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Telephon +49 (0) 7223 / 941 - 0 • Fax +49 (0) 7223 / 2 69 50
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Editorial: Ralf Stopp, Christa Siefert

Layout: Vera Westermann
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Foreword

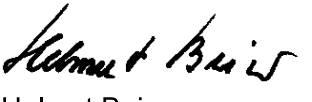
Innovations are shaping our future. Experts predict that there will be more changes in the fields of transmission, electronics and safety of vehicles over the next 15 years than there have been throughout the past 50 years. This drive for innovation is continually providing manufacturers and suppliers with new challenges and is set to significantly alter our world of mobility.

LuK is embracing these challenges. With a wealth of vision and engineering performance, our engineers are once again proving their innovative power.

This volume comprises papers from the 7th LuK Symposium and illustrates our view of technical developments.

We look forward to some interesting discussions with you.

Bühl, in April 2002



Helmut Beier

President
of the LuK Group

Content

CO_2 Compressors

New Technology for Cool Heads and Warm Feet

Willi Parsch
Bernd Brunsch

Introduction

Automotive air-conditioning systems make a significant contribution both to comfort and passive safety. Studies show that the ability to concentrate decreases significantly at interior temperatures above 25 °C.

As with power steering before it, air-conditioning will continue on its victory trail all the way to the compact car class in the years to come. More than 90% of vehicles are expected to be equipped with air-conditioning.

This advantage comes at a high price, because fuel consumption can increase up to 20% when the outside temperature is just 27 °C. This makes the end customer unhappy because fuel economy is significantly less than the values determined in the driving cycles [1].

In addition to the increased CO_2 output from higher fuel consumption, today's coolants also play a role in the greenhouse effect. Since as early as 1994, automakers have sought more environmentally friendly alternatives.

Their research has unearthed a coolant for use in automotive air-conditioning systems that is among the oldest coolants of all: CO_2 [2].

Why CO_2?

Generally only non-combustible coolants are used for climate control in automotive air-conditioning systems. Otherwise, small amounts entering the passenger compartment could become a potential explosion hazard.

After R12 was banned over 10 years ago due to its high global warming potential, there is now a growing movement calling for a ban on R134a as well. Thus the governments of Austria and Denmark are considering prohibiting its use beginning in 2004 and encouraging the substitution of other coolants through tax advantages.

The justification for such considerations is shown in a comparison of these three coolants with regard to their global warming potential (GWP) and the increased fuel consumption caused by the system.

By switching from an R12 to an R134a coolant, it was possible to reduce the equivalent CO_2 output per vehicle by approximately two thirds. This evaluation includes both increased fuel consumption and damage due to air-conditioning system leaks. Switching to CO_2 (R744) provided a further reduction of more than one half.

name	**HCFC (R12)**	**HFC (R134a)**	**CO_2 (R744)**
ozon damage	**yes**	**yes**	**no, it is natural gas**
global warming potential	**GWP = 8100**	**GWP = 1300**	**GWP = 1**
CO_2 emissions from operation	**2600 kg / car**	**2600 kg / car**	**1800 kg / car**
CO_2 GWP-equivalent over service life	**8100 kg / car**	**1300 kg / car**	**0,5 kg / car**
Σ	**10700 kg / car**	**3900 kg / car**	**1800,5 kg / car**

banned in 1990 **2005 due to technical advantages**

Fig. 1: *Environmental Compatibility of Coolants*

In addition to the environmental benefits, the following technical advantages of this new coolant have become increasingly important over time:

- Significantly reduced fuel consumption per annum
- Better cooling performance
- Useful for supplemental heating with a heat pump

System Design

System Layout and Special Features of CO_2

The most important difference from traditional air-conditioning systems using R134a is the additional internal heat exchanger.

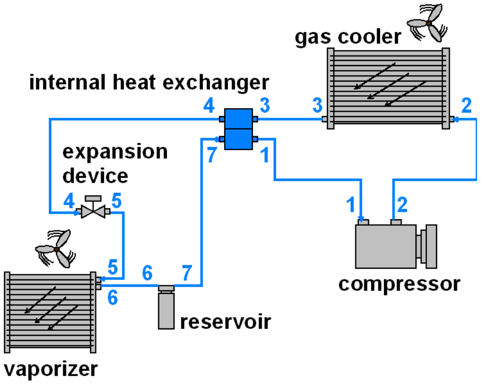


Fig. 2: Schematic Illustration of the CO_2 Coolant Circulation

This heat exchanger is needed [3], [4] because the CO_2 air-conditioning system works with hypercritical heat loss above 31 °C, as illustrated in the following pressure/enthalpy diagram.

The following factors consider only the hypercritical case because this is the more frequent operating condition. After compression 1 - 2 to a hypercritical pressure, the gas is cooled 2 - 3 in a gas cooler, which takes the place of the condenser. There is no condensation in

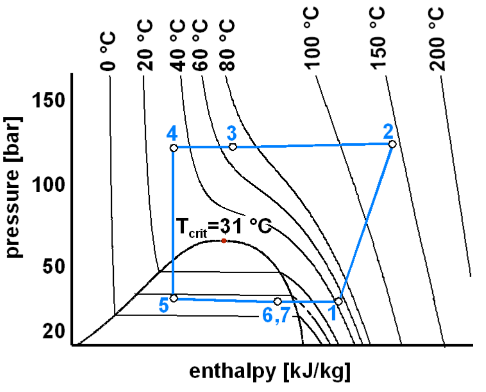


Fig. 3: p-h Diagram of the Coolant Circulation

the hypercritical range. The gas is cooled further 3 - 4 in the downstream heat exchanger, however, there is still no condensation.

The isenthalpic expansion 4 - 5 in the throttle goes below the boiling point into the saturated steam range. Here, the saturated steam portion is condensed and the liquid phase is endothermically evaporated 5 - 6 as much as possible in the evaporator, which brings about the actual cooling effect. In Point 6, there is saturated steam with a very high degree of saturation. After the saturated steam has flown through reservoir 7, which is used for volume control, it absorbs heat in the internal heat exchanger 7 - 1, overheats and thus reaches the starting point again 1.

The internal heat exchanger is necessary because there is normally no overheating control with CO_2, and the evaporator cannot completely evaporate the coolant.

The high pressure with which the system works is also notable here. The advantage is that a smaller flow volume is required to achieve the desired cooling performance because CO_2 has a higher density.

This advantage can be used to increase the cooling performance as well as to reduce the size of the units or lower the losses in the lines.

Thus it was shown that a 25% [5] reduction in fuel consumption can be achieved in certain operating cycles.

In contrast to today's systems, it is also technically useful to operate as a heat pump. See the designs in Section *Heat Pump* for more information on this topic.

Based on design changes, the compressor, despite the high pressure, is lighter and smaller than the current R134a compressor.

The high pressure level, however, affects the required strength of all of the air-conditioning system components [6], [7].

The physical and chemical properties of CO_2 make for another major departure from the current systems. Due to the very small size of the CO_2 molecule, it diffuses very easily through most of the seals currently used.

In addition, its influence on the behavior of the lubricants used and thus on the service life of the compressor is not insignificant.

Requirements for Coolant Compressors

The system properties indicated for CO_2 as a coolant in cooling and heating circulation require completely new compressor designs.

From the very beginning at LuK, development focused on the specific requirements of CO_2.

Initially, standard production compressors for R134a were redesigned for CO_2. Once acceptable operating times were achieved, the focus was shifted to systematically optimizing the entire CO_2 air-conditioning system.

The primary requirements for this type of compressor are:

- Maximum pressures of 135 bar on the high-pressure side
- Very high suction pressures of approximately 40 bar, which are highly variable
- Taking into consideration the fact that CO_2 has a significant impact on the properties of the lubricant used
- Development of special sealing elements for CO_2

In addition to the system properties, there are typical vehicle requirements that still must be addressed:

- Compact installation space
- Low weight
- Low air- and structure-borne noise
- Highest possible efficiency
- Low manufacturing costs

Since clutchless compressors are becoming more common in vehicle applications, the necessary drag power must also be one of the criteria considered in the compressor design.

Technical Description of the Compressor

Today's usual compressor designs for R134a can, in theory, also be used for CO_2. Spiral, vane and axial piston compressors are generally used.

The disadvantage of spiral and vane compressors is that displacement control is not practical with them. It is, therefore, only possible to regulate the mass flow rates using cyclical operation or a bypass valve. Cyclical operation using a magnetic clutch adds to the cost of the clutch and its control requirements. With the bypass solution, particularly at lower cooling outputs, high amounts of energy are lost through the bypass.

It is standard to show the relationship between drive performance and cooling performance as the coefficient of performance (COP). The higher the value, the more effectively the compressor is working. See also figure 4 on this topic.

Piston displacement-controlled axial piston machines currently provide the better alternative with R134a, for the reasons cited above. For CO_2 systems, this principle was applied from the beginning, because it represents the state of the art.

design	vane compressor	spiral compressor	axial piston compressor
mass flow control	bypass	bypass	by variable piston stroke
air-conditioning system coefficient of performance, compressor with maximum flow (COP)	 (2)	 (2)	 (2)
air-conditioning system coefficient of performance, compressor with controlled flow (COP)	 (1)	 (1)	 (1,8)
noise			
cost			

Fig. 4: *Main Selection of Compressors*

The piston displacement of this compressor is controlled by adjusting the stroke of the pistons, which are driven via an angular-moving pivot ring.

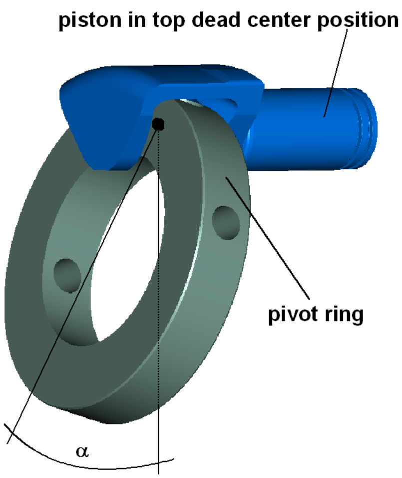


Fig. 5: *Compressor Design, Pivot Ring*

The effective setting and thus the supply volume of the piston is represented by the pivoting angle α. The work output is transferred from the shaft to the pistons via the bolts according to the principle of direct force input. Due to the type of piston connection, only axial forces occur, which allows for low-friction adjustment of the pivot ring. The spring force of the sliding sleeve defines the pivot ring setting during a stop.

The pivot ring is set primarily by the equilibrium of the gas forces on the piston.

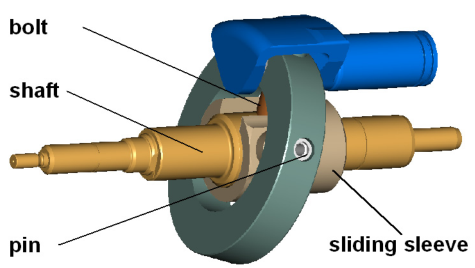


Fig. 6: *Compressor Design, Adjustment*

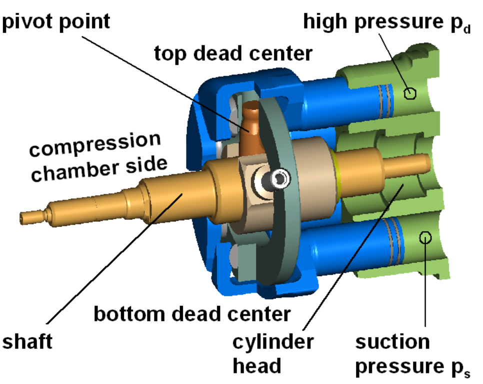


Fig. 7: *Compressor Design, Total System*

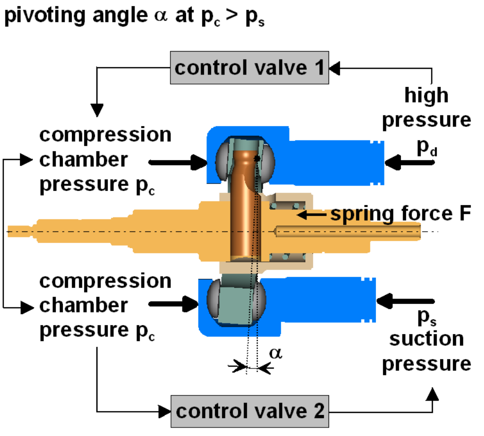


Fig. 8: *Adjusting the Piston Displacement $p_c > p_s$*

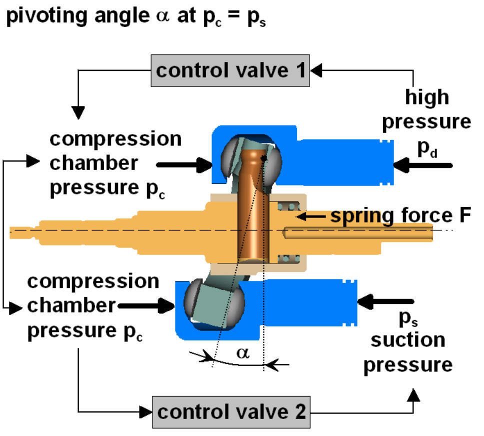


Fig. 9: *Adjusting the Piston Displacement $p_c = p_s$*

Due to its geometry, the high pressure p_d has no influence because it impinges directly on the pivot point. For adjustment, there must be an equilibrium between the suction pressure p_s, compression chamber pressure p_c and the spring force F (figure 8).

Thus, the control element used is the compression chamber pressure on the back of the piston. The pressure is controlled using valves that charge and bleed the compression chamber according to defined criteria (control valve 1 and 2).

Selecting the Pivot Mechanism

Selecting the optimal pivot mechanism for CO_2 operation was a focal point of the development activities. The pneumatic connections of the drive significantly influence the overall behavior of the compressor.

Illustrated here is an example of the conflicting goals between transferring the drive force from the shaft to the pistons and having the pivot mechanism set for the lowest possible friction.

In addition, the following goals were also taken into consideration for optimizing the compressor:

- Adjustment kinematics that are independent of speed, meaning no forward and backward mass moments of inertia through the entire speed range ($M_{free} = 0$)
- As constant a dead-center piston position as possible in the supply stroke to avoid losses in clearance volume over the entire adjustment range
- As little unbalance as possible in the entire pivoting angle range

The above parameters led to the development of the pivot ring. With this part, the force input from piston forces comes together in the pivot point. This means that the top dead center positions of the piston remain constant throughout the entire pivot range.

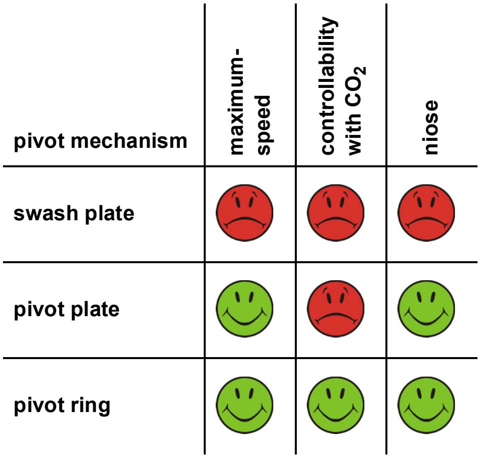


***Fig. 10:** Selecting the Pivot Mechanism*

With other pivot mechanisms, the unbalance is only compensated for in a single setting. In all other positions and during the actual pivoting movement, mass moments of inertia occur due to unbalance.

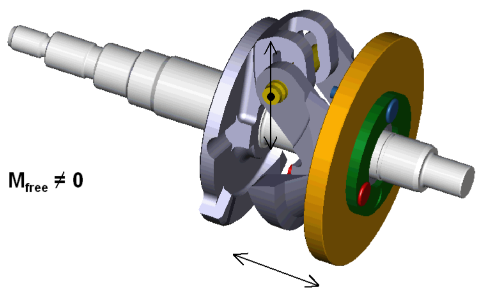


***Fig. 11:** Adjusting the Pivot Plate*

Due to the geometry of the pivot ring, the unbalance throughout the entire pivot range is almost fully compensated for.

Since the focal point of the pivot ring throughout the entire pivot range is on the center axis of the drive shaft, compensation for mass moments of inertia remains (M_{free} = 0).

This allows for a decoupling of the flow and speed, meaning that even during sudden acceleration, there is no piston displacement adjustment.

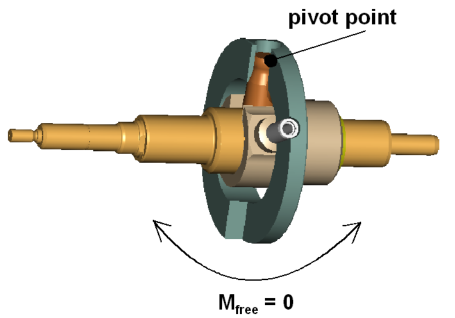


***Fig. 12:** Adjusting the Pivot Ring*

Selecting the Housing Design

The following requirements were at the forefront when determining the housing design and the materials used:

- Pressure resistance and creep resistance under operating points typical for CO_2
- Housing temperatures up to 180 °C with simultaneous high compression chamber pressures of 35 bar
- Continuous pressures of 50 - 70 bar when the system is stopped
- When designing the housing, the special requirements of the sealing elements must be taken into consideration
- Porosities lead to leaks in cast parts much sooner than when using R134a

Based on safety aspects, the first compressor designs used steel housings. When development with CO_2 began, there was only a little validation data available relating to actual application temperatures and possible overloads. Safety against bursts could be improved considerably due to the ductility of the steel materials used. The compressor designs ran smoothly from the beginning with these housings.

In the design selection for a mass-produced coolant compressor, the housing design was re-evaluated. As with the R134a coolant compressors currently used, a forged steel housing and a die-cast aluminum housing were compared. After comprehensive analyses of the functionality and manufacturing costs, the steel housing was chosen.

Steel housings have advantages with regard to pressure resistance and temperature resistance. However, compared to aluminum housings, steel housings have two serious disadvantages: higher weight and cutting expense due to less precise blank manufacturing.

Using a very special steel housing design, these fundamental disadvantages were transformed into advantages. The steel housings now used are lighter than an aluminum design. By using a housing pipe design with the simplest geometry, the steel housing can now be formed very close to the final contour.

The functionality was placed directly in the region of the cylinder head so that comparable designs can be implemented for different housings. See figures 7 and 14 on this topic.

Lubrication and Shaft Sealing

The lubrication and the shaft sealing represent the greatest challenges in developing a new CO_2 coolant compressor. This is caused by the quite variable high solubility of the oils used, the highly variable density of CO_2 and the high operating pressures.

The high solubility means that lubricating oil can enter the solution on the high-pressure side. The dissolved oil is transported through the air-conditioning system by the coolant. The oil is separated out on the suction side due to the vast pressure deviations between the suction side and the high pressure side such that the oil can no longer be supplied back. This can result in a complete loss of oil in the compressor.

Another potential hazard when there is oil loss in the compressor is the piston displacement control when charging and bleeding the compression chamber. There is potential for leakage from the compressor compression chamber to the coolant system due to a direct link to the suction side of the system.

This problem was resolved by using a double oil separator system with a throttle in the compressor.

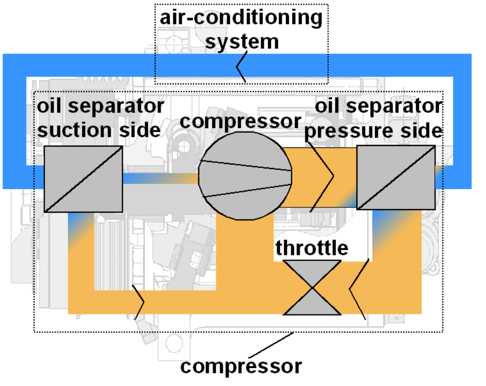


Fig. 13: *Double Oil Separator System*

The shaft is sealed with a floating ring seal because conventional elastomer seals could not withstand the high compression chamber pressures of up to 60 bar.

The principle of a floating ring appears quite simple at first glance, however, the exact configuration of such seals is somewhat difficult.

The actual sealing element is an annular surface between the static and the shaft-side part. This surface must meet high demands with regard to precision and surface quality. It seals via a capillary column filled with oil. This column is defined by the hydraulic compensation of the sealing element as well as the width of the sealing surface.

The goal is a sealing clearance that only just supports. If the gap is greater, oil transport through the column increases, resulting in impermissibly high basic leakage. If the supporting clearance is too small, friction increases before the sealing surface runs with mixed friction and destroys its surface.

With CO_2 coolant compressors, there is also the issue of highly variable suction pressures, which have considerable influence on the compression chamber pressures. The compression chamber pressures define in large part the seal compensation, which results in a considerable configuration problem. Neither accident safety nor an acceptable compromise between basic leakage and low friction could be found using a hydrodynamic basic configuration.

Therefore, new ceramic materials were developed, which can withstand operation in the mixed friction area. This makes possible a lower lubricant clearance during normal operation. Under extreme load, the sealing elements still run in the mixed friction area.

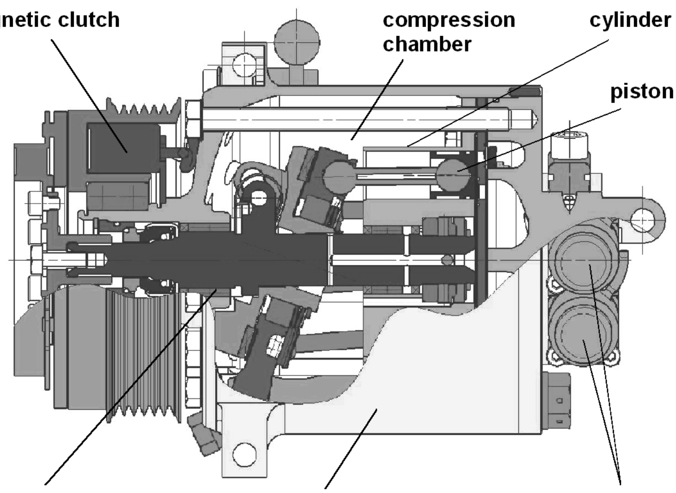


Fig. 14: Overall Profile of the Coolant Compressor

The necessary cooling with coolant oil was achieved by specially designing the sealing environment.

Performance Data on the Coolant Compressor

Piston Displacement and Space

When designing the compressor piston displacement, the geometric flow must be correspondingly smaller due to the high density of CO_2.

To achieve the same level of cooling, only about 13% of the flow needed with an R134a coolant compressor is required with CO_2. [8] Today's standard 170 cm^3 compressor for R134a can be replaced with a powerful 22 cm^3 CO_2 compressor.

The cooling capacity of today's air-conditioning systems is limited by the compressor piston displacement. Piston displacements above 170 cm^3 are not currently possible because there is not sufficient space for a suitable compressor in the engine compartment.

CO_2 compressors offer new options in this arena. Currently, the maximum piston displacements are 33 cm^3, which provides approximately 50% more cooling capacity.

Efficiency, Delivery Rate, Drive Torques

The most important evaluation criteria are the isentropic and volumetric efficiency. The volumetric efficiency, or delivery ratio, primarily takes into account the losses from leaks and re-expansion. The losses from thermal transport and the flow losses are taken into consideration in the isentropic efficiency.

The major influences on efficiency recognized during development were the clearance volume and the suction gas heating. CO_2 has significantly better thermal transport properties than R134a. This means that the heated compressed gas transfers heat to the suction gas. In order to keep this gas to heat transfer low, the suction gasses must be isolated as in high-performance combustion engines.

On the other hand, the CO_2 process works with a significantly lower pressure ratio than R134a – four times as low – which increases efficiency. The overall efficiency of CO_2 processes is considerably higher than the values for R134a.

Results from Vehicle Testing

Performance During Cooling Operation

As indicated in Section *System Design*, there are enormous technical advantages to using CO_2 aside from the environmental aspects. The increased performance capacity can be demonstrated relatively quickly during cooling operation. The CO_2 systems showed significant advantages, particularly when the outside temperature was less than 30 °C.

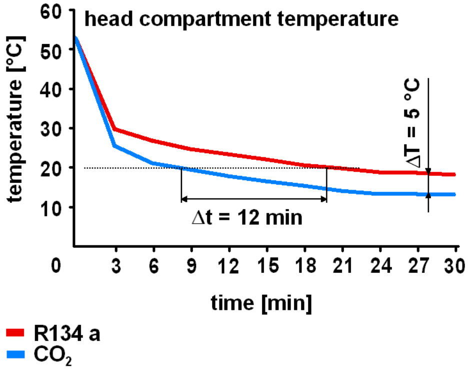


Fig. 15: Comparison of the CO_2 and R134a Cooling Curves

The performance capacity of the system is quickly reduced when outside temperatures are very high due to a worsening rate of exchange in the gas cooler. This disadvantage can be minimized, however, by optimizing the gas cooler and through specific installation positions in the engine compartment as well as by enlarging the compressor capacity [9].

The two systems were compared in a mid-size car in 'stop-and-go' city traffic. Considerably better cooling times and lower temperatures were achieved with CO_2, even at very high outside temperatures. See figure 15.

Heat Pump

The use of an air conditioner as a heat pump is not a primary use of CO_2. These systems have repeatedly been studied in the past. So far, satisfactory results have yet to be achieved.

The cause was identified as the varying gas densities of the coolant used at low temperatures. While the density of R134a is reduced to 8.32 kg/m^3 at -15 °C, the value for CO_2 is still 60.2 kg/m^3. If the density is too low, operation as a heat pump is no longer technically practical.

Another heat source is needed in addition to the existing components when operating as a heat pump. The engine exhaust gasses, outside air or coolant water come into play here.

The advantage with the thermal absorption from the coolant water is that – if the heat exchanger is properly dimensioned – the engine's warm-up phase can be reduced because the simultaneously running compressor also contributes to it. The heat pump can provide high heat output within a short time.

A schematic illustration of the entire air-conditioning system with the necessary components is shown in figure 17.

The effectiveness of a CO_2 heat pump was compared to a mass produced heating system in a vehicle test. The mass produced system was equipped with supplemental heaters as part of its design in order to reach an acceptable heat output.

In general, the CO_2 heat pump cut the warm-up times in half.

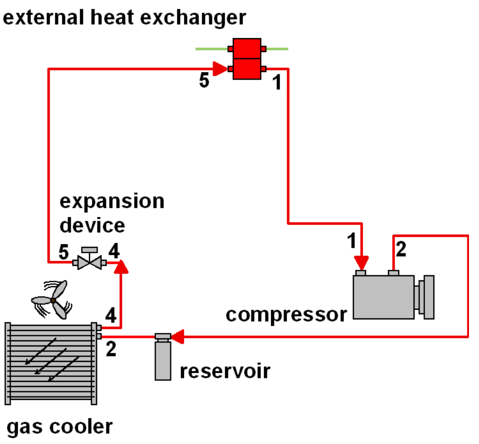


Fig. 16: *CO_2 Heat Pump Function*

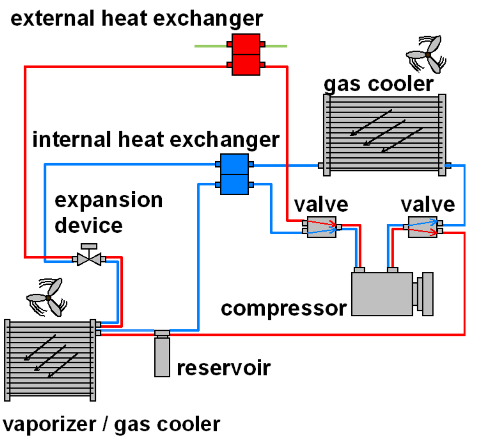


Fig. 17: *Overall CO_2 Air-Conditioning System*

Fuel Consumption

At extreme temperatures, above 35 °C, the efficiency of the CO_2 air-conditioning system was significantly reduced in comparison to R134a. The cooling advantages previously demonstrated were only achieved with higher compressor output.

At moderate temperatures, the advantages in efficiency were clearly applicable. The lower friction of the CO_2 compressor is attributed to the small piston displacement and the resulting smaller friction diameter of the engine parts.

Measurements indicate major advantages with CO_2 when the air-conditioning system was running the entire time. See figure 19, left.

Even measurements taken with the air-conditioning system switched off showed that savings could be achieved as compared to R134a when a clutchless compressor is used [10].

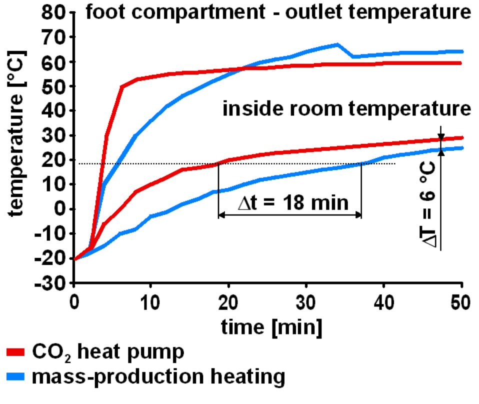


Fig. 18: *Comparison of Warm-up Times*

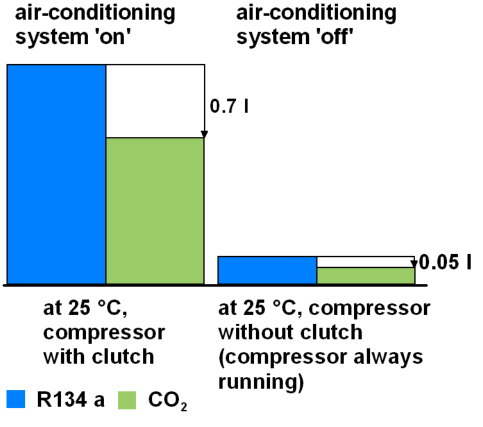


Fig. 19: *Comparison of Advantages in Fuel Consumption*

Summary and Future

The development of CO_2 air-conditioning systems was originally initiated for environmental reasons. The major technical advantages were quickly detected in addition.

We can now start from the premise that CO_2 air-conditioning systems cool significantly better and, in most situations, consume considerably less fuel than R134a systems.

However, the special properties of CO_2 and the high system pressures, among other issues, necessitate the development of new compressors.

Based on the current state of the art, the principle of the axial piston compressor can be further improved with the development of a pivot ring.

Based on the higher density of CO_2 and the use of steel housings, the space requirements can be reduced significantly as compared to R134a compressors with the same cooling capacity.

Supplemental heat will become mandatory as demand for comfort increases and combustion engines are optimized further. The function as a heat pump can make the use of electric supplemental heating units superfluous. With proper configuration, the heating capacity is significantly higher than with current heating systems.

It is anticipated that mass production of CO_2 air-conditioning systems will begin in 2005 and that all new developments in vehicles will be based on CO_2 air-conditioning systems beginning in 2010.

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