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Testing and Modelling of a Bisplit Refrigeration System

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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 behaviour of the two thermostatic expansion valves used is examined. The modelling of a bisplit chiller is also studied and a first model validation is given.

NOMENCLATURE

t	Temperature	[C]	U	internal energy	[J]	el	Electric
h	Specific enthalpy	[J kg^{-1}]	τ	time	[s]	ev	Evaporator
$\dot{M}$	Mass flow rate	[kg s^{-1}]				ex	Exhaust
$\dot{Q}$	Power	[W]	Subscripts			h	Hot
$\dot{W}$	Power	[W]				int	Internal
UA	Global heat transfer coefficient	[W K^{-1}]	amb	Ambient		r	Electric heaters
			c	Cold		su	Supply
c	Specific heat	[J kg^{-1} K^{-1}]	cd	Condenser		W	Water

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 space required 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 twofold. The first aim is to develop a general model of a two-evaporator refrigeration system (« bisplit » system); this model should help the user to design properly a whole bisplit chiller thanks to an appropriate selection of the basic components. A test bench of a bisplit system is operational at the Laboratory of Thermodynamics. A series of about thirty tests has already been carried out; these tests are useful for a first validation of the developed model. The second aim is to observe the behaviour of the two thermostatic expansion valves used in the bisplit system.

DESCRIPTION OF THE TEST BENCH

The bisplit refrigeration system is shown in Figure 1; the refrigerant used is R22.

The main components are as follows:

(i) an hermetic scroll compressor installed inside a calorimeter;
(ii) a shell-and-tube water-cooled condenser followed by a liquid receiver and a subcooler;
(iii) two evaporating circuits connected in parallel. Each circuit is composed of a thermostatic expansion valve (external pressure equalization) and a direct expansion coil;

(iv) a capacity regulator with hot gas bypass;
(v) an auxiliary heat exchanger used to adjust the compressor inlet temperature in order to keep the discharge temperature within an admissible range.

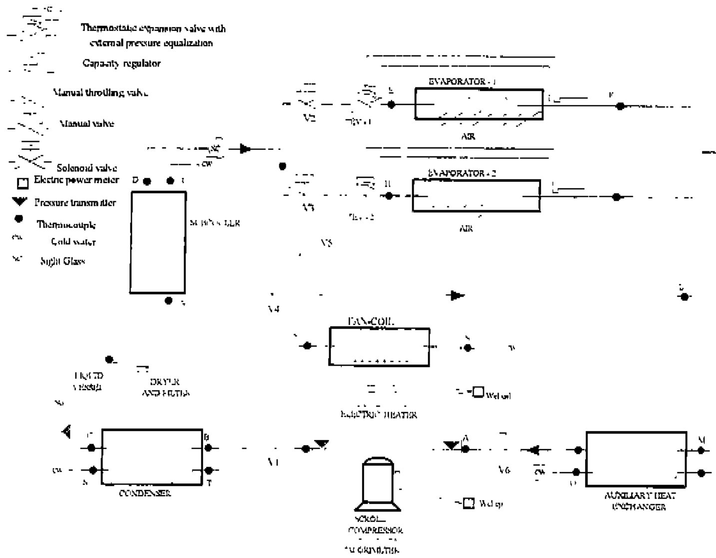

Figure 1 - Test bench of a bisplit refrigeration system

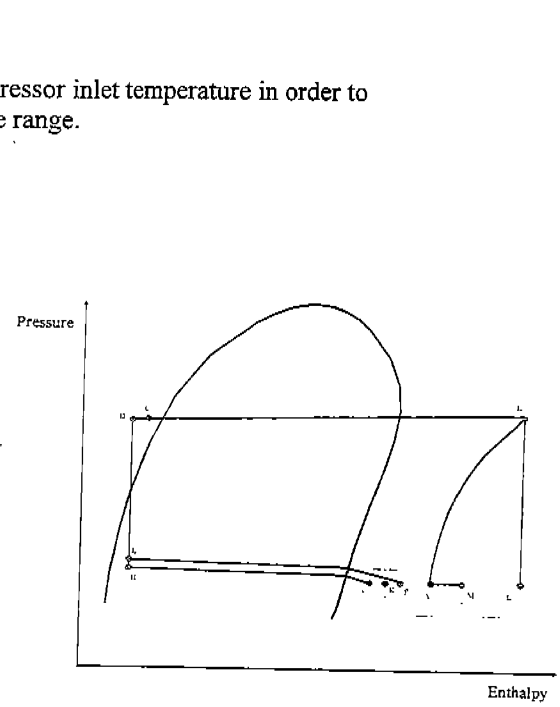

Figure 2 - Thermodynamic cycle

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 the air up before it enters the coil: the setting of these heaters permits 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) before being cooled down in the subcooler. The expansions occur in the two thermostatic valves and two-phase refrigerant is obtained at points E and H (the difference between the two evaporating temperatures is very small: it is only due to the different pressure drops occurring in the evaporators). The refrigerant evaporates as it passes through the coils and the two streams are mixed at the evaporator outlets (points F and I); the resulting stream (point K) is again mixed with the refrigerant coming from the hot gas bypass circuit. 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

A series of thirty one tests were 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 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 condensing temperature is changed by a manual adjustment of the cooling water flow rate passing through the condenser.
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 superheating ($\Delta T_{sh,1}$ and $\Delta T_{sh,2}$) degrees 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).

bounds	$t_{ev,1}$ [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 total refrigerant mass flow rate passing through the evaporating circuits can be estimated in two different ways.
A first estimation is given by the two energy balances of the evaporators. The refrigerating capacity of each evaporator is obtained by means of the balance over the whole air loop (Figure 3a):

$$\dot{Q}_{ev} = \dot{W}_{el,fan} + \dot{W}_{el,r} - \dot{Q}_{amb,h} - \dot{Q}_{amb,c} - \frac{dU}{d\tau} \tag{1}$$

where $\dot{Q}_{amb,h} = (UA)_h \cdot (t_{int,h} - t_{amb})$ and $\dot{Q}_{amb,c} = (UA)_c \cdot (t_{int,c} - t_{amb})$ represent the ambient losses of the hot and cold parts of the air loop; $t_{int,h}$ and $t_{int,c}$ are the air average temperatures inside the hot and cold parts of the air loop; $\frac{dU}{d\tau}$ is the rate of change of internal energy.
It is necessary to avoid condensation inside the evaporators otherwise an unknown quantity of condensate should be taken into account into equation (1). The purpose of the capacity regulator is to control the evaporating pressure in order to avoid condensation.
The refrigerant mass flow rate through each evaporator is given by the following relationship:

$$\dot{M}_{ev,i} = \frac{\dot{Q}_{ev,i}}{h_{ex,ev,i} - h_{su,ev,i}} \tag{2}$$

where the subscript ***i*** refers to the first or to the second evaporator.
The total flow rate is the sum of the two previous values.

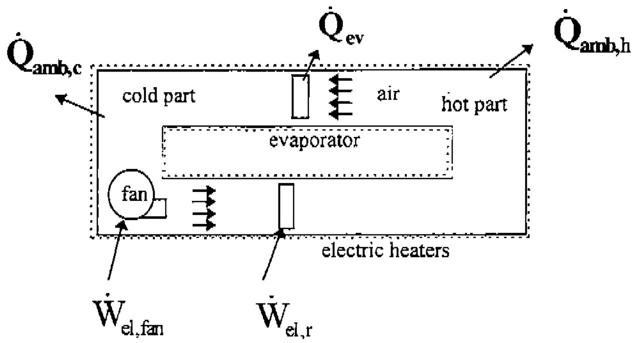

Figure 3a - Control volume of the air loop

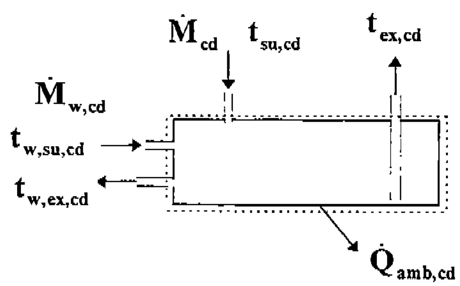

Figure 3b - Control volume of the condenser

A second estimation is given by means of the energy balance applied to the condenser (Figure 3b):

$$\dot{M}_{cd} = \frac{\dot{M}_{W,cd}\; c_w \left(t_{W,su,cd} - t_{W,ex,cd}\right) + \dot{Q}_{amb,cd} - \frac{dU}{d\tau}}{h_{ex,cd} - h_{su,cd}} \tag{3}$$

The difference between the two estimations is always below 3 %.

ANALYSIS OF THE TEST RESULTS

Figure 4 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.
As can be seen, two kinds of oscillations are encountered at the evaporator outlets.

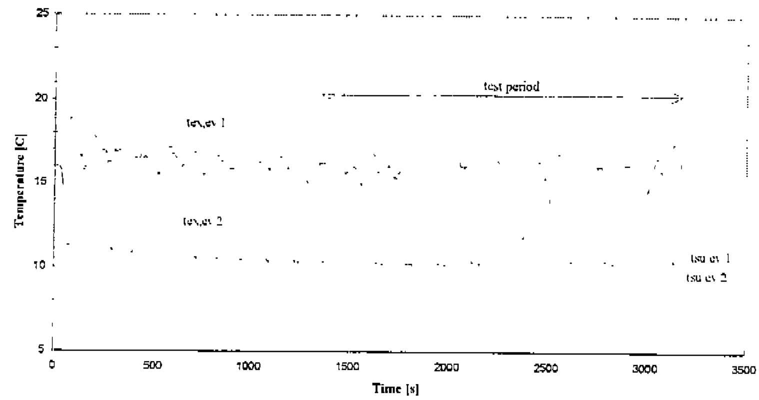

Figure 4 - Variations in the refrigerant temperatures at the evaporator inlets and outlets

The regular periodic oscillations observed for the second evaporator are due to control « hunting » [1,2]: 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 [3].

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) [4,5,6] 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, 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.

MODELLING OF THE BISPLIT SYSTEM

The modelling of a bisplit chiller is essentially based on that of a simple 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) [7] 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 [8] (reciprocating, screw or scroll types), heat exchangers [9] (air-cooled condensers and direct expansion coils) and expansion devices [10] (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. For instance, twelve parameters are needed in the coil model [9]; six parameters are introduced in the laws giving the heat transfer coefficients of refrigerant (during two-phase evaporating flow) and air (for both dry and wet regimes) while six other parameters characterized the laws giving the total pressure drops for both fluids. In the same way, five parameters are introduced into the law giving the port area of the thermostatic expansion valve as a function of bulb temperature, equalization pressure and pressure drop across the valve [10].

All the models are very reliable since they 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. For this first approach, the hot gas bypass circuit is not considered in the modelling.
The bisplit model must allow an accurate prediction of the refrigerant mass flow rates through both evaporating circuits; thus, it become 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 given operating point:
  - Evaporators: inlet dry bulb temperatures, dew-point temperatures and inlet velocities of air;
  - Condenser: inlet temperature and mass flow rate of the heating fluid (air or water) as well as the subcooling degree.

Once these data are imposed, the values of four independant variables are required in order to calculate the whole refrigerating cycle. These four independant variables are: the pressure and enthalpy of the refrigerant at the compressor inlet and the refrigerant enthalpies at the evaporator outlets.The model should be able to deal with superheated refrigerant as well as two-phase refrigerant at the evaporator outlets; that is why enthalpies are chosen as independant variables since they remain continuous functions when passing through the saturation curve.
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 that at the evaporator inlet) leads to the determination of the refrigerant mass flow rate passing through each evaporator (simple isenthalpic expansions are considered for these subsystems 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.

Four independant equations must be verified. Two equations stand for mass and energy conservations during the mixing of the two refrigerant streams at the evaporator outlets; the two other equations stand for mass conservation inside each evaporating circuit (the mass flow rates passing through the evaporator and the thermostatic valve must be equal).
The generalized Newton-Raphson method (with four independant variables) is used in order to solve simultaneously this set of equations. This method is well adapted to solve highly non-linear systems; moreover it has a quadratic convergence order.

Although the model of the whole bisplit chiller is already operational, the model validation is focused here on the subsystem constituted of the two evaporating circuits in parallel from each other.
The data required for the subsystem simulation are as follows:

(i) Parameters associated to the evaporators and thermostatic valves (this implies preliminary parameter identifications);
(ii) Data associated to the given operating point:
  - inlet temperature, dew-point temperature and inlet velocity of air for both evaporators;
  - pressure, temperature and quality of refrigerant at valve inlet;
  - total refrigerant mass flow rate passing through the evaporating circuits.

A resulting drawback of only considering this subsystem is that the validation is somewhat artificial since the total mass flow rate is imposed.
Figure 5 shows the relative errors made on the refrigerating capacity predictions as a function of the actual refrigerating capacities for both evaporators. As can be seen, most of the relative errors lie within $\pm$ 8.5 %. However, three tests (No. 4, 26 and 29) lead to quite important relative errors. This is probably due to the relatively bad identification of the thermostatic valve parameters for these particular tests. When using the valve model with the identified parameters, the following relative errors made on the mass flow rate predictions are obtained (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 %. Thus, the simulation results obtained can be considered as quite satisfactory.

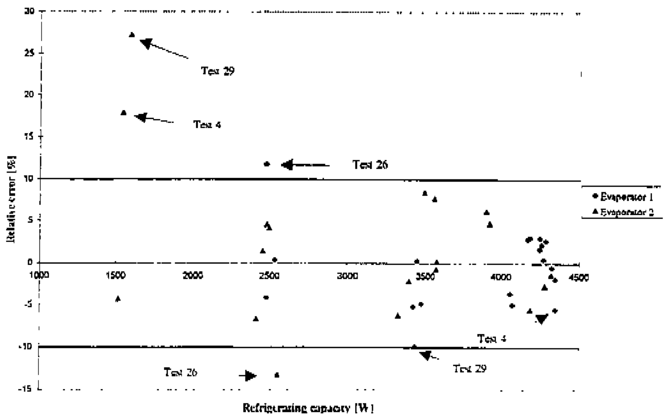

Figure 5 - Relative errors made on the refrigerating capacity predictions

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 outlet. 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 model of the whole bisplit chiller has been developed and a first validation leads to quite satisfactory refrigerating capacity predictions.

In a near future, new tests with varying refrigerant charges will be carried out in order to determine the charge effect on the system global performance. The refrigerant charge will also be taken into account in the bisplit chiller model.

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