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A NEW INSTALLATION FOR PART LOAD TESTING OF AIR TO WATER SINGLE STAGE CHILLERS AND HEAT PUMPS

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ABSTRACT

Nowadays in Europe, air to water heat pump COP and EER, both in the cooling and heating mode, are rated at full load in steady state conditions. In real operating conditions, the water temperature varies between a low and a high thresholds used by the control system of the heat pump. In order to evaluate COP and performances in dynamic conditions for partial loads, a new facility has been set up at the Climatron (EDF R&D, Thermal Systems Group) that enables to supply variable load.
The experimental test bench is made of the heat pump under test, a heating or cooling system providing heat or coolness to a water loop. The water loop design permits to change the inertia by changing the volume of water stored in water tanks. The heat pump under test is installed in a climatic chamber permitting to simulate outside climatic conditions (temperature and humidity).
The Climatron test facility enables to vary all the parameters that influence the energy efficiency of the air to water reversible heat pump: the outside air temperature and humidity (heating mode), the water temperature set point and the control dead band.
Tests performed show the inefficiencies at partial load linked to the usual control system of air-to-water heat pump. Results are reported in the cooling mode and the origin of inefficiencies are analyzed.

NOMENCLATURE

CC : coolig capacity (kW)
Cp : calorific capacity of water kJ kg^{-1}
DB : symmetrical dead-band that frames the temperature set point ()
EP : electric power (fan and compressor) (W)
Tie : inlet water temperature at the evaporator of the chiller (°C)
Toe : outlet water temperature at the evaporator of the chiller (°C)

1. INTRODUCTION

In Europe, nowadays, vapor compression cycles are tested in stationary conditions (CEN, 1998), at full load. Manufacturers may give, within the equipment catalog, performances at full load for varied sources temperatures to extend the characterization range; part load efficiency is never considered.
On the contrary, part load characterization is required in two American Standards dealing respectively with air-to-air conditioners, (ANSI/ASHRAE 116,1995), and capacity staged chillers (ARI, 1998). In both standards, when load is inferior to the smallest capacity step available, a simple law translates the degradation of efficiency with the cycling phenomenon. Efficiency, for given sources temperatures, is supposed to decrease linearly with load. This trend translates the cycling efficiency drop linked to on-off control. When the vapor compression cycle is switched on, capacity does not reach instantaneously its nominal capacity. This capacity loss is supposed to decrease the cycle efficiency. (O'neal, 1991) showed this behavioural law could be described, depicting the capacity growth by an

exponential model with a single constant time and the variation of the cycle times with load by an ideal thermostat. A priori, two reasons may lead to suppose that the efficiency law used for air-to-air units is not valid for air-to-water units :

- for chillers, the capacity growth time constant should be lower than for air-to-air air conditioners given that heat exchange coefficients are higher at the evaporator (water versus air),
- for air-to-air conditioners, when the compressor is switched off, fans are too ; for chillers and air-to-water heat pumps, the water pump is still on when the compressor is switched off. This means that a part of "free" cooling capacity can be recovered when the chiller is stopped (Parken, 1977).

So as to investigate the efficiency variations at part load, given the evident lack of data for dynamic behaviour of chillers and heat pumps, a test bench has been built up.

2. TEST BENCH DESCRIPTION

2.1 Test cells

The test bench has been designed to simulate variations of thermal load and water temperature. It is based on climate cells already existing, ambient temperature can be controlled over a wide temperature range from –15°C to 50°C and specific humidity can be maintained at high levels to perform heat pump tests under frosting conditions for typical European winter climates.

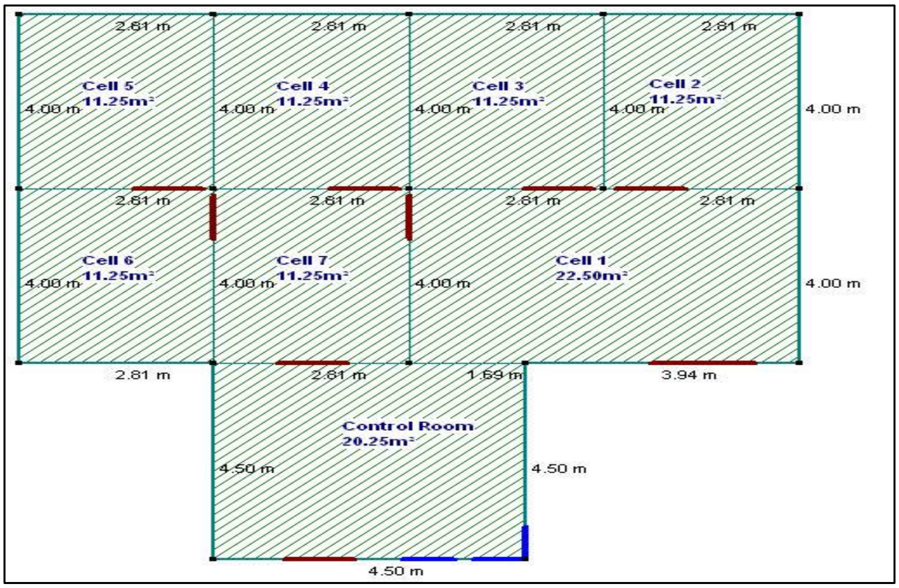

Figure 1: Global view of the test cells and control panel room of the CLIMATRON laboratory / EDF

The "Enceinte n°1" of the laboratory « Climatron » is composed of seven climate cells whose areas are 11,25m² (6 cells) or 22,25m² (1 cell).

A test cell is a room made up with :

- heat insulated partition walls and doors,
- an inclined ground for condensate evacuation,
- two ceiling ventilators for air supply and return,
- one heat exchanger on the supply air path, to cool or heat the air supplied centrally by the control system.

The air entering a specific cell pass trough an air to glycol water heat exchanger. The temperature of the glycol water in the heat exchanger is controlled by a thermal resistance for heating and a glycol water to glycol water heat exchanger that enables to cool the loop. The external inlet temperature of this heat exchanger is controlled by a three-way valve that enables to decrease the inlet glycol water temperature until –25 °C (water glycol temperature supplied by the central chilling plant of the laboratory).

The test cell is instrumented as follows:

- a temperature sensor, Pt 100 type, for the measurement of the air temperature in the test cell,
- a hygrometer.

Both measurement devices are installed at the inlet of the air heat exchanger of the heat pump, and are dedicated to the control of temperature and hygrometry of the test cell. For steady state testing purposes, only one cell is used to test a chiller or a heat pump.
Control parameters (PID parameters for example) must be adapted for each specific set of air temperatures and hygrometry to ensure stabilised conditions. PID parameters for temperature control highly affect the stabilisation time of the system and are not reproducible from one cell to another or from one temperature condition to another: Parameters for integration and derivation periods must be modified for each testing point.
The stabilisation of parameters was a difficult problem to solve for this specific test campaign. Given that the goal is to study a cycling phenomena, two states occur : the « on » mode and the « off » mode. In the "on" mode, the heat pump fan is blowing. But in the off mode, the fan is stopped. The ratio between the mass air flow extracted by the control system and the heat pump is not high enough to ensure the steadiness of the inlet air temperature in dynamic conditions ; important disturbances occur. Moreover, one single series of PID parameters leads to important stabilisation time when shifting from one mode to another.

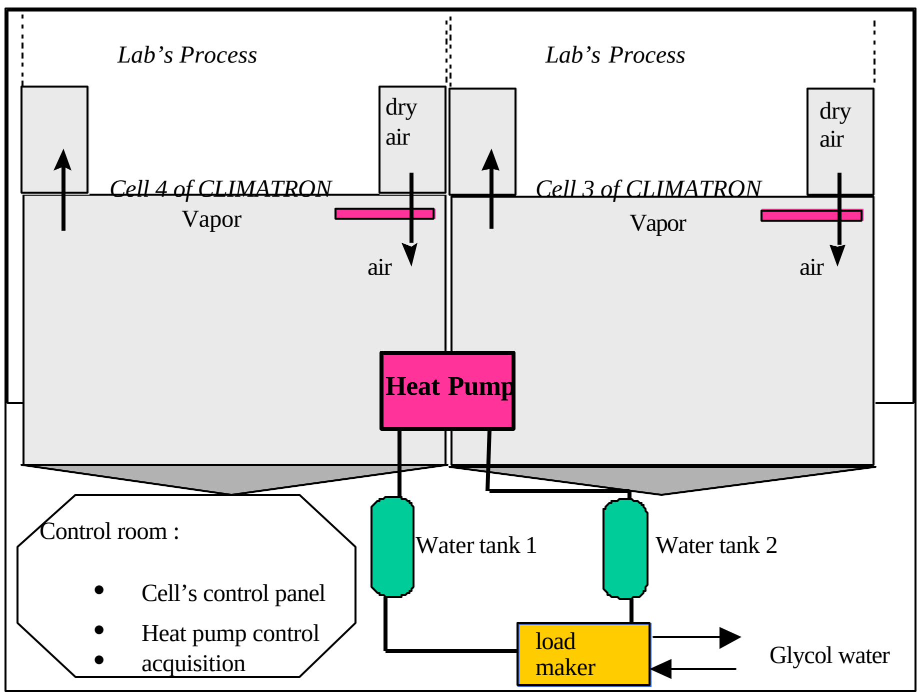

Figure 2: Scheme of the complete installation

To solve this problem two cells have been used. A rectangular aperture has been made between cells number three and four (Figure 2). The heat pump takes the air from the cell 4 and blows it into the cell 3. It avoids the air path disturbances when the heat pump is on, and enables to supply constant air temperature at the inlet of the heat pump air heat exchanger.
With the aperture solution, another problem occurs : the air flow rate of the fan heat pump (from cell 4 to cell 3) is of the same order of magnitude than the air flow rate supplied by the process to the cell 4. Consequently, the heat pump standard air flow rate cannot be supplied and efficiency drops. That is the reason why temperatures of cells 1 and 7 are also controlled and doors opened between cells 4 and 7, and 1 and 7.

2.2 Supplying a constant "compensation load" on the water loop

The heating/cooling floor system of a house (typical application range for the heat pump being tested) is simulated by a closed water loop which includes the plate heat exchangers of the heat pump and of the load control system (the heat rejected by the pump on the water loop is part of the load created on the water loop for both modes). Heat is supplied directly on the water loop by a resistor. The system designed to supply coolness on the water loop is very similar to the one used by the cell air treatment. Coolness is supplied by the water to glycol water heat exchanger (Figure 3) whose glycol water inlet temperature is controlled by a three-way valve. The chilled glycol water is generated by the process of the laboratory.

During the partial load tests, the outlet water temperature of the load maker is controlled so that the thermal capacity of the system (Figure 3) remain constant and equal the required capacity set point. To that extent, a calculator has been adapted. From the measurement of the inlet and outlet temperatures of the Vulcatherm (the physical load maker), and of the water flow rate, by an ultrasound flow-meter, the calculator instantaneously (each second) calculates the capacity of the compensation load. The control of the Vulcatherm (normally designed to supply an instantaneous constant leaving temperature) has been shunted. The capacity value measured is sent to the controller which commands the three-way valve and the resistor. Each 5 seconds, the controller adjusts the 3 way valve position and the resistor power in order to fit the capacity measured and the capacity set point.

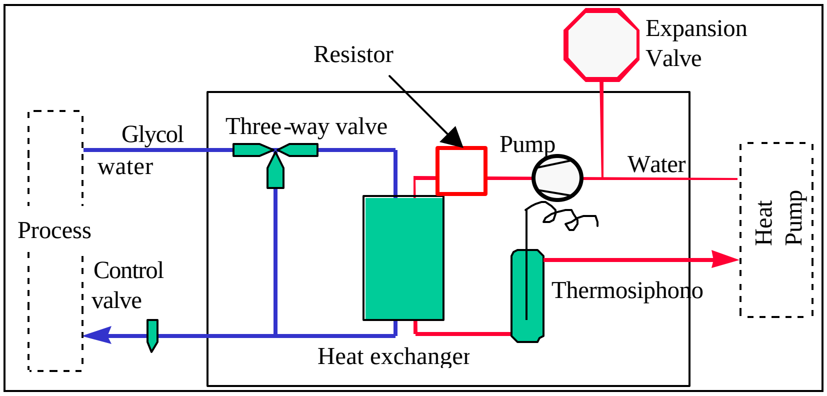

Figure 3: View of the thermal load maker (or Vulcatherm)

The controller uses a classical PID scheme whose parameters have to be set. The controller activates the coolness source and the heat source with the same command law. But results of the same command are quite different depending whether heat or cold is required. Moreover, PID control should be adapted whether the chiller is on or off. It has been found that the best compromise was to keep a rudimentary proportional action law. The system reacts fastly ; stable operations are ensured for both on and off periods.

2.3 Water loop volume variation

The stability varies also according to load and inertia conditions. Three different water tank configurations are tested to simulate water loop inertia variations (Figure 4) :

- the 75 l water tank alone, it represents about 11 l/kW (calculated in reference to the cooling capacity for the stationary conditions, Tic = 30 °C, Tie = 13.5°C) water inertia, water temperatures can be stabilized for all load simulated,
- the 150 l water tank alone, it represents about 21 l/kW water inertia, some stability problems occur for higher thermal loads simulated,
- both water tanks, it represents about 31 l/kW water inertia, water temperatures can be stabilized for all load simulated.

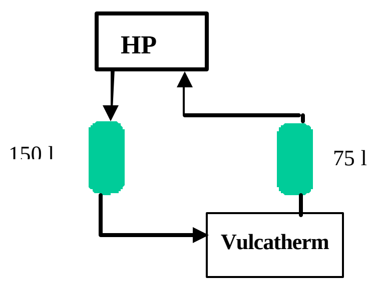

Figure 4: Water circuiting

The 75 l water tank is a common domestic hot water tank. To ensure stratification in the heating mode for hot water supply, hot water is added by below, enters directly the volume and leaves at the upper level. For normal operation, entering water temperature is lower than the temperature inside the tank which is heated by an internal resistor. When the chiller is on, colder water enters the water tank. Thus, stratification appears resulting in temperature inertia. On the contrary, when the chiller is off, hotter water enters by below and mixing occurs.

For the 150 l water tank, water enters by above and leaves by below. When the chiller is on, water temperature is homogeneous. When the chiller is off, hotter water is added by above thus stratification appears.

2.4 About the tested heat pump

The heating capacity of the reversible HP selected for the tests is of 7.9 kW for an outdoor temperature of 7°C and 9.3 kW when operating in cooling mode (conditions T1 of ISO 5151), and is equipped with a scroll compressor. The working fluid is R-407C.
Air measurement apparatus are:

- temperatures at the inlet and outlet of the air coil measured by 6 Pt 100 (4 at the inlet and 2 at the outlet),
- the air humidity is measured at the air coil inlet by a sensor for dew temperature (which is verified by a capacitive sensor indicating the relative humidity),
- air coil pressure losses are measured by a differential pressure sensor; measurement points are installed on the coil width.

The HP electric input power (fan motor + compressor) is measured with a precision of 0.5%.

3. EXPERIMENT RESULTS

The test bench is used here in the cooling mode only. Part load results in the heating mode are reported by (Flach-Malaspina, 2004). The set of experiments presented has been led with constant inlet air temperature of 30 °C at the condenser. The chilled water set point temperature is also maintained constant at 13.5 °C ; load and inertia are varied. The air flow rate of the heat pump is set constant, the head pressure control being shunted.

3.1 Full load reference

So as to establish the reference for part load testing results and be able to compare full load and part load performances, testing is first performed within the stationary conditions required by the European standard (CEN, 1998) at full load. A one hour period with stability criteria is first required before recording the performances for another hour. Average efficiency is calculated according to equation (1).

$$\mathrm{EER} = \frac{\sum_t \mathrm{CC(t)}}{\sum_t \mathrm{EP(t)}} \tag{1}$$

Cooling capacity and electric power (fan and compressor) are fitted versus water inlet temperature in °C. Regression coefficients are superior to 0,999 in both cases (inlet water temperature was successively set at about 10, 15, 20 and 25 °C). Regression results are presented in Table 1.

Regression coefficients Y = A Tie + B		
	CC	**EP**
A	0.21210	0.06120
B	4.40983	2.17510

Table 1: Full load stationary performances for varied inlet chilled water temperature

3.2 Dynamic testing results

The acquisition sampling rate is fixed at 5 s. The average efficiency is also calculated using equation (1) ; however, the integration time must equal a complete cycle period. The set point control (the inlet water temperature is controlled) is set at 13.5 °C for all experiments. The dead-band (DB) is either 1 K or 3 K. The test bench enables to properly set constant the load at the evaporator. Repeatable cycles are obtained as appears on Figure 5. Temperatures are not maintained within the dead-band because of inertia. When the chiller starts, the inlet water temperature still increases. The same phenomena can be observed when the chiller stops. Varying the water volume inertia, for instance a real circuit would be made of more piping and less water storage, modifies the water temperature overtaking : the more piping the more important the overtaking. Moreover, overtaking also varies with load since load determines the degree of stratification of each water storage, (whenever stratified) (Figure 5 and 6).
The comparison of Figures 5 and 6 also shows that the average inlet and outlet temperatures on a cycle are higher when load raises. It is to be linked to variation of both overtaking primarily but also to the control used by the heat pump. A thermocouple is used to measure the inlet water temperature at the evaporator. At very low loads and low

inertia, water temperature evolves fastly when the chiller is on (Figure 5). Thus, compressor and fan are stopped about 0.5 °C lower than the parameters require.

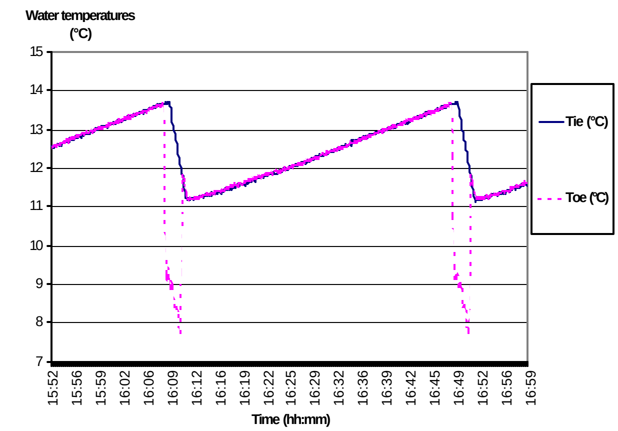

Figure 5: Inlet and outlet evaporator temperatures, 11 l/kW inertia, dead-band 1 K, thermal load 0.4 kW

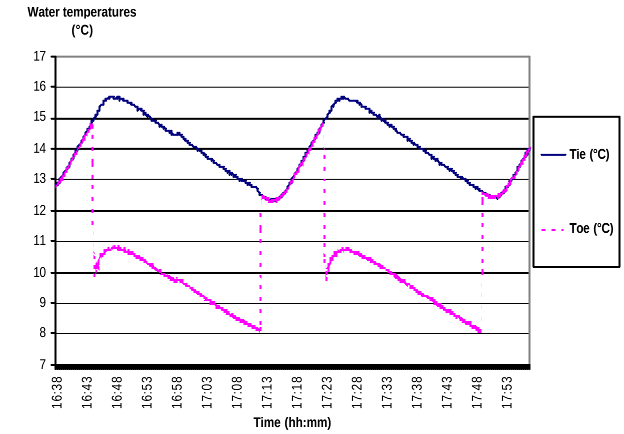

Figure 6: Inlet and outlet evaporator temperatures, 31 l/kW inertia, dead-band 3 K, thermal load 5.4 kW

3.3 Part load efficiency

Six different series have been led for various inertia and dead-band :

- 11 l/kW, dead-band 1 K
- 11 l/kW, dead-band 3 K
- 21 l/kW, dead-band 1 K
- 21 l/kW, dead-band 3 K
- 31 l/kW, dead-band 1 K
- 31 l/kW, dead-band 3 K

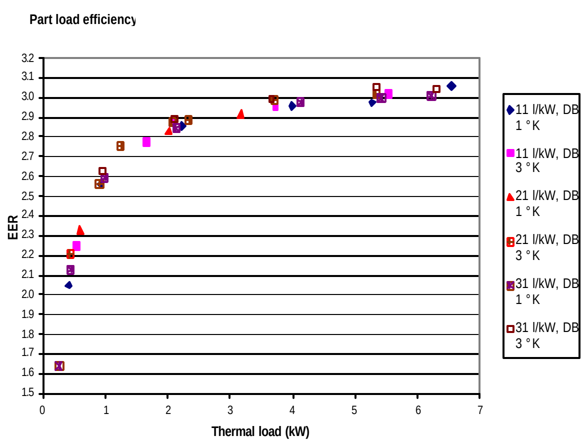

Figure 7: Part load efficiency for various inertia and DB

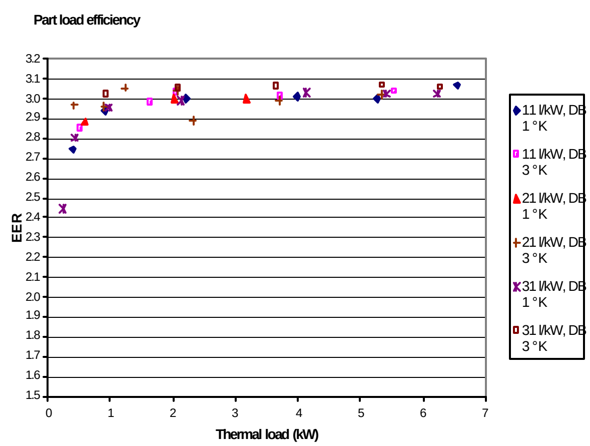

Figure 8: Part load efficiency without sleep power consumption, for various inertia and DB

Results are presented on Figure 7: for all inertia conditions, efficiency decreases with load. The curve shape is similar to what had been observed by (Henderson, 2000) and (Anglesio, 2001): efficiency tends to zero when load decreases. For 0.2 kW load, efficiency drop is about 50 %. The effects of the variation of inertia and dead-band are less visible. However, even if the order of magnitude is lower, it appears that the lower the inertia and dead-band, the lower the efficiencies for all loads.

4. ANALYSIS OF PART LOAD INEFFICIENCIES

The impact of the sleep power consumption appears to be the main efficiency loss cause for the heat pump tested. On Figure 8, resultant efficiencies, calculated without taking into account the sleep power consumption, decrease far more slowly with load. The sleep power consumption is only 54 W, it is to say about 2.5 % of the nominal total electric power (compressor and fan) figured Table 1. However, the off period are longer and longer when load decreases while the on period decrease rapidly.
As had been mentioned previously, results have been difficult to obtain for series 21 l/kW because of stability problems. It explains the greater dispersion around the general tendency for the series [21 l/kW, dead-band 3 K] (Figure 8).
Starting capacity time constants are lower than for air-to-air conditioners (Henderson, 2000), in step with what had been guessed. They range between about 10 and 20 s (Figure 10). For the cooling capacity, the stopping time constants are approximately half of the starting ones, between 5 and 10 s (it is equivalent to say that half the energy needed to establish the R-407C pressures is recovered when the heat pump is stopped). Moreover, electric power does not rise instantaneously neither. Time constants should also be considered. These facts explain that the cycling degradation apart from sleep power losses is very low.
Nevertheless, the general shape tendency without taking into account the sleep power consumption (Figure 8) also exhibits an efficiency degradation when load decreases. Two reasons enable to explain that degradation:

- the evolution with load of both high and low overtaking,
- the evolution with load of the dynamics of the cooling capacity when the chiller is starting.

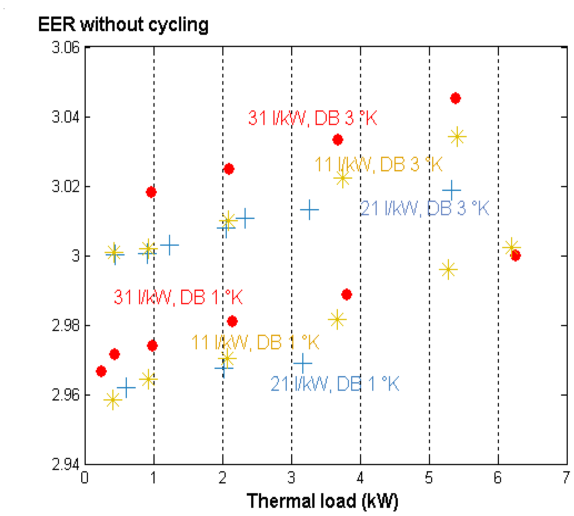

Figure 9: Steady-state performance for average dynamic water inlet temperature at the chiller evaporator

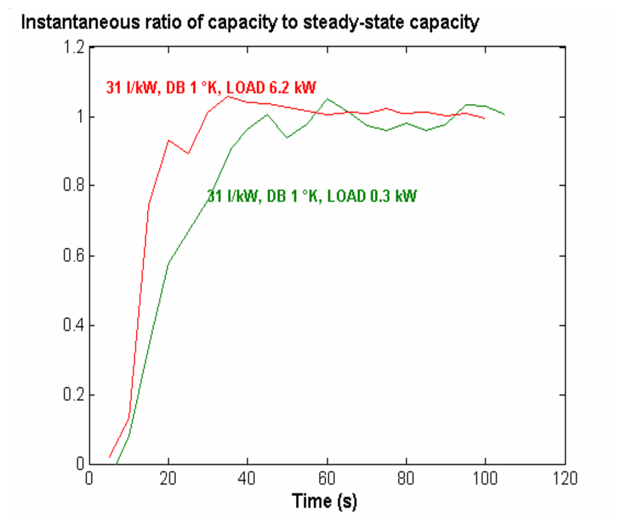

Figure 10: Ratio of the cooling capacity to the steady-state capacity at starting for the series [31 l/kW, DB 1 K]

On Figure 9, the efficiencies calculated according to Table 1 results (interpolation of steady state efficiencies while varying the inlet water temperature at the chiller evaporator) for the average inlet water temperature of each test are plotted versus the thermal load for the 6 series. Both the linear decrease of efficiency with load and the relative positions of the series of the Figure 8 can be explained.
It appears on Figure 8 that for very low loads, despite the fact the electric sleep power has been removed, sharp efficiency degradation remains. Explanation is again to be searched in the dynamics of the water temperature at the inlet of the evaporator of the chiller. At very low loads, the overtaking at starting is brief (Figure 5) whereas it is longer at higher loads (Figure 6). For the starting period, the water temperature inlet is lower than it is at higher loads. It results in lower capacity and thus in increased cycling degradation. This phenomenon is represented on Figure 10: the ratio of the instantaneous capacity to the steady-state capacity calculated for the instantaneous measured capacity is plotted. Higher loads correspond to higher capacity when starting. In the (O'neal, 1991) modelling, it would correspond to a rapid increase of the time constant of the capacity at starting.

6. CONCLUSIONS

A new installation to perform part load tests in dynamic conditions has been designed. An air-to-water heat pump has been tested in the cooling mode for various loop inertia and temperature control dead-band. The efficiency at part load sharply decreases with load. The inefficiency of the on-off control is mainly due to the electric sleep power during off cycle periods, even if it is only around 2 % of the full load nominal electric power of the compressor. This analysis extends the results previously published by (Henderson, 2000) and (Anglesio, 2001) for air-to-air conditioners. Moreover, it has been shown that the cycling inefficiencies linked to on-off starting losses are far lower for the single stage air-to-water heat pump tested.

With the configuration described here, the experimental loop is limited to single stage air-to-water reversible heat pumps up to 10 kW. But the loop was also mounted for capacity staged air cooled chillers with nominal cooling capacities ranging from 50 kW up to 150 kW. For capacity staged equipment, more information can be gained since load control is more complicated. In case part load stages efficiencies differ from full load efficiencies (typical air-to-water tandem scroll chiller with a single refrigerant circuit), the dynamic testing not only measures cycling inefficiencies but also the possible bad use of the available stages. It has been shown that all chillers were not able to use correctly the capacity stages available (Rivière, 2002). In the case mentioned, the unit was only able to cycle from full load and not to operate on the reduced capacity stages, about 20 % more efficient than the full load stage.

Part load testing enables to approach the in-situ behaviour. As opposed to nowadays standards whose testing methods are stationary, it enables to consider also:

- control capability (is my chiller able to supply chilled water at 7 °C ?)
- control "intelligence" (does my chiller knows it owns capacity stages more efficient than others, does he use that information ?)
- dynamic losses including cycling on-off and parasitic sleep power efficiency impact.

REFERENCES

Anglesio, P., Caon, S., Caruso S., 2001, Determinazione delle prestazioni energetiche di condizionatori elettrici a due unità in aria invertible : determinazione delle prestazioni energetiche, *CDA*, n° 02/2001 : p 24-35.

ANSI/ASHRAE 116,1995, Methods for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps.

ARI (Air-Conditioning and Refrigeration Institute), 1998, Standard 550/590, Water Chilling Packages using the vapor compression cycle.

CEN (European Committee for Standardization), 1998, « Air conditioners and heat pumps with electrically driven compressors – Cooling mode" , Cen / *Cenelec Standard*, EN 12055.

Flach-Malaspina, N., Lebreton, J.M., Clodic, D., 2004, Performances of a new air-to-water system with controlled heat capacity, *International Refrigeration and Air Conditioning Conference at Purdue*, R085, July 12-15.

Henderson, H.I., Parker, D., Huang Yu, J., 2000, Improving DOE-2' RESYS routine: User Defined Functions to Provide More Accurate Part Load Energy Use and Humidity and Humidity Predictions, http://gundog.lbl.gov/.

O'Neal, D.L., Katipamula, S., 1991, Performance degradation during on-off cycling of single-speed air conditioners and heat pumps: model development and analysis', *ASHRAE Transactions*, part B, p. 316-323

Parken, W.H., Beausoleil, R.W., Kelly, G.E., 1977, Factors affecting the performance of a residential air-to-air heat pump, *ASHRAE Transactions*, Vol. 83, Part 1.

Rivière, P., Adnot, J., Becirspahic, S., Marchio, D., Colomines, F., Seasonal efficiency of primary air conditioning systems, *Proceedings of the AFCE conference*, Ecole des Mines de Paris, Sept 2002.