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THE EFFECTS OF NON-CONDENSABLE GASES IN HOUSEHOLD REFRIGERATORS

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ABSTRACT

It is well known that the presence of non-condensable gases inside a compression vapour refrigerating circuit introduces an additional thermal resistance at the condenser and can spoil the energy efficiency of the system. However this problem so far has been investigated mainly for shell ant tube condenser of large capacity and scarce information is available on small systems, as it is the case of household appliances where the internal volumes are extremely reduced and therefore a very small amount of non-condensable gas has large effect. Moreover non-condensable gas behaves differently when condensation takes place outside tubes (shell and tube condensers) or inside tubes (condensers of small appliances); in the first case all heat transfer area is wrapped by a gas layer, whereas in the second case non-condensable gas is collected at the end of the tube.
The effect of non-condensable gas in this work is experimentally investigated by injecting controlled amounts of air inside a refrigerating circuit and by recording the thermal and electric variables during different modes of operation (steady state and cyclic running).
The tested refrigerating circuit is part of an appliance (a household refrigerator for fresh foods) and was slightly modified to simplify the analysis: the modification consists in the removal of the suction line/capillary tube heat exchanger.
The presence of non-condensable gas was found to spoil energy efficiency, since brings about an increase in condensing pressure and a concomitant decrease in evaporating temperature, although larger liquid subcooling partially compensate for the first negative effects.
The temperature distribution along condensers and evaporator tubes points out flooding of condenser and starving of evaporator: this is why performance is worsening, the more so as the fraction of non-condensable gases increases. In its turn, this behaviour is caused by the clogging effect of bubbles of gaseous mixture (air and refrigerant vapour) that enter the capillary tube.
The experiments show that the minimum allowable amount of non condensable gas is significantly lower than the one so far deemed to be safe by manufacturers.

1. INTRODUCTION

The effect of the presence of non-condensable gases during the condensation process of a refrigerant is a topic extensively investigated (especially from a theoretical perspective) in the classical case of condensation outside circular tubes; the well known result is that an additional thermal resistance arises, deriving from a diffusion process of the vapour inside the layer of non-condensable gas that tends to concentrate close to the condensing surface, being carried by the centripetal motion of the vapour toward the tube. Under a different point of view, Nusselt theory explains this penalisation in terms of a decrease in the driving potential, due to the lower saturation temperature at the external interface of the falling film of the condensate, as an effect of the partial pressure of the non-condensable gas. It is manifest that small fraction of non-condensable gases make large damage, since they are not uniformly distributed inside the internal volume of the refrigerating systems, but gather close to the surface where condensation takes place. To get a deeper insight on this subject, the reader can refer to the wide review paper of Jensen (1988) or to the specific sections of handbooks, as the ones by Burghardt (1993) and Webb (1995).

As far as the present authors are aware, in the case of the condensation inside tubes, in the open literature there is not available nearly any contribution, maybe because this topic entails technological aspects that are not well known by scientists and academics, but are of the utmost interest for the refrigeration industry.
When a liquid receiver is present at the condenser outlet, as it is recommended for predictable performance of the refrigerating circuit, all non-condensable gas accumulates inside this volume, since a liquid seal prevents the gas from escaping, while the vapour motion draws it inside this trap. This circumstance makes the last portion of the condenser tubes flooded and induces liquid subcooling. As a matter of fact, only a convenient subcooling can prevent liquid from flashing when it expands, entering a volume where the vapour partial pressure is lower than the liquid saturated pressure; on the other hand, vapour cannot develop inside the receiver, as it cannot escape from it neither can condensate since this vessel is virtually adiabatic. A moderate amount of non-condensable gas is therefore not much detrimental to performance, since a certain amount of flooding of condenser can be useful.
The circuit investigated in this work is typical of a household appliance and does not include a liquid receiver, as dictated by the common practice for refrigerating systems using a capillary tube as throttling device. The analysis of the effect of non-condensable gas on the operation of the system will be proposed in the following section 3 where the experimental results will be discussed; the variation of the operational variables as a function of the mass fraction of non-condensable gas shows clearly how this presence affects the performance of the system. In general, when condensation is inside tubes, the non-condensable gas is carried away from the heat transfer surfaces and its detrimental effect does not stem from heat transfer penalisation, but from disturbance of the refrigerant flow.

2. EXPERIMENTAL SETUP

2.1 The tested appliance

The experimental analysis has been carried out on two single door refrigerators of 320 litres internal volume (see Figure 1). The model of the two appliances is the same but they differ because one is the appliance on the market, equipped with a Suction Gas Heat Exchanger (SGHX), i.e. with part of the capillary tube inserted inside the suction line of the compressor, while the other is a simplified version without SGHX. This latter appliance was made to simplify the analysis of the effect of non-condensable gas on the performance of the system; the length of the capillary tube was reduced to compensate for the lack of the SGHX.
The condenser is a coil pipe bond to a louvered plate (see Figure 2) with equivalent length of 10.2 m and internal volume of 91 cm^3.
The compressor is an ACC HQT 55AA model lubricated with mineral oil. The service tube of the compressor was modified in order to allow injecting a controlled amount of “doping” air; a special manifold was welded to it (see Figures 2 and 3) that has three “T” joints with the free ends closed by rubber plugs, sealed with plastic adhesive, for safety. The end of the manifold is closed with an Hansen fitting.
The evaporator was not changed, it is embedded in the back wall of the internal compartment and the circuit has equivalent length of 17.7 m and internal volume of 527 cm^3.
The compressor is controlled by ON/OFF operation using the original thermostat of the appliance, bond to the evaporator plate.
For both appliances the circuit is charged with the nominal amount of refrigerant R600a (isobutane), 40g (±0.2 g).

2.2 Measurement system

Copper-constantan thermocouples (T type) were used as temperature sensors. The estimated accuracy of the entire temperature measurement system is ± 0.36°C. 21 and 5 thermocouples were placed on the pipe wall, respectively of the condenser and of the evaporator for tracing the temperature profiles. Six thermocouples were used to monitor the temperature of the compressor shell and adjacent pipes (bottom shell, cover, suction and discharge tube). Three thermocouples were put inside the refrigerator compartment according to UNI/EN/ISO 7371 (2000). Two thermocouples were put in the climatic room according to UNI/EN/ISO 7371 (2000) and CECED (2000).
Pressures were measured at the suction and discharge sides of the compressor and at the dryer, just before the capillary tube. The accuracy of the transducer (Druck PMP 4070, FS = 10 barA for suction side and FS = 20 barA for discharge side) was ± 0.04% FS, as declared by the manufacturer. The estimated accuracy of the entire pressure measurement system is ± 0.01 barA.
Electrical parameters (absorbed power, current and applied voltage) were recorded through a power analyser with an estimated accuracy of the entire measurement system of ±0.71 W for the power, ±3mA for the current e ±0.3 V for the voltage.

The tests were carried out inside a climatic room built according to specifications of UNI/EN 153 (1997) and CECED (2000).

Figure 1: The tested single door refrigerator

Figure 2: The louvered plate condenser

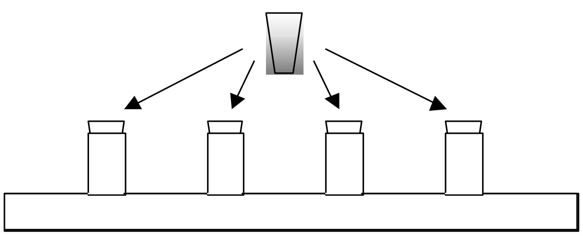

Figure 3: The doping special pipe

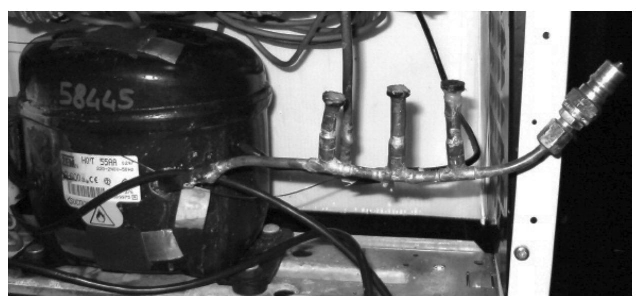

Figure 4: The doping pipe connected to the compressor

2.3 Continuous Running test

Continuous running tests are carried out conforming to ANSI/AHAM HRF-1-1988 (1988); the compressor runs continuously (no thermostatic control) at least for 24 hours or until thermal equilibrium is established. The equilibrium is reached when for 5 hours the measured temperature of the refrigerator compartment does not differ more than ±0.5 °C from the mean value and when there is no significant deviation of other important thermodynamic parameters, according to CECED, 2000.
Thereafter the data from the measurement sensors are acquired for an hour at intervals of 20 seconds. The resulting outputs are the time-averaged values of the different variables. The ambient temperature was 32°C (± 0.2°C).

2.4 Energy Consumption test

For the energy consumption test, the refrigerator is tested at fixed thermostat setting and ambient temperature of 25°C (± 0.2°C). The compressor cycles at least for 24 hours or until thermal equilibrium is established as specified in UNI/EN 153 (1997) and in UNI/EN/ISO 7371 (2000). The equilibrium is reached when for 24 hours the measured temperature of the refrigerator compartment does not differ more than ±0.5 °C from the mean value (CECED, 2000).
Thereafter the data from the measurement sensors are acquired for 24 hours at 20 seconds intervals. Typical test results are energy consumption expressed in kWh/d (accuracy: ± 0.02kWh/d), on and off times of the compressor (accuracy: ±0.13s), run time percentage and the time-averaged values of suction and discharge pressure during the on time.
The test is repeated for different thermostat settings to determine by linear interpolation the final results referred to the refrigerator compartment mean temperature of 5 °C, according to UNI/EN 153 (1997) and in UNI/EN/ISO 7371 (2000).

2.5 Doping procedure

To assure the injection of an exact amount of non-condensable gas, before each doping the circuit is evacuated until a pressure lower than 0.2 mbar is reached; thereafter the circuit is charged with the nominal refrigerant amount 40g (±0.2 g).
The refrigerator is put inside the climatic room at 25°C (± 0.2°C) and relative humidity of 50% and switched on as described in section 2.4. When thermal equilibrium is established and the compressor is running, the desired air volume (± 1.0 cm^3) is injected with a delicate syringe (50 cm^3 total capacity), piercing one of the rubber plugs of the pipe welded to the service tube. After the injection, the plug is resealed with a suitable adhesive to prevent any unwanted air entrance. The injected air mass is calculated from the volume syringed into the circuit, being known the actual values of temperature and pressure inside the syringe.

3. EXPERIMENTAL RESULTS

3.1 Tests on appliance without Suction Gas Heat Exchanger

As already mentioned, the refrigerating circuit was simplified by removing the tube-in-tube heat exchanger made up inserting the capillary tube inside the suction line of the compressor. This action was aimed at simplifying the analysis of the system.

Continuous running tests were performed at different ambient temperature, but the most significant experimentation, to which refer data in Table 1 and in Figures 5 and 6, was conducted at 32°C. The recorded data show clearly the effect of the injected air:

- at the condenser, increasingly with the molar fraction of air, it can be observed:
  - i) increasing of pressure at the discharge side of compressor;
  - ii) increasing of the condensed subcooling;
  - iii) increasing of the wall temperature in the saturated region of the condenser, according to pressure;
- at the evaporator, increasingly with the molar fraction of air, it can be observed:
  - i) lowering of pressure at suction side of the compressor;
  - ii) increasing vapour superheating at the evaporator outlet;
  - iii) lowering of the wall temperature in the saturated region of the evaporator, according to pressure;

The conclusion is straightforward: as the molar fraction of air increases, a certain amount of refrigerant charge is steadily transferred from evaporator to condenser. The obvious result is an increase in condensation temperature, in subcooling of liquid, in superheating of vapour, as well as a decrease in evaporation temperature.
The cause of this behaviour can be ascribed to a clogging action on the capillary tube carried out by non-condensable gases: if we compare the reference case (no air) with the maximum molar fraction of air (1.46%), we must notice that, even though subcooling is increased from 4.7 to 13.5°C and the pressure difference at the compressor is increased from 5.69 to 6.95 bar, the mass flow rate through the capillary tube is reduced.
On the other hand, it is well known that the presence of some residual vapour at the capillary inlet results in lower mass flow rate, the more so as the vapour quality increases The same happens when non-condensable gases are present inside the considered system. As condensation goes on, the vapour quality decreases and the flow pattern from annular flow tends to stratified flow, plug flow and finally bubble flow, but, unlike the case of condensation of a pure fluid, bubbles cannot collapse, because contain an air/vapour mixture and condensation stops when the partial

pressure of the refrigerant becomes equal to that of the surrounding liquid; so bubbles are carried inside the capillary tube and clog the flow. From the temperature profiles shown in Figure 5, it seems that two phase change is not any more isothermal when the amount of doping air is beyond a certain level: this is possible, as a consequence of the variation in partial pressure of the components when vapour is progressively condensed.
Energy efficiency suffers a penalisation from the worsening of the operating conditions induced by non-condensable gases; although the input power is not subjected to great variation, it must be noticed that the temperature of refrigerator compartment is progressively increasing with the fraction of air present inside the circuit (with a limited exception moving from 0.29 to 0.59, maybe due to inaccurate testing procedure or measurements). Therefore the correct reference for power consumption is not the original test with no air, but the condition with no air and the same internal temperature.
Energy consumption tests were performed at different ambient temperature and in Table 2 are reported the data collected at 25 and 32°C ambient temperature. Here the energy consumption data obtained at different air fractions are directly comparable, since the internal temperature of the refrigerator is the same, being automatically adjusted by the thermostatic control through different run times of the compressor; the effect of the injected air appears much more detrimental for energy efficiency than it was expected and tends to become more important as the ambient temperature is increasing. For the definition of the time depending variables reference is made to section 2.4.

Table 1: Continuous running tests results for the appliance without SGHX (suction gas heat exchanger)

Continuous Running test

Air molar fraction	**[%]**	**0.00**	**0.29**	**0.59**	**1.17**	**1.46**
Air mass	[g]	0.00	0.06	0.12	0.24	0.29
Room temperature	[°C]	32.0	32.1	32.1	32.0	31.9
Refrigerator compartment mean temperature	[°C]	-7.4	-6.6	-6.9	-6.1	-4.1
Suction pressure	[barA]	0.71	0.66	0.69	0.68	0.65
Discharge pressure	[barA]	6.4	6.5	6.9	7.4	7.6
Evaporation temperature (compressor suction)	[°C]	-20.7	-22.0	-21.0	-21.3	-22.4
Condensation temperature (compressor discharge)	[°C]	47.4	47.9	50.3	53.3	54.2
Middle condenser temperature	[°C]	45.5	45.6	47.2	47.3	46.8
Dryer temperature	[°C]	40.8	36.9	35.3	33.2	33.3
Input power	[W]	59.5	57.7	59.0	59.4	58.6

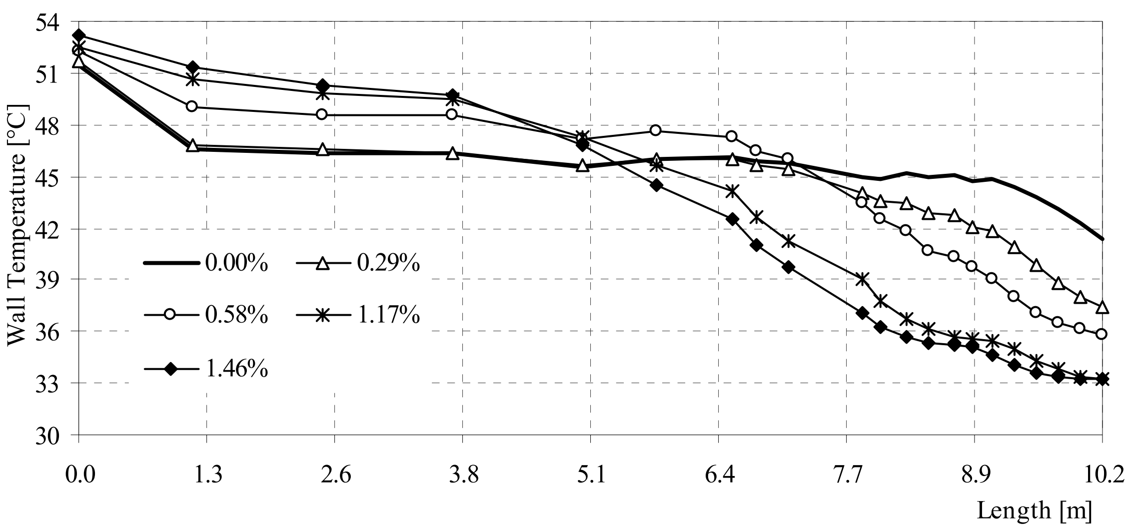

Figure 5: Condenser wall temperatures in continuous running tests with the appliance without SGHX

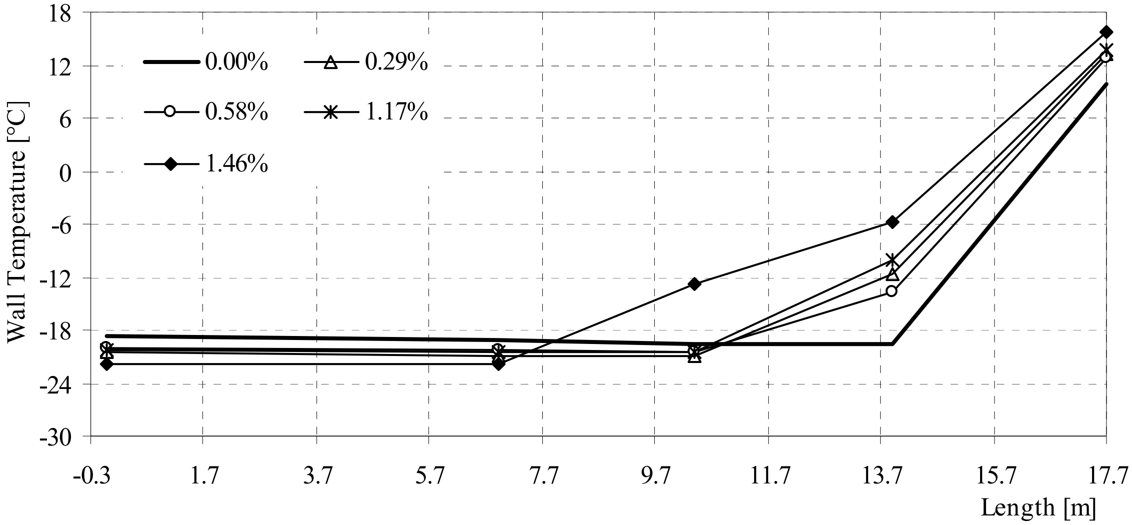

Figure 6: Evaporator wall temperatures in continuous running tests with the appliance without SGHX

3.2 Tests on appliance with Suction Gas Heat Exchanger

Continuous running tests were performed at 32°C ambient temperature and the results are similar to the ones of the appliance without SGHX, as it can be seen by comparing the temperature profiles in Figure 5 and in Figure 7. It is worth noting that in the reference case (no air present) the system performs better with SGHX than without it and operates with nearly no flooded area at the condenser.
Energy consumption tests were performed at 25°C ambient temperature. Since the presence of the Suction Gas Heat Exchanger makes the system more sensitive to non-condensable gases, tests were performed varying the molar fraction of doping air by smaller increment than in the previous case (no SGHX). The results are shown in Table 3. From comparison with the similar results for the case without SGHX, it can be inferred that such a device improves energy efficiency in the basic case, but the advantage is steadily decreasing as the fraction of non-condensable gases inside the circuit is increasing.
The diagrams in Fig. 8 and 9 offer a clear and direct representation of the effect of non-condensable gases on compressor run time, energy consumption and pressure ratio; all the profiles are in good agreement with the theoretical analysis exposed in 3.1.

Table 2: Energy consumption results for the appliance without SGHX (suction gas heat exchanger)

Energy Consumption test									
Air molar fraction	**[%]**	**0.00**	**0.29**	**0.59**	**1.46**	**0.00**	**0.29**	**0.59**	**1.46**
Air mass	[g]	0.00	0.06	0.12	0.29	0.00	0.06	0.12	0.29
Room temperature	[°C]	25.0				32.0			
Compartment mean temperature	[°C]	5.0				5.0			
Suction pressure	[barA]	0.72	0.66	0.63	0.62	0.86	0.80	0.76	0.65
Discharge pressure	[barA]	5.2	5.4	5.5	6.4	6.5	6.6	6.8	7.4
Evaporation temp. (comp. suction)	[°C]	-20.0	-22.0	-23.1	-23.5	-15.7	-17.5	-18.7	-22.4
Cond. temp. (comp. discharge)	[°C]	38.9	40.7	41.4	47.3	47.9	48.5	49.7	53.2
Energy Consumption	[kWh/day]	*0.57*	*0.59*	*0.64*	*0.68*	*0.78*	*0.79*	*0.86*	*1.01*
Energy Consumption Variation	[%]	-	*3.5*	*12.3*	*19.3*	-	*1.3*	*10.3*	*29.5*
Run Time	[%]	38.6	41.7	47.4	49.5	48.0	51.5	57.1	72.0
Run Time Variation	[%]	-	8.0	22.8	28.2	-	7.3	19.0	50.0

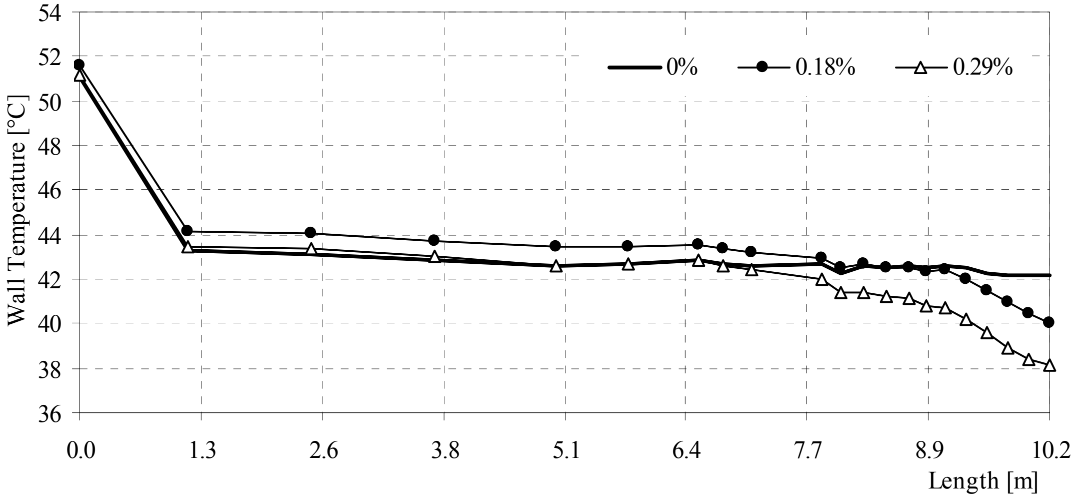

Figure 7: Condenser wall temperatures in continuous running tests with the appliance with SGHX

Table 3: Energy consumption results for the appliance with SGHX (suction gas heat exchanger)

Energy Consumption test **Air molar fraction**	**[%]**	**0.00**	**0.06**	**0.12**	**0.18**	**0.24**	**0.29**	**0.47**	**0.76**	**0.99**
Air mass	[g]	0.000	0.012	0.024	0.035	0.047	0.059	0.094	0.153	0.200
Room temperature	[°C]					25.0				
Compartment mean temperature	[°C]					5.0				
Suction pressure	[barA]	0.87	0.84	0.83	0.82	0.77	0.73	0.66	0.64	0.62
Discharge pressure	[barA]	5.4	5.45	5.4	5.5	5.49	5.6	5.7	6.1	6.2
Evaporation temp. (comp. suction)	[°C]	-15.7	-16.3	-16.6	-16.9	-18.4	-19.7	-22.4	-23.1	-23.8
Cond. temp. (comp. discharge)	[°C]	40.8	41.0	40.9	41.5	41.2	42.1	42.9	45.3	46.2
Energy Consumption	[kWh/day]	*0.491*	*0.492*	*0.500*	*0.523*	*0.529*	*0.536*	*0.585*	*0.604*	*0.630*
Energy Consumption Variation	[%]	-	*2.5*	*4.2*	*9.0*	*10.2*	*11.7*	*21.9*	*25.8*	*31.3*
Run Time	[%]	31.7	31.70	31.50	34.00	34.30	35.90	41.10	43.30	*46.00*
Run Time Variation	[%]	-	2.3	1.6	9.7	10.6	15.8	32.6	39.7	*48.4*

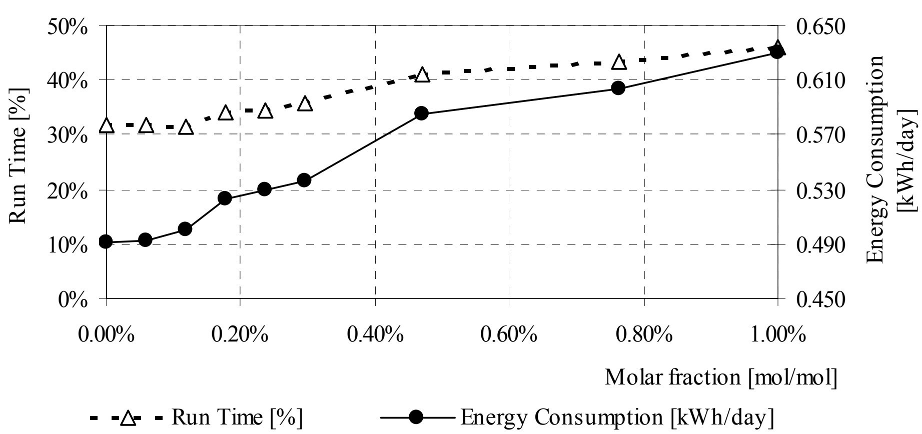

Figure 8: Energy Consumption and Run Time with SGHX

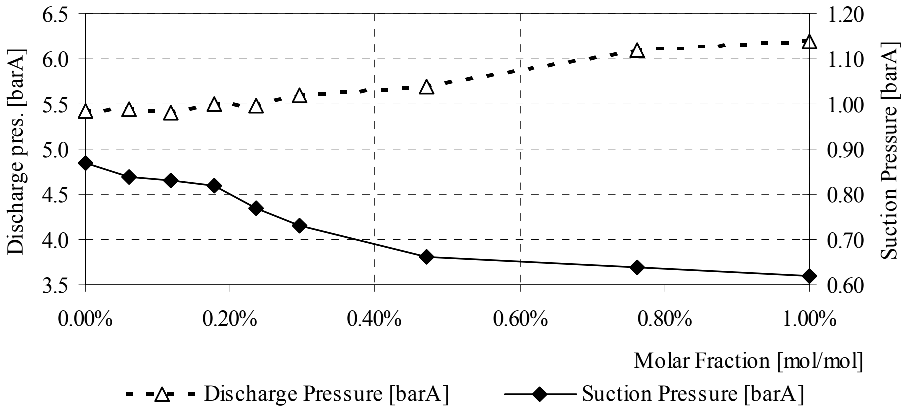

Figure 9: Suction and discharge pressure in energy consumption tests with SGHX

4. CONCLUSIONS

Tests conducted on a household appliance have demonstrated that a significant penalisation, maybe more than expected, arises from the presence of incondensable inside the refrigerating circuit; this penalisation concerns both energy efficiency and refrigerating capacity.
The detrimental effect of non-condensable gases can be theoretically explained, not so much in terms of a direct action resulting in worse heat transfer process during condensation, but in terms of decrease in capacity of the throttling device. Unfortunately a capillary tube cannot possess an intrinsic capability of controlling the mass flow rate and therefore, under such circumstances, induces flooding of condenser and starving of evaporator that are responsible for the degraded performance.

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