Testing and Modelling of a Bisplit Refrigeration System
Analysis of the Refrigerant Charge Effect on the System Global Performance

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ABSTRACT

This work focuses on the testing and modelling of a bisplit refrigeration system. This system is composed of two evaporating circuits (each circuit consists of an expansion device and an evaporator) connected in parallel inside the same refrigeration unit. The test bench developed in the Laboratory of Thermodynamics is briefly described and the influence of the refrigerant charge effect on the system global performance is examined. The modelling of a bisplit chiller is also studied and a first model validation is given.

INTRODUCTION

The most simple refrigeration system is made up of four basic components: an evaporator (low pressure level), a compressor, a condenser (high pressure level) and an expansion device. This kind of system is usually used when only one cooling demand is needed. However two or more cooling demands would require two or more simple chillers which is obviously not the optimal solution: this would considerably increase the required space as well as the global plant cost. That is why many air conditioning systems often comprise several evaporators connected in parallel inside the same refrigeration unit: a « multisplit » system is obtained. A drawback of this system is that it requires quite a complicated control strategy.
For instance, the use of a multisplit system is recommended for the household air conditioning and for the food storage in supermarkets.

The objective of this work is double. The first aim is to design and develop a test bench of a bisplit system and to carry out steady-state tests with different refrigerant charges. These tests must lead to a good understanding of the actual charge effect on the system global performance. The second aim is to develop a general model of a two-evaporator refrigeration system (« bisplit » system); this model should help the user to design a whole bisplit chiller properly thanks to an appropriate selection of the basic components.

This work is the continuation of a first study described in [1].

DESCRIPTION OF THE TEST BENCH

The bisplit refrigeration system is shown in Figure 1; the refrigerant used is HCFC-22.
The main components are as follows:

- (i) a hermetic scroll compressor installed inside a calorimeter;
- (ii) a coaxial water-cooled condenser. This condenser is fixed to a force transducer so that the refrigerant charge inside the condenser can be measured during the tests;
- (iii) a condensing pressure regulator located on the water circuit and connected to the compressor discharge. This regulator adjusts the water flow rate through the condenser in order to obtain the desired condensing pressure;
- (iv) two evaporating circuits connected in parallel. Each circuit is composed of a thermostatic expansion valve (external pressure equalisation) and a direct expansion coil;
- (v) an evaporating pressure regulator located at the outlet of one evaporator. This regulator allows the system to operate with two different evaporating temperatures;
- (vi) a capacity regulator with a hot gas bypass;
- (vii) an auxiliary heat exchanger used to adjust the compressor inlet temperature in order to keep the discharge temperature within an admissible range.

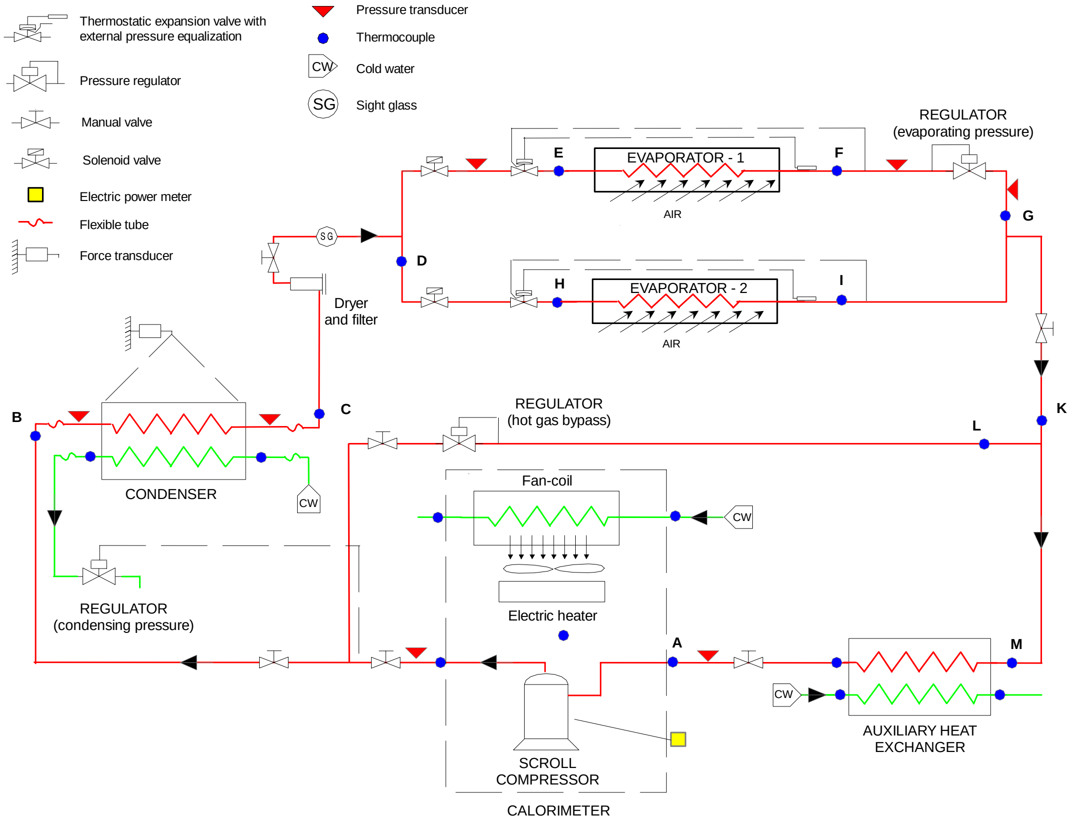

Figure 1 - Test bench of a bisplit refrigeration system

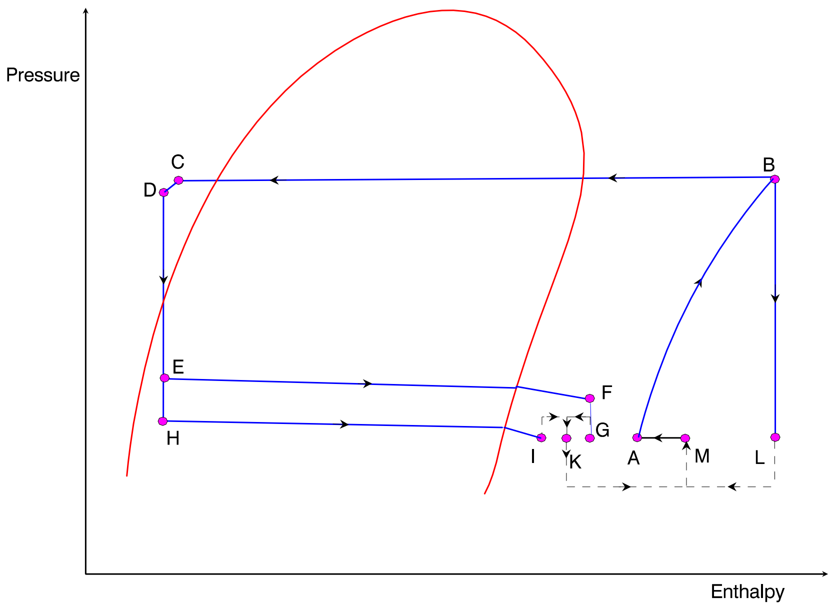

Figure 2 - Thermodynamic cycle during a bisplit system operation

The particularity of this bisplit system lies in the use of thermostatic expansion valves. Usually capillary tubes or electronic valves are used in multisplit systems.

Each evaporator is installed inside a closed air loop; air circulation is achieved by means of a fan. A series of electric heaters are used to heat up the air before it enters the coil: the setting of these heaters allows us to adjust the thermal load applied to the coil.

The thermodynamic cycle can be represented on a pressure-enthalpy diagram (Figure 2).
Compression of the superheated vapour takes place between points A and B. Condensation begins at point B; the refrigerant leaves the condenser at point C (refrigerant is then subcooled or two-phase) and enters the expansion valves at point D, after a slight pressure drop. The expansions occur in the two thermostatic valves and two-phase refrigerant is obtained at points E and H. The refrigerant evaporates as it passes through the coils and superheated vapour is obtained at points F and I. An additional expansion, between points F and G, occurs through the evaporating pressure regulator and the two streams are mixed. The resulting stream (point K) is again mixed with the refrigerant coming from the hot gas bypass circuit (point L). Finally the vapour is cooled down in the auxiliary heat exchanger (from point M to point A) before entering the compressor.

DESCRIPTION OF THE PERFORMED TESTS

Two distinct series of tests are undertaken. The first series (chronologically) aims at analysing the general behaviour of a bisplit system; the chiller is filled with a constant refrigerant charge. The experimental results are used in order to validate the model. The second series considers several refrigerant charges in order to determine whether or not the charge has an influence on the system performance.

The test bench was slightly modified between the two series. For the first series of tests, a shell-and-tube condenser followed by a liquid receiver and a subcooler are used. A considerable amount of liquid refrigerant is located inside these three components. In particular, the liquid receiver acts as a refrigerant tank which could possibly induce some perturbation on the charge distribution throughout the whole system. Thus it was decided to replace these three components by a single coaxial condenser (as shown in Figure 1) when charge effect must be analysed.

The first series of thirty one tests is carried out with the bisplit system under different load conditions. The cooling capacities are imposed by simply adjusting the electric power supplied to the heaters located inside the air loops. A typical test sequence is carried out by imposing a given load in one air loop while the load applied to the other air loop is progressively increased from 1 kW up to 4 kW (or, conversely, decreased from 4 kW down to 1 kW). Different settings (i.e. static superheat settings) are also applied to the thermostatic expansion valves. The evaporating pressure is controlled by means of the capacity regulator which always keeps this pressure above a minimum value; indeed, too low evaporating temperatures could cause water condensation on the air side of the evaporators.

The table below shows the variation domain for evaporating ($t_{ev,1}$ and $t_{ev,2}$) and condensing (t_{cd}) temperatures, subcooling (ΔT_{sc}) and superheats ($\Delta T_{sh,1}$ and $\Delta T_{sh,2}$) and refrigerating capacities ($\dot{Q}_{ev,1}$ and $\dot{Q}_{ev,2}$). Some tests are performed with only one evaporating circuit in operation ($\dot{Q}_{ev} = 0$ for the closed evaporating circuit).

	$t_{ev,1}$ [C]	$t_{ev,2}$ [C]	t_{cd} [C]	ΔT_{sc} [K]	$\Delta T_{sh,1}$ [K]	$\Delta T_{sh,2}$ [K]	$\dot{Q}_{ev,1}$ [W]	$\dot{Q}_{ev,2}$ [W]
minimum	5	5	31	4.8	0	0	0	0
maximum	17	17	58	12.7	24	24	4300	4300

The second series of twenty-four tests is carried out with seven different refrigerant charges, varying from 1.7kg to 2.4kg. A very important point is to verify that the system is free from any leakage before starting the tests. The test bench is first pressurised with nitrogen and a pressure of about 20 bar is maintained for ten hours; the system is then vacuumed (down to 0.2 bar) for another ten-hour period.
A scale is used to weigh the refrigerant added to the system.

Since the main goal of these tests is to analyse the refrigerant charge effect, it is necessary to maintain the same operating conditions for all the charges: eight so called « standard tests » are performed.
The standard conditions are imposed as follows:

(i) Condenser: the supply water temperature is adjusted by means of electric heaters and a constant water flow rate is imposed by a small pump (during these tests, the condensing pressure regulator is bypassed). The following values represent respectively the average and the standard deviation obtained for all the standard tests.

$$\mathbf{t}_{\mathbf{w},\mathbf{su},\mathbf{cd}} = 13.2\,\mathbf{C} \pm 0.5\mathbf{K}$$
$$\dot{\mathbf{M}}_{\mathbf{w},\mathbf{cd}} = 0.06696\ \mathbf{kg\,s}^{-1} \pm 0.4\%$$

(ii) Evaporators: a constant dry air temperature is imposed at both evaporator inlets by means of voltage converters connected to the electric heaters. All the tests are performed with the same fan velocity.

$$\mathbf{t}_{\mathbf{a},\mathbf{su},\mathbf{ev}1} = 19.0\,\mathbf{C} \pm 0.1\mathbf{K}$$
$$\mathbf{t}_{\mathbf{a},\mathbf{su},\mathbf{ev}2} = 14.2\,\mathbf{C} \pm 0.1\mathbf{K}$$
$$\dot{\mathbf{M}}_{\mathbf{a},\mathbf{ev}1} = 0.733\mathbf{kg\,s}^{-1} \pm 4\,\%$$
$$\dot{\mathbf{M}}_{\mathbf{a},\mathbf{ev}2} = 0.732\,\mathbf{kg\,s}^{-1} \pm 5\%$$

As explained in the next section, the dry air mass flow rate is not directly measured but is calculated by means of an energy balance which leads to higher standard deviations than for measurements.

(iii) Thermostatic expansion valves: both valves have a fixed setting so that nearly constant superheats are measured for all the standard tests.

$$\Delta\mathbf{t}_{\mathbf{SH}1} = 6.5\mathbf{K} \pm 0.4\mathbf{K}$$
$$\Delta\mathbf{t}_{\mathbf{SH}2} = 6.6\,\mathbf{K} \pm 0.4\mathbf{K}$$

(iv) Bypass capacity regulator: this regulator is adjusted for each test so that a constant pressure is imposed at the compressor inlet. Therefore, the same evaporating temperatures are reached for all standard tests.

$$\mathbf{p}_{\mathbf{su},\mathbf{cp}} = 5.77\,\mathbf{bar} \pm 0.04\,\mathbf{bar}$$
$$\mathbf{t}_{\mathbf{su},\mathbf{ev}1} = 6.3\mathbf{C} \pm 0.2\mathbf{K}$$
$$\mathbf{t}_{\mathbf{su},\mathbf{ev}2} = 5.2\,\mathbf{C} \pm 0.2\mathbf{K}$$

In addition to these standard tests, other tests with constant refrigerant charge are carried out under different load conditions. These tests are useful for model validation purposes.

DESCRIPTION OF THE ENERGY BALANCES

Evaporators

The control volume of one evaporator air loop is represented in Figure 3.

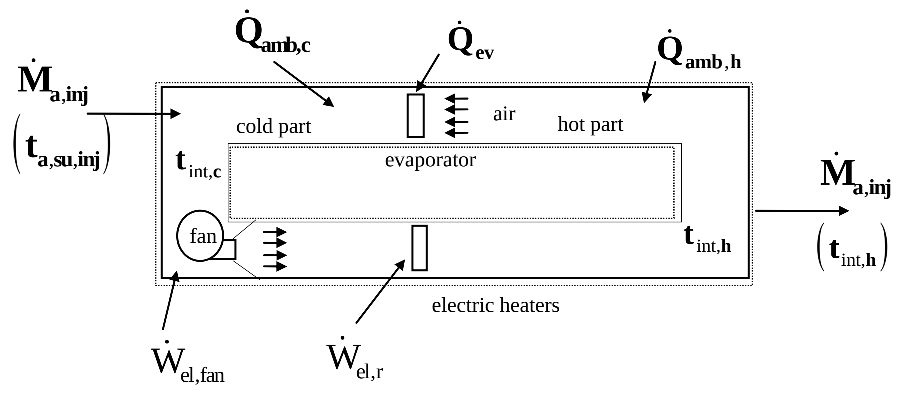

Figure 3 - Control volume of the whole air loop

It is necessary to avoid condensation inside the evaporators otherwise an unknown quantity of condensate should be taken into account in the energy balance. The purpose of the capacity regulator is to control the evaporating pressure in order to avoid condensation. However if low evaporating temperatures (5 C, for instance) are required, dry air must be injected in order to reduce the dew point temperature inside the loop.

Since most of the cold part is at the atmospheric pressure, it is assumed that the air leakage occurs only in the hot part of the loop. The refrigerating capacity, $\dot{Q}_{ev}$, of each evaporator is obtained by means of the energy balance over the whole air loop. This balance leads to the following relationship in steady-state regime:

$$\dot{Q}_{ev} + \dot{W}_{el,fan} + \dot{W}_{el,r} + \dot{Q}_{amb,h} + \dot{Q}_{amb,c} + \dot{M}_{inj}\, c_{pa}\,(t_{a,inj} - t_{int,h}) = \left(\frac{dU}{d\tau}\right)_{loop} \tag{1}$$

where:

$\dot{Q}_{amb,h} = (UA)_h.(t_{amb} - t_{int,h})$ and $\dot{Q}_{amb,c} = (UA)_c.(t_{amb} - t_{int,c})$ represent the ambient losses of the hot and cold parts of the air loop; $t_{int,h}$ and $t_{int,c}$ are the average air temperatures inside the hot and cold parts of the air loop.

Note that the balances are always written with the following convention: a heat flux or an electrical power is positive if it is provided to the system.

The dry air mass flow rate through the coil is calculated from the energy balance of the hot part (Figure 4).

$$\dot{M}_{a,ev} = \frac{\dot{W}_{el,r} + \dot{Q}_{amb,h} + \dot{M}_{a,inj}\, c_{pa}\left(t_{a,su,r} - t_{int,h}\right) - \left(\frac{dU}{d\tau}\right)_{hot}}{c_{pa}\left(t_{a,su,ev} - t_{a,su,r}\right)} \tag{2}$$

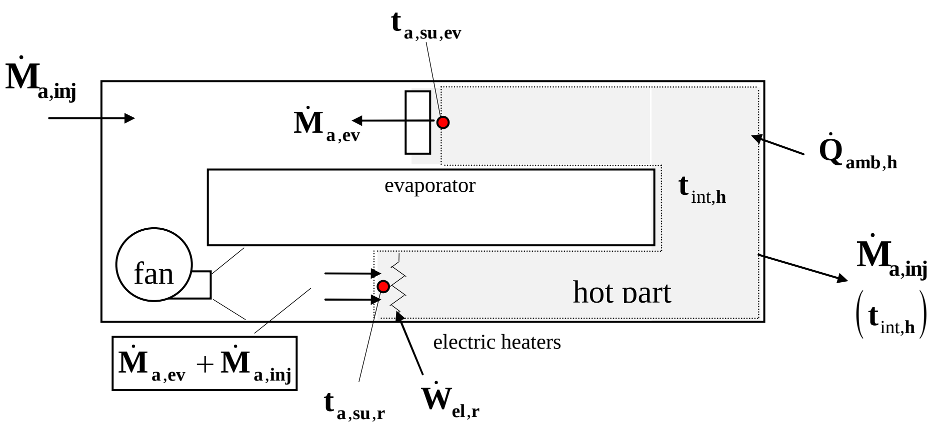

Figure 4 - Control volume of the hot part

An energy balance applied to each evaporator (Figure 5) leads to the refrigerant mass flow rate passing through it.

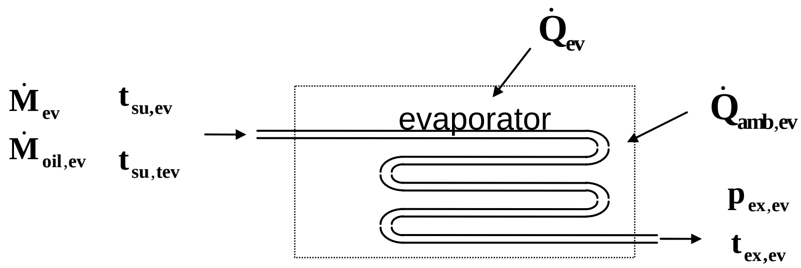

Figure 5 - Control volume of the evaporator

In addition to the refrigerant, it is assumed that lubricating oil also circulates inside the whole system.
The following equation is obtained:

$$\dot{M}_{ev} = \frac{\dot{Q}_{ev} + \dot{M}_{oil,ev}\,\bar{c}_{oil}\left(t_{su,tev} - t_{ex,ev}\right) + \dot{Q}_{amb,ev} - \left(\frac{dU}{d\tau}\right)_{ev}}{h_{ex,ev} - h_{su,ev}} \tag{3}$$

It is important to specify that the oil temperature remains constant during the expansion process so that the temperature at the valve inlet must be considered in the balance.
More details about this observation are given in [2].

Condenser

Figure 6 shows the control volume of the coaxial condenser.
The refrigerant mass flow rate through the condenser is given by:

$$\dot{M}_{cd} = \frac{\dot{M}_{w,cd}\,c_w\left(t_{w,ex,cd} - t_{w,su,cd}\right) + \dot{M}_{oil,cd}\,\bar{c}_{oil}\left(t_{ex,cd} - t_{su,cd}\right) - \dot{Q}_{amb,cd} + \left(\frac{dU}{d\tau}\right)_{cond}}{h_{su,cd} - h_{ex,cd}} \tag{4}$$

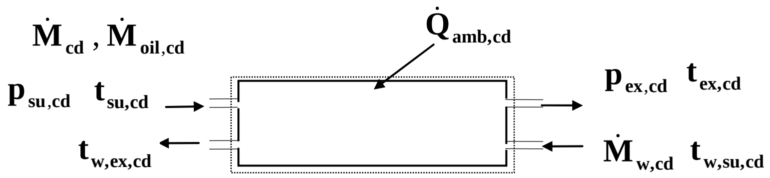

Figure 6 - Control volume of the condenser

Compressor and bypass mixing

The scroll compressor is installed inside a calorimeter which permits an accurate estimation of the compressor ambient losses. the energy balance applied to the calorimeter (Figure 7) can be written as follows:

$$\dot{Q}_{amb,cp} = \dot{M}_{w,cal}\,c_w\left(t_{w,ex,cal} - t_{w,su,cal}\right) - \dot{W}_{el,cal} - \dot{Q}_{amb,cal} + \left(\frac{dU}{d\tau}\right)_{cal} \tag{5}$$

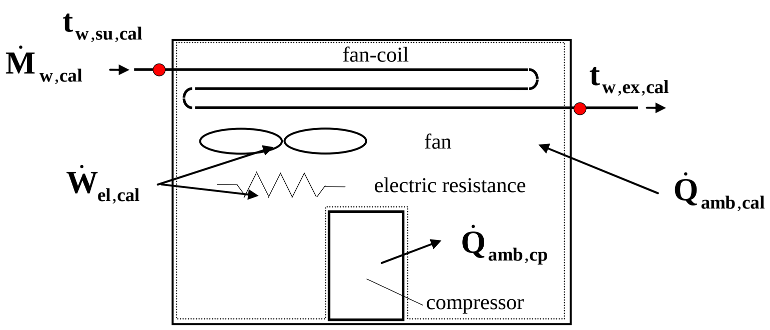

Figure 7 - Control volume of the calorimeter

The refrigerant mass flow rate through the compressor is given by (Figure 8):

$$\dot{M}_{cp} = \frac{\dot{W}_{el,cp} + \dot{Q}_{amb,cp} + \dot{M}_{oil,cp}\ \bar{c}_{oil}\left(t_{su,cp} - t_{ex,cp}\right) - \left(\frac{dU}{d\tau}\right)_{cp}}{h_{ex,cp} - h_{su,cp}} \tag{6}$$

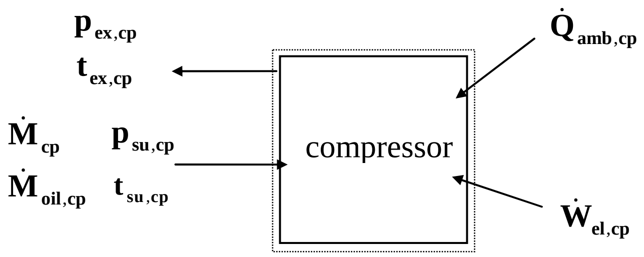

Figure 8 - Control volume of the compressor

A last energy balance can be written at the bypass mixing (mixing between the flow coming from the evaporators and the flow coming from the hot gas bypass), which is represented in Figure 9.

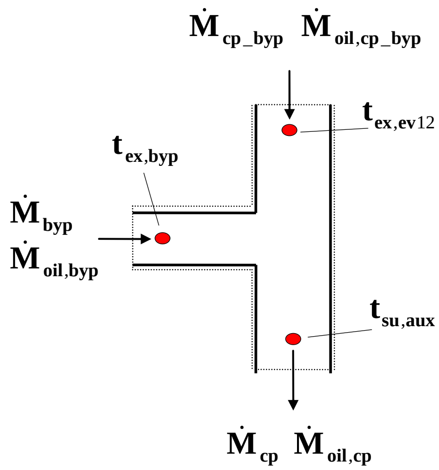

Figure 9 - Control volume of the bypass mixing

The mass flow rate coming from the evaporators can be easily calculated:

$$\dot{M}_{cp,byp} = \frac{\dot{M}_{cp}\left(h_{ex,byp} - h_{su,aux}\right) + \dot{M}_{oil,cp,byp}\left(\bar{c}_{oil}\ t_{ex,ev12} - h_{ex,byp}\right) + \dot{M}_{oil,byp}\left(\bar{c}_{oil}\ t_{ex,byp} - h_{ex,byp}\right) + \dot{M}_{oil,cp}\left(h_{ex,byp} - \bar{c}_{oil}\ t_{su,aux}\right)}{h_{ex,byp} - h_{ex,ev12}} \tag{7}$$

As can be seen, this last relationship requires the mass flow rate through the compressor (Equation (6)).

Calculation of the refrigerant mass flow rates

It is possible to check the accuracy of the energy balances by comparing the calculated flow rates. The following relative errors are estimated:

$$\varepsilon_{cd\_ev} = \frac{\dot{M}_{cd} - \left(\dot{M}_{ev1} + \dot{M}_{ev2}\right)}{\dot{M}_{cd}} \tag{8}$$

$$\varepsilon_{cd\_cpbyp} = \frac{\dot{M}_{cd} - \dot{M}_{cpbyp}}{\dot{M}_{cd}} \tag{9}$$

Figure 10 shows these two relative errors for the second series of tests when the oil circulation is neglected in each component.

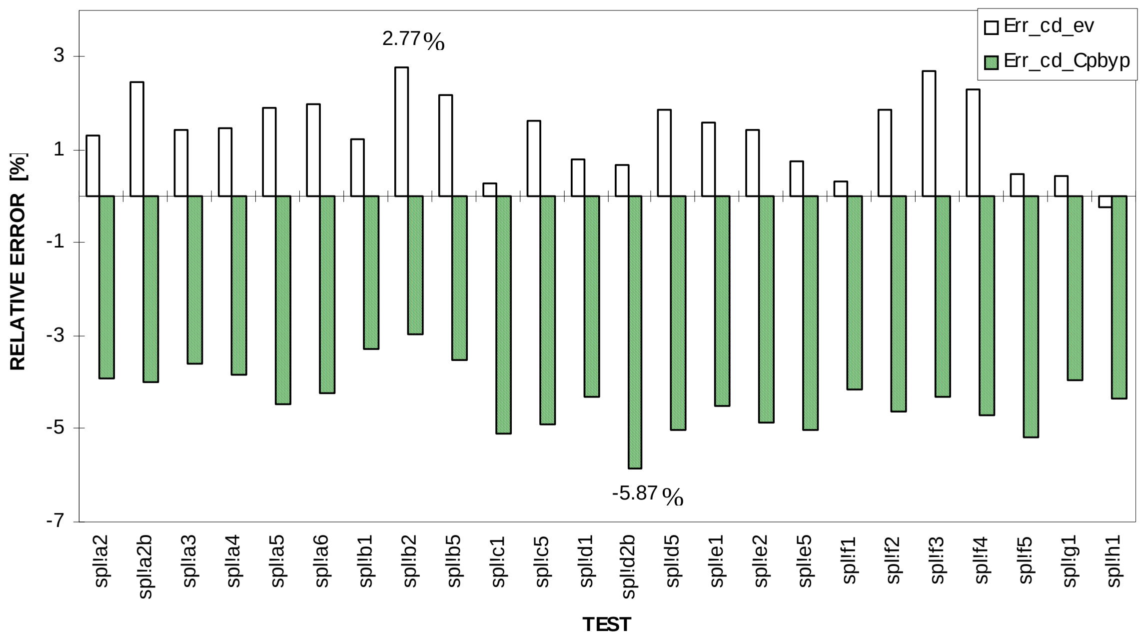

Figure 10 - Relative errors assuming no oil circulation.

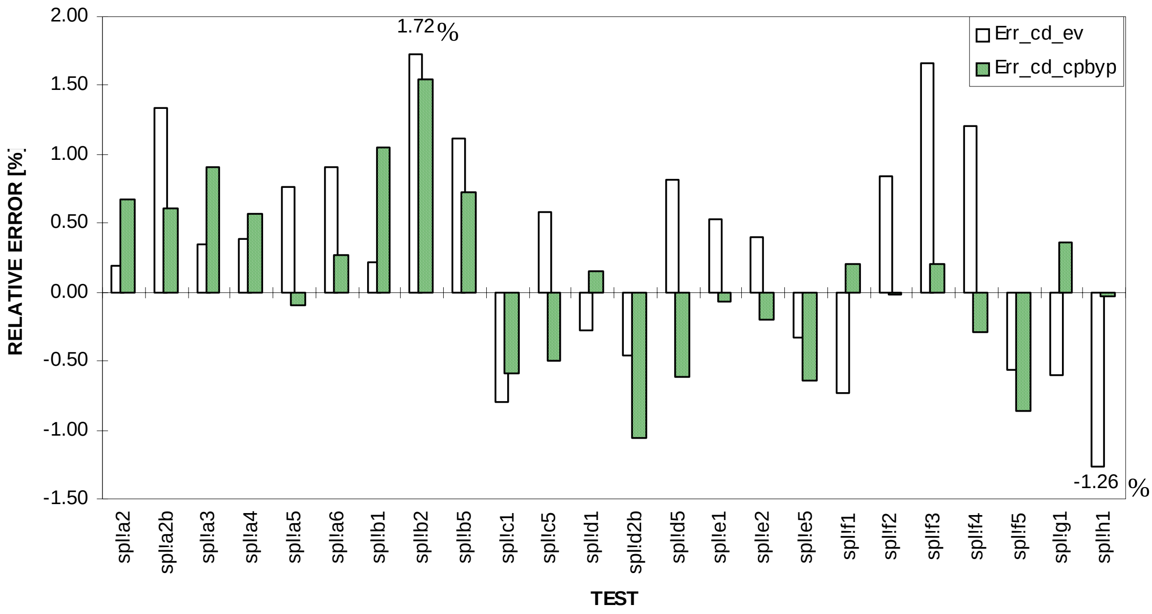

Figure 11 - Relative errors when oil circulation is considered .

There is an excellent agreement between the condenser and evaporator energy balances (mean relative error: 1.4%), the condenser balance always leading to the highest prediction. The mass flow rate calculated from the compressor-bypass mixing energy balance is always overestimated in comparison with the one through the condenser (mean relative error: 4.4%). However the bypass mixing balance leads to the less reliable result: an accurate balance requires homogeneous temperatures inside the pipes which is certainly not the case in this particular situation.

It is very interesting to introduce the effect of oil circulation.
It is assumed that the ratio of oil flow rate to the refrigerant flow rate is constant for all the components. An optimization method is then applied in order to calculate the ratio that minimizes the global relative error.

Figure 11 shows the results obtained for an optimal ratio of about 0.015, which is a quite realistic value.
An appreciable improvement is obtained and the relative errors do not exceed 1.7%.
Of course, the best justification of these results would be to experimentally measure the oil flow rate during a steady-state operation; unfortunately such a test has not been carried out so far. However it is clear that the oil effect should be taken into account in all energy balances when accurate flow rate estimations are required.

ANALYSIS OF THE TEST RESULTS

First series of tests

Figure 12 shows typical variations in the refrigerant temperatures at the evaporator inlets and outlets. The refrigerating capacities obtained for this test are 4.3 kW for the first evaporator and 1.5 kW for the second evaporator; however, the same behaviour is observed for all the tests.

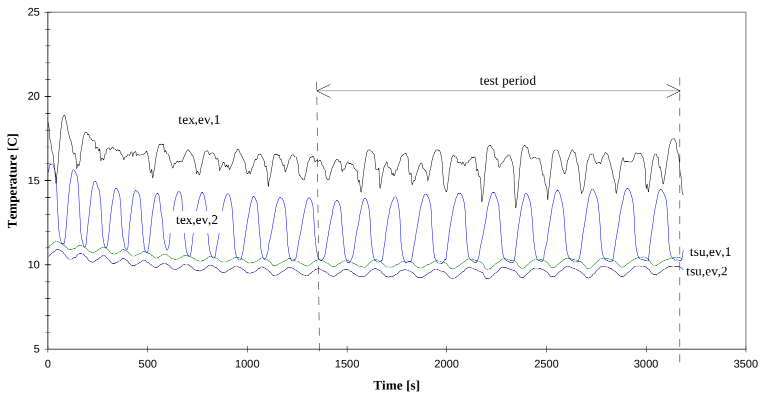

Figure 12 - Variations in the refrigerant temperatures at the evaporator inlets and outlets

As can be seen, two kinds of oscillations are encountered at the evaporator outlets.
The regular periodic oscillations observed for the second evaporator are due to control « hunting » [3,4]: first, the thermostatic expansion valve closes too much which leads to too high a superheating, then the valve progressively opens and some liquid refrigerant can leave the evaporator (the minimum outlet temperature becomes almost equal to the evaporating temperature); the valve detects too low a superheating and begins to close again. These oscillations can reach an amplitude of about 5 K. The reason for this phenomenon is that the thermostatic expansion valve can not adapt to a wide range of flow conditions [5].
The irregular oscillations appearing for the first evaporator are probably due to random fluctuations of the liquid dry-out point (i.e. the mixture-vapour transition point) [6,7,8] combined with the evaporator arrangement inside the air loop. The evaporator is supplied with two-phase refrigerant from its upper part while superheated refrigerant is collected in the lower part. This arrangement permits the oil discharge by gravity effect, but it could also result in the appearance of liquid refrigerant at the evaporator outlet. From time to time the thermocouple detects the presence of liquid droplets which causes irregular oscillations of the outlet temperature. This temperature begins to decrease but does not reach the droplet temperature (i.e. the evaporating temperature) due to the inertia of the thermocouple; then the temperature rises again when liquid has disappeared.

Of course, one has to take care when considering the energy balances applied to the evaporators: the possible presence of liquid refrigerant at the evaporator outlets could cause the outlet enthalpies to be poorly estimated. However, this is not the case with these tests since a good agreement between evaporator and condenser balances is obtained.

Standard tests (second series of tests)

During the « standard » tests, the refrigerant charge varies between 1.7 kg to 2.4 kg, (40% change). At the beginning of these tests, the thermostatic expansion valve settings are adjusted so that no oscillation is encountered at the evaporator outlets over the whole range of loading conditions (refrigerating capacity varying between 1 kW and 4 kW for each evaporator); these settings are maintained constant during all the standard tests.

It should be noted that two tests are performed at 2.3 kg charge. One is performed at the beginning of the experiments and the other at the end. The repeatability of the first test proves that the charging operation is properly carried out.

The effect of charging on the subcooling measured at the condenser outlet is shown in Figure 13. The subcooling increases continously as the charge also increases; however this influence seems to be less important at high-charge conditions.

Figure 14 is in complete agreement with this observation: the subcooling increases as the refrigerant charge inside the condenser increases.

Figure 15 shows that both superheats tend to decrease slightly as the refrigerant charge increases. This result indicates that the valves are able to maintain a relatively constant superheat for all the performed tests.

The influence of charging on the refrigerant flow rate through the condenser is illustrated in Figure 16. The system has a peak at 2.04 kg. A 11.2 % undercharge (1.81 kg) results in a 5.8 % reduction in flow rate and a 12.4 % overcharge (2.29 kg) produces a 9.5 % reduction in flow rate. Exactly the same behaviours are observed for the flow rates passing through each evaporator.

The variation of the total refrigerant capacity (evaporator 1 and evaporator 2) is shown in Figure 17. The behaviour is similar to that observed for the flow rate. Once again the same behaviours are observed for each refrigerant capacity.

A maximum capacity is reached at the optimal charge (2.04 kg) and overcharge as well as undercharge lead to almost the same capacity decrease: a 11.2 % undercharge (1.81 kg) results in a 7.1 % reduction in capacity and a 12.4 % overcharge (2.29 kg) produces a 5.6 % reduction in capacity.

The coefficient of performance (COP) (ratio of the total refrigeranting capacity to the compressor power consumption) is illustrated in Figure 18. A very small variation (about 3.5 %) is observed for the whole range of charging conditions. The maximum value is still obtained for the optimal charge but no general trend comes out.

In conclusion, careful attention must be paid to the total refrigerant charge inside a chiller. For given operating conditions, there is always a refrigerant charge leading to optimal system performance. Morever important undercharge or overcharge can lead to bad operation of the system. Two-phase refrigerant can occur at the condenser outlet for too high an undercharge, leading to a very important reduction in refrigerating capacity.
If the amount of refrigerant is too high, an unstable behaviour of the thermostatic expansion valves is likely to occur since the superheat tends to become too small. Such an observation is shown in Figure 19. This test (non standard test) corresponds to a charge of 2.29 kg and the refrigerating capacities are equal to 4 kW and 3 kW. As can be seen the steady-state regime is almost obtained, but suddenly the superheats tend to decrease and an unstable behaviour of the thermostatic expansion valves occurs.
If charge is still added liquid refrigerant can enter the compressor.

This study does not consider the effect of the condenser water inlet temperature. This effect is investigated in previous researches [9,10,11] dealing with the refrigerant charge impact on monosplit air-conditioner performance. However, exactly the same behaviours are observed over the whole range of temperatures applied to the condenser and optimal performances are obtained for almost the same refrigerant charge.

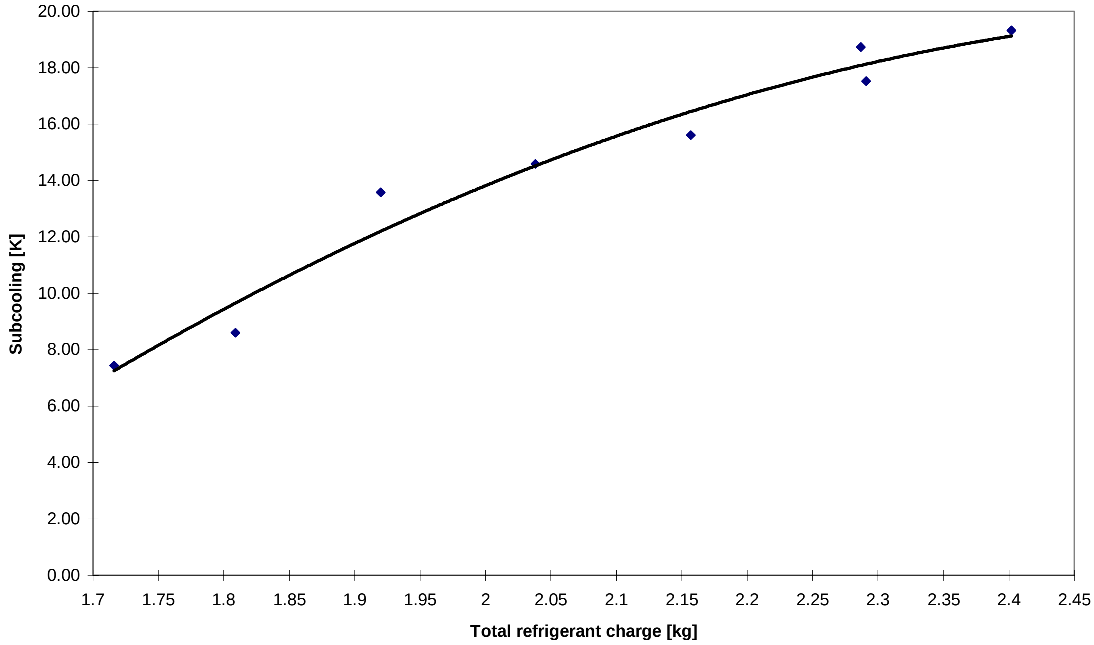

Figure 13 - Effect of the total refrigerant charge on the subcooling.

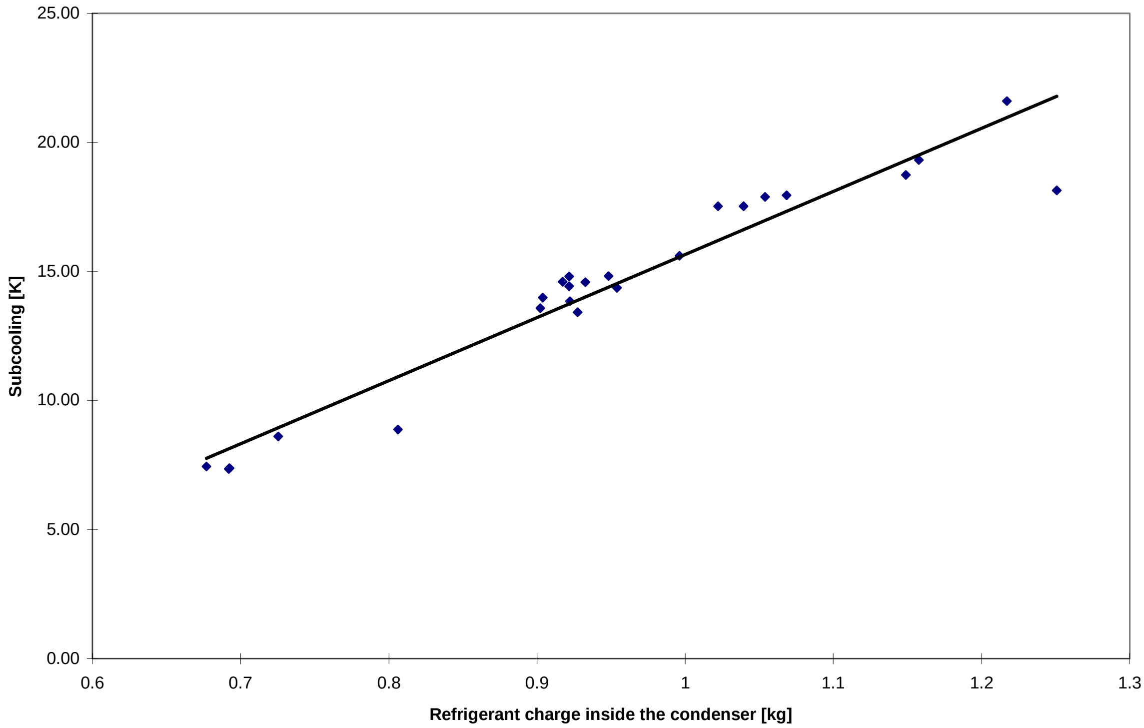

Figure 14 - Variation of the subcooling with the refrigerant charge contained in the condenser.

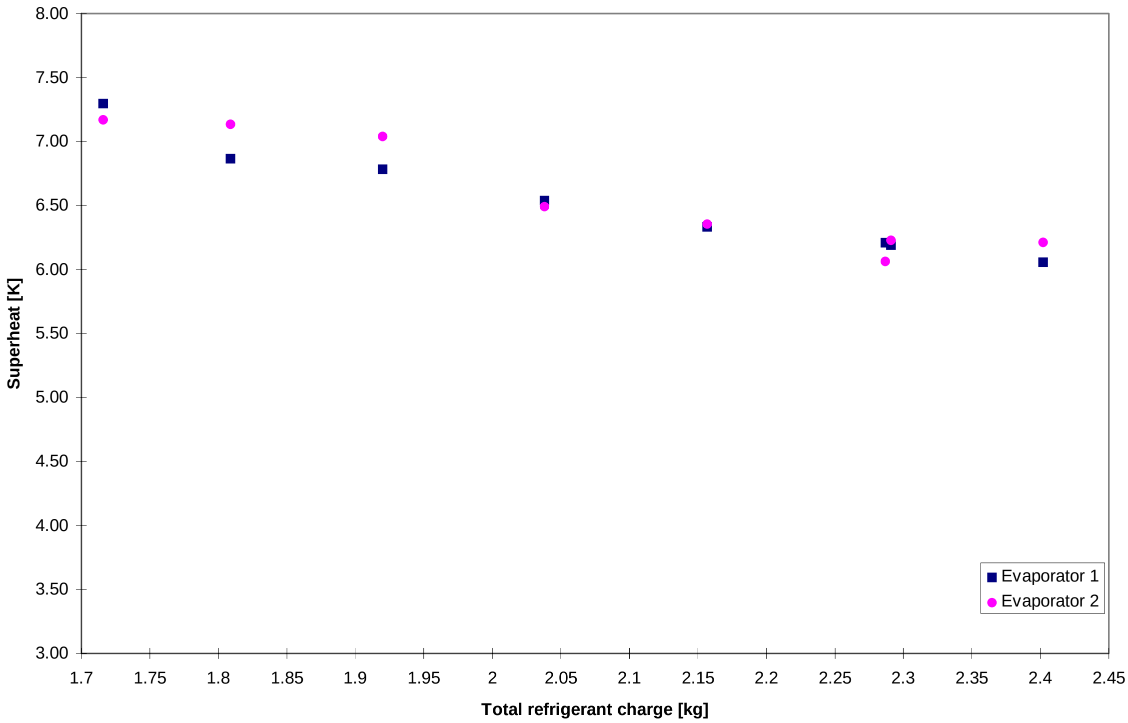

Figure 15 - Effect of the total refrigerant charge on the superheats.

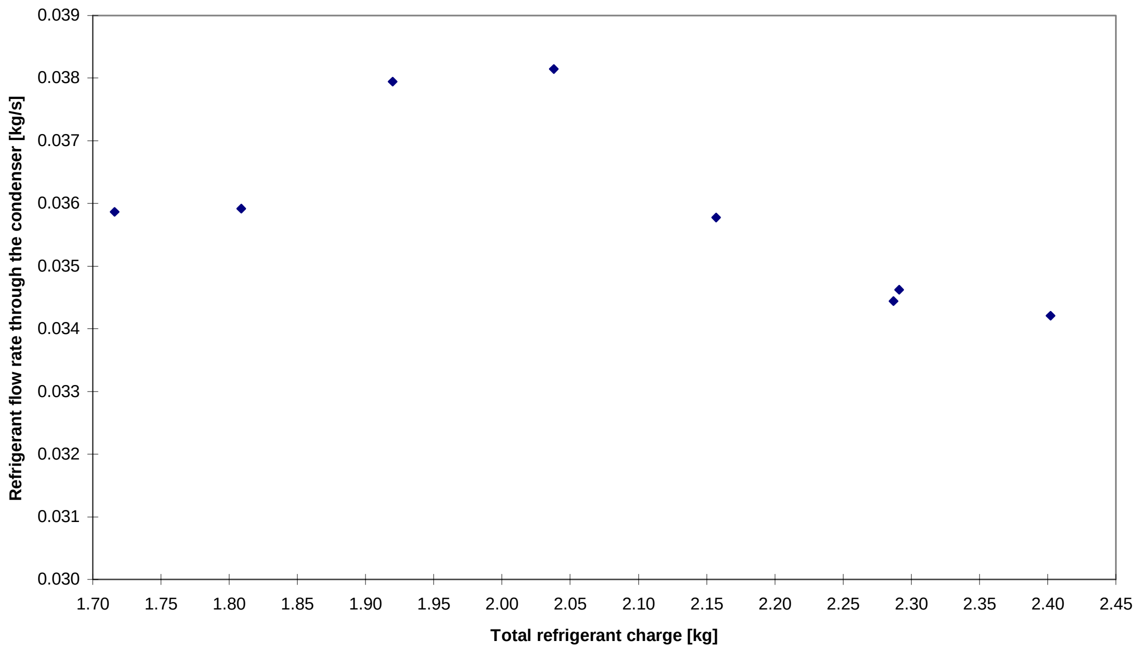

Figure 16 - Effect of the refrigerant charge on the flow rate through the condenser.

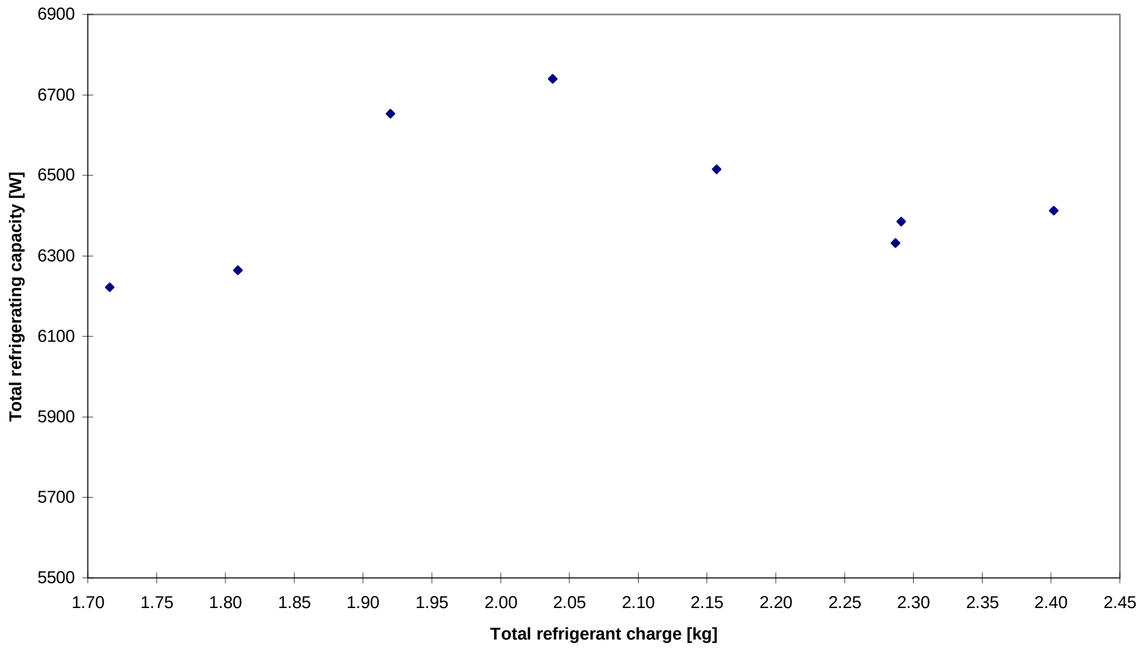

Figure 17 - Effect of the refrigerant charge on the total refrigerating capacity.

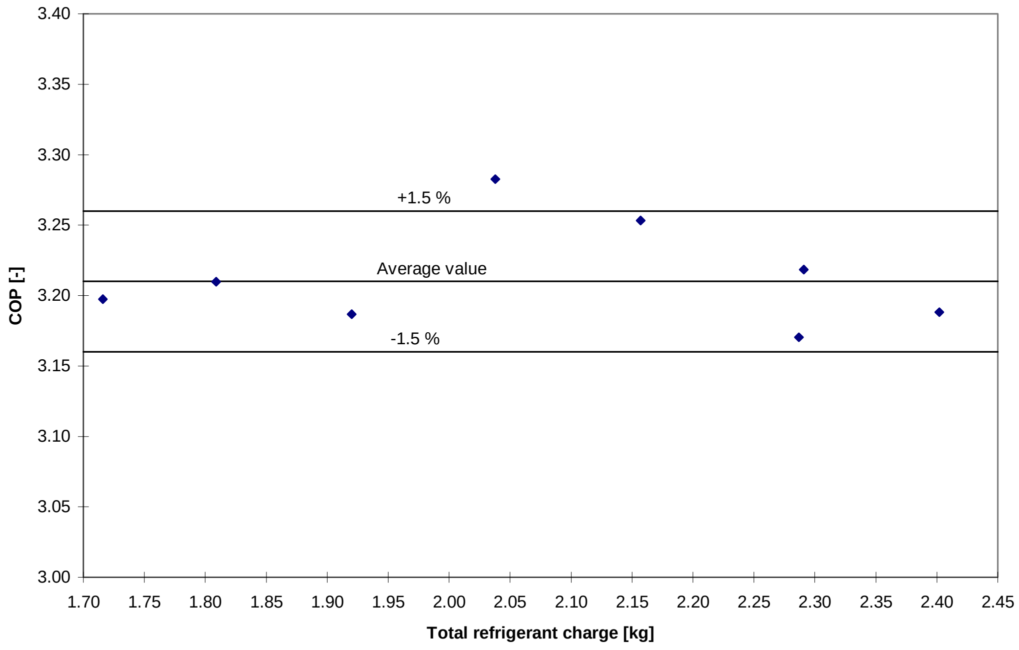

Figure 18 - Variation of the COP with the total refrigerant charge.

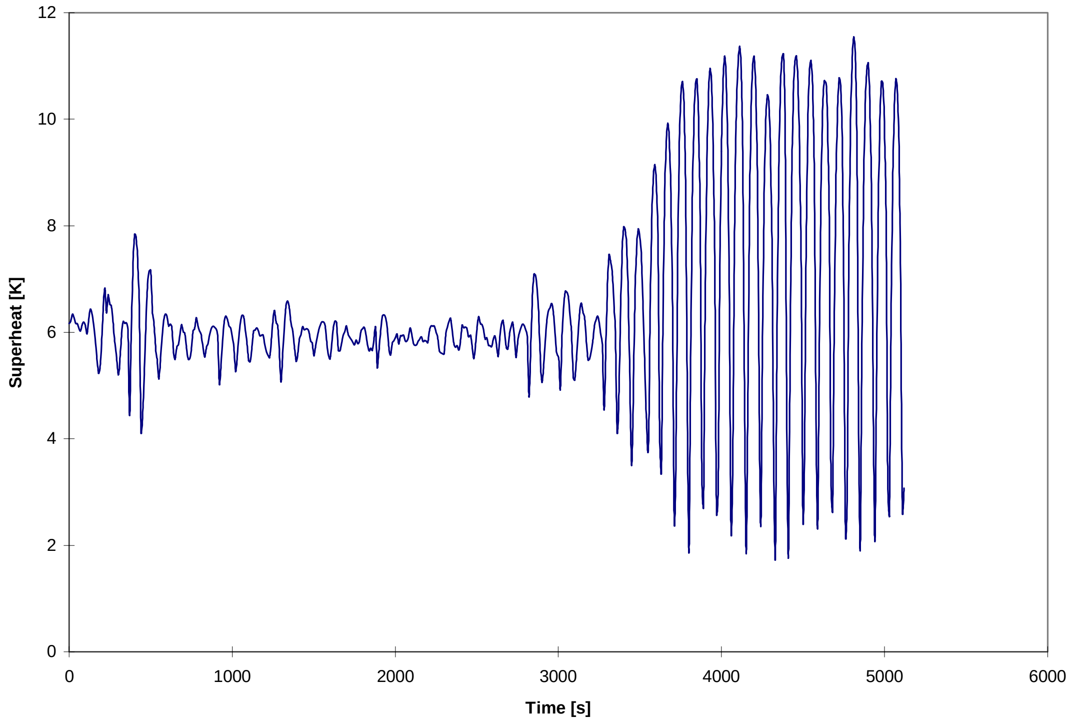

Figure 19 - Variation of the superheat for the first evaporator (charge=2.29 kg).

MODELLING OF THE BISPLIT SYSTEM

Another objective of this work is to develop a general model of a bisplit system. This model is essentially based on the model developed for a basic chiller (one evaporator only).
Indeed, chiller modelling has been widely investigated as part of the MoMo project conducted by CETIAT for several years. MoMo (Modular Modelling) [12] is a modular software using a Windows interface that should help the user to design compression refrigeration units. The complete chiller modelling is based on existing and detailed models of compressors [13] (reciprocating, screw or scroll types), heat exchangers [14,15] (air-cooled as well as water-cooled condensers and direct expansion coils) and expansion devices [16] (thermostatic valves and capillary tubes). Although all the models have a strong physical meaning, each of them requires a series of parameters that must be previously identified on the basis of existing test results.

The finned tube evaporator and thermostatic expansion valve models are considered in order to illustrate the concept of parameter identification.

Evaporator model

The heat transfer for forced convection evaporation in tubes is calculated using the correlation proposed by Pierre (see reference[17]):

$$\mathbf{Nu} = \mathbf{E}\ [\mathrm{Re}_l^{\ 2}\ \mathbf{K}]^{\mathbf{q}} \tag{10}$$

where Re_l is the liquid Reynolds Number;

$\mathbf{K} = \dfrac{\Delta \mathbf{x}\ \mathbf{h_{fg}}}{\mathbf{g\ L}}$;

$\Delta \mathbf{x}$ is the quality change during evaporation;

$\mathbf{h_{fg}}$ is the heat of vaporization evaluated at the evaporating pressure;

$\mathbf{g}$ is the acceleration due to gravity;

$\mathbf{L}$ is the length of the two-phase region.

The coefficients **E** and **q** proposed by Pierre are only valid for particular flow configurations, evaporating temperatures and quality changes. Therefore, these two parameters are identified for each evaporator.

The same procedure is used to model the heat transfer on the air side. A classical method consists in calculating the heat transfer coefficient through the Colburn j-factor; this factor is assumed to depend only on the air Reynolds number:

$$\mathbf{j} = \mathbf{A}\ \mathrm{Re}_{\mathbf{air}}^{\ \mathbf{n}} \tag{11}$$

Once again, the coefficients **A** and **n** are identified for each regime (wet and dry) on the basis of available test results.

Similar identifications are used to model the pressure drops on both sides. More information can be found in [15].

Thermostatic expansion valve

The refrigerant mass flow rate through the valve is calculated as follows:

$$\dot{\mathbf{M}} = \Omega\sqrt{2\,\rho_{\mathbf{l}}\Delta\mathbf{p}_{\mathbf{tev}}} \tag{12}$$

Where

Ω is the port area;

$\rho_{\mathbf{l}}$ is the density of the subcooled liquid;

$\Delta\mathbf{p}_{\mathbf{tev}}$ is the pressure drop through the valve.

The port area can be estimated as a function of the operating conditions by means of a force balance applied to the diaphragm [16]:

$$\Omega = \mathbf{C}_1\,\mathbf{T}_{\mathbf{b}} + \mathbf{C}_2\,\mathbf{T}_{\mathbf{b}}^{\ 2} + \mathbf{C}_3\,\mathbf{p}_{\mathbf{eq}} + \mathbf{C}_4\,\Delta\mathbf{p}_{\mathbf{tev}} + \mathbf{C}_5 \tag{13}$$

where

$\mathbf{T}_{\mathbf{b}}$ is the bulb temperature, expressed in K;

$\mathbf{p}_{\mathbf{eq}}$ is the internal or external equalization pressure;

$\mathbf{C}_1$ to $\mathbf{C}_5$ are coefficients depending on the valve geometry and spring stiffness.

Therefore, the simulation model is based on a previous identification of five parameters (C1 to C5), which is valid for a particular position of the adjusting screw.

All the models have been subjected to numerous validations over the past few years.
The thermodynamic properties of the most widespread refrigerants (R22, R12, R134a, R407c and R404a) are also integrated into the MoMo software so that it becomes very easy to predict the impact of the refrigerant change on the system global performances.

The modelling of a bisplit chiller requires two additional models in comparison with the classical one-evaporator chiller: a second thermostatic expansion valve is coupled to a second evaporator, forming a new evaporating circuit which is connected in parallel to the first one. In a first approach, the hot gas bypass circuit is not considered in the modelling.

The bisplit model must lead to an accurate prediction of the refrigerant mass flow rates through both evaporators; thus, it becomes possible to determine the refrigerating capacities available for a given operating point.
The data required for the simulation of the whole bisplit chiller are as follows:

(i) Parameters associated to the components (compressor, condenser, evaporators and thermostatic expansion valves). This implies preliminary parameter identifications;
(ii) Data associated to the operating point:

- Evaporators: inlet dry bulb temperatures, dew-point temperatures and inlet velocities of air;

- Condenser: water inlet temperature and mass flow rate as well as the subcooling.

It is important to note that imposing the subcooling in a simulation is quite restrictive since this value is usually unknown. This difficulty can be overcome if the refrigerant charge is taken into account in the modelling. This improvement will be introduced in a subsequent version of the model.

Three main steps are considered for the calculation of the refrigerating cycle:

(i) Compressor/condenser subsystem: an iterative calculation (the subcooling predicted with the condenser model must be equal to the imposed value) leads to the determination of the total refrigerant mass flow rate;
(ii) Evaporator/expansion valve subsystems: an iterative calculation (the pressure at the valve outlet must be equal to the pressure at the evaporator inlet) leads to the determination of the refrigerant mass flow rate passing through each evaporator (adiabatic expansions are considered so that no valve model is required at this step);
(iii) Thermostatic expansion valves: the operating conditions obtained in the previous step are used to calculate the refrigerant mass flow rate through each thermostatic valve.

The complete system can be reduced to a set of independent equations that can be solved simultaneously by means of the generalized Newton-Raphson method. This method is well adapted to solve highly non-linear systems (quadratic convergence order).

Although the model of the whole bisplit chiller is already operational, the model validation is focused here on the evaporators/thermostatic expansion valves subsystem.
The subcooled refrigerant state at the valve inlet (pressure, temperature) as well as the total refrigerant mass flow rate must be given in addition to the aforementioned data.
Only the tests from the first series are considered in this validation.

Figure 20 shows the relative errors made on the refrigerating capacity predictions as a function of the actual refrigerating capacities for both evaporators.

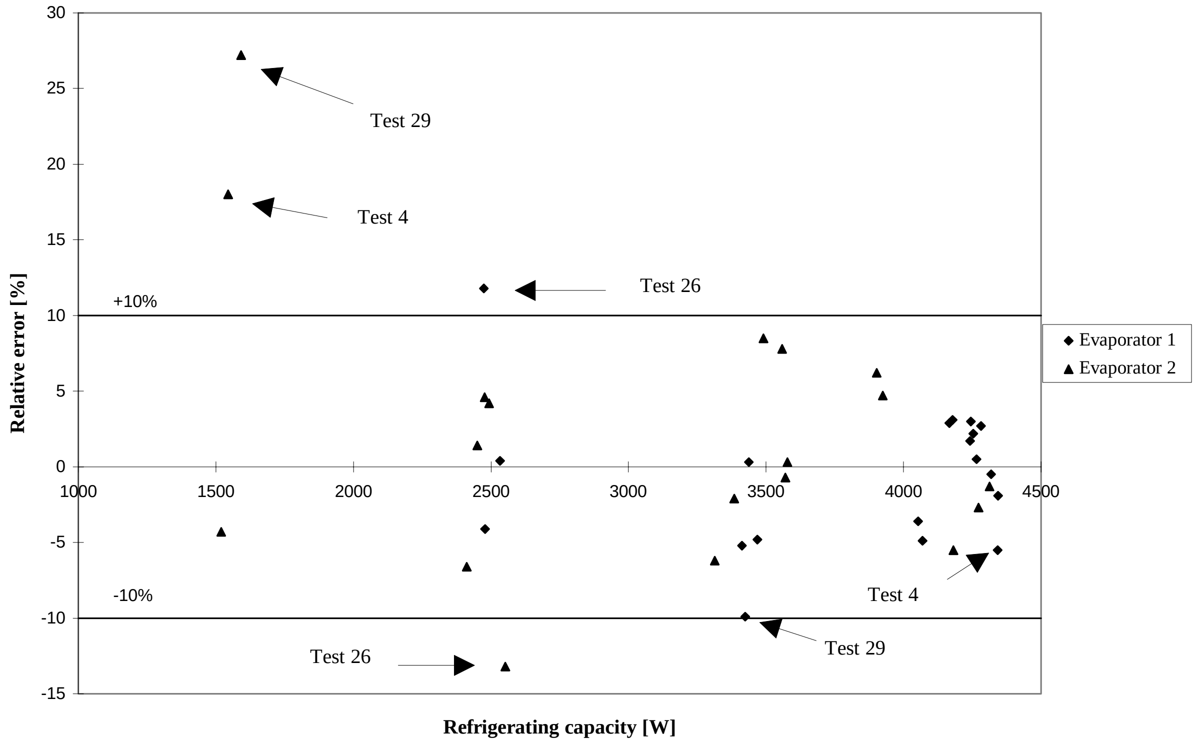

Figure 20 - Relative errors made on the refrigerating capacity predictions

As can be seen, most of the relative errors lie within ± 8.5 %. However, three tests (No. 4, 26 and 29) lead to quite important relative errors. This is probably due to a relatively bad accuracy of the thermostatic valve model for these particular tests. The following relative errors made on the mass flow rate predictions are obtained when using only the valve model (respectively, for valves No. 1 and 2): -7.6 % and 11 % (test No. 4); 31.6 % and -4.6 % (test No. 26); -28.8 % and 6.5 % (test No. 29). For the other tests, the thermostatic valve model predicts the mass flow rate with a mean relative error of about 5 %. On the other hand, the evaporator model predicts the refrigerating capacity with an accuracy lower than 5%. The simulation results can be considered as quite satisfactory.

The predicted superheats are compared with the actual values in Figure 21. The general trend is good although a quite important scattering is observed for some small superheats.

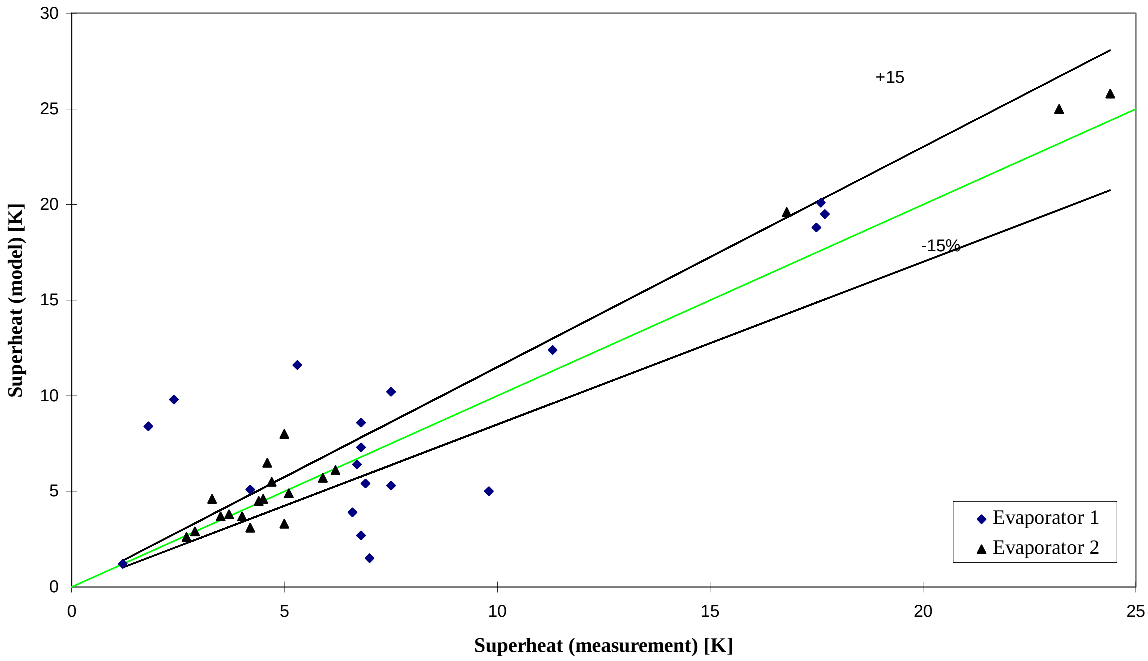

Figure 21 - Measured and simulated superheats.

CONCLUSION AND FUTURE STUDIES

A test bench of a bisplit chiller is operational at the Laboratory of Thermodynamics. The main advantages of this system in comparison with standard one-evaporator chillers are the reduced space requirement and the reduced plant cost; the main drawback is the quite complicated control strategy. Bisplit systems are essentially used in the air conditioning plants.

The particularity of this bisplit system lies in the use of thermostatic expansion valves. Two kinds of oscillations are encountered at the evaporator outlets. The regular periodic oscillations of the refrigerant temperature are due to control « hunting »; the irregular oscillations are probably due to random fluctuations of the liquid dry-out point combined with the evaporator arrangement inside the air loop. However, these oscillations did not affect the accuracy of the energy balances applied to the evaporators.

A series of tests is performed in order to investigate the refrigerant charge effect on the system global performance. This analysis shows that there is always a refrigerant charge leading to optimal system performance. For instance, a 11.2 % undercharge results in a 7.1 % reduction in total refrigerating capacity and almost the same orders of magnitude are obtained for overcharging conditions. Thus, the refrigerant charge proves to be an important parameter when optimizing the bisplit chiller performance.

A model of the whole bisplit chiller has been developed and a first validation leads to quite satisfactory refrigerating capacity predictions. In a near future, the refrigerant charge will also be taken into account in the bisplit chiller model.

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NOMENCLATURE

h	Specific enthalpy	[J kg^{-1}]
p	Absolute pressure	[bar]
t	Temperature	[C]
$\dot{M}$	Mass flow rate	[kg s^{-1}]
$\dot{Q}$	Power (heat flux)	[W]
$\dot{W}$	Power (electrical)	[W]
U	internal energy	[J]
UA	Global heat transfer coefficient	[W K^{-1}]
$c_{(p)}$	Specific heat (at constant pressure)	[J kg^{-1} K^{-1}]
τ	Time	[s]

Subscripts

a	Air
amb	Ambient
aux	Auxiliary heat exchanger
byp	Bypass circuit
c	Cold
c	Calorimeter
cd	Condenser
cp [W]	Compressor
el	Electric
ev	Evaporator
ex	Exhaust
h	Hot
inj	Injection
int	Internal
r	Electric heaters
su	Supply
tev	Thermostatic expansion valve
w	Water

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