CO_2 - A FUTURE REFRIGERANT?

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1. INTRODUCTION

The increasing interest in CO_2 as refrigerant in light commercial applications has created a demand for small compressors and other components. For CO_2 cooling at ambient temperatures above 30 ºC, the thermodynamic properties of CO_2 require the transcritical cycle, which has gas cooling instead of condensation. This implies, not only a different control strategy compared to normal systems, but also a compressor capable of operating with high differential pressures. As found earlier this implies that a favorable compressor choice is a reciprocating type with piston rings, as any leakage of the cylinder needs to be minimized to achieve a high efficiency /1,2/. The high volumetric efficiency of CO_2 in combination with the need for good cylinder sealing seems to limit the possibility to design the compressors for very low refrigeration capacities.

As mentioned /3/, a development of a small CO_2 compressor and valve for cooling capacities in the range of 400 W to 1,2 kW refrigeration capacity at -10 ºC was launched. The development has progressed in sequential steps since the first initiative in early 2001 and tested in various applications and ambient conditions. This paper describes the development and resulting performance characteristics of a small compressor working with CO_2 as the refrigerant and contributes to the evaluation of CO_2 as a future refrigerant.

2. COMPRESSOR

Due to the thermodynamic properties of CO_2 the effects influencing the compressor efficiency are different compared to those of conventional refrigerants like HFCs. Static pressure losses and heat transfer losses in the cylinder are of minor importance for the efficiency /2/. Also the leakage is known to be a driving factor in the determination of a suitable type of compression mechanism. Due to the large difference between the suction and the discharge pressure, cylinder leakage is critical for the compression process performance. This requirement led to the choice of a single piston compressor with piston rings. Another important factor was found to be the heat transfer outside the cylinder- especially the suction gas heating inside the suction plenum. The efficiency of the present compressor was - compared to earlier compressor versions - significantly increased by the reduction of heat transfer between the cold suction gas and the compressor /1,2/.

The compressor, as being developed so far is shown in *Figure 2*. Various strokes in 3 different models presently offer a capacity range from 400 W to 1200 W refrigeration at -10 ºC evaporation and 32 ºC compressor return and high side heat exchanger outlet temperatures.

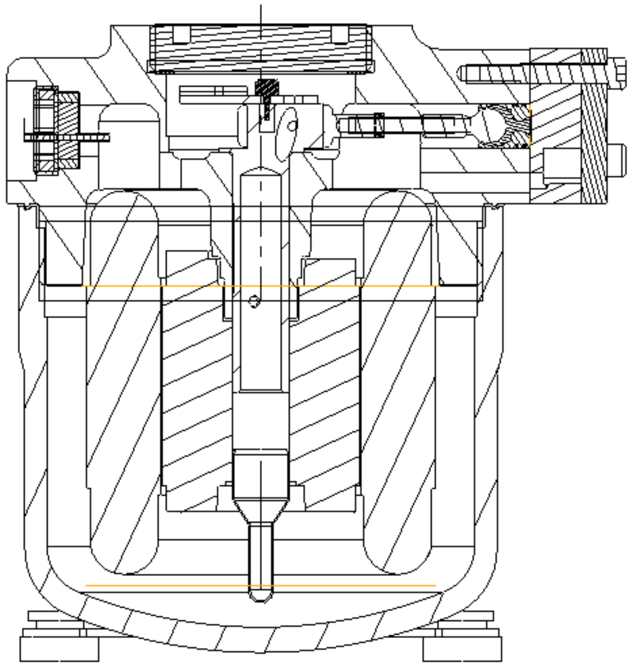

Figure 1: CO_2 compressor concept

As shown in *Figure 1* a thread connects the compressor block and the shell. The power feed through and the lid on top of the compressor are also held to the block by applying a thread. Apart from this, the overall compressor design and especially the drive and lubrication concept correspond to the design of traditional HFC/HC hermetic type compressors for light commercial application. A standard single-phase motor, as used today in HFC/HC hermetic type compressor, is applied. Motors for different voltage and frequency are available. As mentioned before, suction gas heating is, besides cylinder leakage, a major source of energetic and volumetric losses of the CO_2 compression process /2/, so a direct suction is choosen. Not considering valve losses, the pressure drop in the suction chamber is negligible due to the high suction pressure and therefore hardly contributes to volumetric process losses.

3. SYSTEM PERFORMANCE

A number of compressor prototypes for have undertaken several independent performance tests in applications such as bottle coolers, vending machines and heat pumps. The compressor performance data has been analyzed in some detail and a polynomial fit for the efficiency map has been derived. It covers isentropic and volumetric efficiencies as a function of discharge and suction pressures. This data is used for an analysis of the performance of CO_2 systems.

The refrigeration process implemented in the applications described here is a transcritical cycle with a one stage compression and expansion. An important parameter for this cycle is the control of the high side pressure. A steady state analysis, assuming isenthalpic expansion and gas cooler exit temperature of 3 K above ambient temperature can be made based on the compressor data.

Figure 2 shows the results of such a steady state analysis for -10 °C evaporation temperature and different high side pressures. The ambient temperature is 32 °C. The cycle has an inherent characteristic discharge pressure for either optimisation of COP or cooling capacity (Qe) and we clearly identify these pressures. At the optimal COP point of 1.43 we get a discharge pressure of 87 bar with a corresponding cooling capacity Qe of 859 W. At a pressure of 97 bar we find the maximum cooling capacity of 894 W with a corresponding COP of 1.36. This simulation reflects an ideal situation with continuously running compressor. In a real application there are several effects playing in to change the optimal discharge pressure setting. For example the cycling introduces losses and control valves may have slight dependencies on the refrigerant mass flow etc. In addition to these effects there may be a difference in optimal pressure depending on the specific condition that the application experiences, e.g., if it is a pull down or if we are looking at a steady state condition.

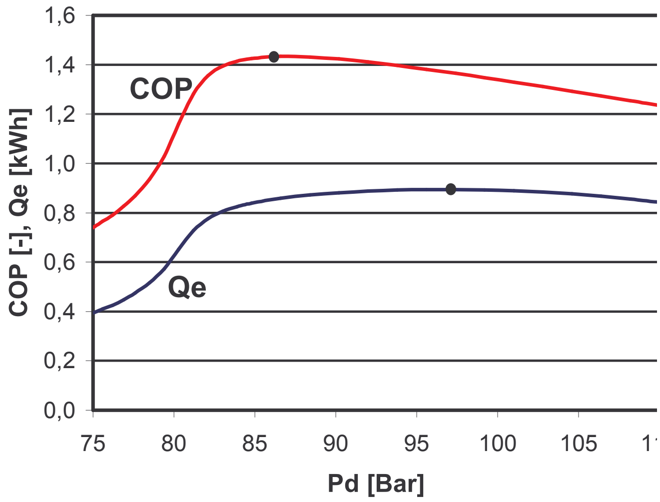

Figure 2 Steady state analysis of the trans-critical CO_2 cycle

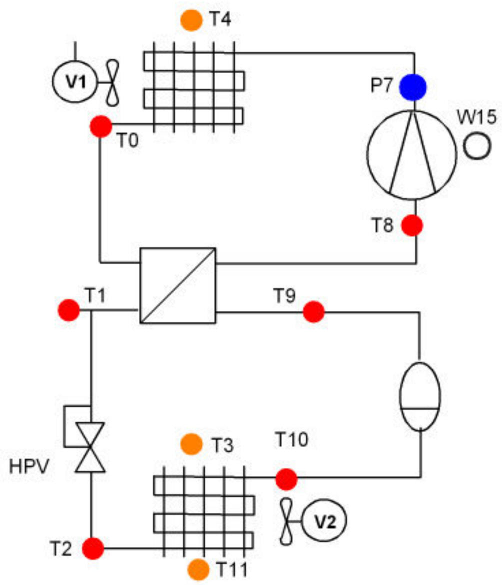

Figure 3 Schematic diagram of the implemented CO_2 cycle.

Figure 3 shows a schematic diagram of the refrigeration cycle implemented in an bottle cooling applications. The individual components are the compressor; a gas cooler and an internal heat exchanger; a mechanical filter in connection with the high pressure control expansion valve was developed in parallel to the compressor; the evaporator is a normal fin & tube type suited for the pressures experienced in CO_2 systems and the accumulator is placed in the cold area close to the evaporator and has the function of accommodating filling change due to changes of ambient condition, discharge pressure setting or evaporator pressure changes. It is important for the CO_2 cycle that the gas cooler exit temperature should be as close to ambient as possible.

The internal heat exchanger has a twofold purpose. On the discharge side it contributes to reduce the temperature before expansion - thereby increasing Qe, and on the suction side it ensures that excess liquid from the accumulator is evaporated before returning to the compressor. The high pressure valve is an active controller keeping the discharge pressure at a set level acting as an internal relief valve to suction side. At the same time the valve acts as the expansion device. Several test were performed on the coolers. Based on the steady state analysis described above, discharge pressures in the range of 85-95 bar were used. The best result yielded an about 35 % energy saving compared to R134a baseline, which represents a state of the art machine as being placed in the market today. The data were recorded at an ambient temperature of 32 °C and 65 % relative humidity.

4. CONCLUSION

In conclusion, the implementation of a transcritical CO2 cycle in bottle cooling equipment applications was demonstrated. The results have shown an improved energy efficiency compared to baseline commercial R134a capillary tube controlled systems. A series of measurements have confirmed the existence of an optimal high pressure setting for ambient temperatures above 30 ºC. The optimal pressure corresponds to the theoretical prediction from a steady state analysis. We have found stable performance for different ambient temperatures.

A CO_2 compressor concept for cooling capacities in the range of 400 W to 1,2 kW refrigeration capacity at -10 ºC is development. In addition a mechanical high pressure controller was designed, which ensures system operation at optimal conditions and at the same time enhances the safety of the system, as it acts as an internal relief from the high pressure to the low pressure side. The development for both key components has progressed in sequential steps since the first initiative in early 2001. The present designs showed promising results regarding performance and reliability, so that an industrialization process is in progress.

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