A WET-VAPOUR INJECTION AND VARIABLE SPEED SCROLL COMPRESSOR AIR TO WATER HEAT PUMP MODEL AND ITS FIELD TEST VALIDATION

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Abstract: The European government has recognized air, water and ground as renewable energy sources and has indicated heat pumps as the most effective technology to make use of them. As a consequence a new labeling scheme will clearly compare performances of heat generators by comparing the primary energy consumption factor. ErP (Energy related Products) directive provides also the methodology and reference normative by which obtaining the η % value, which is directly related to the seasonal performance factor of the heat pump measured through the EN14825.
The first part of this paper shows results of field testing air to water heat pumps from different installations across Europe. These heat pumps are equipped with the latest development of variable speed scroll compressor with Wet-Vapour Injection technology. The second part of the paper presents a semi-empirical model. The model is based on compressor data and heat pump parameters, such as pump and fan power consumption. Further parameters like heat exchanger temperature differences and defrost impact on the efficiency are taken into account. Field test data are used to validate the semi-empirical model in order to calculate seasonal coefficient of performance (SCOP) of the air to water heat pump. A comparison with EN14825 calculation method is presented.

Key Words: Semi-empirical model, variable speed, wet injection, vapor injection, heat pump SCOP, EN14825, scroll compressor

1 INTRODUCTION

In 2012, Emerson Climate Technologies has officially released high efficiency variable speed scroll compressors, called ZHW, featuring vapour injection and wet-vapour injection. Those compressors are qualified with R410A refrigerant and are controlled by an Emerson drive. An integrated Refrigerant Module for Heating (RMH) has been developed using this technology. This is a refrigerant circuit designed for use in split air-to-water heat pump applications, where the evaporator is located outdoors. RMH integrates the compressor ZHW, its drive, the indoor refrigerant circuit and its controller (RCC).
Since end of 2011, twelve field tests of split air-to-water heat pumps incorporating RMH have been installed .
These field tests give the opportunity to analyze the performance of air-to-water split heat pumps, and estimate their performance by modeling the complete system.
This paper presents selected results from the field tests, a semi-empirical model of an air to water heat pump using a variable speed scroll compressor with Wet-Vapour Injection technology and its validation. A further comparison with the EN14825 method is also included.

2 SYSTEM DESCRIPTION

The air-to-water heat pump system schematic is shown on Figure 1. The indoor unit includes the Refrigerant Module for Heating RMH with the RCC controller, the hot water tank and water pumps. A separate hot water tank for domestic hot water (DHW) is necessary when both heating and DHW are to be supplied. The RCC receives the heating capacity demand from the main heat pump system controller, and sets the compressor speed and control valves to match the demand. Pressure and temperature sensors are connected to the RCC, so that it can communicate the measurements to the main system controller. The outdoor unit consists of the evaporator fan coil unit, with necessary refrigerant and electrical connections. The system uses vapour and wet-vapour injection. A reversing valve is included in the refrigerant circuit and is used to provide reverse cycle defrost. The drive, necessary to control the compressor speed, is cooled down by subcooled liquid refrigerant from the outlet of the economizer heat exchanger. Two internal controllers (RCC indoor and outdoor), connected to each other and to pressure and temperature measurements, are used to control three electronic expansion valves (heating, cooling and injection).

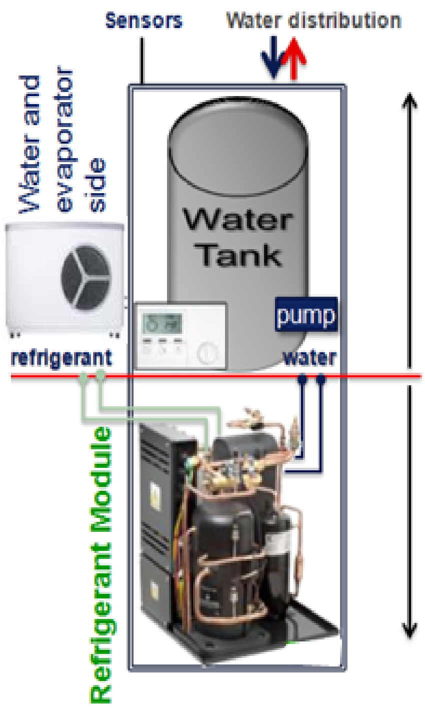

Figure 1: Schematic of the tested heat pump.

Vapour and wet-vapour injection technologies

One of the most important ways to optimize the seasonal efficiency of an Air-to-Water heat pump system is to limit the need of any back-up heater by cold outdoor temperature. With a standard scroll compressor, the heating during cold period is generally limited by the discharge temperature limit of the compressor, especially when high water temperature is requested.

Vapour injection technology, commonly called EVI, can push the compressor operating limits by cooling down the refrigerant gas at an intermediate level of the compression through an additional injection stub on the compressor. The vapour injection scroll compressor is for use with an economized vapour compression cycle heat pump (showed on Figure 2). In addition to enlarging the compressor operating envelope, this cycle offers the advantages of a higher heating capacity and a better COP than a conventional cycle. Both the heating capacity and the COP improvement are proportional to the temperature lift and this technology offers best results at high condensing operation where capacity and efficiency are most needed.

Additional gas cooling and therefore even larger operating envelope can be obtained with liquid injection. The additional cooling liquid can be injected along with the gas through the injection stub. This is called wet-vapour injection. Figure 3 shows the p/h diagrams of wet vapour injection and its operating envelope benefit. Note that wet injection is carried out only at the level of the intermediate injection port. The main evaporator superheat should remain positive enough to avoid liquid droplets at the compressor suction port.

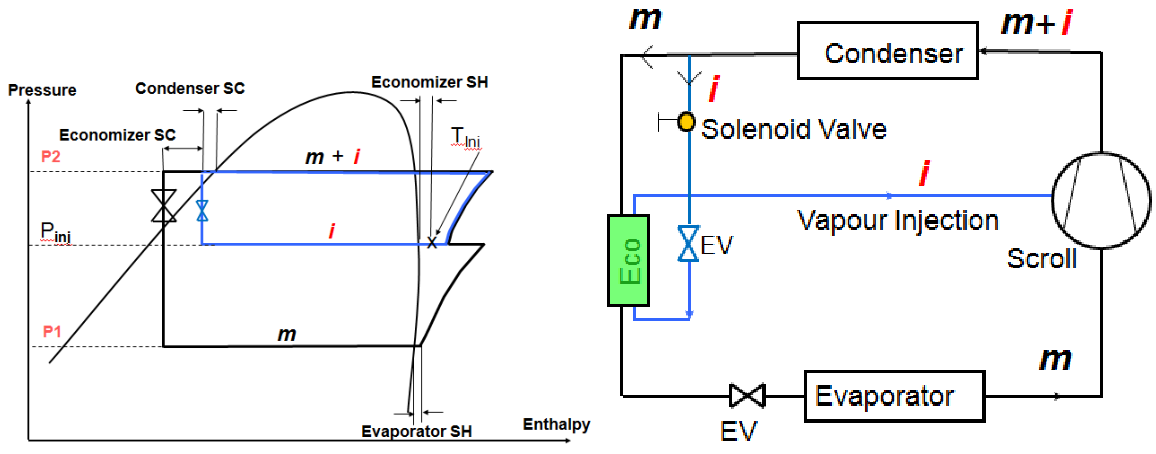

Figure 2: Vapour Injection Technology Ph Diagram and Circuit Schematic

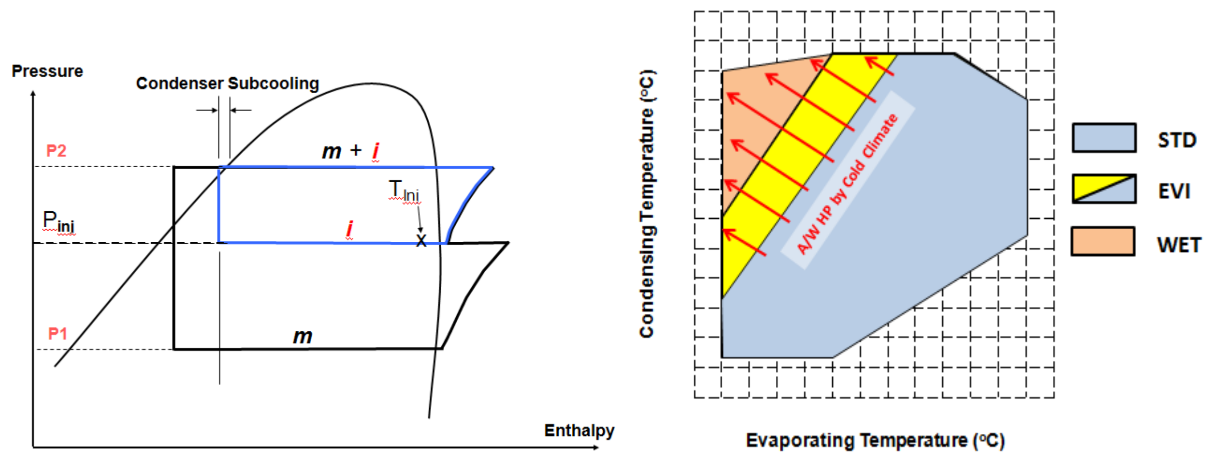

Figure 3: Wet-Vapour Injection Technology Ph Diagram and Compressor envelope extension

3 FIELD TEST DESCRIPTION

There are 12 monitored sites distributed in Europe. The locations situated in Belgium, Germany and Czech Republic are representative of Mid-European average climate conditions. They are shared between floor heating, radiator heating, with or without buffer tank, with or without domestic hot water production, and different split line lengths. A summary of the field test sites is shown in Table 1.

All heat pumps have been equipped with the following measurements: refrigerant and water circuit pressures and temperatures, water mass flow rate, electrical power consumed and compressor speed.
A data logging system receives the measured data through the RCC controller by Modbus and stores them every 4 seconds. The heating power demand curve is also logged.
The COP is measured in two ways. Real COP is calculated from measured total heat pump power, and water temperatures and mass flow. Secondly, RCC can also calculate the COP from the nominal performance curves of the compressor stored within it. From those and considering the measured pressures and temperatures, RCC continuously calculates the total theoretical electrical power consumed (including fan and water pump powers) and the

heating capacity delivered. RCC integrates that “theoretical” COP along the time to obtain the seasonal COP. It accounts the defrost periods by considering heating capacity as negative during defrost. The difference between the two ways of calculating COP is usually very small. More information about the field test can be found in Maggioni and Proserpio (2013) and Dardenne et al (2013a).

Table 1: Field test sites summary

#	Description	Split Line Length [m]
1	Radiator + Buffertank 200l + Solar DHW pre-buffer tank	20
2	Radiator Heating + Buffer Tank 200l	8
3	Radiator Heating + Buffer Tank 200l	23
4	Radiator Heating	8
5	Radiator Heating Low Temp	15
6	Floor Heating + Solar DHW pre-buffer tank	8
7	Radiator Heating	12
8	Radiator Heating + combined Solar DHW & Heating Buffer tank	12
9	Radiator Heating + Buffer Tank 200l	7
10	Floor & Radiator Heating + Buffer Tank 200l	15
11	Radiator Heating	22
12	Floor & Radiator Heating	18

4 FIELD TEST RESULTS

Figure 4 shows the operational condition distribution for Site #2 for the period September 2012 to August 2013. This site is radiator heating and DHW. The hours accumulated in the heating (radiator mode) and DHW production are shown in these diagrams. The size of the marker corresponds to the duration at that particular operating condition. For radiator heating, the dominant conditions are in the central part of the compressor envelope with only a very few hours requiring wet injection. Application of compensated control is apparent in the reduction of condensing temperatures at higher ambient conditions. DHW shows a nominal temperature of 55°C without any requirement of wet injection (not shown here).

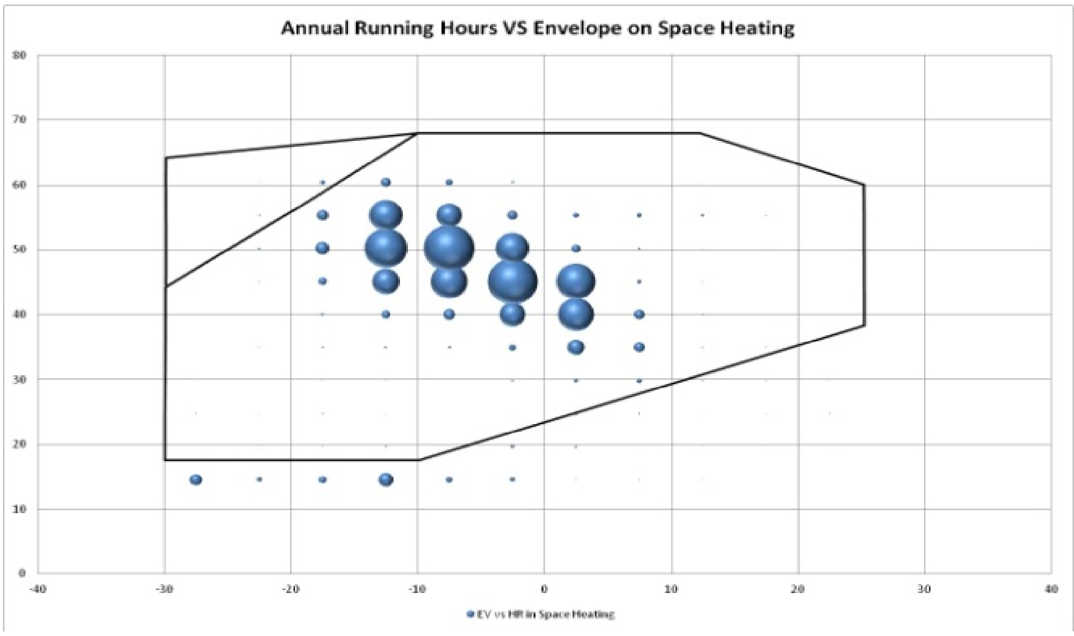

Figure 4: Number of hours at various operating points during the heating season: Space heating

Figure 5 shows a combination of data from the 5 first sites for the complete winter season 2012-2013. The HCOP showed here is the one calculated by RCC and in all cases includes the defrost allowance, water and fan pump powers. The SCOP for the 12 field test was in a range from 3 to 3.25. This seasonal HCOP is the summation of heat delivered and electrical power requirement, and includes defrost cycles.

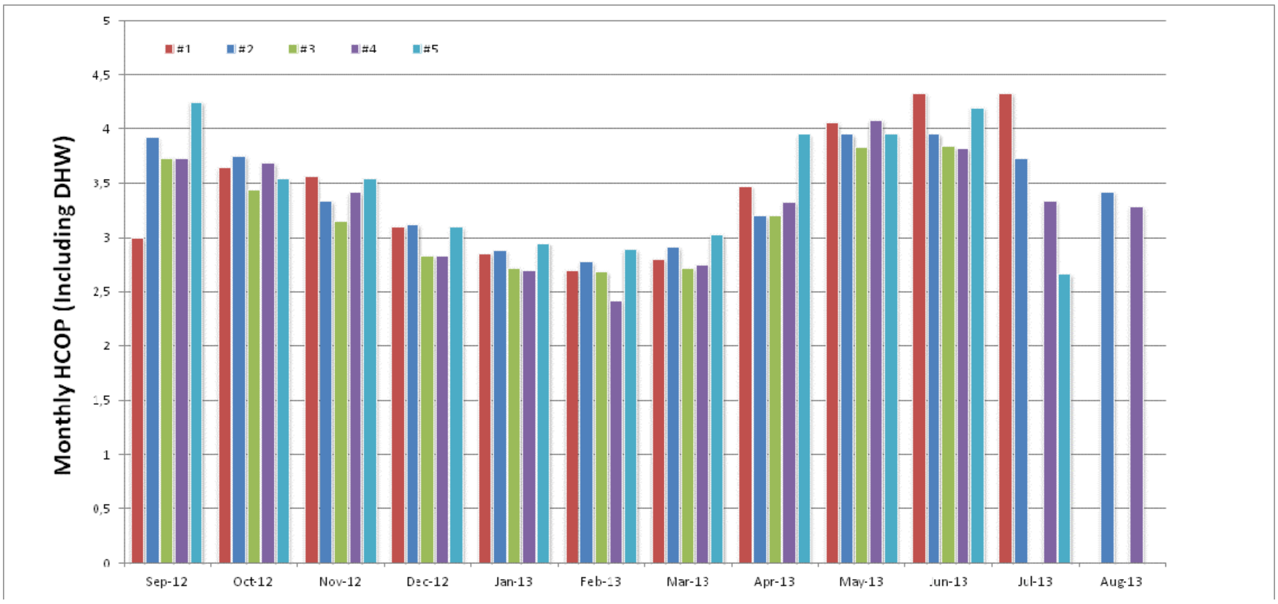

Figure 5: Combined 5 sites results for the winter season 2012-2013.

Defrost Considerations
The RCC controller determines the need for start and finish of defrost from the measured system parameters. The monitoring system calculates and accounts for any negative heat input to the heating circuits, i.e. heat drawn from the heating circuits to enable the defrost. Figure 6 shows the defrost degradation factors derived from the measured data. That factor is calculated as the ratio between real HCOP with defrost and the one without taking care of defrost, for given periods. Each point represents a period of half a day. The spread of the points is due to variable air humidity level, which is not measured on the various sites.

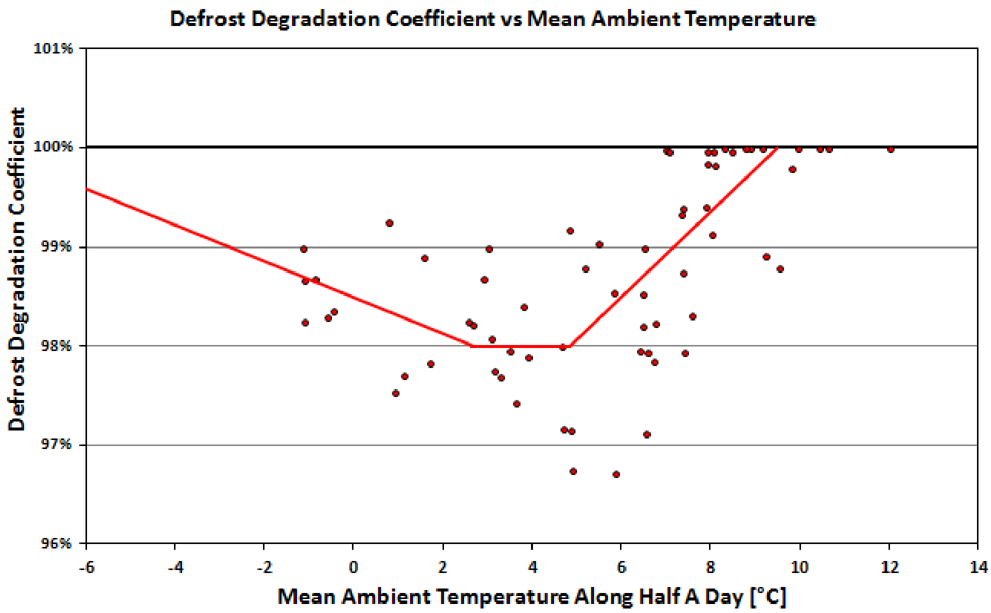

Figure 6: Defrost degradation coefficient function of the outdoor air temperature.

4. AIR-TO-WATER HEAT PUMP SYSTEM MODEL OVERVIEW

A steady-state semi-empirical model of the split air-to-water heat pumps installed in field tests was developed. The general inputs, outputs and parameters of the model are shown on Figure 7. The inputs of the model are the heating demand (which is an output of the main system controller), the return water temperature, the hot water mass flow and the ambient temperature. It aims to predict the compressor speed, the different refrigerant temperatures along the cycle, the leaving water temperature, the total electrical power consumed by the heat pump and the system coefficient of performance. Some parameters related to the compressor, the evaporator fan and the heat exchangers are provided. More information on the model can be found in Maggioni and Proserpio (2013) and Dardenne et al (2013b).

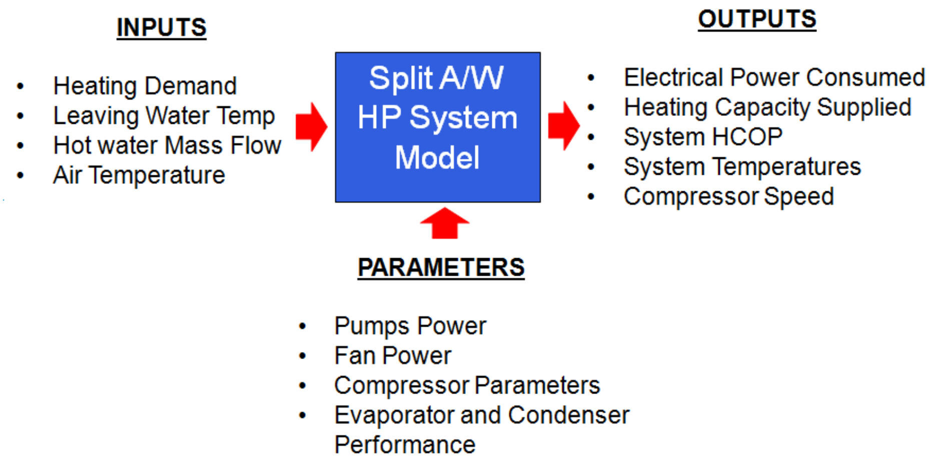

Figure 7: General overview of the split air-to-water heat pump system model.

4.1 Scroll Compressor Model

The compressor model is based on the model proposed by Winandy et al (2002). This steady-state semi-empirical thermodynamic model allows for the calculation of the mass flow rate, the compressor electrical power and the discharge temperature of the compressor without intermediate injection. The model divides the compression four parts as shown in Figure 8:

1. Isobaric heating up of the refrigerant before the inlet of the scroll (su to su1)
2. Isentropic compression to an adapted pressure determined by the built –in volume ratio of the scroll (su1 to ad).
3. Adiabatic-isochoric compression corresponding to the opening of the compression chamber to the discharge plenum where the losses due to over- or under-compression develop (ad to ex1).
4. Isobaric cooling down, through the shell, of the refrigerant from the outlet of the scroll to the discharge port of the compressor (ex1 to ex).

The model was modified by the following: firstly to include the variable speed effect, the power losses were made dependent on the compressor speed and internal leakage, also speed dependant, was added. Secondly the model was modified to integrate the vapour injection as shown on Figure 8.

The internal leakage mass flow rate M_{leak} mingles with the scroll suction mass flow. It was found inversely proportional to the square root of the speed and proportional to the pressure difference between the discharge and the suction like shown in equation eq. (1) where M_{leak0} is a parameter to optimize, N the rotation speed, and Δp the pressure difference between compressor discharge and suction. It reflects the effect of the centrifugal force increase (induced by speed increase) on the contact between the lower and upper scroll flanks.

$$M_{leak} = M_{leak0} \,.\, (N_{max}/N)^{(1/2)} \,.\, \Delta p / \Delta p_{max} \qquad \textbf{(Eq. 1)}$$

To model the vapour injection, a second stage compression volumetric efficiency was defined (ε_{v2}), in order to determine injection mass flow. This represents volumetric efficiency after closure of injection holes and is given by eq. (2) where M_{su} represents the suction mass flow, M_{inj} the injection mass flow and v_{inj} the refrigerant specific volume at injection conditions.

$$\varepsilon_{v2} = [(M_{su} + M_{leak} + M_{inj}) \cdot v_{inj}] \,/\, \text{Displacement} \qquad \textbf{(Eq. 2)}$$

By validation on the ZHW16 performance data, ε_{v2} was observed dependent on the rotation speed, the suction pressure, and the pressure ratio between compressor injection and suction.

Injection flow is modelled as quasi incompressible flux through a nozzle, as shown in eq. (3) where A_{inj} is the fixed flow section area of injection holes, C_d the coefficient of contraction, ρ_{inj} the density at injection conditions, and Δp_{inj} defined at eq. (4) represents the difference between the injection pressure and the mean scroll pocket pressure p_{int} during injection process (eq. (5)).

$$M_{inj} = A_{inj} \cdot C_d \cdot (2 \cdot \rho_{inj} \cdot \Delta p_{inj})^{(1/2)} \qquad \textbf{(Eq. 3)}$$

$$\Delta p_{inj} = p_{inj} - p_{int} \qquad \textbf{(Eq. 4)}$$

$$p_{int} = \alpha_{inj} \cdot p_{su} \qquad \textbf{(Eq.5)}$$

The injection temperature and pressure are determined by a mass and heat balance of the economizer, presuming fixed injection superheat, and approach which is representative of the real system. A_{inj} C_d and α_{inj} are optimized by validation on compressor.

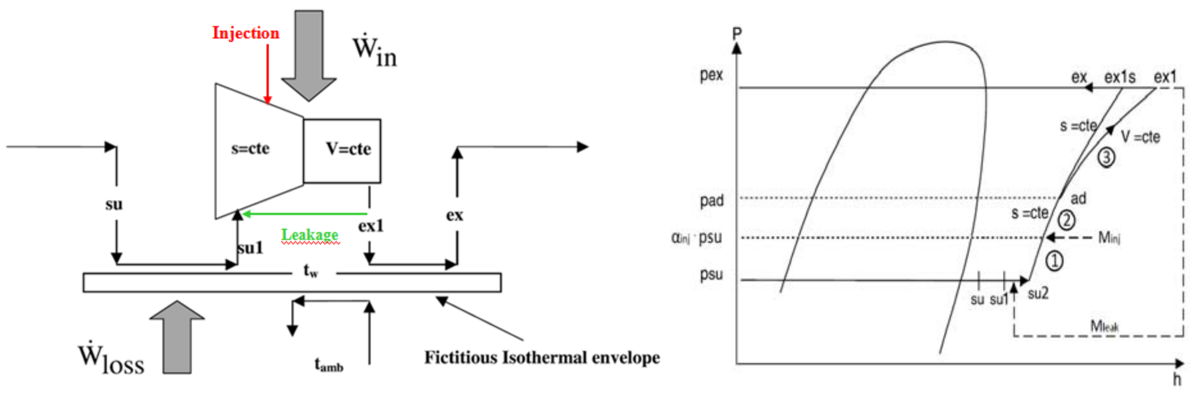

Figure 8: Compressor model based on Winandy et al (2002) adapted to variable speed with vapour injection

The compressor model was validated based on 77 test performance data at different operating conditions covering the whole compressor operating envelope and speed range. The results of the validation are shown in Figure 9.

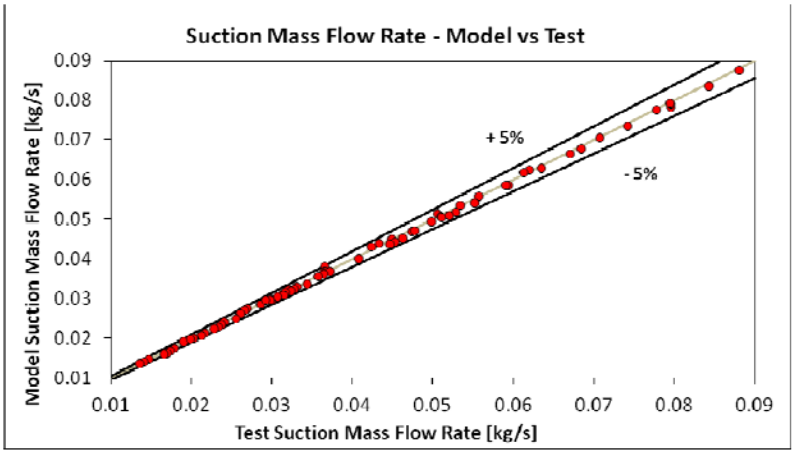

Figure 9: Validation of the compressor model in term of suction mass flow rates

The compressor suction mass flow rate and power was estimated within ± 5%, discharge line temperature within ± 5K and injection mass flow rate within ± 10%.

4.2 Condenser Model

The condenser is a plate heat exchanger. It is considered as a steady-state two-zones counter-flow heat exchanger featuring a total heat transfer surface of A_c supplied by the heat exchanger manufacturer. In the first zone, the refrigerant gas is desuperheated and the global heat transfer coefficient $U_{c,1}$ depends on the refrigerant mass flow rate. In the second zone, the refrigerant is condensed and slightly subcooled and its heat transfer coefficient was considered constant. The heat transfer surface of both zones is variable. The whole condenser model is solved following the ε-NTU method.
The absolute error obtained on the dew condensing temperature when comparing the model with steady-state conditions in the field tests is within +- 1K, as shown on Figure 10.

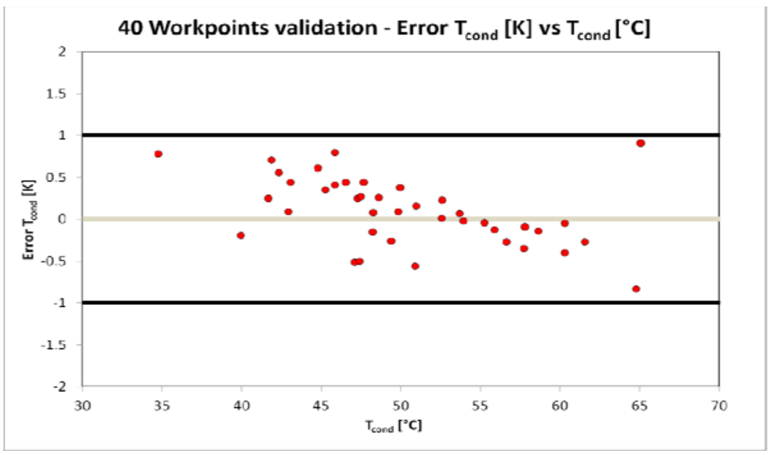

Figure 10: Absolute error on dew condensing temperature

4.3 Evaporator Model

The evaporator was modeled following the method detailed by Pennati C. et al. in 2011. This is a steady-state one-zone model where the heat transfer coefficient varies with air and refrigerant mass flows. The inlet air relative humidity is an input, and it is fixed to 95% at the outlet. Figure 11 shows the results obtained in term of evaporating temperature.

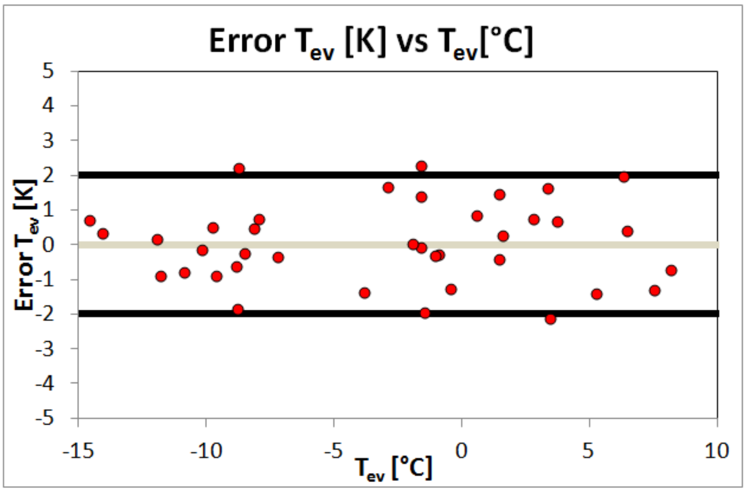

Figure 11: Absolute error on dew evaporating temperature between the model and the field tests data.

Defrost is considered through a defrost degradation coefficient that is applied to HCOP. That coefficient accounts the “negative” heating capacity and the electrical power consumed during defrost. From field tests data shown in Figure 6, a defrost correction was established based on the outdoor air temperature (red line).

5. HEAT PUMP SYSTEM MODEL VALIDATION

The heat pump system model has been applied to field test data. On Figures 12 and 13 are presented some comparisons between the model and the real field test data for two cycles: a radiator heating cycle and a DHW heating cycle. The comparison shows the good estimation of the model with most of the time, temperatures within ±5K and rotation speed, cooling capacity and total electrical power consumed within 5%. During fast transient periods like heat pump start, the error is much higher. This is mainly due to bad estimation of the evaporating and condensing temperature, itself due to heat exchangers dynamics in the field not taken into account in the model.

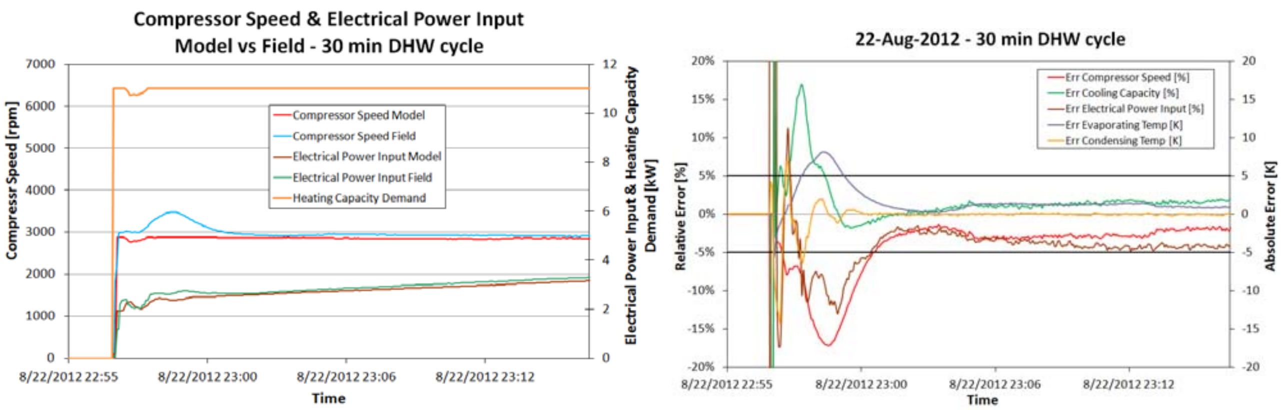

Figure 12: Validation of the model along a DHW heating cycle.

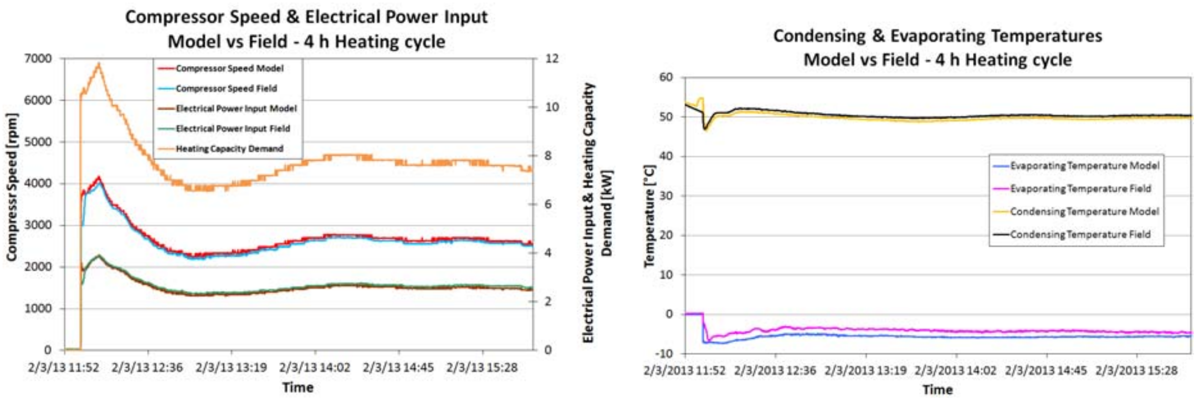

Figure 13: Validation of the model along a radiator heating cycle.

5.1 Comparison with the Model

In order to compare the model predictions with actual test data, specific days, i.e. 24h periods were chosen on site #3 as representative of a range of heating conditions. Both radiator heating and DHW production were considered. The HCOP and total system energy input based on total seasonal heat demand were calculated by means of the model and compared to test data.. Test HCOP is the real one based on field water and electrical power measurements. These results are presented in Table 2. It can be seen that the comparison is very good with errors within the tolerances expected.

Table 2: Comparison of heating COP derived from the model and the test.

Days	COP_{real}	COP_{model}	Relative Error [%]	Mean $T_{outdoor}$
20-Jun-2012	3.19	3.18	-0.4%	18 °C
22-Aug-2012	3.16	3.31	+4.5%	19 °C
22-Sept-2012	3.43	3.51	+2.4%	10 °C
18-Nov-2012	3.12	3.24	+3.8%	9 °C
03-Feb-2013	2.84	2.84	+0.0%	5°C

5.2 Comparison with EN14825 Method

The Seasonal COP, SCOP can be calculated using the EN14825. SCOP calculated with this methodology will directly be used to define unit labeling classes. This new standard considers three different climate zones for heating: warmer, average and colder. Each zone sets a minimum outdoor temperature and yearly profile. The standard also defines four different water temperature regimes for different applications: 35, 45, 55, and 65°C.
Finally, stand-by losses and auxiliary power consumption are also taken into account in the SCOP calculation.

The average climate has been chosen, as most representative for the labeling classes. In order to compare results to field test #2, average climate and high water temperature have been selected. The compressor model is based on a 20 coefficients polynomial derived from compressor rating tests.
In order to calculate SCOP some hypotheses have been made:

- Fan power consumption proportional to compressor speed and cooling capacity
- High temperature (55ºC) compensated water curve
- Air coil and condenser constant temperature differences profile (10K air / 0.5K water)
- Constant auxiliaries power consumption, including water pump (fixed speed type)

EN14825 average climate has a minimum outdoor air temperature of -10ºC and a defined hourly profile. This profile combined with a build heating load (linear from a maximum at -10ºC and zero at 16ºC) gives the amount of heating energy that is required by the building. EN14825 defines the specific conditions at which a heat pump system should be tested in order to calculate the Coefficient of Performance (COP). These tests and building load information are thereafter combined to calculate the overall seasonal performance of the unit. As mentioned, SCOP is dependent on the building load. In this simulation, a monovalent application has been selected, which means the heat pump can supply the entire building energy requirement without any use of back up heating system. It may also be noticed that a capacity modulated heat pump is less sensitive to building load variation and therefore the same heat pump can be applied in a wider range of dwellings without impacting significantly the overall performance.

The results are summarized in Table 3 and it shows that both numerical models are in good accordance with field test data. Note that field test data includes DHW, with a set point equal to the maximum water temperature. In case of site #2, it is set to 55ºC. In any case total energy spent for DHW is approximately 7% of the total.

Table 3: Comparison of seasonal COP derived from the model, the test and EN14825.

	Field Test#2	Model	EN14825
Location/Climate	Belgium	EU Average	EU Average
Evaporator DT	-	One Zone Model	Fixed 10K
Condenser DT	-	Two Zones Model	Fixed 1K
Fan Control Logic	80% - 50%	80% - 50%	100% - 50% Speed linked to cooling capacity (Power Consumption @100% 500W)
Water Pump & Aux	-		Fixed 65W
Weighted Fan/Pump/Aux Power Consumption	150W	150W	130W
Defrost	-	Test Curve	2% COP de-rate @Ambient between -7 to 7ºC
Cycling Coefficient	-	1	0,95
Crankcase Heater	Yes	Not included	Yes
Max Water Temperature	55	55	55
Monovalent	Yes	Yes	Yes
Max Load	-	15 kW	15 kW
Total Heating Energy	39959 kWh	30985 kWh	30990 kWh
Running Hours (year)	2554	4174	4157
SCOP	**3.05**	**3.3**	**3.11**

6 CONCLUSIONS

Field test sites for the heat pumps incorporating R410A vapour injected variable speed compressors with wet injection capability have been installed in 12 locations in Europe. The SCOP for each of the 12 field tests was in a range from 3 to 3.25. This seasonal COP is the quotient of heat delivered and electrical power requirement, and includes defrost cycles.
A semi empirical model of an air to water heat pump has been presented in this paper. The model computes the heat pump performance given the heating demand curve and the water and air conditions. That model integrates a compressor model adapted to vapour injection and speed modulation, a two-zone physical model of condenser, and a physical model of evaporator. Wet injection hasn't been studied as it concerns only a limited number of hours along the year, especially in European warm and average climates. The validations of the compressor, evaporator and condenser show errors within 5% range. The system model has been validated on RMH field test running in Belgium. The results show good correlation between the model and the field data except in fast transient operation. Among next steps of improvement of the model are the consideration of frost formation on the evaporator, and a better representation of the vapour and wet injection.

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