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PERFORMANCES OF A NEW AIR-TO-WATER HEAT PUMP SYSTEM WITH CONTROLLED CAPACITY

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ABSTRACT

Current European air-to-water heat pumps for residential heating are designed with a single compressor and a simple on-off system in order to control the heating capacity. Using the same heat exchangers a new heat pump system equipped with 2 compressors of different capacities has been designed in order to reach better heating capacity and higher energy efficiency.
Tests have been carried out in a dedicated test room in order to compare the energy consumption of a typical European heat pump system (baseline) and the new one with 2 compressors.
In order to analyze independently the gains related to the compressor technology, and to the control system, a first series of tests has permitted to measure volumetric, isentropic and global efficiencies of the 3 compressors installed in the 2 heat pumps under comparison. Then global tests on the two heat pumps have been performed in order to measure the energy consumption for different heat loads permitting to define the seasonal efficiency during the heating period.
This paper presents a technical solution permitting to reach better energy performances. Energy gains are evaluated by tests permitting to calculate seasonal energy performances.

1. INTRODUCTION

Variable capacity is one of the solutions to improve energy performances of thermodynamic systems. Variable capacity permits to adapt the heat pump operation to the nearest of the building heating needs and to reduce energy consumptions. Different technical options are available: some are mechanical (the simplest being to use several compressors in parallel), but variable speed operated by the frequency variation of the electric motor constitutes the reference.

Variable speed is extensively used in Japan where Japanese manufacturers have a significant technological advance. In Japan only air-to-air equipment use variable speed. However, in European residential and non-residential buildings, Heat Pumps (HP) are mostly air-to-water systems and operate with fixed speed.

2. TEST CELL AND MEASUREMENT DEVICES

Because of the lack of data on performances of air-to-water HPs operating at partial load, a measurement campaign has been performed. A typical air-to-water HP with low specified capacity, currently available in France for floor heating/cooling with a single compressor, has been selected for tests.

The tests have been performed at EDF in Moret, in a dedicated climatic chamber named "Climatron". The Climatron permits to reproduce a large range of thermal loads and variable water temperature as well as outdoor temperature and humidity. The temperature and humidity of the test cell can be controlled over a temperature range from -15°C to 50°C.

The test cell is instrumented as follows:

- a temperature sensor, Pt 100 type, for the measurement of the air temperature in the test cell, with a precision of ± 0.04 K,

- a hygrometer.

Both measurement devices are installed at the inlet of the air heat exchanger of the HP, and are dedicated to the control of temperature and hygrometry of the test cell.

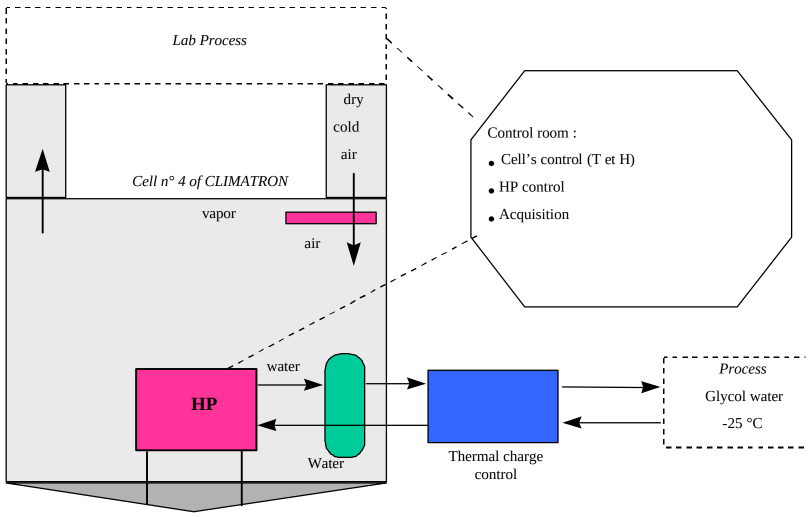

Figure 1: Global view of the air-to-water HP in the test room.

The air evaporator of the air-to-water HP is installed in the climatic chamber where defined air temperature and humidity are generated. The plate heat exchanger (condenser) of the HP is connected to a water circuit, which flow rate and temperature are controlled to simulate the thermal load of a residential house. Figure 1 shows the layout of the testing facilities.

During full load tests, the water temperature at the HP inlet is controlled in order that the heating needs remain higher than the heating capacity that the HP can supply to obtain a constant temperature at the HP outlet.
The water circuit is instrumented to permit:

- the measurement of the HP heating capacity,
- the water loop control.

Thermocouples are dedicated to control, and the platinum sensors to record the heating capacity.

3. ENERGY PERFORMANCES OF THE 2-COMPRESSOR HP

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.

This baseline HP has been modified in a prototype equipped with two scroll compressors of staged capacity, operating in tandem. Two collectors have been installed respectively at the suction and the discharge ports, to connect the compressors in parallel to the refrigerating circuit. The oil equalization has been realized directly on the oil casing by connecting the two hermetic compressors. Pressures have also been equalized in the same way. A bi-flow electronic expansion valve and a controller have been installed. This electronic expansion valve operates in both fluid directions, with dedicated temperature sensors. Pressure sensors have been installed at the inlet and outlet of each component (fin evaporator, plate condenser, compressors, expansion valve and 4-way valve) for the measurement of suction and discharge pressures, and pressure losses. The HP operates with R-407C.

The HP is controlled through a computer and a control/command interface written in C, permitting to define defrosting sequence and to change the operating parameters on the water circuit.

Data used for the calculation of the heating capacity on the water circuit are:

- the mass flow rate measured on the circuit close to the controller by an electromagnetic flow-meter (precision 1%),

- water temperatures at the HP inlet and outlet, measured by intrusive Pt 100 by pair and doubled by surface thermocouples; the precision of the measurement is ± 0.04 K.

Air measurements 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%.
Concerning the refrigerant:

- surface thermocouples are installed at the inlet and outlet of all components,
- two pressure sensors are installed at the high-side and low-side pressures.

Those measurements permit to calculate the superheating and the sub-cooling with a precision of 0.5°C.

3.1 Test results

Table 1: Test results of the reference HP.

Measurements	Units	A-7E35	A-3E35	A0E35	A3E35	A7E35	A12E35	A15E35
Te_air (average)	°C	-6.9	-3.3	-0.2	3.1	7.4	12.2	14.9
Ts_air (average)	°C	-8.5	-5.2	-2.3	0.1	3.5	6.8	9.0
Te_eau (average)	ºC	31.8	31.4	30.0	30.5	29.9	29.8	29.9
Ts_eau (average)	ºC	34.6	34.4	33.2	34.2	34.7	35.2	35.6
Vol Flow rate	l/h	1339	1339	1342	1343	1306	1345	1344
Heating time	s	2379	2381	2384	2376	7333	No defrost	No defrost
Defrost time	s	153	190	177	173	209		
Cycle duration	s	2532	2571	2561	2549	7542		
Pdefrost losses	kW	6.1	6.7	8.4	8.6	9.9		
Pe defrost	kW	1.7	1.5	1.5	1.5	1.5		
Edefrost lost	kJ	935.6	1277.7	1485.1	1493.8	2073.8		
Edefrost abs	kJ	258.7	294.4	257.4	256.0	306.3		
P_{HTF}_water w/o defrost	kW	5.18	5.57	5.91	6.88	7.69	8.41	8.85
Pe w/o defrost	kW	2.16	2.17	2.12	2.19	2.21	2.20	2.21
COP w/o defrost	W/W	2.40	2.57	2.78	3.14	3.48	3.83	4.01
P_{HTF}_water with defrost	kW	4.44	4.62	4.89	5.81	7.20	NA	NA
Pe total with defrost	kW	2.13	2.12	2.07	2.14	2.19		
COP with defrost	W/W	2.08	2.18	2.36	2.72	3.29		
Psuc_cpr_ff	kPa	320	338	358	409	442	491	516
Pdisch_cpr_ff	kPa	1 646	1 656	1 624	1 682	1 685	1 692	1 714

Evaporating and condensing pressures are represented as a function of the outdoor air and water temperatures. Heating capacities and input powers of the HP are measured. Losses due to defrosting have been separated so that they can be compared to the losses measured on the prototype HP. Those results are represented in Figure 2.

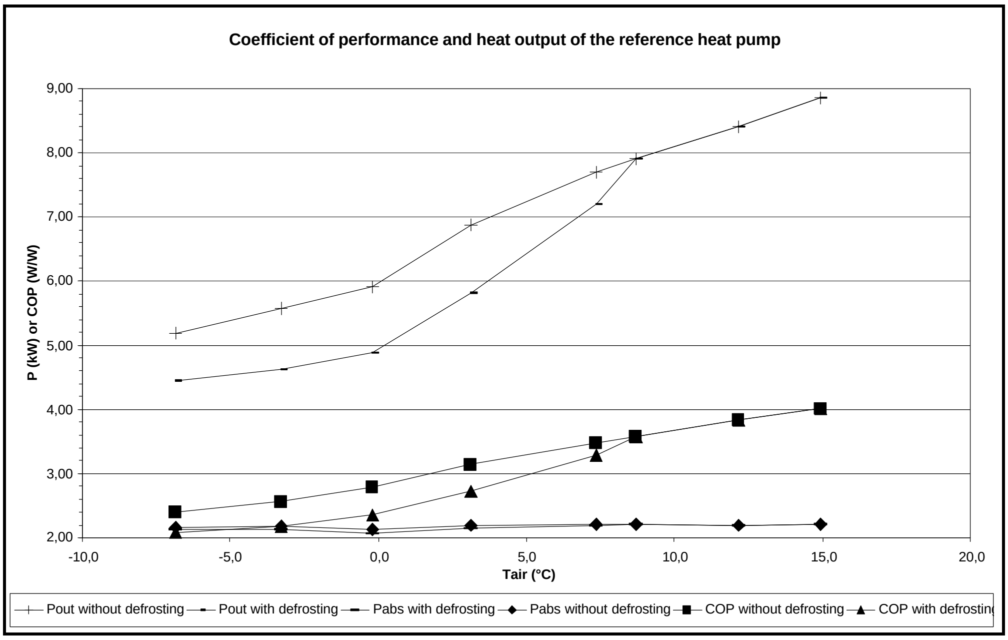

Figure 2: Input power, heating capacity and COP of the baseline HP
as a function of the outdoor air temperature.

The baseline HP operates with a constant set water temperature at the outlet (fixed in the tests at 35°C).

3.2 Test results of the two-compressor HP prototype

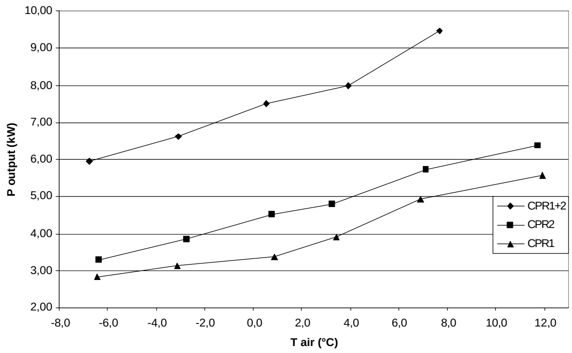

Figure 3: Heating capacity as a function of outdoor temperature.

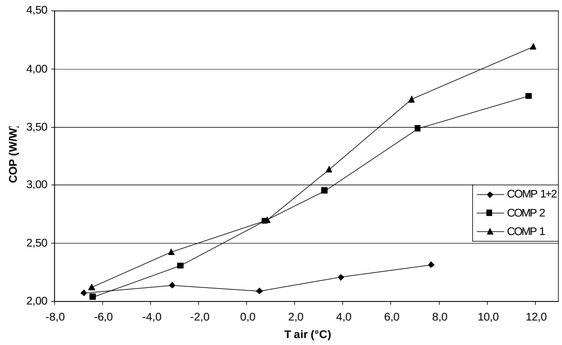

Figure 4: HP COP as a function of outdoor temperature

Performances of the HP prototype are presented in Figures 3 and 4. They represent the heating capacities, the input power, and COPs as a function of the outdoor temperature and the system operation mode. The diamond curves correspond to the operation with the two compressors working in parallel; the square curves to the most powerful compressor operating alone, the triangle curves to the smallest compressor operating alone.

Performances and output powers do not include losses due to defrosting. Some measurement points have been difficult, for example when both compressors were operating in parallel with a water temperature at the HP outlet of 35°C, and an outdoor temperature of 10 to 12°C. The heating capacity is so high that the high pressure is higher than the maximum admissible threshold of 3000 kPa, and so the high-pressure pressostat stops the HP system.

The heating capacity when the two compressors operate in parallel is much higher than the single compressor baseline HP. So the new prototype HP can deliver the needed heating capacity at a lower outdoor temperature down to –7°C. The electric additional resistor heating is eliminated permitting a higher seasonal energy performance, as it will be seen further.

3.2.1 Defrosting control
For the prototype HP, four defrosting modes are available. Between -7 and + 3°C, defrosting can be performed by reversed cycle either with both compressors operating in parallel or one or the other of the 2 compressors. For an outdoor air temperature of 2°C, defrosting has been performed using ventilation with all compressors stopped and the by-pass of the expansion valve as indicated in Argaud (2001).

After analysis, the best defrosting process consists in using the less powerful compressor at –7°C, and the more powerful compressor for higher outdoor air temperatures. From 2°C, the most efficient defrosting process is to use ventilation with expansion valve by-pass. Even though the defrosting time is longer (6.09 min.), the energy consumption is only due to the fan operation (100W) and to the sleep power operation (50W).

3.2.2 Energy gains
The heating needs of a house are proportional to the outdoor temperature: $B = GV.(T_{\text{int}} - T_{ext})$. For a new house of 150 m², the heat loss coefficient GV is of 220 W/K. When an indoor air temperature of 20°C is desired, the heating need can be quantified according to the outdoor temperature and compared to the output powers of the baseline HP and the prototype HP according to the operation mode (for a constant water temperature at the outlet of 35°C).

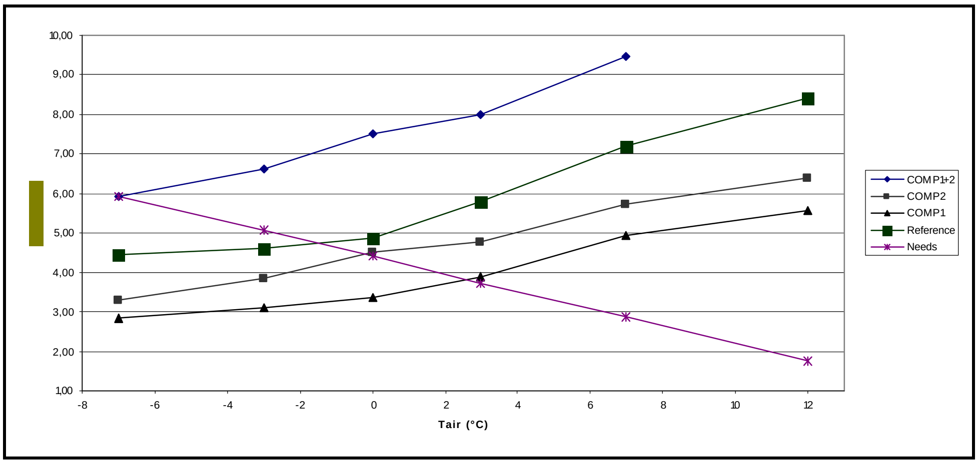

Figure 5: Heating capacities and heating needs
of the baseline and the prototype HPs.

Figure 5 presents the heating capacity of the baseline HP (large square curve), and of the prototype HP (diamond curve), as a function of the operation mode and the outdoor temperature. The heating capacity of the prototype HP at -7°C is sufficient to compensate the house heat losses, so the use of additional electric resistor heating is avoided. The square and the triangle curves represent heating capacities of the prototype HP operating respectively with the most and the less powerful compressors. The prototype HP operation over the range of the outdoor temperatures will be as follows:

- between -7 and 0°C, both compressors operate in parallel,
- between 0 and 3°C, the most powerful compressor operates alone,
- above 3°C, the less powerful compressor operates alone.

The heating capacity of the baseline HP is sufficient to compensate the heat losses when the outside air temperature is higher than -2°C. Above this temperature, an additional electric heater will increase the heating capacity in order to compensate the building heat losses. The efficiency of the additional electric heater is assumed to be of 1. Figure 6 presents the instantaneous efficiency gains and their types as a function of the outdoor temperature for the prototype HP compared to the baseline HP, and operating according to the principles mentioned previously.

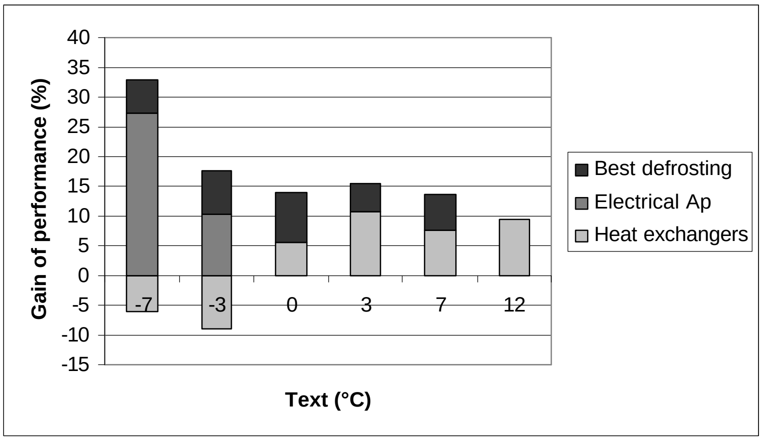

Figure 6: Energy gains and energy losses as a function of the outdoor temperature.

The proposed control shows an efficiency gain from 8 to 26 % compared to the baseline HP. Those gains have been analyzed and three types have been identified.

- The gain by optimized defrosting. It has been evaluated by comparing defrosting losses on the prototype HP using the proposed control to the defrosting losses on the same HP when assuming a defrosting process identical to the baseline HP.
- The gain due to the avoidance of the additional electric heating. This gain consists in designing the HP for 100% of the heating needs.
- The gains/losses of heat exchangers. Heat exchangers have been designed to operate with the compressor of the baseline HP. Two cases can be distinguished.
  When the prototype HP operates with the 2 compressors in parallel, the outdoor air coil and the plate heat exchanger are undersized, which entails energy losses from 6 to 9% for outdoor temperatures of -7°C and -3°C respectively.

  On the contrary, when the HP operates at partial load, with only one compressor, heat exchangers are relatively oversized, which permits a gain from 5 to 12% compared to the baseline HP.

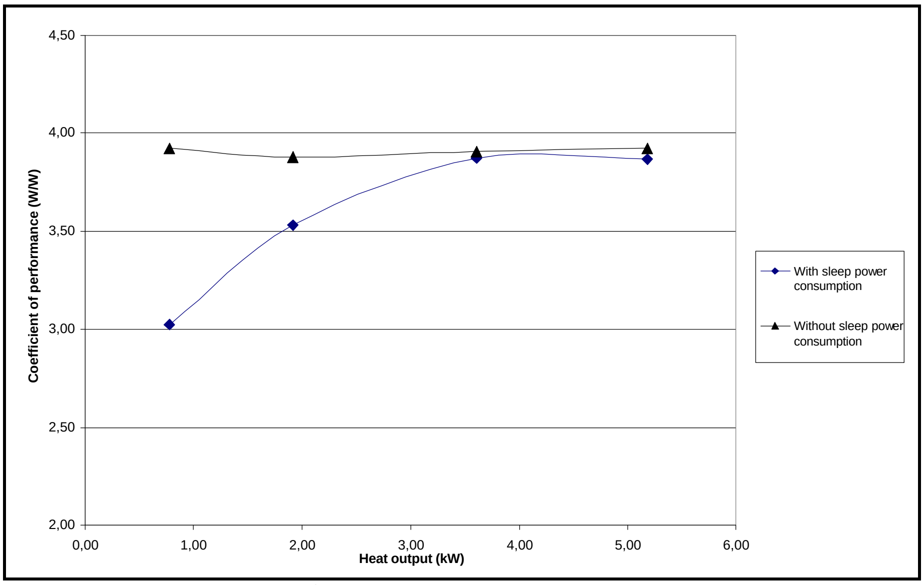

Figure 7: Comparison of HP COP with and without sleep power accounting.

3.2.3 Energy losses for operation at partial load

In a study of the same baseline HP as the one analyzed in this paper, Rivière (2004) has shown that when the HP operates in cycling for a thermal load lower than the specified one, losses due to the HP cycling are nil. On the contrary, during the standstill period the HP consumes in the range of 50 W, which is the sleep power consumption. Figure 7 presents the variation of the HP COP when taking into account the sleep power consumption or not. Those losses due to the sleep power operation depend on the percentage of the standstill period.

The prototype HP using two compressors in parallel can have up to 3 levels of heating capacities. It is capable to adapt more precisely its operation to the heating needs. A simulation over a full heating period shows that the total time of the standstill periods is shorter.

4. CONCLUSIONS

The two-compressor technology has been adapted on an existing HP and tested. Performances of such a system compared to the baseline HP show an increase in instantaneous performances between 8 and 26%. Energy gains have been identified, and then quantified.

Capacity variation with two compressors requires a control strategy that takes carefully into account the differences in efficiencies depending on the required heating capacity. When designing a HP with 2 compressors the heating needs can be totally fulfill for the lowest outdoor temperature. Heat exchangers have to be designed for this maximum refrigerant mass flow rate in order to get a high efficiency system.

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