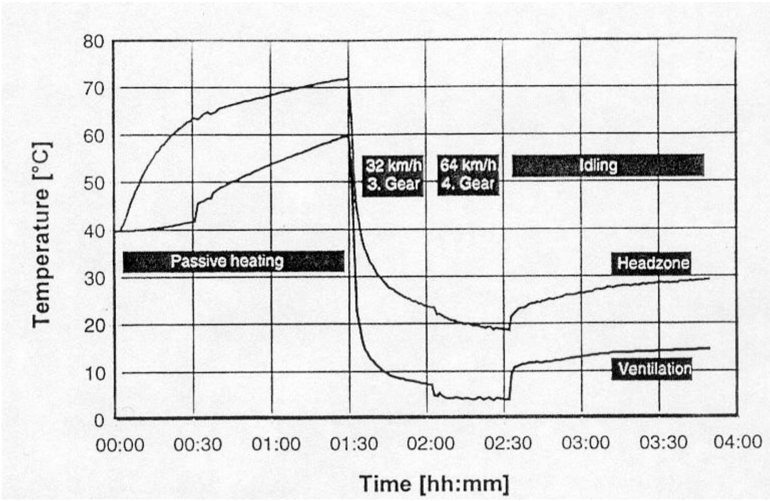


Figure 1 – Passive heating and pull down-down curves with CO_2 system

- The considerably worsened ventilation of the gas cooler leads to an increase in the high-side pressure and the outlet temperature at the rear of the gas cooler, and with that, to a higher enthalpy at the entry of the internal heat exchanger and the evaporator.

Comparison CO_2 / R-134a

The tests carried out on the car allow the comparison of CO_2 and R-134a systems.

Figure 2 shows that for both systems, the dynamic of cooling down is the same. At 32 km/h, the R-134a systems reach a cabin temperature that is 2K below the CO_2 system. This is due to the fact that the ventilation air flow rate for the CO_2 system is lower since the evaporator outlet air temperature is the same for both systems.

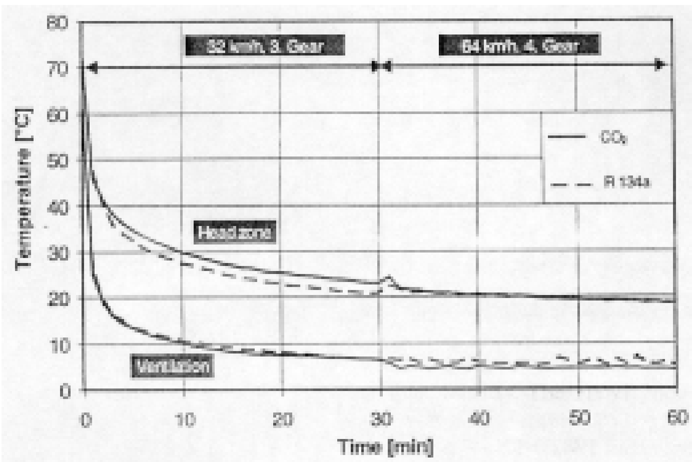


Figure 2 – Pull down curves of the CO_2 system compared to R-134a (series system)

At 62 km/h the evaporator outlet air temperature is 2K lower for the CO_2 system so the temperature inside the car is the same.

Another prototype has been constructed during the project using the same compressor. It was tested and achieved very promising results. At pull down conditions the system shows the same results as the ones obtained with the R-134a system; and at high ambient temperature, the CO_2 system showed significant advantages.

Figure 3 shows the inside and ventilation outlet temperatures reached for various driving conditions (stabilized tests).

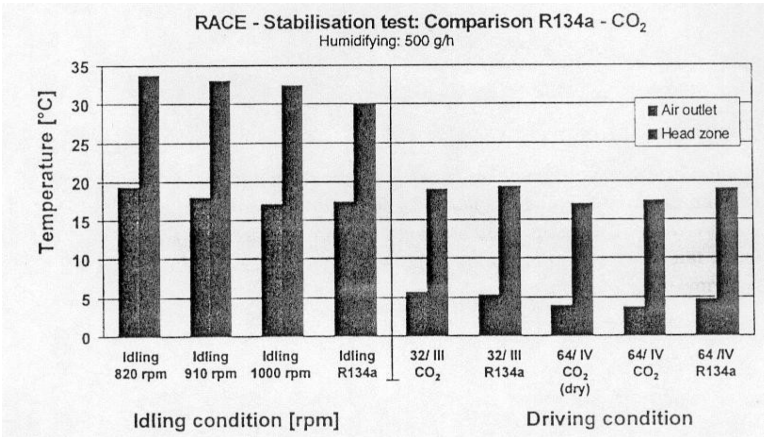


Figure 3 – Stabilization tests : comparison R-134a – CO_2 (Tests at 40 °C, 950 W/m^2 and 500g/h air moisture)

System evaluation

Environmental aspects

CO_2 used as refrigerant has no direct contribution to the greenhouse effect (with a GWP of 1), since it is gained as a waste product of the chemical industry.

To be adopted, the CO_2 systems must show better energy efficiency than R-134a systems. The investigations carried out show that:

- CO_2 systems present sufficient cooling capacity to reach a high level of comfort in all climatic regions ;
- CO_2 MAC systems present fuel consumption comparable to R-134a systems in all climatic regions under steady state conditions ;
- CO_2 MAC presents a significantly lower TEWI than R-134a systems,
- the estimated extra weight of 3kg for CO_2 system has no remarkable influence on the fuel consumption.

With variable displacement compressors, the COP of R-134a systems increases by 20%. But late test benches show that the optimization of CO_2 components offers the same potential of improvement.

Safety aspects

The high pressure of the CO_2 is the major problem for CO_2. However, the high volumetric cooling capacity of CO_2 leads to small internal volume and consequently to a low value of the product of pressure times volume inside the refrigerant container and lines. Problems can occur during maintenance work but constructive changes can minimize the risk of injury.

Another aspect is the CO_2 toxicity. The MAK (max. allowable concentration) for working place is 1% vol, and so the respiratory system can be affected by values of 3 to 5% vol. Therefore sensors are mandatory in order to prevent the entire charge of CO_2 to be released in the passenger compartment sensor.

Technological aspects
Many efforts have been made to develop adequate heat exchangers. The main efforts now concerns the compressor and its lubrication. Several developments are needed until the compressor will fulfill the conditions for a series production.

Economical aspects
CO_2 systems are more expensive than R-134a systems because of supplementary components (internal heat exchanger, high pressure control and CO_2 sensors). Hoses with metallic diffusion barrier are also expected to add extra costs.

Conclusions
The RACE project showed very promising results for CO_2 systems and motivate the car industry to continue the development of such systems for MAC. Many problems have to be solved before the system could be introduced into the market:
- Safety problems,
- Maintenance,
- System efficiency,
- Quality aspects,
- Price.

A1.5 - Comparison of automotive air-conditioning systems operating with CO_2 and R-134a

Title	Comparison of automotive air-conditioning systems operating with CO_2 and R-134a
Authors	M. Preissner, B. Cutler, S. Singanamalla, Y. Hwang, R. Radermacher
Subject	Experimental comparison of R-134a and CO2 systems
Type of document	Paper
Company	Center of Environmental Energy Engineering
Source	IIF-IIR Commission B1, B2, E1 and E2, Purdue University
Date	July 2000
Number of pages	185-192

The main subject of this paper consists in the experimental comparison of R-134a and CO_2 systems on a test bench. Both systems have been tested under the following conditions:

- Ambient air temperature: 25°C to 45°C,
- Compressor rotation speed: 1000 and 1800 RPM.

Introduction

The use of CO_2 as refrigerant has been revived in the last decade and it has been considered for use in stationary and MAC air-conditioning systems. R-134a used in actual systems contributes to the global warming when released to the atmosphere. CO_2 would present the following advantages:

- No refrigerant recovery,
- Independent COP and capacity control,
- Potential for heat pump.

The challenge for CO_2 systems is long-term system leak tightness, the choice of appropriate oil and safety issues which are not addressed yet.

Experimental setup

Tests have been performed on a test bench presented in the paper. The dimensions of the components used are presented Table 1.

Table 1 : Dimension of components

Component	Refrigerant	Face area (dm^2)	Depth (dm)	Core volume (dm^3)	Displacement (cm^3)
Evaporator	R-134a	4	0.8	3.2	
	CO2a	4.2	0.76	3.19	
	CO2b	4.67	0.6	2.8	
Condenser / Gas cooler	R-134a	25.4	0.2	5.08	
	CO2a	25.3	0.23	5.82	
	CO2b	28.4	0.17	4.83	
Compressor	R-134a				155
	CO2				20.7

Two CO_2 systems were tested. They are named CO_2a and CO_2b but have the same piston compressor with a fixed swept volume. The only changes concern the heat exchangers.

The front area of the gas cooler of the CO_2a system is equivalent to the R-134a condenser, whereas the one of the CO_2b system is slightly larger.

All systems were tested according to the conditions presented Table 2.

Table 2 : Test conditions

Indoor			Outdoor			Compressor Speed (RPM)
Temperature (°C)	Relative humidity (%)	Air flow rate (m3/h)	Temperature (°C)	Relative humidity (%)	Frontal Air velocity (m/s)	
27	50	580	25 35 40 45	40	Idling : 1.0 driving : 2.5	Idling : 1000 driving : 1800

Results and discussion

The results obtained during the test campaign are presented Figure 1 (cooling capacity) and Figure 2 (COP).

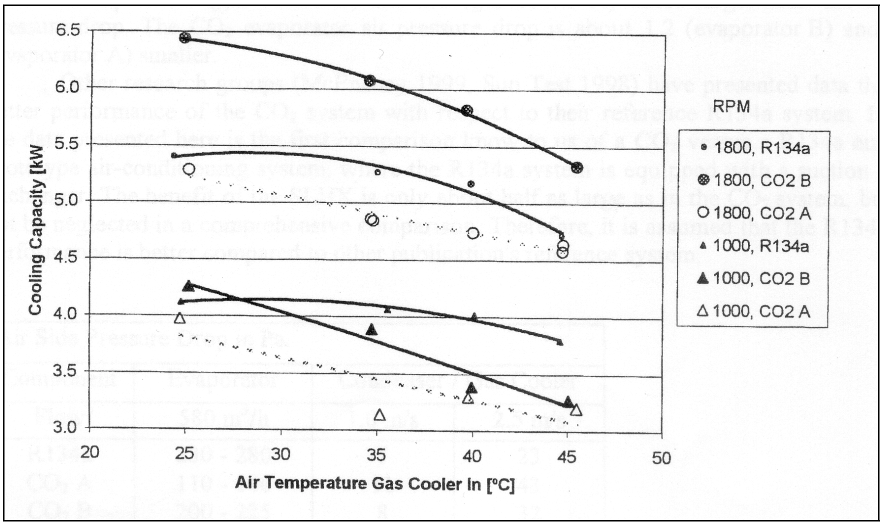


Figure 1 – Cooling capacity

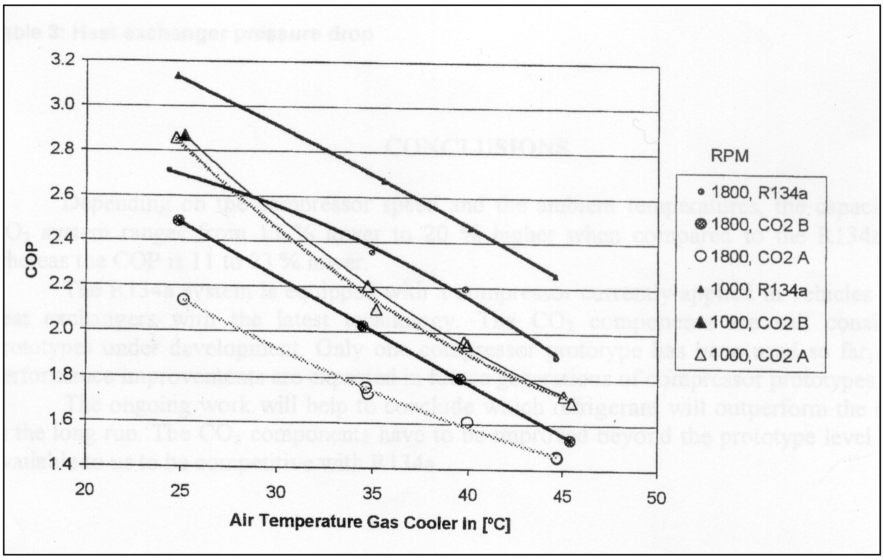


Figure 2 - COP

The first observation is that the CO_2b system is more efficient than the CO_2a one.

For the R-134a system, the cooling capacity ranges from 4.1 to 3.8 (kW) (25°C to 45°C) at 100 RPM and 6.4 to 5.3 at 1800 RPM. At the higher compressor speed, the refrigerant mass flow rate increases. Therefore, the evaporating temperature is lower to transfer the increased cooling capacity. The cooling capacity of the CO_2b system is 0 to 13% lower than the R-134a system (25 to 45°C) at 1000 RPM and 20 to 13% at 1800 RPM.

The COP of the R-134 system ranges from 3.1 to 2.3 (1000 RPM) and from 2.7 to 1.9 (1800 RPM). For both compressor speeds, the CO2b system COP is 8% lower at 25°C and at 45°C, the COP is 23% lower when idle, and 19% lower under driving conditions.

It must be noticed that during the tests, the R-134a system was fitted with an internal heat exchanger. The benefit is only about half as large as in the CO_2 system, but should not be neglected for comprehensive comparison.

Conclusions

Depending on the tests conditions, the cooling capacity of the CO_2 system ranges from 13% lower to 20% higher when compared to the R-134a system, whereas the COP is 11 to 23% lower.

A1.6 - A critical look at R-744 and R-134a mobile air conditioning systems

Title	A critical look at R-744 and R-134a mobile air conditioning systems
Authors	M.S Bhatti
Subject	Theoretical comparison
Type of document	Paper
Company	Delphi Harrisson Thermal systems
Source	SAE International Congress
Date	February1997
Number of pages	25

In this paper, the main point developed by the author is the comparison of the energy performances of R-134a and R-744 MAC systems, and their environmental influence through the TEWI calculation.

Presented in this paper is the critical assessment of the R-744 system including its COP under realistic operating conditions, and its high operating pressure 5 to 12 times higher than R-134a ones.

Even if the GWP of CO_2 (GWP = 1) seems negligible when compared to the R-134a (GWP = 1300), its indirect impact related to the energy consumption is much higher than R-134a system ones.

The author stars the paper with an historical background: the R-744 was used as a refrigerant from 1850 to the invention of chlorofluorocarbon in 1928.

System performances
If today R-134a systems use a sub-critical cycle, it shall be noticed that R-744 cycles are super-critical with very high working pressures.
In this study the calculations are performed both for realistic as well as idealized MAC.
To compare R-134a and R-744 systems, it is crucial to take into account the electrical energy used for fan operation (specially condenser fans).

Idealized systems performances
Table 1 shows the evolution of the cycle COP (without blowers energy consumption). For each system, the COP is calculated taking into account the ambient air temperature and the cooling capacity.

Table 1 – COP evolution of R-744 and R-134a AC system

Road Load Condition	Idle		Down-the-Road	
Refrigerant	R-744	R-134a	R-744	R-134a
Outside air temperature, ^{o}F	120	120	100	100
Outside air relative humidity, %	22.4	22.4	40	40
Outside air absolute humidity, lb_m H_2O/lb_m dry air	0.0167	0.0167	0.0167	0.0167
Evaporator air volume flow rate, ft^3/min	250	250	250	250
Evaporator air mass flow rate, lb_m/min	16.6	16.6	17.3	17.3
Conditioned air temperature, ^{o}F	70	70	50	50
Condenser/gas cooler air volume flow rate, ft^3/min	600	600	2,000	2,000
Condenser/gas cooler air mass flow rate, lb_m/min	40	40	138	138
Condenser/gas cooler air out temperature, ^{o}F	149	147	134	113
Refrigerant charge, lb_m	0.917	2.125	0.917	2.125
Refrigerant mass flow rate, lb_m/min	4.26	3.66	7.08	5.92
Compressor suction pressure, psia	852	86	653	60
Compressor suction temperature, ^{o}F	70	70	50	50
Compressor discharge pressure, psia	2,000	186	1,500	139
Compressor discharge temperature, ^{o}F	182	124	165	105
Compressor isentropic efficiency	1.0	1.0	1.0	1.0
Compressor power, HP	1.20	0.59	2.24	1.04
Condenser/gas cooler out pressure, psia	2,000	186	1,500	139
Condenser/gas cooler out temperature, ^{o}F	120	120	100	100
Condenser/gas cooler temperature effectiveness	1.0	1.0	1.0	1.0
Condenser/gas cooler cooling capacity, Btu/min	270	243	477	426
Evaporator in pressure, psia	852	86	653	60
Evaporator in temperature, ^{o}F	70	70	50	50
Evaporator out pressure, psia	852	86	653	60
Evaporator out temperature, ^{o}F	70	70	50	50
Evaporator temperature effectiveness	1.0	1.0	1.0	1.0
Latent load on evaporator, Btu/min	20	20	175	175
Sensible load on evaporator, Btu/min	199	199	207	207
Evaporator cooling capacity, Btu/min	219	219	382	382
Compressor power, Btu/min	51	25	95	44
Compressor power transmission loss, Btu/min	0	0	0	0
A/C blower power, Btu/min	14	14	14	14
System power, Btu/min	65	39	109	58
Cycle COP	4.26	8.80	4.00	8.71
System COP	3.37	5.62	3.50	6.59

Table 2 - Theoretical cycle COP for R-134a and R-744 systems depending on ambient air temperature

T ambient (°C)	Cooling capacity (kW)	COP R-744	COP R-134a	COP ratio
15.6	0.7	16.9	48.4	0.35
21.1	1.4	6.4	24	0.39
26.7	2.2	6.3	15.5	0.4
32.2	2.9	4.6	11.3	0.4
37.8	3.6	4.0	8.7	0.46
43.3	4.4	2.9	7	0.42

The theoretical COP calculation shows that the R-134a COP is 2.2 to 3 times higher than the R-744 COP. These results prove that R-134a systems present a higher improvement potential.

Realistic system performances

Table 3 shows the predicted performance of the realistic R-744 and R-134a systems. The performance comparisons are for identical cooling capacities both under idle and down the road conditions.

Table 3 – Predicted performance of R-744 and R-134a AC systems under realistic conditions

Road Load Condition	Idle		Down-the-Road	
Refrigerant	R-744	R-134a	R-744	R-134a
Outside air temperature, °F	120	120	100	100
Outside air relative humidity, %	22.4	22.4	40	40
Outside air absolute humidity, lb_m H_2O/lb_m dry air	0.0167	0.0167	0.0167	0.0167
Evaporator air volume flow rate, ft^3/min	250	250	250	250
Evaporator air mass flow rate, lb_m/min	16.6	16.6	17.3	17.3
Conditioned air temperature, °F	70	70	50	50
Condenser/gas cooler air volume flow rate, ft^3/min	600	600	2,000	2,000
Condenser/gas cooler air mass flow rate, lb_m/min	40	40	138	138
Condenser/gas cooler air out temperature, °F	162	159	123	118
Refrigerant charge, lb_m	0.917	2.125	0.917	2.125
Refrigerant mass flow rate, lb_m/min	5.65	6.06	9.04	6.97
Compressor suction pressure, psia	713	67	518	44
Compressor suction temperature, °F	64.4	61.6	41.0	38.5
Compressor discharge pressure, psia	2,125	401	1,750	283
Compressor discharge temperature, °F	251	219	256	201
Compressor isentropic efficiency	0.70	0.65	0.65	0.60
Compressor power, HP	3.83	3.49	9.06	4.60
Condenser/gas cooler out pressure, psia	2,125	381	1,750	269
Condenser/gas cooler out temperature, °F	133	176	116	122
Condenser/gas cooler temperature effectiveness	0.90	0.50	0.90	0.50
Condenser/gas cooler cooling capacity, Btu/min	380	371	759	580
Evaporator in pressure, psia	792	82	578	53
Evaporator in temperature, °F	64.4	67.1	41.0	43.5
Evaporator out pressure, psia	792	74	578	48
Evaporator out temperature, °F	64.4	61.6	41.0	38.5
Evaporator temperature effectiveness	0.90	0.90	0.85	0.85
Latent load on evaporator, Btu/min	20	20	175	175
Sensible load on evaporator, Btu/min	199	199	207	207
Evaporator cooling capacity, Btu/min	219	219	382	382
Compressor power, Btu/min	162	148	382	195
Compressor power transmission loss, Btu/min	5	4	12	6
A/C blower power, Btu/min	26	26	26	26
System power, Btu/min	193	178	420	227
Cycle COP	1.36	1.48	1.03	1.96
System COP	1.13	1.23	0.91	1.68

Results show that the compressor power consumption is 1.10 times the R-134a AC system power consumption under idle conditions and 1.96 times the power absorbed by the R-134a AC system under down the road conditions.

A comparison of the cycle COP for idealized AC systems with the cycle COP under realistic conditions show that the idealistic values are significantly higher and thus due to the fact that irreversibilities were not taken into account. The disparity between idealized and realistic COP cycle, COP of the R-744 AC system is seen to be less than that for the R-134a AC system. This is due to the fact that the compressor discharge pressure of the realistic system has been optimized as described by Inokuty op cit. If a similar optimization is carried out for the R-134a compressor, the disparity between idealized and realistic will narrow and the realistic R-134a AC system COP will be even higher than the idealistic R-744 AC system COP.

Measurement of the degree of optimization of the compressor is the ratio of the actual compressor discharge pressure to the idealized compressor. It is seen from the tables above that for the R-744 AC system, the discharge pressure ratios are respectively 1.06 and 1.17 under idle and down-the-road conditions, whereas the corresponding ratios for the R-134a AC system are 2.16 and 2.04, which attest to the extent of unrealized optimization with the R-134a system.

Global warming impact

The hypothesis taken to calculate the warming effect are listed Tables 4 and 5.

The author assumes that the refrigerant charge is 1000g for R-134a AC system and 430g for R-744 AC system. The emission rate is taken equal to 5.5% of the charge per year for both systems.

Table 5 shows the gases produced during the combustion of gasoline, and that impact on the greenhouse effect.

Table 4

A/C system	Refrigerant charge, gm	Emission rate, gm/yr
R-134a	1,000	55
R-744	430	24

Table 5

Greenhouse gas	Emission rate, gm/gallon
R-744 (CO_2)	8,868
CH_4	0.29
N_2O	0.15

Table 6 shows the EPA Test particular procedure. With an average speed of 33 MPH, the air-conditioning system is on for 100 hours during the year.

Table 7 shows the fuel consumption expressed in gallons of R-134a and R-744 AC systems. The results obtained is the sum of the fuel needed to operate the system and the fuel used to run the AC system.

The annual fuel consumption of an R-134a is taken equal to 23.5 gallons; 3 gallons are used to run the system (this value is within the range of 20 and 25 estimated by auto manufacturers around the world).

To calculate the fuel consumption of the R-744 system, the COP ratio compared to R-134a system is used. Moreover the assumption that the R-744 system is 50% higher than R-134a system is done.

Table 6

Greenhouse gas	Emission rate, gm/gallon
R-744 (CO_2)	8,868
CH_4	0.29
N_2O	0.15

Table 7

	Fuel consumption, gallons		
A/C system	To operate A/C system	To carry A/C system	Total
R-134a	20.5	3.0	23.5
R-744	35.3	4.5	39.8

Table 8

The global warming potential and the atmospheric lifetime of various gases are presented in Table8.

Greenhouse gas	Atmospheric life, yrs	GWP (mass basis)
R-744	50 - 200	1.0
CH_4	14.5	24.5
N_2O	120.0	320
R-134a	14.0	1,300
R-12	102.0	8,500

The GWP index varies with the timeframe called integration time horizon over which the gas is compared to CO_2.

A shorter timeframe emphasizes the climate forcing potential of shorter lived gases while the longer timeframe is more representative of the cumulative effect of a gas in climate over its lifetime.

Table 9 and Table 10 show the global warming impact of R-134a and R-744 systems according to the calculations performed.

Table 9

Greenhouse gas	GWP	Absolute amount, kg/yr	R-744 equivalent amount, kg/yr
Direct Impact			
R-134a	1,300	0.0550	71.5000
Indirect Impact			
R-744 CH_4 N_2O	1 24.5 320	208.4051 0.0068 0.0035	208.4051 0.1670 1.1280
Total Impact			
R-134a + R-744 + CH_4 + N_2O	-	208.5	281.2

Table 10

Greenhouse gas	GWP	Absolute amount, kg/yr	R-744 equivalent amount, kg/yr
Direct Impact			
R-744	1	0.0240	0.0240
Indirect Impact			
R-744 CH_4 N_2O	1 24.5 320	353.3559 0.0116 0.0060	353.3559 0.2831 1.9126
Total Impact			
R-744 + CH_4 + N_2O	-	355.4	355.6

The calculations are made including direct and indirect impacts for each system.

Table 11 presents a direct comparison between the R-134a system and the R-77 system.

Table 11

Equivalent warming	R-134a system	R-744 system	Ratio
Direct	71.5	0.0	0.0
Indirect	209.7	355.6	1.7
Total	281.2	355.6	1.3

The direct impact of the R-134a system is equal to 71.5 kg of CO_2 per year, whereas the direct impact for R-744 is 0. This was calculated assuming a annual leakage of 55 g/year for the R-134a system.

For the indirect (i.e related to the fuel over consumption due to the use of the AC system) it appears that there is a huge difference between the two systems since the R-744 impact is 70% higher than the R-134a one. At the end the global warming impact of the R-744 is 30% higher than the R-134a one.

During the previous calculations, the COP of the R-134a systems was taken equal to 1.63.

Table 12 analyzes the influence of the COP on the global warming impact. If the COP is equal to 3, which can be achieved with some optimization, the global warming impact of the R-134a system would be 45% lower than the R-744 system impact.

Table 12

	TEWI, kg/yr		
R-134a COP_{sys}	R-134a system	R-744 system	TEWI ratio
1.0	395	356	1.11
1.5	296	356	0.83
2.0	247	356	0.69
2.5	217	356	0.61
3.0	197	356	0.55

Table 13

	TEWI, kg/yr		
m_{134} gm/yr	R-134a system	R-744 system	TEWI ratio
0	211	356	0.59
55	283	356	0.79
110	354	356	0.99
165	426	356	1.20

Table 13 presents the analysis of the influence of refrigerant leakage on the global warming impact. If it is taken equal to 0, the global warming impact of the R-134a system will be decreased to 211kg of CO_2 per year. On the other hand, if the leaks reach 165g/yr, the global warming impact of the R-134a system will be 20% higher than the R-744 system one.

Table 14 shows that the influence of the R-744 system weight on the TEWI is negligible.

Table 14

R-744 m_{sys}, kg	TEWI, kg/yr R-134a system	R-744 system	TEWI ratio
13.6	283	343	0.83
15.0	283	346	0.82
17.5	283	350	0.81
20.0	283	355	0.80
22.5	283	360	0.79

Table 15

$\dot{q}_{cool}$, Btu/min	TEWI, kg/yr R-134a system	R-744 system	TEWI ratio
100	155	121	1.28
200	212	203	1.04
300	269	284	0.95
400	326	365	0.89
500	382	447	0.85

Table 15 presents the influence of the cooling capacity on the TEWI. If the average cooling capacity needed is below 200 Btu/min (3.5 kW) the global warming impact will be lower for the R-744 system. Above 200 Btu/min, the warming impact of the R-134a system will be lower.