

THE INTERNAL HEAT EXCHANGER IN A TRANSCRITICAL CO₂ REFRIGERATOR: AN EXPERIMENTAL STUDY

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SUMMARY

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 (CO₂) is non-toxic, non-flammable, has zero ozone depletion potential and negligible global warming potential as refrigerant. Referring to a transcritical CO₂ 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.

Key-words: carbon dioxide, transcritical cycle, internal heat exchanger

INTRODUCTION

Carbon dioxide has zero ODP and negligible GWP, excellent heat transfer coefficients [1] compatibility with material of refrigeration system, very low cost. Unfortunately the operating pressure is high because the refrigeration cycle is transcritical and the carbon dioxide needs completely new design of system components [2,3]. The energetic performance pertaining to the transcritical carbon dioxide cycle is smaller than the energetic performance obtainable using the classical refrigerant in sub-critical cycles [4]. For this reason it is necessary to experiments methodologies able to improve these performances; to this aim an internal exchanger can be used. In this heat exchanger the carbon dioxide at the outlet of the gas-cooler exchanges heat with the carbon dioxide going out from the evaporator before entering into the compressor.

The internal heat exchanger (IHE) is often employed in refrigerating systems. Generally the cooling of the refrigerant going out from the gas-cooler prevents flash gas at the expansion valve and the superheating of the suction gas avoids that liquid refrigerant from the evaporator enters in the compressor. On the other hand, the thermodynamic efficiency could be improved employing an internal heat exchanger, chiefly in applications which require low suction temperature [5]. 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, 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 [6]. The evaluation criterion can be settled as follows:

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

When this inequality is verified, the adoption of a 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:

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

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 [7] 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 [8].

EXPERIMENTAL EQUIPMENT

Figure 1 shows the experimental plant in the Thermodynamics Applied laboratory at University of Salerno. Figure 2 shows a sketch of the experimental plant. Basically there are two single-stage hermetic reciprocating compressors, an oil separator, an air gas-

cooler, a liquid capacity, an air evaporator, an electronic expansion valve, an electronic regulated back pressure valve. To study the influence of the “flash vapour” a bypass with an auxiliary compressor has been mounted. 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 increases improving the low pressure side of the system because the evaporator is fed with liquid only [9,4]. The main compressor is a semi-hermetic compressor.

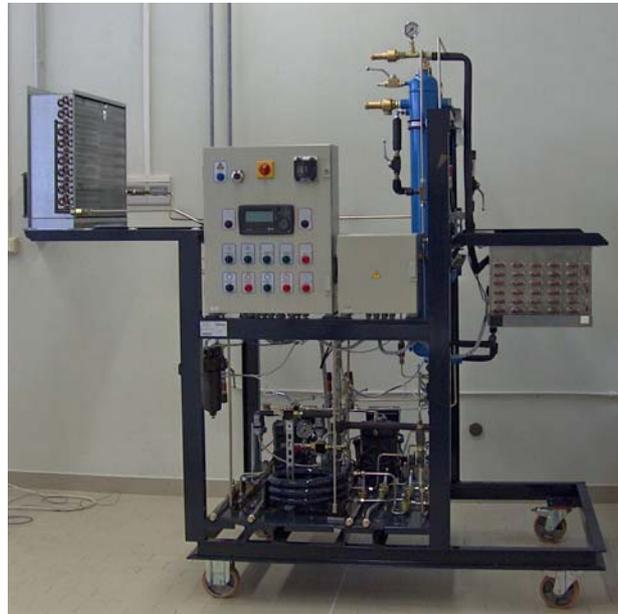


Figure 1. The experimental plant.

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 3000W. 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 gascooler 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.

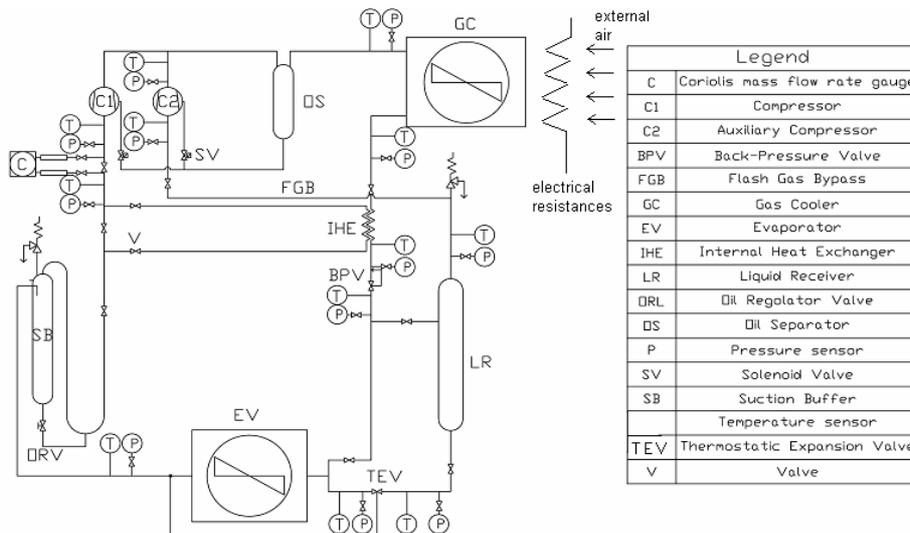


Figure 2. Sketch of the experimental plant.

SYSTEM MONITORING

The plant has been instrumented to evaluate the performances of the whole plant and of the components one by one. Both the pressure and the temperature of the carbon dioxide 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 Figure 2. The electric power of the compressors is measured. Temperatures were measured by means of four-wire 100 Ω platinum resistance 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 to 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. Two meters 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:

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

With reference to the accuracy of the Coefficient of Performance, an analysis has been accomplished according to the procedure suggested by Moffat [10]. 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 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 [12] environment.

RESULTS

In order to varying the ambient temperature the airflow entering at the gas-cooler has been warmed by variable electrical resistances in the range 25°C ÷ 40°C; as reported above, the lamination process has been obtained by means of the back pressure valve and by means of the electronic expansion valve.

The tests were carried out keeping a carbon dioxide evaporating temperature of about 2°C. In Figure 3 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.

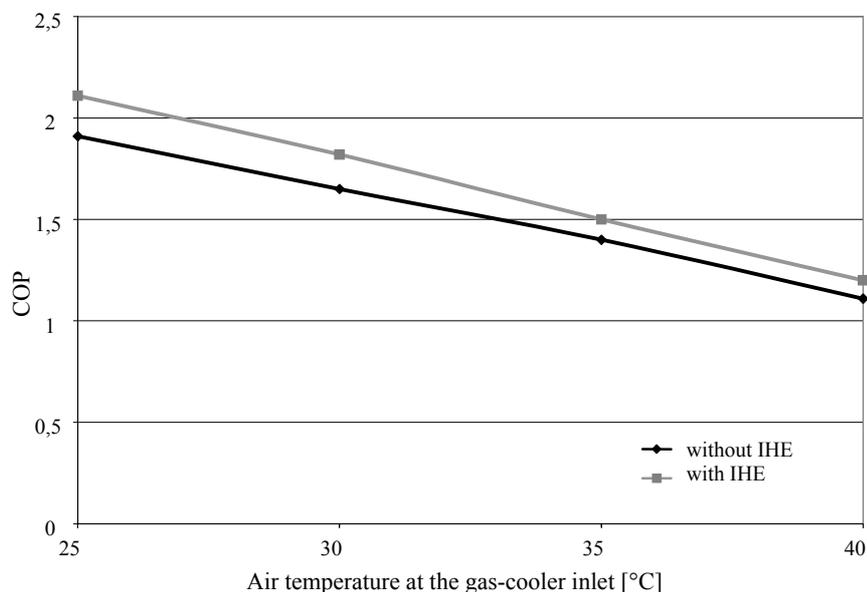


Figure 3. Coefficient of Performance versus the air temperature at the gas-cooler inlet (with and without IHE)

The Coefficient of Performance is evaluated like the ratio between the refrigeration power to the electric power supplied by the compressor as reported previously. It is seen that the COP of the cycle using the

internal heat exchanger in all tests is higher of about 10% than the COP referred to the cycle without internal heat exchanger; this circumstance confirms the precision of the method reported in the introduction. It is possible

to explain this improvement in addition to the considerations deriving by the first law of the thermodynamics using the second law of the thermodynamics thanks to the properties exergy; 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[11]. Obviously this circumstance explains both the improvement of the COP decreasing the temperature at the outlet of the gas-cooler and the higher value of the COP when the internal heat exchanger is used. Besides it has been noted that when the internal heat exchanger is used, 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 the evaporator can be completely used for the evaporation phase considering the internal heat exchanger because the carbon dioxide is superheated in the internal heat exchanger. In Figure 4 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.

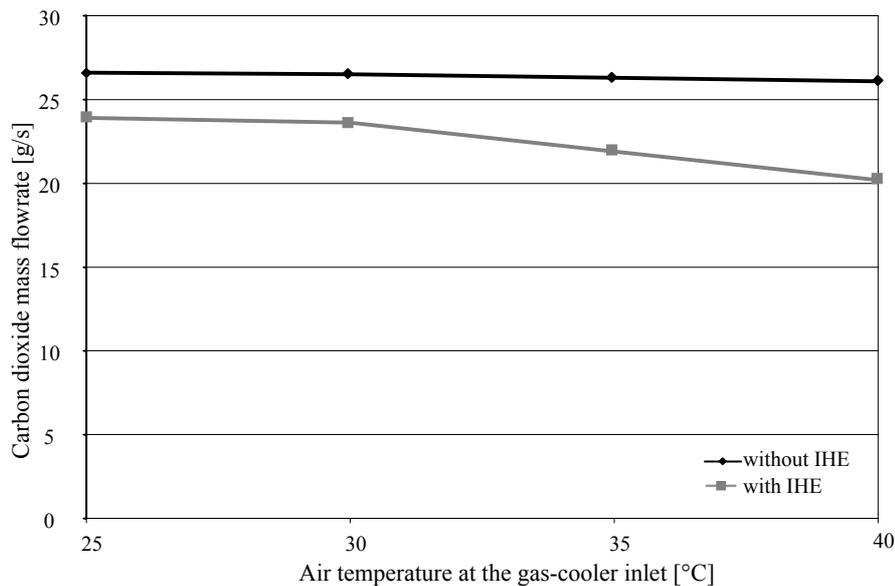


Figure 4. Carbon dioxide mass flow rate versus the air temperature at the gas-cooler inlet (with and without IHE).

It has been noted that without the internal heat exchanger the refrigerant mass flow rate practically is constant because the thermodynamic conditions at the suction of the compressor are fixed by the thermostatic valves and the compression ratio remains practically unvaried modifying the air temperature at gas-cooler. On the contrary using the internal heat exchanger, when the temperature of the air at the outlet of the gas-cooler increases, augments the temperature of the carbon dioxide at the exit of the internal heat exchanger since the heat exchange area remains constant; for this reason the temperature of the carbon dioxide at the compressor suction increases together with the specific volume of the carbon dioxide decreasing therefore the mass flow rate.

In Figure 5 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 outlet of the gas-cooler decreases in consequence of a decrease of the air temperature at the inlet of the gas-cooler: in this case the enthalpy vaporization in the evaporator is augmented. However when the internal heat exchanger is used also the better exploitation of the evaporator increases the enthalpy vaporization. It has been noted that the refrigeration power increases although the mass flow rate of the carbon dioxide decrease as discussed below. Obviously if the COP improves using the internal heat exchanger the increase of the compression work is smaller than the augmentation of the enthalpy vaporization in the evaporator.

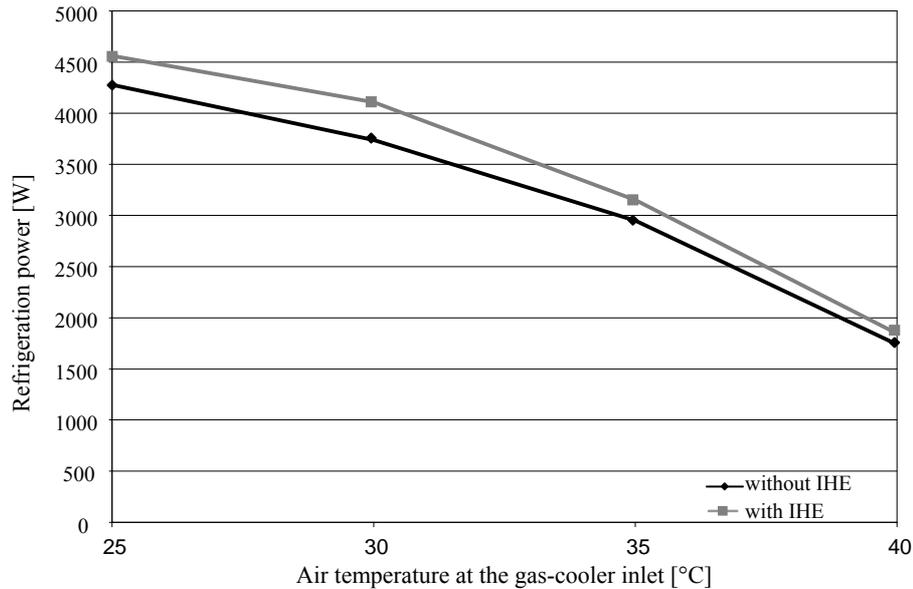


Figure 5. Refrigeration power versus the air temperature at the gas-cooler inlet (with and without IHE).

CONCLUSIONS

In this paper an experimental transcritical carbon dioxide refrigerator working as a classical split-system for air residential 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 inlet of the gas-cooler. The COP is better of about 10% when the internal heat exchanger is used. 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 the carbon dioxide in a transcritical cycle to cool air in residential applications.

NOMENCLATURE

Symbols

c	specific heat	[kJ/kgK]
COP	Coefficient of Performance	[-]
h	enthalpy	[kJ/kg]
IHE	internal heat exchanger	
\dot{m}	mass flow rate	[kg/s]
T	Temperature	[°C]
\dot{W}	Electrical power supplied to compressor	[W]

Subscripts

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

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