Refrigerant Charge and Ambient Temperature Effects on the Refrigeration Cycle of a Small Capacity Food Freezer

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REFRIGERANT CHARGE AND AMBIENT TEMPERATURE EFFECTS ON THE REFRIGERATION CYCLE OF A SMALL CAPACITY FOOD FREEZER

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ABSTRACT

An experimental study of a typical small capacity food freezer has been carried out to determine the refrigeration cycle. The temperatures, pressures and electric power of the plant were measured and some thermocouples were also positioned inside the hermetic compressor unit to improve the description of the refrigeration cycle. The cycle was drawn and its peculiarities were pointed out. An experimental sensitivity study concerning two parameters was performed: the refrigerant charge and the ambient temperature. A transient analysis was then performed with and without a thermostat switch. The inside temperatures of the freezer, the electric power consumption and the temperature variations of the refrigerant were analyzed, throughout the cyclic phenomenon.

1. INTRODUCTION

The vapor-compression cycle is almost universally employed for small capacity freezers. The knowledge of the thermodynamic cycle is of fundamental importance to understand the behavior of all the system components and to choose where to operate to improve the performance. An experimental study has been carried out to "build" the thermodynamic cycle of a chest freezer. The only difference, in comparison to the standard refrigeration circuit, is the compressor, which was modified to allow some internal temperature measurements, that are useful to improve the description of the cycle with respects to that obtained with a previous analysis (Anglesio and Torchio, 2002). Of all the different working conditions of the freezer, only two were analyzed: the refrigerant charge and the ambient temperature. These conditions were changed and their effects were discussed. A transient analysis was also performed to show the behavior of the system.

2. EXPERIMENTAL SET UP

The physical quantities that were measured are: temperatures, pressures and electric power. Type T thermocouples (copper-constantan) temperature sensors were used as these are particularly suitable for low temperatures: a first group of thermocouples was placed along the refrigerating circuit, in order to know the thermodynamic cycle of the refrigerant. A second group had the purpose of measuring both the ambient and the inside air temperatures. Two pressure values were measured in the refrigerating circuit, one in the outlet of the condenser and one in the inlet of the compressor. Finally, the voltage, the current and the phase angle were measured to know the electric power of the motor-compressor. All the signals were sent to a digital voltmeter and to a computerized data acquisition system. The experiments were performed in a climate chamber (9 kW heating, 5 kW cooling) that was designed to test household refrigerators and freezers and which is capable of providing ambient temperatures ranging from 10 to 50°C.

2.1 Compressor Instrumentation

The positioning of the temperature sensors near the compressor suction or discharge is not easy because the entire motor-compressor was hermetically sealed in a welded steel shell. Measurements in hermetic compressors have

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already been performed (Cavallini *et al*., 1996) (Rigola *et al*., 2000) with appropriate compressor modifications. In the here illustrated experimental set up the positioning of the temperature sensors inside a mass-production compressor has been possible as the hermetic shell was cut into two parts, the upper part is removable and can be closed thanks to a special flange. Four thermocouples were placed inside the shell (Figure 1): N°1 was positioned on the inner suction muffler and its signal was used for the inlet temperature in the cylinder when the thermodynamic cycle was drawn; two thermocouples were placed on the part of the discharge line inside the hermetic shell, N°2 was placed closer to the cylinder discharge orifice than N°3 and N°4 was placed on the cylinder head. The thermocouples cross the shell through the vacuum pressure feedthroughs.

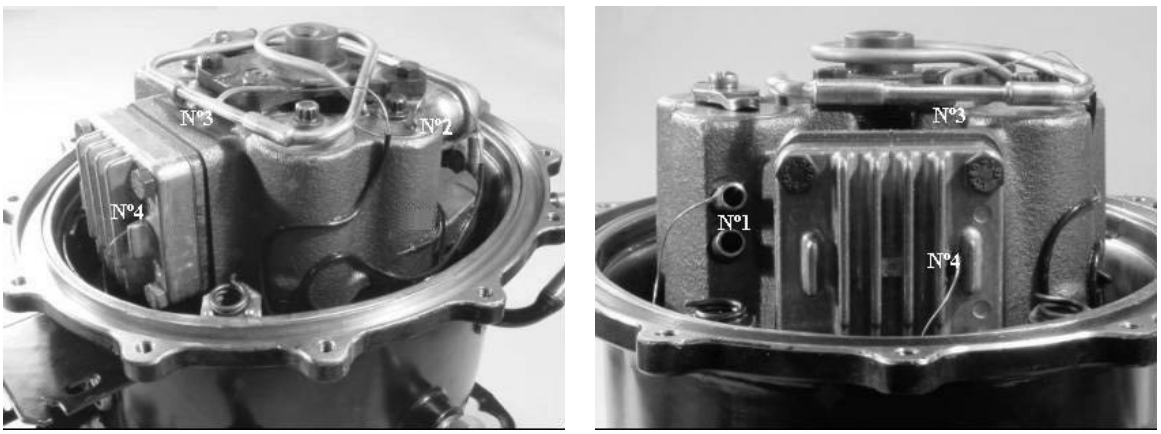

Figure 1: Views of the instrumented compressor (the numbers indicate the positions of the internal thermocouples)

2.2 Freezer Characteristics

Figure 2 shows the views and dimensions of the freezer that is used in the present investigation. This is a chest freezer with a design internal temperature of -25°C. Its storage capacity is separated in two different cold rooms (0.687 m^3 and 0.101 m^3, respectively). Some thermocouples were placed in both the first and the second cool rooms; for this study the temperature averaged over the volume of the cool rooms was assumed as the inside air temperature.

A schematic plan of the refrigeration system is shown in Figure 3. The freezer has a single stage vapor-compression refrigerating system that works with R404A (the usual refrigerant used in this kind of freezer). The compressor is a welded hermetic reciprocating compressor: the piston displacement is 12 cm^3, and the design power is 450 W. The condenser is air-cooled. It is make up of 18 copper tubes with aluminum fins and its dimension is 225x240x44 mm. The expansion device is a capillary tube: the internal diameter is 0.9 mm and the length is 3500 mm; a part of the capillary tube is welded to the suction line for heat exchange purposes. The evaporator is a copper tube coil inside the freezer cabinet: the diameters are 6.0 mm internal, 6.8 mm external and the length is about 55 m. The cabinet walls are made of two zinc sheets with polyurethane foam in the middle. The evaporator tube was placed in contact with the innermost sheet before the space between the sheets was filled with foam.

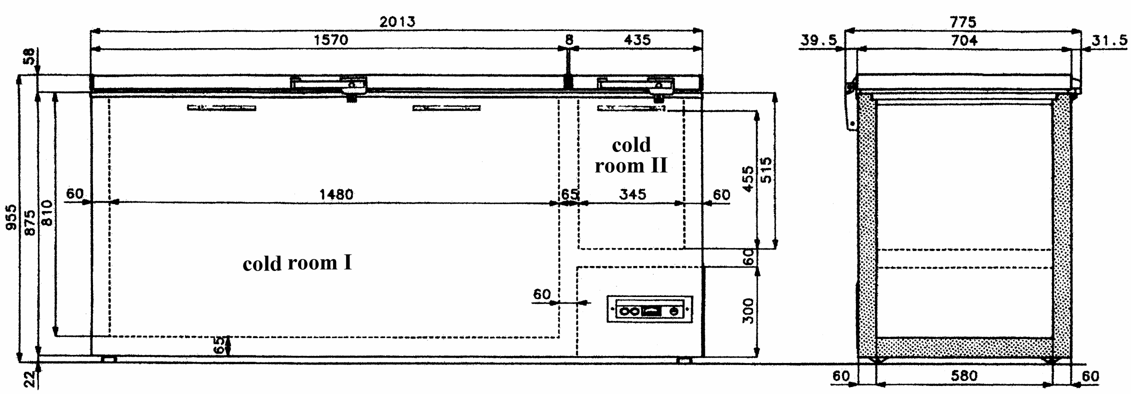

Figure 2: Freezer geometry

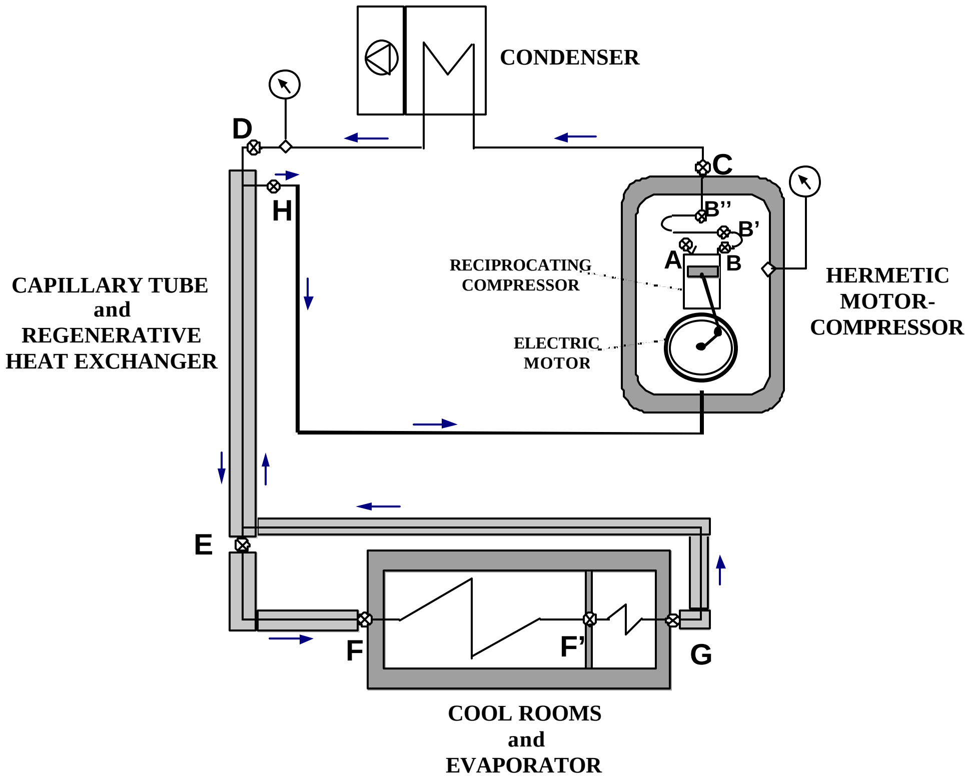

Figure 3: Schematic plan of the refrigerating circuit; the symbol ⊗ shows where the temperatures were measured (each letter correspond to a measurement), the symbol ? shows where the pressures were measured.

3. RESULTS AND DISCUSSION

3.1 The Thermodynamic Cycle

Each experiment was carried out imposing both an ambient temperature and a refrigerant charge. When the thermostatic switch was turned off, after a transient period, the refrigeration system arrived at steady state conditions. The thermodynamic cycle, with nominal charge (m=0.260 kg) and ambient temperature t_a=30°C, is represented in these conditions in the pressure-specific enthalpy diagram (Figure 4). The absorbed heat can be divided into two parts: **FG**, where the heat is absorbed from the freezer cold rooms (the useful refrigerating effect), and **GH**, where the heat is exchanged in the regenerative heat exchanger, which gives an useful superheating as long as there is no liquid refrigerant present in the suction gas that enters in the compressor (state **H**). In comparison to the tests that were illustrated in a previous study (Anglesio and Torchio, 2002), this paper also deals with some thermodynamic states of the refrigerant inside the hermetic compressor, that have been labeled with the letters **A**, **B**, **B'**, **B''**, which correspond to temperature sensors N°1, N°4, N°2 and N°3, respectively. It is assumed that the temperature measured on the cylinder head (thermocouple N°4) is the highest temperature in the cycle (letter **B**). The state **E** is determined with an energy balance of the regenerative heat exchanger while the **EF** process is assumed isenthalpic. The compression process **AB** is not adiabatic, as it occurs inside the shell where the temperature of the refrigerant is 84.6°C; this heat exchange is useful to reduce the discharge temperature (ca. 135.3°C) with respect to the theoretical compression (reversible and adiabatic, hence isentropic), that the discharge temperature would have at 175.8°C (labeled $\mathbf{B_{is}}$ in the diagram). The refrigerant, which flows in the discharge line inside the hermetic shell, continues the cooling: $t_{B'}$ =119.4°C, $t_{B''}$ =108.5°C while outside the shell t_C =74.1°C. It also emerges that there is a further superheating (**HA**) before the refrigerant enters the cylinder which offers a double advantage: it protects the compressor from an eventual residual liquid–phase (this is particularly important when the system has no regenerative heat exchanger) and it cools the electric motor. Finally, a pressure drop (ca. 0.5 bar) in the evaporator (**FG**) was noted because of the head losses due to its small diameter and considerable length.

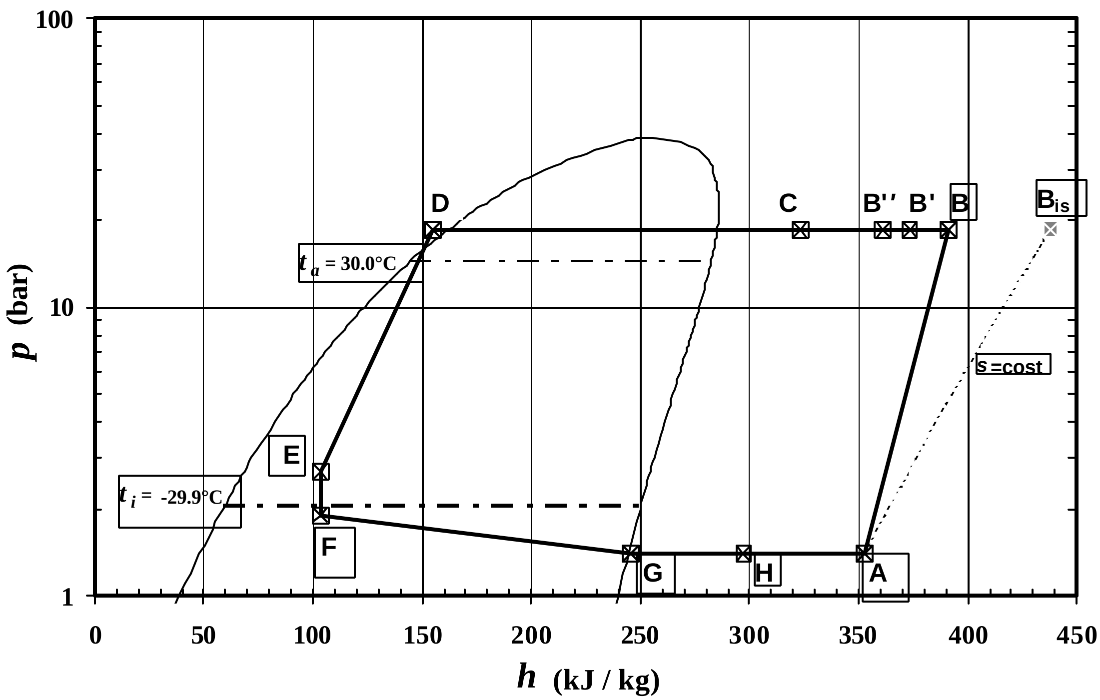

Figure 4: Thermodynamic cycle (R404A) in the pressure-enthalpy diagram with t_a=30°C and m=0.260 kg. The crosses refer to the process statepoints (the same as Figure 3). The dotted-dashed lines labeled t_i refer to the average temperature in the cool rooms and t_a refers to the ambient temperatures. The dotted line refers to an isentropic process.

3.2 The Refrigerant Charge Effect

In this analysis the tests were carried out with different charges while the ambient temperature was fixed (30°C) and the thermostatic switch was turned off. Figure 5a shows the influence of the charge both on the temperature of the cool rooms and on the evaporator temperatures. The evaporator temperatures were measured at three significant points: at the inlet in the first cold room (**F**), at the passage between the first and the second cold rooms (**F’**), and at the outlet from the second cold room (**G**). The optimal condition is when the charge is 0.260 kg. The temperature of the refrigerant in fact decreases when it advances in the evaporator, and only a limited amount of superheating (a few degrees) is present in the second cold room (**G**). The air temperature in the cool rooms decreases as it passes from the first cool room to the second one. For charges lower than 0.260 kg, the temperatures are not always compatible with freezer food conservation: with a charge of 0.215 kg the refrigerant temperature is already -17.6°C in the passage between the cold rooms and the temperature is about -7.5°C in the outlet from the second cold room. The air temperature in the second cool room is very high: -11.1°C. Therefore, with low charges a great superheating of the refrigerant already occurs inside the cold rooms, as was also observed in (Levis *et al.*, 1996). On the contrary, for charges greater than 0.260 kg, there is no improvement with respect to the evaporator temperatures.

In short, three possible consecutive superheatings were identified before the inlet of the compressor cylinder: one at the last part of the evaporator tube inside the cold rooms (its size depends on the charge), another at the regenerative heat exchanger and the last one at the shell of the hermetic compressor, due to the heat exchange with the electric motor.

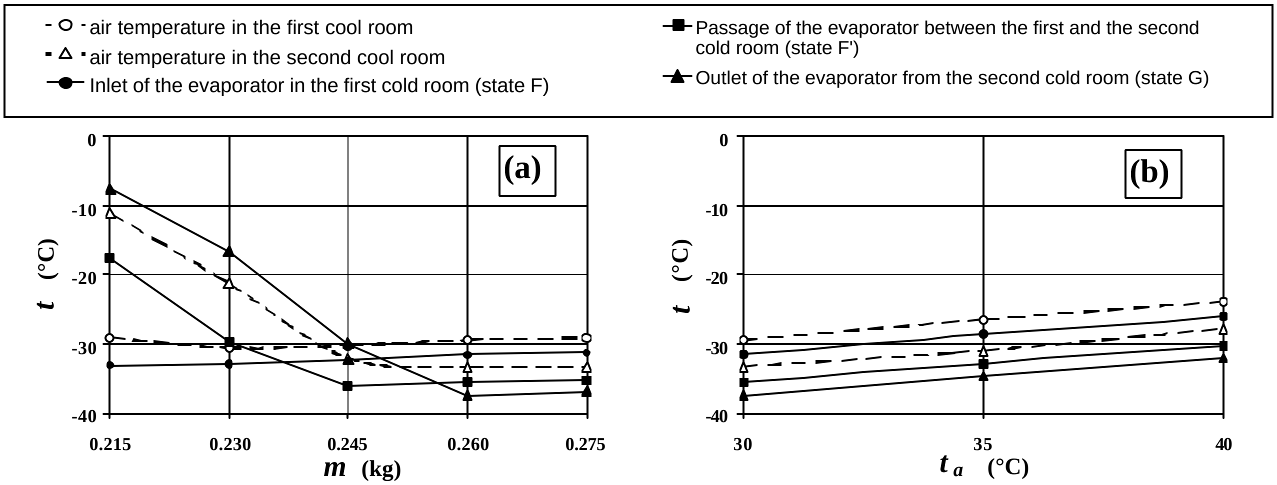

Figure 5. (a) Evaporator temperatures and temperature of the cool rooms versus refrigerant charge with a constant ambient temperature at 30°C. (b) Evaporator temperatures and temperature of the cool rooms versus ambient temperature with a constant charge at 0.260 kg.

3.3 The Operating Conditions Effect

Having fixed the refrigerant charge, the evaporator temperatures were analyzed while the operating conditions were changed. First, the ambient temp erature was studied within the 30°C to 40°C range; Figure 5b shows that the design conditions of this freezer are respected for each analyzed ambient temperature: the temperature in the cool rooms, as well as the temperatures in the evaporator, are nearly always lower than -25°C (the design temperature), and the second cold room is always colder than first one by about 4°C. The only load that is present in these tests is due to the heat exchange dispersion through the freezer walls, since these freezers are designed to control the internal temperature and should not be used to freeze fresh food.

The testing of these freezers is often performed in steady conditions, that is, with the thermostatic switch was turned off. Instead, in usual working conditions, the thermostatic switch is turned on, therefore a study was also carried out in these conditions. The thermostatic switch is controlled by a thermal sensing element placed in the second cool room. The electric motor stops when the temperature is lower than -28°C, and starts when the temperature is higher than -22°C.

Two significant variables versus time are shown in Figure 6: the air temperature averaged over the volume of the cool rooms, and the electric power consumption of the compressor. Two trends are shown for each variable: thermostatic switch turned off and turned on. When the thermostatic switch is turned off (solid lines in Figure 6) after a transient of ca. 5 hours, the system reaches a steady-state condition with t_i= -29.9°C. Instead, when the thermostatic switch is turned on (dashed lines) a typical cyclic phenomenon starts after ca. 3 hours, with period of 35 min; the average value in this time period (dotted-dashed lines) is t_i= -22.8°C. The inside air temperature t_i (the average over the volumes) varies between -20.6°C and -24.5°C, since the thermal sensing element is placed in the second cool room where the minimum temperature is usually reached. The guidelines for the control of the cold chain for quick-frozen foods (IIF/IIR, 1999) give -18°C as the maximum temperature for conservation. This temperature must be maintained in all the points of the product, therefore all the thermocouple signals in the two cool rooms were analyzed. The maximum temperature that was measured is -20.0°C which respects the IIF/IIR guidelines. The minimum temperature that was reached in the cool rooms is -29.2°C, therefore the temperatures change in the freezer both in space and in time, with a maximum variation of 9.2°C. It should be remembered that when there is not a sufficient charge a superheating occurs in the last part of the evaporator and the temperature exceeds the previously mentioned limits, consequently the refrigerating plant runs continuously (no cyclic phenomenon is present).

The electric power is shown in Figure 6: the maximum power is ca. 500 W (peak power) and, if the thermostatic switch is turned off, the electric power is minimum when steady condition are reached (358 W). If we analyze the cyclic phenomenon (switch turned on) we can see that the minimum power (402 W) is greater (12%) than the previous one, since the compressor runs for a shorter period (22 min) than the time that is necessary to arrive at steady conditions (ca. 4h). It is useful to know the average electric power (259 W) to estimate the electric requirements of the freezer, and this is -28% than the steady state conditions.

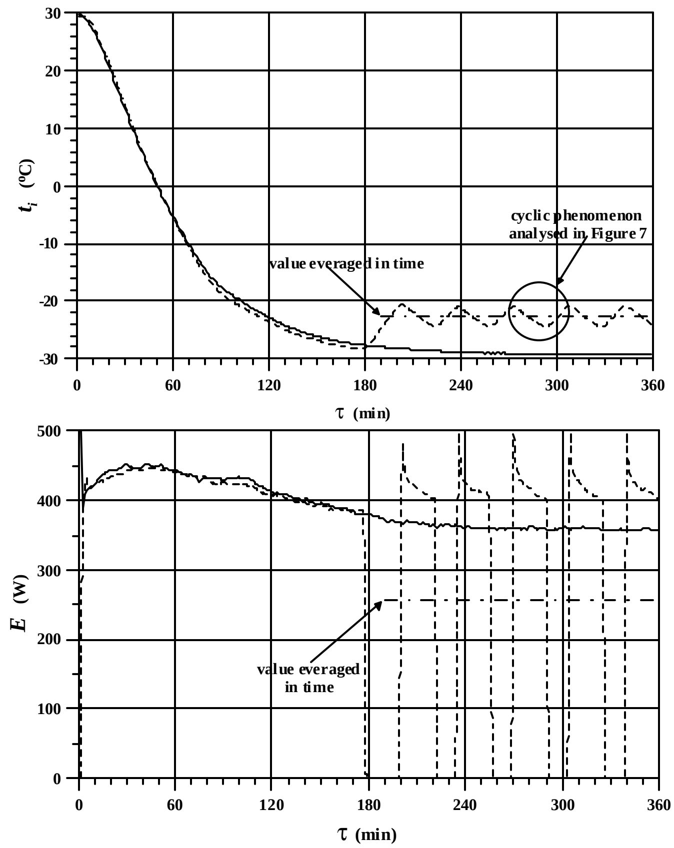

Figure 6: Transient analysis when the thermostatic switch is turned off (solid lines) or turned on (dashed lines). There are two variables versus time: t_i, the air temperature averaged over the volume of the cool rooms and E, the electric power consumption. The dotted-dashed lines refer to the values averaged in time when the thermostatic switch is turned on. In these tests the charge is 0.260 kg and the ambient air temperature is 30°C.

The temperatures that were measured during one cyclic phenomenon are reported in Figure 7. These temperatures are measured by the thermocouples that were placed along the refrigerating circuit (Figure 3). The vertical dashed lines show when the electric motor is turned on or off.

The temperature is uniform at the beginning of the analyzed cyclic phenomenon (time 270 min) inside the hermetic shell (states **A** and **B**) and it is ca. 89°C. After the compressor motor is started, the suction temperature (**A**) initially decreases as some refrigerant, which has a low temperature, arrives at the inlet of the shell (**H**). This temperature then increases to 81°C. The effect of the suction temperature can also be seen on the temperature at the end of the compression process (**B**), which reaches 133°C. The temperature of the refrigerant that come out of the hermetic shell (**C**) is ca. 46°C and it increases at a lower rate than **B** because, as seen in the thermodynamic cycle, a heat exchange occurs between the refrigerant in the discharge line and that in the hermetic shell (**A**). The temperature of **C** tends rapidly to temperature **A** because of the high heat transfer coefficients.

At the beginning of the cyclic phenomenon, the temperature of the condenser outlet (**D**) is equal to the ambient temperature (30°C) and increases to 40°C. Because of expansion, after the first part of the capillary tube (**E**), the temperature decreases by ca. 20°C, and after the second part of the capillary tube (**F**), the temperature again decreases by another 10°C.

The temperatures are practically constant in the evaporator before cycling is started, they progressively decrease at the evaporator inlet (**F**) and at the passage of the evaporator between the first and second cold rooms (**F'**). A superheating occurs at the outlet of evaporator (**G**) that lasts ca. 11 min.

The temperature at the outlet of the regenerative heat exchange (**H**) is very low (ca. 0°C) at first, probably because the heat transfer coefficients are low in the initial period. Subsequently, the superheating increases to a temperature of 25°C.

After the compressor is turned off, the refrigerant temperature in the high-pressure side (**B**, **C**, **D**) undergoes a sudden fall in the first minutes because of the reduction in pressure. Instead, in the low-pressure side (**F**, **F'**, **G**, **H**, **A**) the temperatures increases as the pressure increases. When the compressor is not working, almost all the temperatures tend to the ambient temperature (30°C), with the exception of **H**: its temperature rise could be due to a reflux of the hot vapor from the hermetic shell to the external inlet line where the temperature sensor is placed.

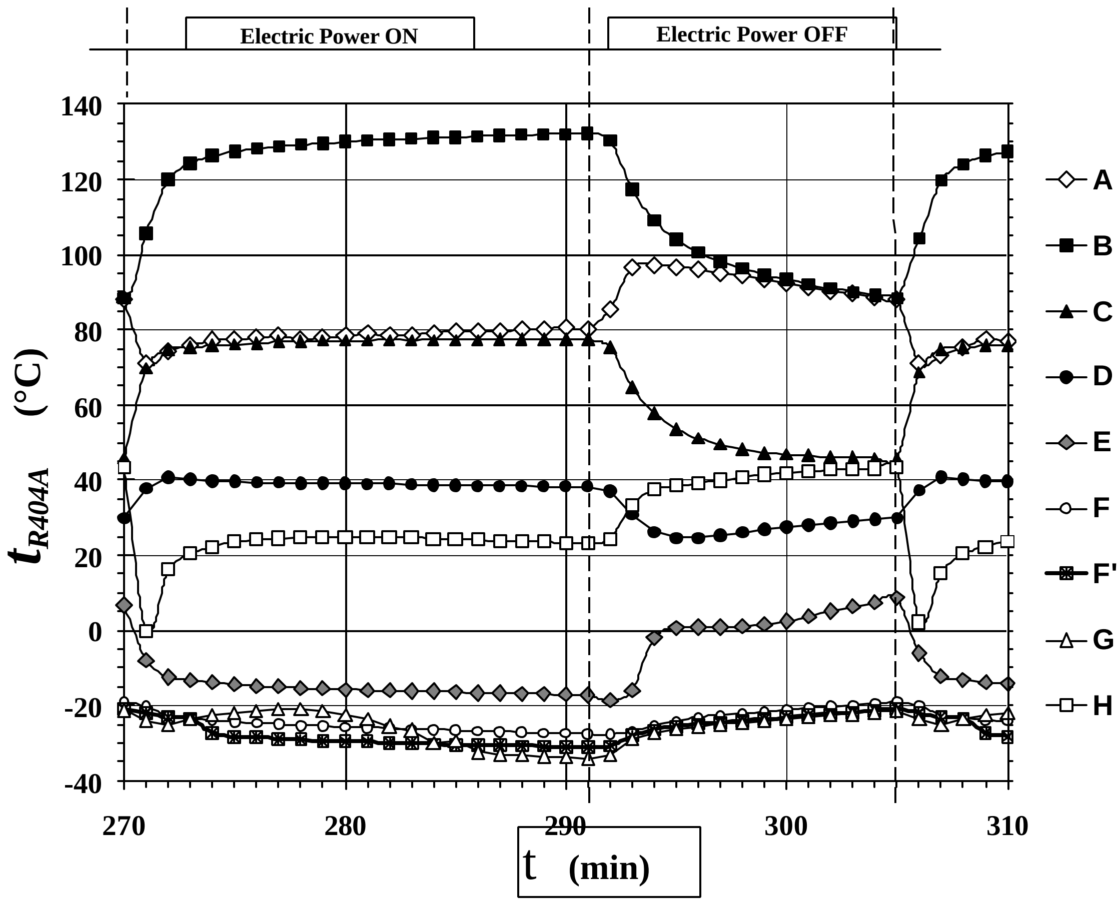

Figure 7: Representation of the refrigerant temperature (the capital letters in the legend refer to Figure 3) when the thermostatic switch is turned on and the cyclic phenomenon is started (see Figure 6). In this test the charge was 0.260 kg while the ambient air temperature was 30°C.

4. CONCLUSIONS

The thermodynamic cycle of a small capacity food freezer has been studied and its peculiarities, with respect to the theoretical single-stage vapor compression refrigeration cycle, have been pointed out:

- regenerative heat exchange with capillary tube;
- pressure drop in the evaporator;
- superheating of the refrigerant in different components (the most important are in the regenerative heat exchanger and in the hermetic compressor shell);
- no adiabatic compression;
- beginning of the desuperheating inside the hermetic compressor shell.

The design conditions were respected when the ambient temperature was changed from 30°C to 40°C. The temperature inside the cool rooms had a significant thermal gradient that must be checked to continuously assure the correct temperature level in all the points. The position of the thermal sensing element of the thermostatic switch is therefore very important.

As far as the refrigerant charge effect is concerned, the test showed that a charge defect, with respect to nominal values, could cause superheating even in the last part of the evaporator, consequently the temperature of the inside cool rooms becomes greater than those allowed for frozen food conservation.
Finally, the analysis of the transient, and in particular the study of the cyclic phenomenon, led to the actual working conditions of this plant both as far as the electric power consumption and the refrigerant temperatures are concerned.

NOMENCLATURE

E	compressor electric power consumption	(W)
h	specific enthalpy	(J/kg)
m	refrigerant charge	(kg)
p	pressure	(bar)
s	specific entropy	(J/kg/K)
t	temperature	(°C)
t	time	(min, h)

Subscripts

a	ambient condition
i	inside freezer condition
is	isentropic process

REFERENCES

Anglesio, P., Torchio, M.F., 2002, Considerazioni su due congelatori per prodotti alimentari di piccola potenza in condizioni di funzionamento nominali, *Il Freddo*, vol. 56, no. 6: p. 32-36.

Cavallini, A., Doretti, L., Longo, G.A., Rossetto, L., Bella, B., Zannerio, A., 1996, Thermal analysis of a hermetic reciprocating compressor, *Proc. International Compressor Engineering Conference*, Purdue University, p. 535-540.

Levins, W.P., Rice, C.K., Baxter, V.D., 1996, Modelled and measured effects of compressor downsizing in an existing air conditioner/heat pump in the cooling mode., *ASHRAE TRANSACTION*, vol. 102, Pt. 2: p. 22-33.

IIF/IIR, 1999, *Control of the Cold Chain for Quick-Frozen Foods Handbook*, IIF/IIR Publishers, Paris, 8 p.

Rigola, J., Pérez-Segarra, C.D., Oliva, A., Serra, J.M., Escribà, M., Pons, J., 2000, Advanced numerical simulation model of hermetic reciprocating compressors. Parametric study and detailed experimental validation., *Proc. International Compressor Engineering Conference*, Purdue University, p. 23-30.

Riaudo, A., 2001, *Determinazione sperimentale delle prestazioni energetiche di congelatori per prodotti alimentari*, Degree Thesis, Politecnico di Torino, Turin.

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