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DISCHARGE PRESSURE OPTIMIZATION FOR CO_2 TRANSCRITICAL CYCLE USING A CAPILLARY TUBE

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ABSTRACT

This work is focused on the evaluation of the performance of a single stage CO_2 reciprocating compressor working on a beverage cooler application. A glass door merchandiser (GDM) was tested to develop a procedure to determine the best combination of capillary tube and refrigerant charge. Fin and tube heat exchangers were used both for the evaporator and the gas cooler. The criteria to choose the combination was the total energy consumption of the system. The theoretical optimum discharge pressure was determined point by point during the "ON" period of the cycle and was compared to the experimental discharge pressure. The results showed that the closer profile to the optimum profile was the best in terms of energy consumption. The system was also tested with R134a and the results were compared showing 26% of energy savings in favor of the CO_2 system.

1. INTRODUCTION

Since environmental regulations for the phase-out of fluorine refrigerants are becoming more severe, alternatives for substituting the current ones for commercial and light commercial applications are more and more needed. The use of hydrocarbons on refrigeration systems is limited due to its flammability, and large amounts of charge inside commercial systems.

In the past few years, many researchers and companies have been working on the development of CO_2 compressors and refrigeration systems as an alternative. Lorentzen (1994) has emphasized in his work the revival of CO_2 as a refrigerant.

Typical applications for CO_2 are low pressure circuit on cascade refrigeration systems, heat pumps for water heating, automotive air conditioners and so on.

One of its characteristics is that it has a very low critical temperature (30.98°C), providing a singular refrigeration cycle operation as shown on figure 1. It shows a comparison of a transcritical cycle with CO_2 between a subcritical cycle with HFC-134a.

It can be seen that there is no condensation on a transcritical cycle, so the heat exchanger for the high pressure side is then called a gas-cooler. Gosney (1982) shows that for a given operating condition, an optimum COP can be achieved by adjusting the system high pressure. A slight change in the outlet temperature of the gas-cooler leads to a lower optimum discharge pressure.

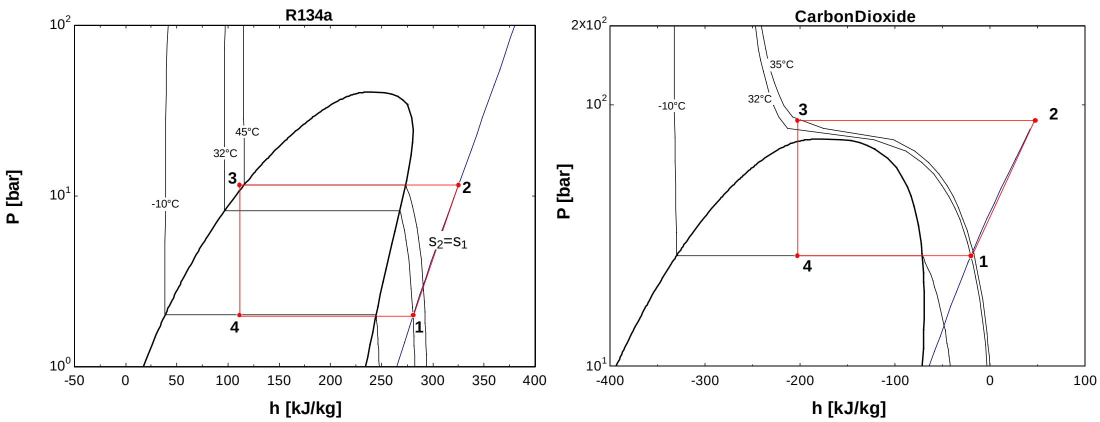

Figure 1. Subcritical and transcritical refrigeration cycles.

With several companies announcing their developments and advances on CO_2 compressors, the interest for applications of this refrigerant on light commercial refrigeration became higher, mostly for beverage cooling and ice cream freezers.

Carbon dioxide has a special characteristics that is its high pressure levels inside the system. As a result for that, many of the work regarding heat exchangers and expansion devices is being done to develop special devices for CO_2 systems. Many companies are applying microchannel heat exchangers, variable expansion devices and also high pressure control devices. This approach will obviously add cost to the system, and put a barrier for the adoption of this refrigerant.

Maciel *et al.* (2005) had previously worked on the development of CO_2 systems for light commercial applications.

DeAngelis and Hnrjak (2005) developed a system for a similar application to the GDM but used a test rig to evaluate the system performance. They found that the CO_2 system performed better than the R134a baseline system, but the concept was different from one used in this work, which would add cost to the system because they used a two-stage compressor and microchannel gas coolers.

The aim of this work is to present results of application tests of CO2 refrigeration system assembled in a GDM, comparing to R134a @32°C. It is also presented a methodology to optimize the combination of capillary tube and refrigerant charge. This methodology puts a special focus on the discharge pressure profile versus the theoretical optimum discharge pressure. A future step is to account the losses of not having an active discharge pressure control device.

2. METHODOLOGY

In order to compare the CO_2 systems, an experimental approach was adopted. The system was obtained with the manufacturer, as well as the baseline results with R134a.

The system was tested on environmentally controlled chambers in accordance with the Brazilian standard ABNT 12863 which is based on ISO 7371. All chambers have a ambient temperature variation of ± 0.5°C.

It is possible to measure power consumption of the system with a wattmeter, which has an uncertainty of ± 0.33 % FS, all uncertainties lead to a typical energy consumption uncertainty of ± 3%.

2.1 Energy consumption optimization

Because of the approach on the development of the system was to use fin&tube heat exchangers and capillary tubes as expansion devices, it was needed to determine what was the optimum combination between refrigerant charge and capillary tube length that would lead to the best value for energy consumption.

To do so, an initially estimated capillary tube length was used and two other tubes were tested, one longer and the other one shorter than the estimation.

To determine this optimum configuration, for each capillary tube the refrigerant charge was varied in order to identify a minimum energy consumption value. This energy consumption was calculated using 4 hours of repeatable cycles after the system was operating at a stable condition, that means repeatable cycle period, power, pressure and temperature levels and profiles.

A schematic diagram of the refrigeration system tested is shown on figure 2.

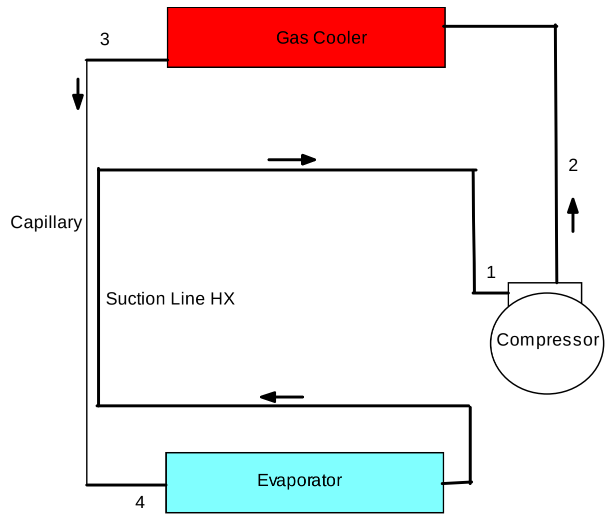

Figure 2. Schematic diagram of the refrigeration cycle.

It's well known that for each transcritical refrigeration cycle condition, there is a value of discharge pressure that provides a maximum COP. From figure 3 it can be noticed this in a *p-h* diagram. This discharge pressure optimum value can be obtained performing a maximization of the COP defined in equation 1.

$$COP = \frac{(h_1 - h_3)}{(h_2 - h_1)_{isentropic}} \tag{1}$$

Where the point 1 is accounting the outlet of the suction line heat exchanger, which is very close to the compressor, and point 3 is considered as an isenthalpic expansion process.

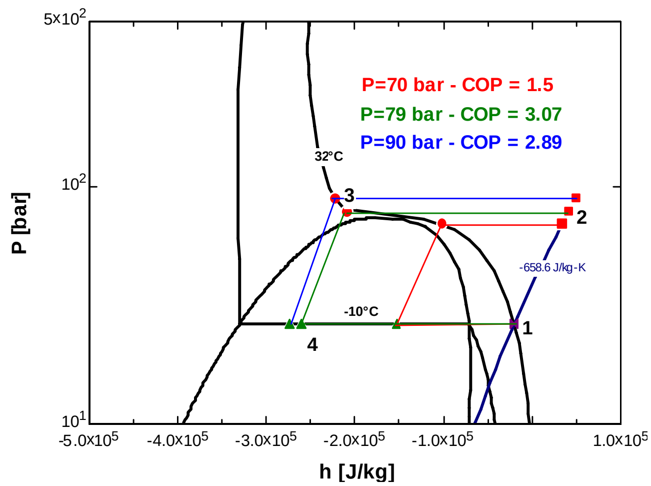

Figure 3. *P-h* diagram of the transcritical cycle showing the optimum discharge pressure.

The maximization is made by varying the discharge pressure and keeping the same temperatures of the points. The isentropic efficiency of the compressor doesn't affect the optimum COP so it can be accounted as a isentropic process.

This procedure is applied to each point of the "ON" period of the cycle to determine an optimum discharge pressure profile, so it can be compared to the real profile and corroborate the decision based on the energy consumption with the theoretical indications of a quasi-optimum discharge pressure profile. The quasi word means that the profile is not self adjustable to the ambient and system operation conditions.

The final value for energy consumption was always determined at the end of a pull-down test for the GDM.

2.2 GDM refrigeration system

2.2.1 Original system and instrumentation
The glass-door merchandiser tested is presented on figure 4. It has a capacity of 540 soda cans, resulting on a refrigerating capacity requirement of around 600W. The system was tested fully charged with products.

Figure 4. GDM tested.

The sealed unit of the refrigeration system was mounted on a split system arrangement. The three capillary tubes tested had the same internal heat exchanger length to keep the same effectiveness.

Several measurements of can temperatures were performed on all the shelves, the four cans in the corners and the central can of the first row of each shelf were monitored. It was also measured the compressor suction temperature located 100mm from the inlet, the discharge temperature located 100mm from the outlet of the compressor, the inlet, outlet and middle temperature of the condenser/gas cooler, the inlet and outlet temperatures of the suction line heat exchanger, compressor body temperature, the inlet and outlet of the suction accumulator, air temperatures at the inlet and outlet of the condenser/gas cooler and evaporator, and also the ambient temperature. All temperatures were obtained using 24AWG T-type thermocouples calibrated at Embraco laboratories with an uncertainty of ± 0.2°C.

Suction and discharge pressures were also measured by transducers with an uncertainty of ± 0.2bar.

2.2.2 CO_2 compressor
The CO_2 compressor used on this analysis has a volumetric displacement of 1.75cc and provided 858W of cooling capacity @ -10ºC/83bar of evaporating temperature and discharge pressure respectively. It operates at 115V/60Hz It has a totally different design concept than for the R134a compressor. Compressor appearance is shown on figure 5.

Figure 5. CO_2 compressor

3. RESULTS

3.1 Refrigerant Charge and Capillary Determination

The first test with the GDM cabinet was to evaluate its energy consumption in the original configuration, which will be discussed later in comparisons.

Then the changes for CO_2 applications were performed, and the system had its refrigerant charge and capillary tube length determined by minimum energy consumption criteria @ 32°C ambient temperature. The results for the optimization are shown on figure 6.

From the graphic it can be seen that there’s always an optimum region where the refrigeration system operates at a lower energy consumption level.

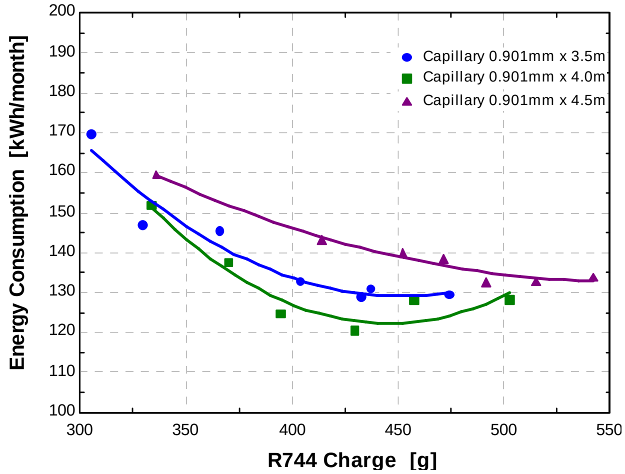

Figure 6. Results for R744 charge and capillary tube optimization.

From the previous theoretical analysis, the optimum discharge pressure can be obtained and plotted against the measured discharge pressure to evaluate the best profile during the “ON” cycle.

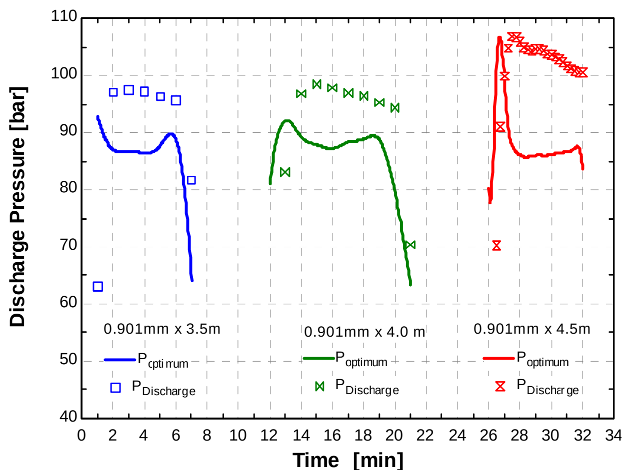

Figure 7. Discharge pressure profiles for various Capillary tubes .

Figure 7 shows this behavior for the best charge values for each capillary tube. Also from the theoretical analysis it can be noticed that the losses in COP are much higher when operating at lower discharge pressures than the optimum value. Figure 8 shows this behavior.

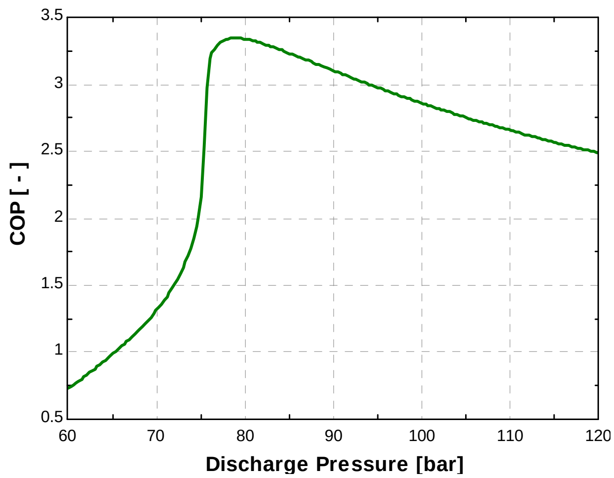

Figure 8. COP vs discharge pressure.

From this figure above and the energy consumption of the system, the optimum combination of refrigerant charge and capillary tube was found to be the 0.908mm i.d. x 4.0m tube with 430g.

Figure 9 shows a comparison between the original R134a system @ 32°C ambient temperature. From that it can be seen that the CO_2 system had a much larger energy efficiency than R134a system. It is important to remark that the HFC system does not uses a top of line compressor in terms of energy efficiency. The authors agree that if the original compressor was replaced the energy consumption could be much lower but this not represents the current situation of the systems in the field. The authors find to be much more fair to compare CO_2 compressor technology with the technology wich s currently wide spreaded in the equipments running in the field.

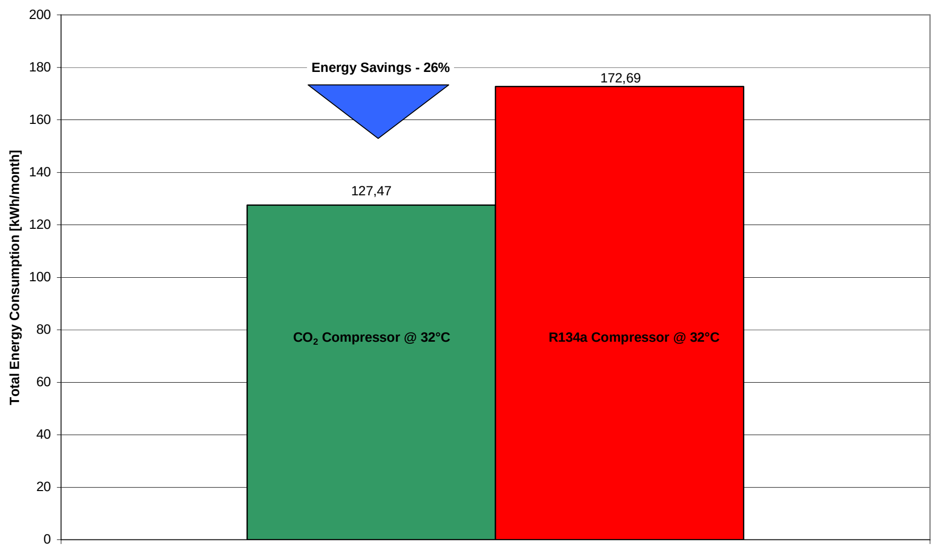

Figure 9. Energy consumption comparison between CO_2 system and R134a.

This difference is mainly because the isentropic efficiency of the CO_2 compressor is far better than the R134a compressors applied in the majority of the light commercial refrigeration market. The CO_2 compressor's isentropic efficiency is nowadays above 0.6, and for R134a compressors typically applied in this field a representative value of isentropic efficiency is around 0.4 ~ 0.5.

4. CONCLUSIONS

A single stage CO_2 reciprocating compressor was assembled and tested on a GDM to compare its performance with a R134a refrigeration system. The results showed 26 % of energy savings in favor of CO_2.

A procedure and criteria to determine the best combination of capillary restriction and refrigerant charge was shown based on the optimum discharge pressure profile during the "ON" period of the cycle. The optimum profile was compared to the experimental profile and the closer profile showed the best energy consumption.

Improvements in order to quantify the losses on COP, cooling capacity and power consumption at "off-design" points shall be addressed on future works.

5. REFERENCES

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