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An experimental evaluation of the transcritical CO_2 refrigerator performances using an internal heat exchanger

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ABSTRACT

The classical substances as hydrochlorofluorocarbons (HCFCs) used as working fluids in the vapour compression plants have to be replaced by new substances because of their ozone depletion potential and their greenhouse effect. Carbon dioxide (CO_2) is non-toxic, non-flammable, has zero ozone depletion potential and negligible global warming potential as refrigerant. Referring to a transcritical CO_2 cycle working as a classical ''split-system'' to cool air in residential applications, the aim of this paper is the evaluation of the energy performances using an internal heat exchanger. The experimental plant employs a semi-hermetic compressor, plate-finned tube type heat exchangers, a back pressure valve electronically controlled and an expansion valve. Besides it is possible to control the flash gas produced in the liquid receiver thanks to another semi-hermetic compressor linked to an inverter. An increase of the coefficient of performance has been found using the internal heat exchanger. The comparison of the coefficients of performance of two cycles, working with and without the internal heat exchanger, is discussed.

Evaluation expérimentale des systèmes frigorifiques au CO_2 transcritique utilisant un échangeur de chaleur interne

Mots clés : Système frigorifique ; Système à compression ; Dioxyde de carbone ; Cycle transcritique ; Expérimentation ; Amélioration ; Performance ; Échangeur de chaleur ; Liquide-vapeur

1. Introduction

The use of natural refrigerant after CFC has been attracting many research institutions and related industries. Among common substances that can be used as refrigerant, such as air, carbon dioxide, and water, carbon dioxide has unique characteristics and almost fulfils all required properties to be used as refrigerant. Carbon dioxide has zero ODP, negligible GWP, excellent heat transfer coefficients (Yoon et al., 2004), compatibility with material of refrigeration system and very

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Nomenclature

Symbols

c	specific heat (kJ/kg K)
COP	coefficient of performance (–)
h	enthalpy (kJ/kg)
IHX	internal heat exchanger
$\dot{m}$	mass flow rate (kg/s)
T	temperature (°C)
$\dot{W}$	electrical power supplied to the compressor (W)

Subscripts

cp	compressor
ea	external air
ev	evaporator
in	inlet
out	outlet
p	constant pressure
va	valve

low cost. At high temperature, the carbon dioxide refrigeration cycle is operating in transcritical mode and at high working pressure because of the specific thermodynamic properties of carbon dioxide (relatively low critical temperature and relatively high critical pressure). This leads to the need of completely new design of system components (Hesse and Kruse, 1993; Huai et al., 2004). The energetic performance concerning the transcritical carbon dioxide cycle is smaller than the energetic performance obtainable using the classical refrigerant in sub-critical cycles (Beaver et al., 1999). For this reason it is necessary to experiment methodologies able to improve these performances; to this aim an internal heat exchanger can be used. In this heat exchanger the carbon dioxide at the outlet of the gas-cooler exchanges heat with the carbon dioxide flowing out of the evaporator before entering into the compressor.

The internal heat exchanger (IHX) is often employed in refrigerating systems. Generally the cooling of the refrigerant flowing out of the gas-cooler prevents flash gas at the expansion valve and the superheating of the suction gas avoids that liquid refrigerant from the evaporator entering into the compressor. On the other hand, the thermodynamic efficiency could be improved employing an internal heat exchanger, especially in applications which require low suction temperature (ASHRAE, 1994). One can note that employing the internal heat exchanger increases the specific refrigerating effect, on the other hand, the specific volume of the refrigerant vapour at the beginning of the compression rises and, as a consequence, the specific compression work increases too. The system coefficient of performance, that is the ratio of the refrigerating effect to the compression work, could be higher or lower than one of a cycle without internal heat exchanger. In the following a simple criterion for evaluating the suitability of using an internal heat exchanger will be applied to carbon dioxide (Aprea et al., 1999). The evaluation criterion can be settled as follows:

$$c_p T_{\mathrm{in,cp}} > (h_{\mathrm{in,cp}} - h_{\mathrm{out,va}}) \quad (1)$$

When this inequality is verified, the adoption of an internal heat exchanger turns out to be advantageous. Furthermore, it is important to fix the suction temperature value under which this occurs. To this aim, the following inequality should also be true when the temperature at the compressor inlet is evaluated using the internal heat exchanger:

$$\frac{\partial(\Delta \mathrm{COP})}{\partial T_{\mathrm{in,cp}}} > 0 \quad (2)$$

One can find out that this is always true when Eq. (1) is verified. On other terms, if the internal heat exchanger allows an increasing COP, this effect will be amplified when the temperature of the working fluid at the outlet of this heat exchanger, that is the suction temperature, increases. Of course, the actual limit would be fixed by the well-known discharge temperature limits. Using a software (Labview) the criterion has been verified in all tests performed on the transcritical plant; these results are in according to the literature reporting the improvement of the energetic performances adopting the internal heat exchanger for the carbon dioxide (Robinson and Groll, 1998).

2. Experimental equipment

Fig. 1 shows a sketch of the experimental plant. Basically there are two single-stage semi-hermetic reciprocating compressors, an oil separator, an air gas-cooler, a liquid receiver, an air evaporator, an electronic expansion valve, and an electronic back pressure valve. To study the influence of the "flash vapour" a by-pass with an auxiliary compressor has been mounted (Elbel and Hrnjak, 2004).

The refrigerant state at the outlet of the expansion device is in a two-phase condition, provided that the fluid crosses the saturated liquid line during the isenthalpic expansion process. For this reason some fraction of the refrigerant flow enters the evaporator in vapour state not having a cooling effect. Using the flash gas by-pass the performances of transcritical carbon dioxide cycle increase improving the low pressure side of the system because the evaporator is fed with liquid only (Beaver et al., 1999; Hanson and Van Essen, 2001). The main compressor is a semi-hermetic compressor and its working pressure range is from 15 bar to 120 bar. As the intermediate pressure is about 65 bar, the working pressure range of the auxiliary compressor is from 55 bar to 120 bar. At evaporating temperature of 5 °C and temperature of 30 °C at the gas-cooler exit when the pressure is 80 bar, the refrigerating power is about 3000 W. An internal heat exchanger between the refrigerant at the compressor suction and the refrigerant at the exit of the gas-cooler has been set up. The lamination process has been obtained thanks to the back pressure valve and to the electronic expansion valve. To fix the air temperature on the gas-cooler and to simulate the external conditions, the air flows under the influence of a blower in an thermally insulated channel where are located some electrical resistances that can be modulated. In Fig. 2 a photo of the plant is shown. In Table 1 the heat exchangers' specifications are reported. In order to investigate on improvement of COP including the IHX, the tests were carried out using a basic plant configuration: main compressor, oil separator, gas-cooler,

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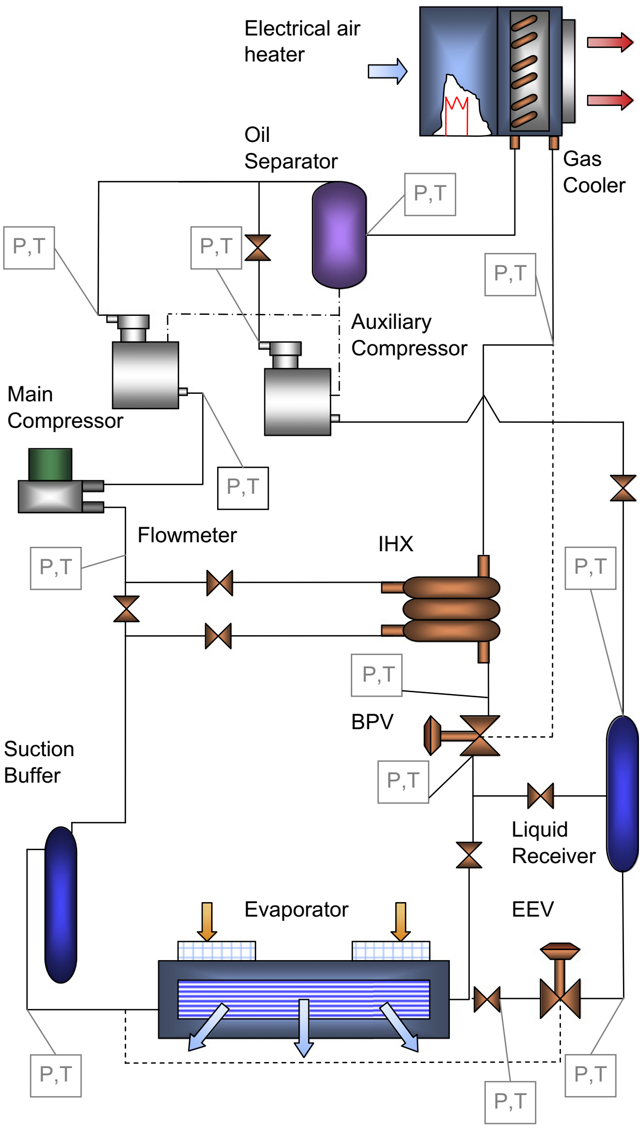

Fig. 1 – Sketch of the experimental plant.

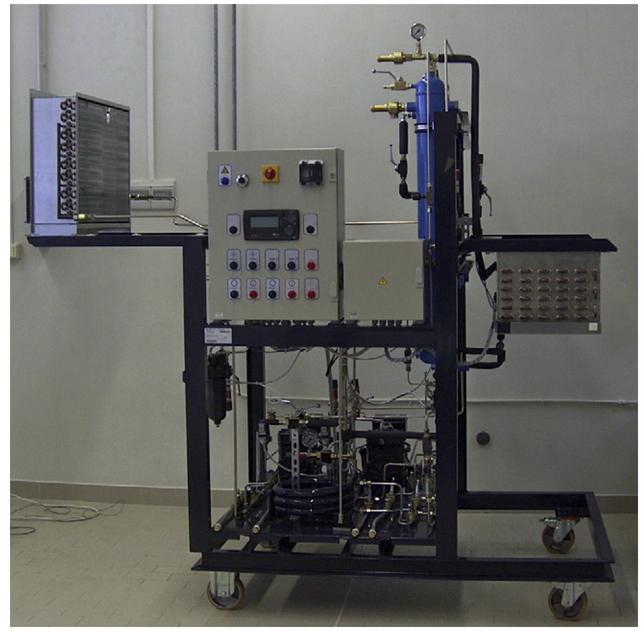

Fig. 2 – Photo of the plant.

EBPV, EEV, liquid receiver, evaporator, IHX (included or by-passed) and suction buffer.

The IHX specifications are reported in Table 2.

3. System monitoring

The plant has been instrumented to evaluate the performances of the whole plant and of the components one by one. The carbon dioxide pressure and temperature are measured at the inlet and at the outlet of each device; the working fluid mass flow rate is monitored at the compressor suction as evidenced in Fig. 1. The electrical power of the compressors is measured. Temperatures were measured by means of four-wire PT 100 thermometers with a declared accuracy of 0.15 °C. The sensors were located outside the pipe, with a layer of heat transfer compound (aluminium oxide plus silicon) placed between the sensor and the pipe to provide good thermal contact. The whole pipe was insulated with 25 mm thick flexible insulation. The system of temperature measurement was checked against a sensor positioned in pocket in a similarly insulated pipework. For various test conditions, the difference between the two measurements has been always less than 0.3 °C. Pressure values are measured by piezoelectric sensors; the current output is recorded directly on the data logger. They were calibrated by the manufacturer in the range of 0–100 bar gauge to an accuracy of 0.4%. Refrigerant mass flow rate was measured using a Micro Motion mass flow meter. Since the mass flow meter could be affected by machine vibration it was mounted on a 25 kg steel plate positioned separately from the plant. Transducers (2 W) were used to measure the electrical power supplied to the compressors. The manufacturers claim, in the range 0.5–6 kW, an accuracy of ±0.2%. The coefficient of performance is evaluated as

$$\mathrm{COP} = \frac{\dot{m}(h_{\mathrm{out,ev}} - h_{\mathrm{in,ev}})}{\dot{W}} \quad (3)$$

With reference to the accuracy of the coefficient of performance, an analysis has been accomplished according to the procedure suggested by Moffat (1985). The resulting accuracy of the coefficient of performance is about 3.8%. The test apparatus is equipped with 16 bit A/D converter acquisition cards

Table 1 – Heat exchangers specifications

Heat exchangers	Gas-cooler	Evaporator
Number of pass	3	11
Number of circuits	4	1
Number of tubes	5	6
Material of tubes	Copper	Copper
Depth of the fins (mm)	0.15	0.7
Material of the fins	Aluminum	Aluminum
Fin pitch (mm)	2.7	3.5

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Table 2 – IHX specifications

Type	Coaxial
Material	Copper
Inner tube (cold flow), mm	10×1
Outer tube (hot flow), mm	35×1.5
Length (m)	4
Pressure drop ($\dot{m} = 25$ g/s), bar	0.5

linked to a personal computer that allows a high sampling rate monitoring all the measures carried out by means of the transducers. The data acquisition software has been realized in a Labview environment.

All tests were run at steady state conditions. Temperature and pressure values in key points of the plant are continuously monitored, in order to check the achievement of steady state conditions. Usually, the start up time required was about 1 h. Steady state conditions are assumed to hold when the deviations of all controlled variables from their corresponding mean values are lower than 0.5 °C for temperatures and 50 kPa for pressures, respectively. At this stage, the test starts and the logging of data with 0.5 Hz acquisition frequency is performed on all channels for 60 s. For each channel, the 30 samples recorded are averaged. After 180 s each sample during 60 s is checked against the corresponding mean values of the two previous samples; when the mean values of the temperatures and of the pressures are within the range reported above the steady state is reached.

4. Results

The airflow temperature entering at the gas-cooler has been warmed by variable electrical resistances to vary the air temperature in the range 25–40 °C; as aforementioned, the lamination process has been realized by both the back pressure valve and the electronic valve.

It can be seen in Fig. 3 that the evaporation temperature has been approximately 5 °C constant during the tests both when the plant runs with and without the internal heat exchanger; the small oscillations are due to the air temperature at evaporator inlet.

The maximum refrigerant temperature at the discharge compressor line has been about 120 °C; as one can foretell, this value has been carried out using the internal heat exchanger, which has caused a further superheating of the refrigerant before it enters in the compressor. It is important to underline the temperature value aforementioned is lower than the maximum working pressure of the compressor.

In Fig. 4 the coefficient of performance is shown as function of the air temperature at the inlet of the gas-cooler when the plant runs with and without the internal heat exchanger.

The coefficient of performance has been evaluated like the ratio between the refrigeration power to the electrical power supplied to the compressor as reported in Eq. (3). Like first result it can be seen that the COP curve slope is negative both using and excluding the IHX; this circumstance is in accordance with Kauf (1999). One can see (Table 3) that using the internal heat exchanger a 10% improvement has carried out; this circumstance confirms the precision of the method reported in Section 1. In addition to consideration based on the first law of thermodynamics one can explain this improvement having recourse to the second law of the thermodynamics. For this purpose an exergetic analysis has carried out; the exergy loss in each component except the evaporator increases with the increasing outlet temperature of the gas-cooler: a significant contribution to the global exergy loss can be attributed to the lamination process (Yang et al., 2005). Obviously this circumstance explains both the improvement of the COP decreasing the temperature at the gas-cooler outlet and the higher value of the COP when the internal heat exchanger is used.

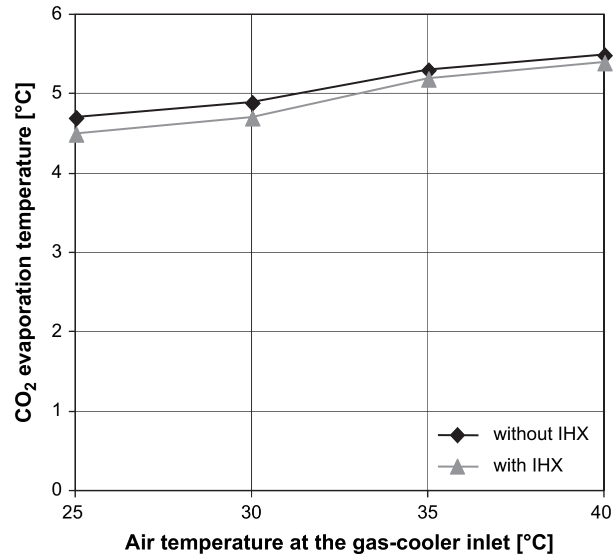

Fig. 3 – Refrigerant evaporating temperature versus the air temperature at the gas-cooler inlet (with and without IHX).

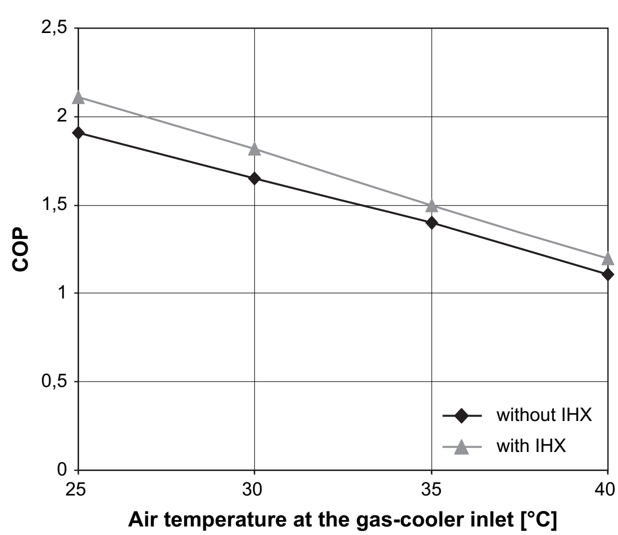

Fig. 4 – Coefficient of performance versus the air temperature at the gas-cooler inlet (with and without IHX).

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Table 3 – COP improving using IHE

T_{ea} (°C)	COP without IHX	COP with IHX	ΔCOP (%)
25	1.91	2.11	10.47
30	1.65	1.82	10.30
35	1.40	1.50	7.14
40	1.11	1.20	8.11

Besides it has been noted that using the internal heat exchanger, a better use of the evaporator has been evidenced also with reference to the superheating of the refrigerant: when the internal heat exchanger is not used the indispensable superheating to avoid refrigerant liquid at the suction of the compressor is obtained in the last coils of the evaporator decreasing therefore the enthalpy vaporization. On the contrary, as aforementioned, using the internal heat exchanger the carbon dioxide is further superheated so much that the evaporator can be completely used for the evaporation phase. It is important to highlight the fact that this measured COP improvement is due to this specific plant, using other design it could obtain different improvement values.

In Fig. 5 the carbon dioxide mass flow rate is reported versus the air temperature at the inlet of the gas-cooler when the plant runs with and without the internal heat exchanger. It has been noted that without the internal heat exchanger the refrigerant mass flow rate is approximately constant; this fact happens because the thermodynamic conditions at the compressor suction are fixed by the thermostatic valves and the compression ratio remains quite unvaried even if the air temperature at gas-cooler changes. On the contrary using the internal heat exchanger, when the temperature of the air at the outlet of the gas-cooler increases, the carbon dioxide temperature at the exit of the internal heat exchanger augments since the heat exchange area remains constant; for this reason the refrigerant temperature at the compressor suction increases together with the carbon dioxide specific volume; consequently the mass flow rate decreases.

In Fig. 6 the carbon dioxide suction temperature versus the air temperature at the inlet of the gas-cooler is reported, both including and excluding the IHX. One can see as using the IHX the suction temperature increases causing a compression work to increase due to reducing of density of the carbon dioxide.

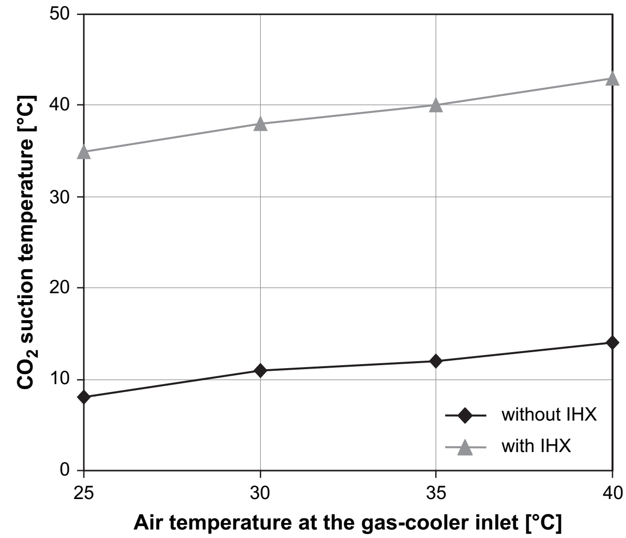

Fig. 6 – Carbon dioxide suction temperature versus the air temperature at the gas-cooler inlet (with and without IHX).

In Fig. 7 the refrigeration power versus the air temperature at the inlet of the gas-cooler when the plant runs with and without the internal heat exchanger is reported. The refrigeration power increases when the refrigerant temperature at the gas-cooler outlet decreases because of a decrease in air temperature at the gas-cooler inlet: in this case the enthalpy vaporization in the evaporator augments. However, using the internal heat exchanger the better exploitation of the evaporator increases the enthalpy vaporization. As aforementioned, it has been noted that the refrigeration power increases although the mass flow rate of the carbon dioxide decreases. Obviously if the COP improves using the internal heat exchanger the compression work increase is smaller than the augmentation of the enthalpy vaporization in the evaporator.

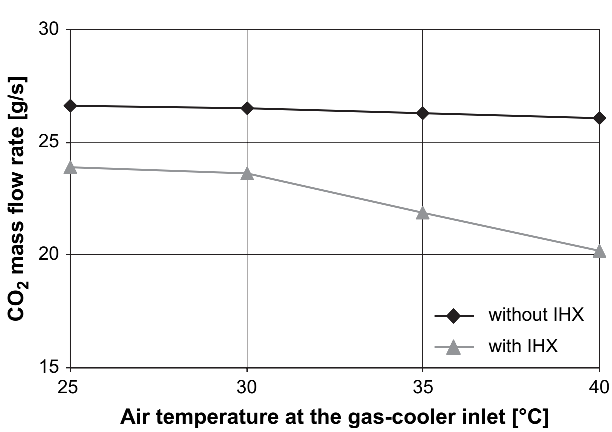

Fig. 5 – Refrigeration power versus the air temperature at the gas-cooler inlet (with and without IHX).

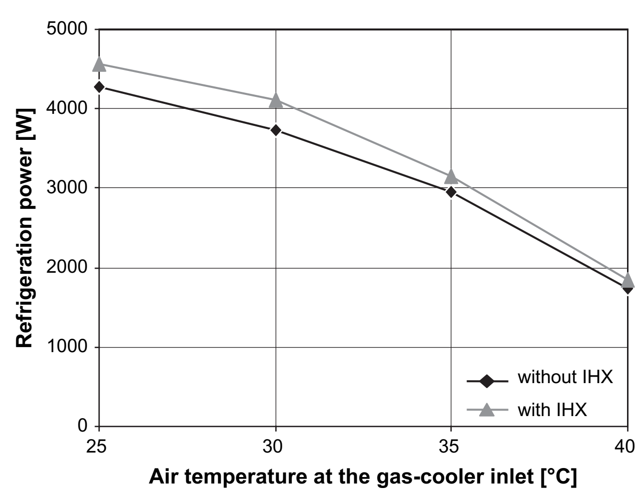

Fig. 7 – Carbon dioxide mass flow rate versus the air temperature at the gas-cooler inlet (with and without IHX).

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5. Conclusions

In this paper an experimental transcritical carbon dioxide refrigerator working as a classical split-system for residential air conditioning has been examined when the internal heat exchanger is used. A simple criterion formulated by the authors for evaluating the suitability of using this heat exchanger has been applied to carbon dioxide: the results indicate the convenience in the use of the internal heat exchanger confirmed by the experimental results. Various working conditions have been considered varying the air temperature at the gas-cooler inlet. Using the internal heat exchanger the COP is 10% better. Some considerations about the COP, the refrigeration power and the carbon dioxide mass flow rate are reported. The experimentation suggests the use of the internal heat exchanger working with the carbon dioxide in a transcritical cycle to cool air in residential applications.

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