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Investigation of field methods for evaluation of air-to-air heat pump performance

AURÉLIE JACTARD

ZELIN LI

Department of Energy and Environment
Division of Building Services Engineering
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MASTER’S THESIS

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Master’s Thesis within the *Sustainable Energy Systems* programme

AURÉLIE JACTARD

ZELIN LI

SUPERVISOR

Roger Nordman (SP)

EXAMINER

Per Fahlén

Department of Energy and Environment
Division of Building Services Engineering
CHALMERS UNIVERSITY OF TECHNOLOGY

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AURELIE JACTARD
ZELIN LI

Master Thesis E2011- 17
Department of Energy and Environment
Division of Building Services Engineering
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

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ABSTRACT

Air-to-air heat pumps have gained an increasing popularity to supply domestic heating and cooling in the building service sector since the 1990s. Due to the challenge from depletion of fossil energy sources as well as global warming, it is essential for the manufacturers to improve the heat pump efficiency in order to satisfy the requirement of living standard without consuming excessive electricity.

This diploma work focuses on studying different air-to-air heat pump COP (Coefficient of Performance) field testing methods and standards. Practical measurements with three test methods were carried out in parallel in the laboratory, namely SP Method, Climacheck Method and Calorimeter Method. Results were collected from both long and short term testing with the corresponding outdoor weather conditions. Hence, heat pump performance characteristics can be studied. Furthermore, uncertainty evaluation and limitation analysis of each testing method were made so that further improvement could be suggested for field measurement. While defrosting is a key factor to ensure high efficiency, the dynamic performance within that period is extremely hard to monitor, and it is critical for the long term HSPF (Heating Seasonal Performance Factor) measurement of air-to-air heat pumps.

In this study it has been discovered that neither the SP Method nor the Climacheck Method is optimal for the field test, due to the inaccuracy during the defrosting process. The Calorimeter Method is feasible to serve as the reference method but it is not suitable for field testing. Results achieved by the SP Method seem to be in accord with what is achieved by the Calorimeter Method. However, the influence of the air volume flow measurement devices upon the testing unit itself, and the bulky equipment setup, make the SP Method infeasible for field testing during long time periods. The Climacheck Method is designed as a convenient way to carry out continuous measurement, with a user friendly interface and less bulky instrument. But it does not work properly if, for instance, the superheat is too low to avoid liquid droplets in the suction line to the compressor. Droplets result in an overestimated isentropic efficiency and consequently the heating capacity is also overestimated. On the other hand, the Climacheck unit warns the user and indicates the cause of the problem.

Based on this study, further improvements could be suggested for each testing method. For the Climacheck method, an additional testing point is available and should be mounted close to the inlet of the condenser. For the SP Method, pre-testing in the lab could be made concerning air volume flow rate under different indoor fan speeds, so that the bulky air collecting system would be avoided for field testing afterwards. Nevertheless, how to measure air volume flow rate accurately, especially under dynamic conditions, remains a challenge for the external COP testing method.

Key words: air-to-air heat pump, COP testing method, defrosting

Contents

Preface

This project is a part of the European program Seasonal Performance factor and Monitoring for heat pump systems in the building sector (SEPEMO-Build) which aims at improving quality of heat pump systems.

This study has been carried out within SP Technical Research Institute of Sweden in Borås, Sweden. The tests were performed between March and May 2011.

We would like to thank our supervisor Roger Nordman and our examiner Per Fahlén for their follow-up and their very valuable advice. We also appreciate the involvement of Ola Gustafsson and Peter Lidbom whose experience has been very helpful.

Moreover, we would like to give our thanks to IVT for supplying heat pumps for testing as well as to Climacheck for their support.

Finally, we express our gratitude to all our co-workers which took part in the project and without whom the tests could not have been conducted.

Borås, June 2011

Zelin Li and Aurélie Jactard

Nomenclature

Roman letters

A	Surface area of the overall climate chamber	m^2
COP_1	Coefficient of Performance for heating purpose	
COP_{1_c}	Coefficient of Performance for heating purpose, Carnot cycle	
COP_2	Coefficient of Performance for cooling purpose	
COP_{2_c}	Coefficient of Performance for cooling purpose, Carnot cycle	
c_p	Specific heat capacity at constant pressure	J/(K·kg)
f	Heat loss factor through the compressor	
h_1	Refrigerant enthalpy at compressor outlet/condenser inlet	kJ/kg
h_2	Refrigerant enthalpy at compressor inlet/evaporator outlet	kJ/kg
h_3	Refrigerant enthalpy at expansion valve inlet/condenser outlet	kJ/kg
h_{oil}	Lubrication oil specific enthalpy	kJ/kg
k	Coverage factor for uncertainty	
$\dot{M}$	Mass flow rate	kg/s
HSPF	Heating Seasonal Performance Factor	
p_1	Discharging pressure	kPa
p_2	Suction pressure	kPa
p_{atm}	Atmospheric pressure	kPa
p_{diff}	Pressure difference through the SP collector manometer	Pa
p_{stat}	Static pressure inside the SP collector	Pa
Q_1	Heat rejected from the condenser	kJ
Q_2	Heat absorbed in the evaporator	kJ
$\dot{Q}_{comp\_loss}$	Heat loss through the compressor	W
$\dot{Q}_{cool}$	Cooling device capacity (the Calorimeter Method)	W
$\dot{Q}_{heat\_r}$	Heating capacity from refrigerant side	W
$\dot{Q}_{heat\_w}$	Heating capacity from water side	W
$\dot{Q}_{heat}$	Heating capacity of the heat pump	W
$\dot{Q}_{loss}$	Heat loss through the climate chamber	W
R	Universal gas constant	J/(mol.K)
RH	Relative Humidity	%

s_x	Standard deviation of x	
$s_{\bar{x}}$	Type A uncertainty	
T_1	Condensing temperature	K
T_2	Evaporating temperature	K
t_{a_amb}	Average ambient air temperature (outside the climate chamber)	°C
t_{a_in}	Inlet air temperature to the indoor unit	°C
t_{a_out}	Outlet air temperature from the indoor unit	°C
t_{a_room}	Average room air temperature	°C
t_{comp_in}	Refrigerant temperature at compressor inlet	°C
t_{comp_out}	Refrigerant temperature at compressor outlet	°C
t_{cond_in}	Refrigerant temperature at condenser inlet	°C
t_{cond_out}	Refrigerant temperature at condensor outlet	°C
t_{exp_in}	Refrigerant temperature at expansion valve inlet	°C
t_{w_in}	Cooling unit supply water temperature	°C
t_{w_out}	Cooling unit return water temperature	°C
TT_1	Temperature from Climacheck sensor (Outlet of compressor/inlet of condenser)	°C
TT_2	Temperature from Climacheck sensor (Inlet of compressor/outlet of evaporator)	°C
TT_3	Temperature from Climacheck sensor (Outlet of condenser/inlet of expansion valve)	°C
U	Overall heat transfer coefficient	W/(m^2·K)
$u_{\bar{x}}$	Combined uncertainty of $\bar{x}$	
$U_{\bar{x}}$	Overall uncertainty of $\bar{x}$	
v	Speed	m/s
$\dot{V}_a$	Air volume flow rate from the indoor unit	m^3/s
$\dot{V}_w$	Cooling water volume flow rate	m^3/s
W_{BU}	Electric work input to the back-up heater	J or kWh
W_{cal_fan}	Electric work input to the air-cooler fan of the chamber	kJ
W_{comp}	Electric work input to the compressor	kJ
W_{fans}	Sum of electric work of the SP fan and the Calorimeter Method fan	kJ
W_{HP}	Total electric work input to the heat pump	kJ or kWh
W_{in_fan}	Electric work input to the indoor unit fan	kJ or kWh
W_{out_fan}	Electric work input to the outdoor unit fan	kJ or kWh

W_{SP_fan}	Electric work input of the circulation fan (the SP Method)	kJ or kWh
$\dot{W}$	Electric power	W
$w_{\bar{x}}$	Type B uncertainty	
x_r	Refrigerant quality	
x_a	Air water vapour ratio	kg_w/kg_a
$\bar{x}$	Average value of x	

Greek letters

α	Void fraction	
ε	Relative difference	
η	Isentropic efficiency	
φ	Air area-specific heat flow rate	W/ m^2
ρ	Air density	kg/m^3

Subscripts

a	Air
av	Average
comp_in	Refrigerant state, inlet to the compressor
comp_out	Refrigerant state, outlet of the compressor
cond_in	Refrigerant state, inlet to the condenser
cond_out	Refrigerant state, outlet of the condenser
evap_in	Refrigerant state, inlet to the evaporator
evap_out	Refrigerant state, outlet of the evaporator
exp_in	Refrigerant state, inlet to the expansion valve
exp_out	Refrigerant state, outlet of the expansion valve
is	Isentropic process
r	Refrigerant
w	Water

1 Introduction

The original idea of a heat pump could be traced back to the 1850s, when William Thomson, also known as Lord Kelvin, discovered that heat could be "pumped" from a colder environment to a warmer one. As he was developing the theory and the device, he had already foreseen the huge application potential in the field of building service as well as refrigeration . The development of heat pump technology has been stimulated in the recent decades, due to the increasingly high requirement of the indoor environment and the serious restraints of the energy consumption. A considerable amount of improvements has been applied to the technology since it first came into being in the 1940s, in order to have a higher efficiency, a more reliable running range as well as less influence on the surrounding environment. A heat pump could be defined as “a machine or device that moves heat from one location (the ‘heat source’) at a lower temperature to another location (the 'heat sink') at a higher temperature using mechanical work or a high-temperature heat source.”

Heat pumps could be sorted into two main categories based on the prime mover they use, compression heat pump and absorption heat pump. The former uses a compressor operated by mechanical power, mostly from electricity, to increase the pressure and temperature of the refrigerant. It is suitable with distributed systems where the scale is relatively small, for instance, the domestic HVAC and refrigeration systems. For the absorption heat pump, instead of using a compressor, a generator, an absorber and a pump are installed. Furthermore, in addition to refrigerant, another fluid known as absorbent is introduced as a medium to lift the pressure and temperature of the refrigerant, by changing the concentration of the solution. Heat input to the generator plays a key role in this cycle, and it is favourable where there is a cheap and abundant heat source. Normally, the COP value of the absorption heat pump is much lower than that of the vapour compression heat pump, and the benefit from accessing the waste heat suggests that this kind of heat pump is only practical in large scale industrial applications or district heating/cooling.

Heat pumps could also be classified by the different heat sources and heat sinks, as illustrated in Table 1.1 below.

Table 1.1: Classification of heat pumps by heat sources

Heat Pump type	Heat source	Heat sink
Ground Source Heat Pump (GSHP)	Ground	Indoor air / Water
Air Source Heat Pump	Outdoor air/Exhaust air	Indoor air / Water
Water Source Heat Pump	Well/surface/ sewage water	Indoor air / Water

The heat pump market has expanded a lot since the 2000s. Indeed some political incentives contributed to that development in the European Union such as subsidy schemes. Nowadays the installation of heat pumps is not only limited to the building of new houses but concerns the renovation of existing dwellings too. In Sweden, the heat pump market is now mature and as a result of the high sales (as described in Figure 1.1), heat pumps were used in about 50 % of single family houses in 2008.

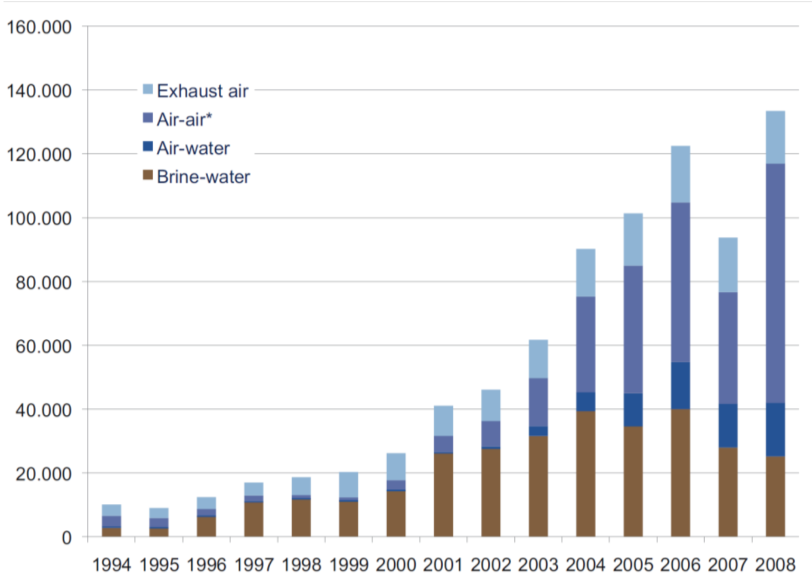

Figure 1.1: Sales of heat pumps in Sweden 1994-2008

As can be seen in Figure 1.2, air-to-air heat pumps represent a significant proportion since that kind of heating system shares 15 % of Swedish single and two family houses according to an estimation by SVEP (Svenska Värmepumpföreningen) and the Swedish Energy Agency .

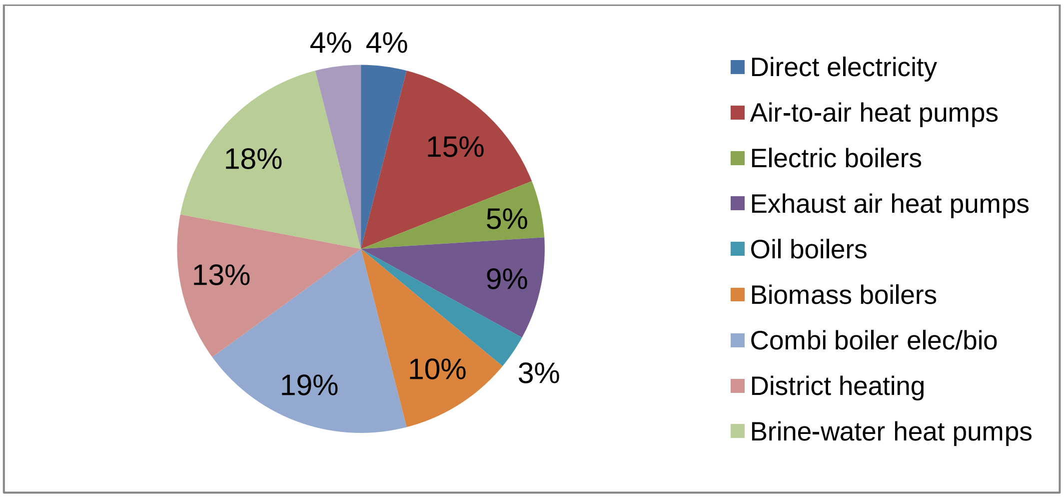

Figure 1.2: Estimated share for different types of heating system in the building stock in single and double family houses in Sweden

Statistics show that air-to-air heat pumps are mostly installed in already existing houses using direct electricity heating since that is the most cost effective option in that case. Air-to-air heat pump installations are also popular in small shops, offices and restaurants.

Accompanied with the growing popularity in the market, problems concerning energy consumption and environment are raised regarding the application of air-to-air heat pumps. It becomes equally important to improve the heat pump performance to be environmentally friendly while satisfying the customer requirements. From this perspective, how to evaluate the COP of air-to-air heat pump in reality becomes important. Meanwhile, from the economic point of view, one of the market barriers to the prevailing market application of air-to-air heat pump is the lack of robust data regarding the performance under real conditions . Normally some parameters related to the function of heat pumps are provided by the manufacturer, and standards have also been developed to regulate the lab testing activities. Nevertheless, in reality the heat pump usually works to some extent below the given data, due to the variety of installation and operating conditions. Hence there are plenty of ECOs (Energy Conservation Opportunities) existing in this industry. Therefore, it is of great significance to develop a proper method for field testing of air-to-air heat pumps in order to achieve HSPF, which could evaluate the quality and efficiency of the unit more rationally.

However, field testing of air-to-air heat pumps is far from mature and still needs improvements, although a number of methods have been developed for this purpose. Currently the challenges are mainly due to the difficulty of properly measuring the air volume flow rate and refrigerant properties during dynamic performance of the heat pump, in particular the defrosting process, and these issues could be of great potential for further investigation.

2 Aim and Scope

The aim of this thesis work is first to review existing air-to-air heap pump COP testing methods and standards, and then to make practical measurements in the lab under different outdoor weather conditions. All the testing methods are supposed to be tested in parallel so that comparisons could be made based on the same preconditions.

The project aims at answering the following questions:

- What are the potentials to improve the studied testing methods?
- Do the test results achieved from each method agree with each other?
- What are the testing uncertainties and limitations of each testing method for field testing?
- How does the air-to-air heat pump practically perform under different outdoor weather conditions?

For each testing method, measurement data are taken from experiments together with the corresponding outdoor weather conditions. The requirements of the testing methods are also investigated.

3 Theory and literature review

3.1 Heat pump working principle

3.1.1 Basic vapour compression process

As illustrated in Figure 3.1, a common vapour compression heat pump is mainly composed of four components: evaporator, condenser, compressor and expansion valve. When the heat pump is running, saturated (or even super heated) refrigerant vapour is sucked into the compressor in which the temperature and pressure are raised; afterwards, it passes through the condenser to exchange heat with the heating medium, which is in the secondary loop. The refrigerant is cooled at the same time until being saturated (or even sub cooled to the liquid region) then it comes to the evaporator by passing through the expansion valve, where expansion occurs ideally without any enthalpy change. This low pressure liquid/vapour mixture refrigerant is then evaporated by absorbing heat from the surrounding cooling media and becomes saturated (or even super heated vapour), when it is followed by another cycle. The difference between heating mode and cooling mode lies in whether heat supply (from the condenser side) or heat removal (from the evaporator side) is required. This will determine whether the indoor unit is the evaporator or the condenser.

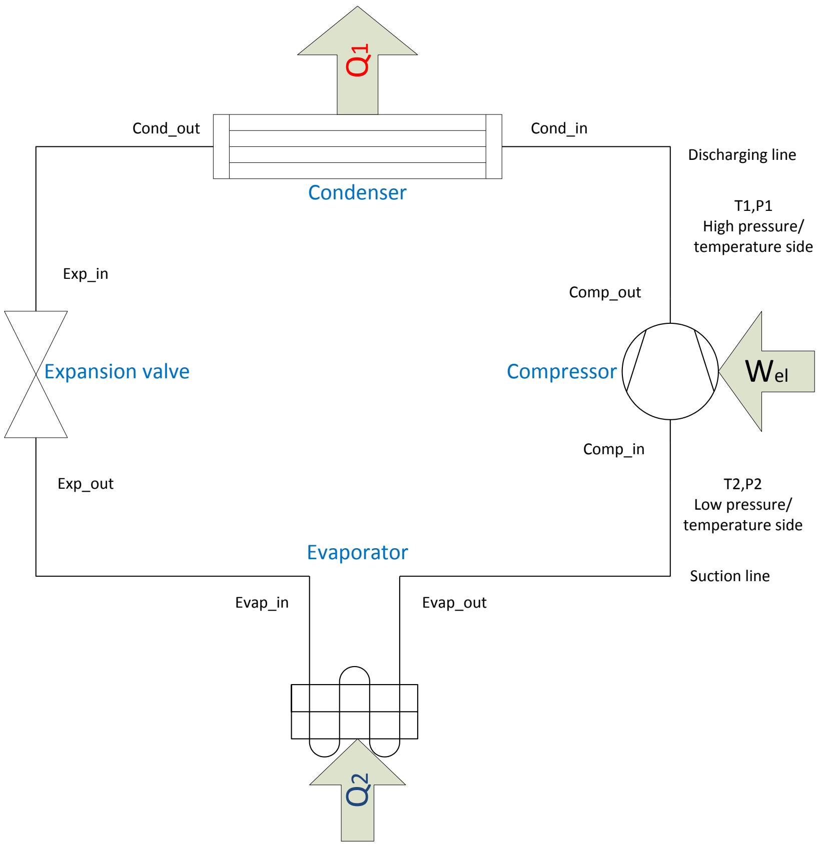

Figure 3.1: Principle scheme of heat pump

The basic vapour compression process assumes an adiabatic expansion and an isentropic compression as well as neither superheating nor sub cooling. This whole process can be plotted in the p-h and T-s diagram as can be seen in Figure 3.2 and Figure 3.3.

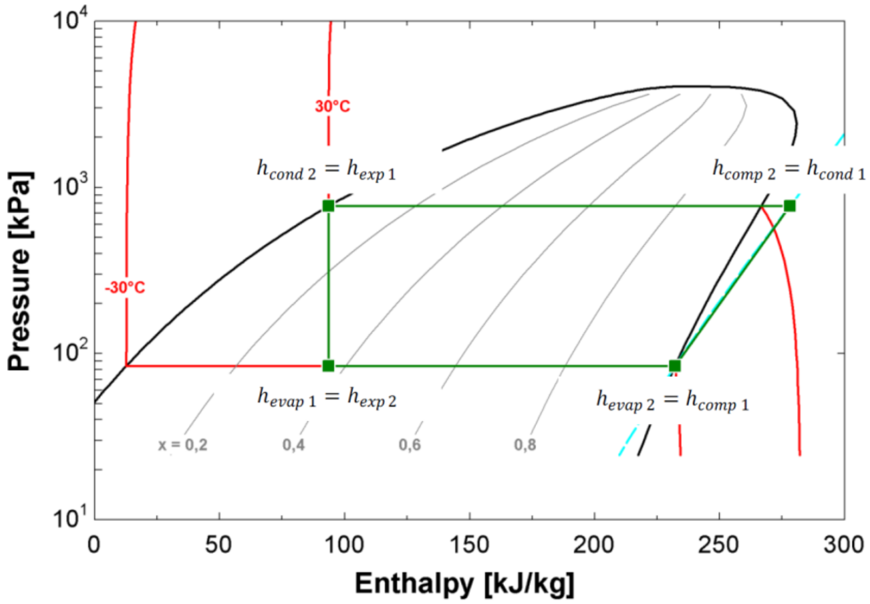

Figure 3.2: A basic vapour compression process illustrated on a p-h diagram

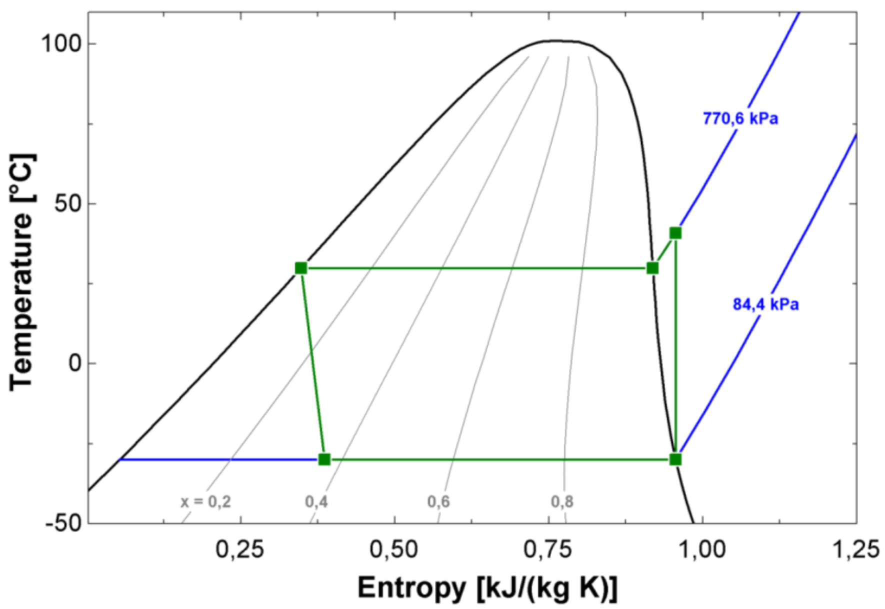

Figure 3.3: A basic vapour compression process illustrated on a T-s diagram

The Carnot cycle represents the ideal (theoretical) case of a process working between two constant temperatures. The main differences with the basic vapour compression process are the isentropic expansion and the isentropic compression as it can be observed in Figure 3.4.

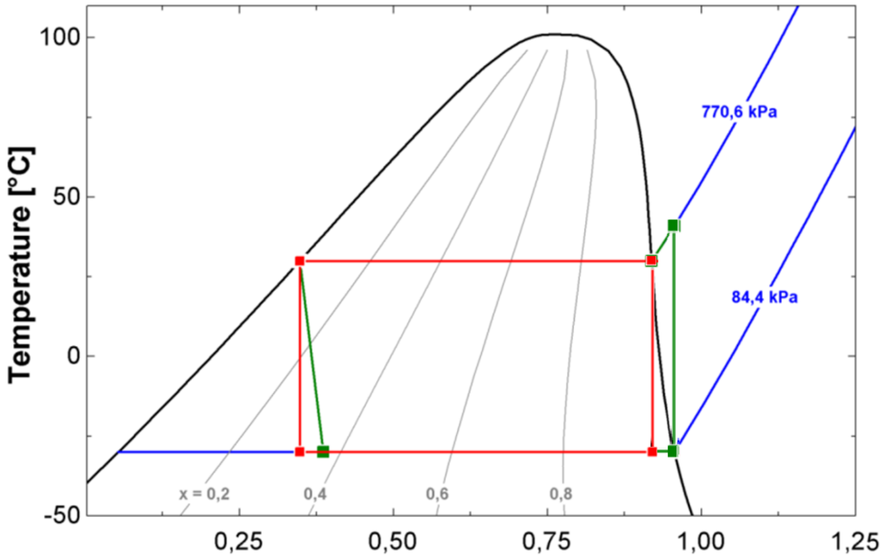

Figure 3.4: A Carnot process (red lines) overlaid a basic vapour compression process (green lines) on a T-s diagram

3.1.2 The defrost cycle

Defrosting is one of the features of an air-source heat pump. Indeed, if the outdoor temperature falls to near (or below) 0°C, the moisture contained in the air passing through the outside heat exchanger will condense, leading to a frost layer on the coil. That phenomenon decreases the efficiency of the heat exchange between the outside air and the refrigerant circulating in the heat pump pipes. Therefore frost must be removed at some point. Different techniques exist to reach that goal such as electric heating, hot gas by-pass and reverse cycle. For the first method, an electric heater is incorporated in the outdoor coil but it is rarely used nowadays. For the hot gas by-pass method, the superheated refrigerant from the compressor is directly passed into the evaporator, skipping the condenser and the expansion device. So this sends hot gas to the outdoor coil to melt the frost. For the reverse cycle, by means of a valve, the refrigerant flow is reversed meaning that the heat pump is working in cooling mode. Therefore, the former evaporator becomes the condenser and vice-versa. Besides, for air-to-air heat pumps, the indoor fan is stopped to avoid blowing cold air into the room. For both methods, during defrost cycle, the outdoor fan, which normally blows cold air over the coil, is shut off in order to melt the frost faster.

Two different methods are used to determine when the unit should go into defrost mode: demand-defrost and time-temperature defrost. In order to detect frost accumulation on the outdoor coil, the control system based on the demand monitors airflow, refrigerant pressure, air or coil temperature and pressure differential across the outdoor coil. Time-temperature defrost is less sophisticated since it starts and ends by either a preset interval timer or a temperature sensor located on the outside heat exchanger. The frequency at which that defrost cycle is initiated depends both on the climate and on the design of the system..

3.1.3 Refrigerants

Another key component of a heat pump is the refrigerant. One of the most important properties of a refrigerant is the saturation pressure/temperature characteristics.

Several parameters are to be taken into account when choosing a refrigerant for heat pump applications. Thus, the refrigerant will ideally:

- be non-toxic (for health and safety reasons)
- be non-flammable (to avoid risks of fire or explosion)
- operate at modest positive pressures (to minimize the pipe and component weights and avoid air leakage into the system)
- have a high vapour density to minimise the compressor capacity and pipe diameters
- be environmentally friendly
- be cheap to produce

Of course, in practice the choice of a refrigerant is a compromise.

Most heat pumps previously used R22 until ozone depletion concerns arose in the 1980s. Due to Montreal protocol, that refrigerant has to be phased out by 2020. Therefore nowadays other refrigerants with lower impact on the ozone layer are being used. R407C (mixture between R-32, R-125 and R134a) offers a wide application range. Despite its higher global warming potential, the larger manufacturers are now adopting R410A for heat pumps below 20 kW due to smaller components and marginal gains in efficiency. Additionally R290 is being used. Because of its good environmental characteristics and high efficiency, the idea of using CO2 as a refrigerant is becoming popular.

During defrost the heat pump is not delivering any heat and therefore unnecessary defrost cycles lead to a reduced general performance of the heat pump over a long period. As a consequence, the demand-frost method is generally more efficient since it starts the defrost cycle only when it is required.

Furthermore, the heat produced by the reverse cycle defrost is lost to the outdoor ambient, reducing the performance of the air source heat pump. Besides the indoor unit is cooling down a bit the room during defrosting inducing afterwards supplemental heat consumed to temper indoor air during the defrost. Indeed, despite the fan is not working, passive heat transfer occurs between hot room air and cold condenser leading to heat removed.

Thus, defrosting duration and frequency are among the most important factors influencing the performance of a heat pump.

3.2 Performance of a heat pump

3.2.1 COP and HPSF

A better knowledge of the factors influencing the heat pump is needed to develop and broaden the use of heat pumps. COP (Coefficient of performance) is one of the most important factors to evaluate the function of the heat pump; it is defined as the ratio between the useful heat delivered and the work supplied to the heat pump. It can be applied both in the heating mode and in the cooling mode. The corresponding equations are equations (3.1) and (3.2) below. According to the first law of thermodynamics, equation (3.3) can be stated.

$$COP_1 = COP_{heat} = \frac{Q_1}{W_{HP}} \qquad (3.1)$$

$$COP_2 = COP_{cool} = \frac{Q_2}{W_{HP}} \qquad (3.2)$$

$$Q_1 = Q_2 + W_{HP} \qquad (3.3)$$

Taking COP_1 for example, the expression could also be transformed into

$$COP_1 = \frac{Q_1}{Q_1 - Q_2} \qquad (3.4)$$

Suppose the heat pump is working at the maximum efficiency, which is Carnot efficiency, equation (3.5) is satisfied accordingly, so that the Carnot COP could also be expressed as equation (3.6) for COP_{1_c} and the corresponding COP_{2_c} is equation (3.7).

$$\frac{Q_1}{T_1} = \frac{Q_2}{T_2} \qquad (3.5)$$

$$COP_{1\_c} = \frac{T_1}{T_1 - T_2} \qquad (3.6)$$

$$COP_{2\_c} = \frac{T_2}{T_1 - T_2} \qquad (3.7)$$

HSPF (Heating Seasonal Performance Factor) is widely accepted as the standard to evaluate the efficiency of heat pump in heating performance, especially for the air source heat pump; by contrast, SEER (Seasonal Energy Efficiency Ratio) is applied in the cooling mode. HSPF is a ratio of heat energy delivered to the electricity energy supplied to the unit.

However, HSPF is not completely identical to the COP value of the heat pump. On the one hand, the COP value is normally disregarding the time span, mainly heating and electricity capacity are compared. However, HSPF concerns more about the average value and the accumulating effect during some definite period of time, preferably a few months or one season, so as to give a more reliable judgment of the efficiency of the heat pump. On the other hand, HSPF takes into consideration the influence from the variable conditions of the heat source, which could significantly affect the performance of the air source heat pump, so it makes more practical sense to introduce HSPF to evaluate the heat pump efficiency in addition to the COP value.

3.2.2 System boundaries

While calculating the HSPF of a heat pump, it is also necessary to clearly define what the system boundary is.

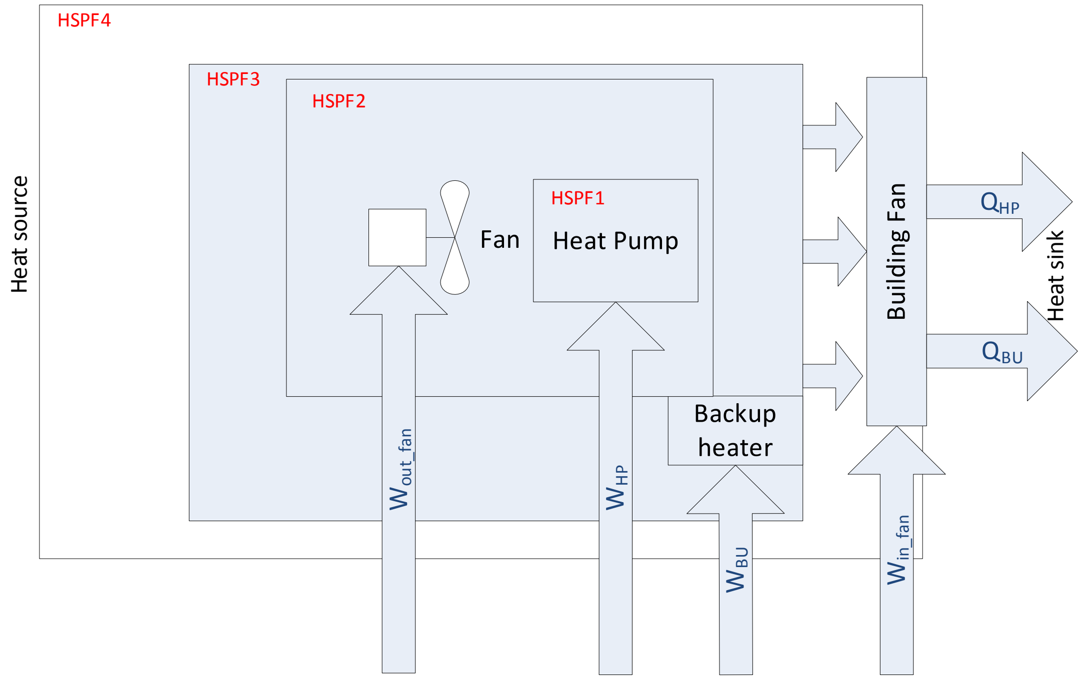

Figure 3.5: System boundaries for heat pump in heating operation

Because air to air systems have different architectures than water based systems, a specific nomenclature has been built by SEPEMO as shown in Figure 3.5 and in equations (3.9) to (3.13)

$$\mathrm{HSPF} = \frac{\sum \mathrm{Q}}{\sum \mathrm{W}} \qquad (3.9)$$

$$\mathrm{HSPF}_1 = \frac{\mathrm{Q}_{\mathrm{HP}}}{\mathrm{W}_{\mathrm{HP}}} \qquad (3.10)$$

$$\mathrm{HSPF}_2 = \frac{\mathrm{Q}_{\mathrm{HP}}}{\mathrm{W}_{\mathrm{out\_fan}} + \mathrm{W}_{\mathrm{HP}}} \qquad (3.11)$$

$$\mathrm{HSPF}_3 = \frac{\mathrm{Q}_{\mathrm{HP}} + \mathrm{Q}_{\mathrm{BU}}}{\mathrm{W}_{\mathrm{out\_fan}} + \mathrm{W}_{\mathrm{HP}} + \mathrm{W}_{\mathrm{BU}}} \qquad (3.12)$$

$$\mathrm{HSPF}_4 = \frac{\mathrm{Q}_{\mathrm{HP}} + \mathrm{Q}_{\mathrm{BU}}}{\mathrm{W}_{\mathrm{out\_fan}} + \mathrm{W}_{\mathrm{HP}} + \mathrm{W}_{\mathrm{BU}} + \mathrm{W}_{\mathrm{in\_fan}}} \qquad (3.13)$$

It should be noticed that the way to divide the system boundary as above is not quite stringent, even though it makes sense with the practical testing case in this specific project. Theoretically, distinctions among heat pump, heat pump system and heat transfer system should be made. Further information regarding this could be achieved from Fahlén's references . Therefore, it is essential to make clear whether auxiliary devices should be taken into account according to different system boundaries; otherwise, the results are ambiguous and unable to be compared under the same fundaments. Furthermore, measurements for different system boundaries also call for different instruments and setups to get all the energy input and output.

3.3 Existing COP testing methods for air-to-air heat pumps

The experienced in situ air-to-air heat pump COP testing methods could be summarized into two big categories, external methods and internal methods.

3.3.1 External methods

External method is based on the traditional measurements of heat transfer on the air side, mainly air flow rate, inlet and outlet temperature, corresponding air properties such as density and specific heat capacity. Based on these data, heating capacity could be calculated according to equation (3.14). In heating mode, relative humidity has very little effect in enthalpy variation, so equation (3.14) can be replaced by equation (3.15).

$$\dot{Q}_{heat} = \dot{V}_a \cdot \rho_{\_a} \cdot \left(h_{a\_out} - h_{a\_in}\right) \qquad (3.14)$$

$$\dot{Q}_{heat} = \dot{V}_a \cdot \rho_{\_a} \cdot c_{p\_a} \cdot \left(t_{a\_out} - t_{a\_in}\right) \qquad (3.15)$$

It is possible to measure heating capacity using the external method to achieve an uncertainty less than 1-2 %. However, in practical installations, normally the expected uncertainty is higher than 5 %, due to restriction while maintaining sufficiently stable conditions and achieving laboratory level accuracies . The biggest challenge is in the air flow rate measurements. Efforts have been taken to investigate the accuracy regarding this issue, and mainly two different testing methods have been developed for external measurements, named as SP Method and Tokyo Gas Method as further explained below.

The SP Method has originally been developed by SP Sveriges Tekniska Forskningsinstitut (Method 1721) for the field testing of non-ducted air-to-air heat pumps. It can be applied both in the heating mode and in the cooling mode with an expected uncertainty below 10 % . The in situ setups are shown in Figure 3.6.

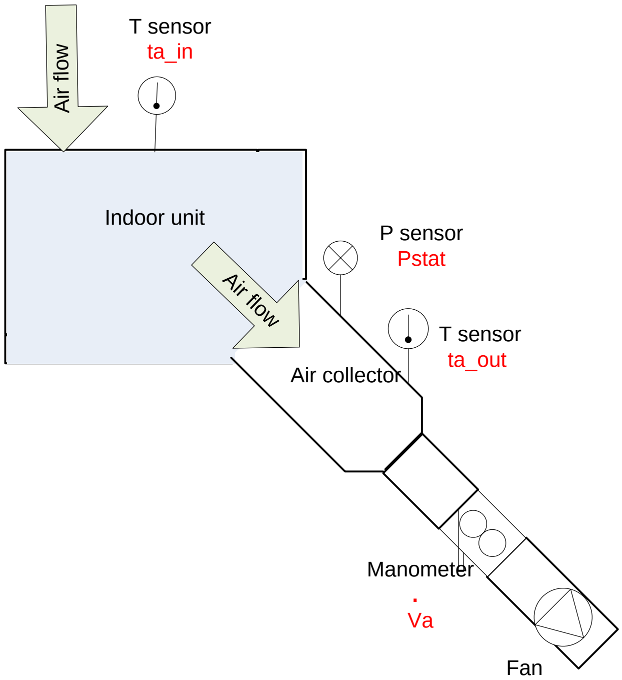

Figure 3.6: Equipment setup for the SP Method

According to the requirements specified in the report , at least four temperature sensors should be applied at both inlet and outlet of the heat pump, they should be evenly distributed in order to get a good average value. Reaching a steady state is extremely important during the testing; as suggested, the test shall not start earlier than 5 minutes after defrosting and should last for at least 10 minutes[1]. During the testing process, the static pressure is to be kept at 0 Pa by adjusting the speed of the SP circulation fan, in order to minimise the influence from the duct and manometer itself. The permitted uncertainty of measurement is shown in Table 3.1.

Table 3.1: Permitted uncertainty of measurement

Magnitude of measurement	**Permitted total uncertainty of measurements**
Air temperature	±1 K
Air flow	±5 %
Atmospheric pressure	±10 hPa
Electric power, electric energy	±4 %
Time	±0.5 %

[1] In the actual tests, the method was used for continuous measurements and the outdoor conditions were varying.

In this method, all the connecting parts shall be well sealed to prevent any air leakage. Insulation of the duct and collector is also very important so that heat loss is minimised; the average value is taken for the inlet and outlet of air temperature, preferably an additional temperature sensor is installed close to the manometer to get the air properties such as density and specific heat capacity; volume flow rate is achieved from a manometer, which takes into consideration the pressure difference of the two testing points. Heating capacity is then calculated referring to equation (3.15). Meanwhile, a power meter is connected concerning the definition of system boundary; here the power meter can either be an electricity power meter or an integrating electric energy meter.

For instance only the power input to the heat pump itself is considered within the system boundary (see Section 3.2.2), and the electricity input is measured as $\dot{W}_{HP}$, then the COP value is calculated as equation (3.16).

$$\mathrm{COP} = \frac{\dot{Q}_{heat}}{\dot{W}_{HP}} \qquad (3.16)$$

The Tokyo Gas Method is another external testing method which was developed by Tokyo Gas Co.,Ltd, Japan. This method is proposed for measurements of split-type package air conditioners, which are very popular in Asian countries like Japan and China.

The heating/cooling capacity measurement is principally based on determining the difference of heat flow before and after the indoor unit. The equipment setup is illustrated in Figure 3.7. The inlet and outlet areas are divided into a certain number of small regions, within which air temperature, humidity as well as velocity are measured. Air velocity of each point is measured by anemometer and then compared with the value measured at the centre position to get a certain ratio. A velocity distribution curve could then be interpolated. Thereafter the air volume flow rate is achieved from the integration of air velocity over the whole area, separately for the inlet and outlet of the indoor unit, and compared with manufacturer's data, in order to get a correction factor which is applied to modify the air velocity measured at each point. Finally, air volume flow rate of each formerly divided small region is further determined. Together with the air property, which is from the tested temperature and humidity, energy flow rate into or from each small region is calculated, as illustrated in equation 3.17.

$$\varphi_a = v_a \cdot \rho_a(t_a) \cdot h_a(t_a, x_a) \qquad (3.17)$$

Integrations are made regarding all the energy flows achieved over the total inlet and outlet area of the indoor unit. Heating capacity is then presented by equation (3.18).

$$\dot{Q}_{heat} = \iint\limits_{A} \left(\varphi_{a\_in} - \varphi_{a\_out}\right) dA \qquad (3.18)$$

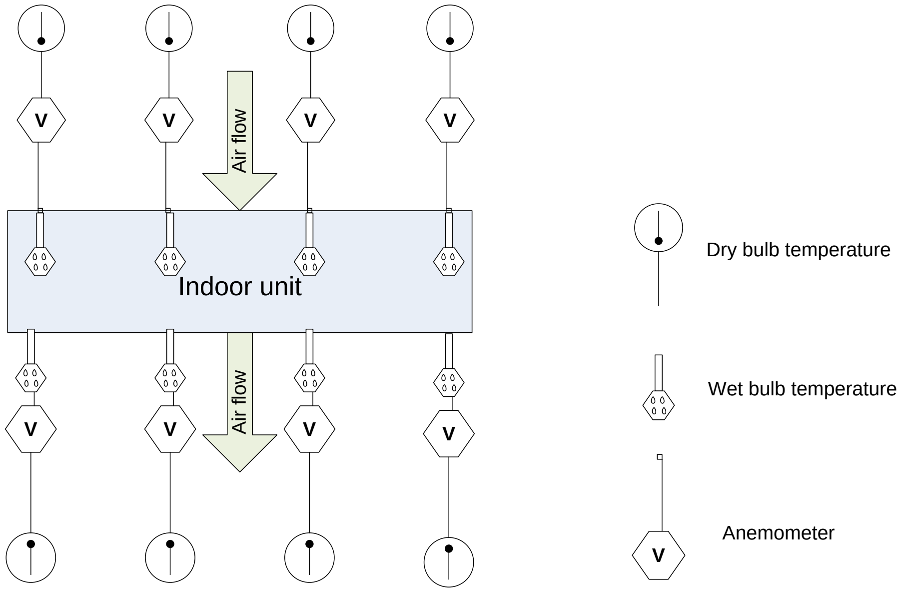

Figure 3.7: Instrument setup for Tokyo Gas Method

The Tokyo Gas Method could be applied not only in heating mode but also in cooling mode, as described in. Experiments were done with the weather conditions recorded both during summer and during winter, so that the SPF could be gotten quite reasonably. There are fewer restrictions concerning the test of air volume flow rate, but the accuracy is relatively low as could be predicted. The measurements could adapt quickly considering load change; however, it is still impossible to get the heat removed during defrosting. Hence it is not the perfect solution for the continuous measurement of HSPF.

3.3.2 Internal methods

Distinguished from external method, the internal method focuses on the refrigerant cycle, measuring the refrigerant temperature and pressure at certain points, so as to get the corresponding enthalpy values, which are applied to calculate the heating capacity. Refrigerant mass flow rate is also necessary for the calculation; it could be achieved in different ways as it will be further explained later. Heating capacity is further determined through equation (3.19)

$$\dot{Q}_{heat} = \dot{M}_r \cdot [h_{cond\_in}(t_{cond\_in}, p_{cond\_in}) - h_{cond\_out}(t_{cond\_out}, p_{cond\_out})] \qquad (3.19)$$

One of the benefits over the external method is the possibility to study the factors which cause the deviation from the expected value more directly, and for this reason, the internal method has the potential to be applied in expert FDD (Fault Detection and Diagnosis) systems.

However, there are also difficulties to have a good value of refrigerant mass flow rate directly, due to the changing of the refrigerant state during operation. Climacheck AB developed a method to calculate the mass flow rate indirectly, with an estimated heat loss factor , while another testing method using Coriolis flow meter has also been developed to overcome this challenge .

The Climacheck Method has been invented by the company Climacheck Sweden AB to analyse the performance of any refrigeration, heat pump or air-conditioning system. It uses a “Refrigeration Performance Analyzer” which focuses on the refrigeration cycle. The method is based on pressure and temperature measurements of the refrigerant as well as the electric power input to the heat pump compressor. Then the mass flow rate of the refrigerant as well as the heating/cooling capacity is determined from thermodynamic calculations.

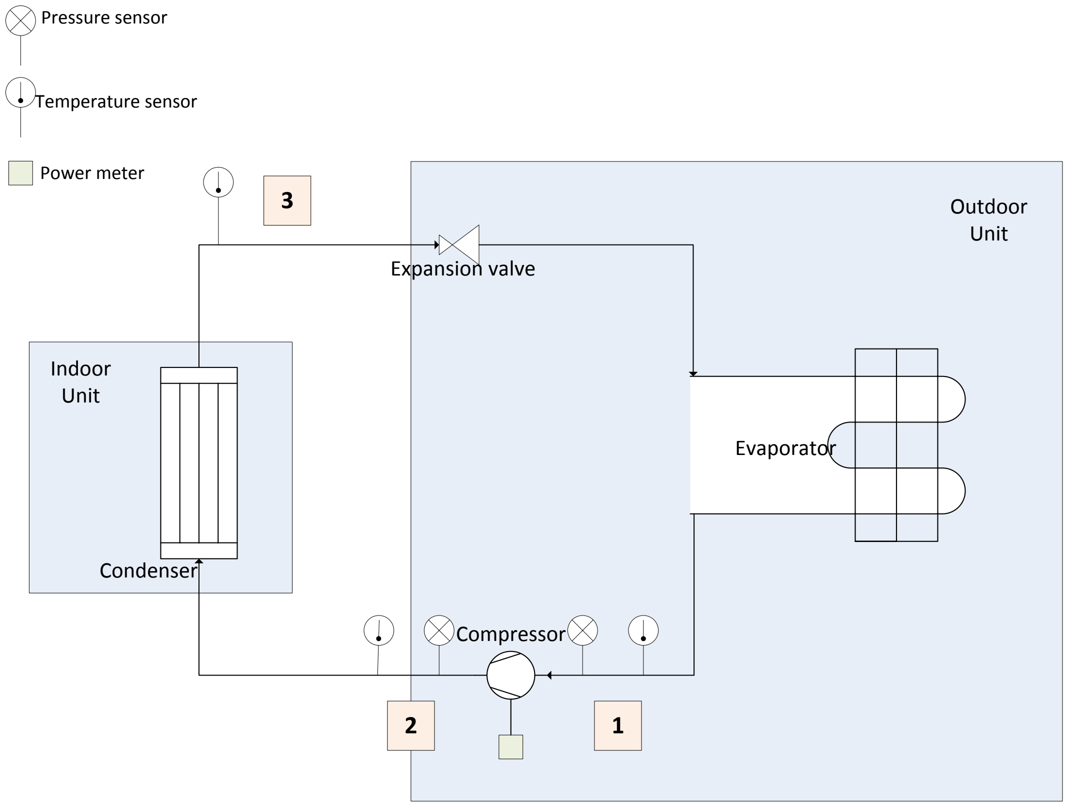

Figure 3.8: Instrument setup for the Climacheck Method

Figure 3.8 shows the instrument setup for the Climacheck Method. The temperature sensor is non intrusive and attached on the surface of the refrigerant pipe with heat transfer compound, aluminium and insulation. It should be installed far enough from the compressor or valves which can lead to a difference between surface temperature and inner temperature of the tube as illustrated in Figure 3.9. Power measurement is only referring to the compressor electricity consumption by connecting current clamps and voltage cables to the corresponding three-phase wires to the compressor. All the sensors shall be connected in accordance with each template which is developed specially for the Climacheck data logger, as in Figure 3.10.

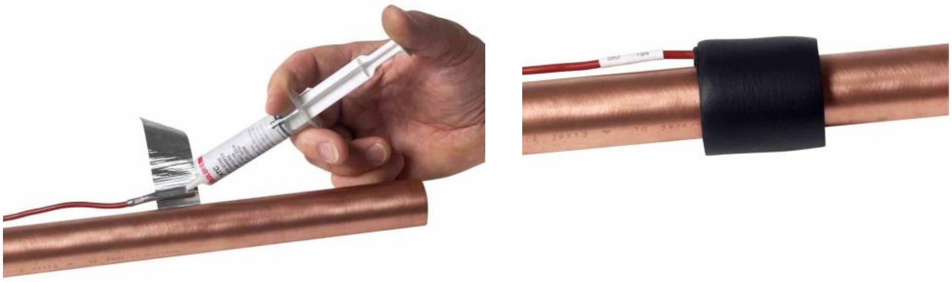

Figure 3.9: Climacheck temperature sensor recommended set up

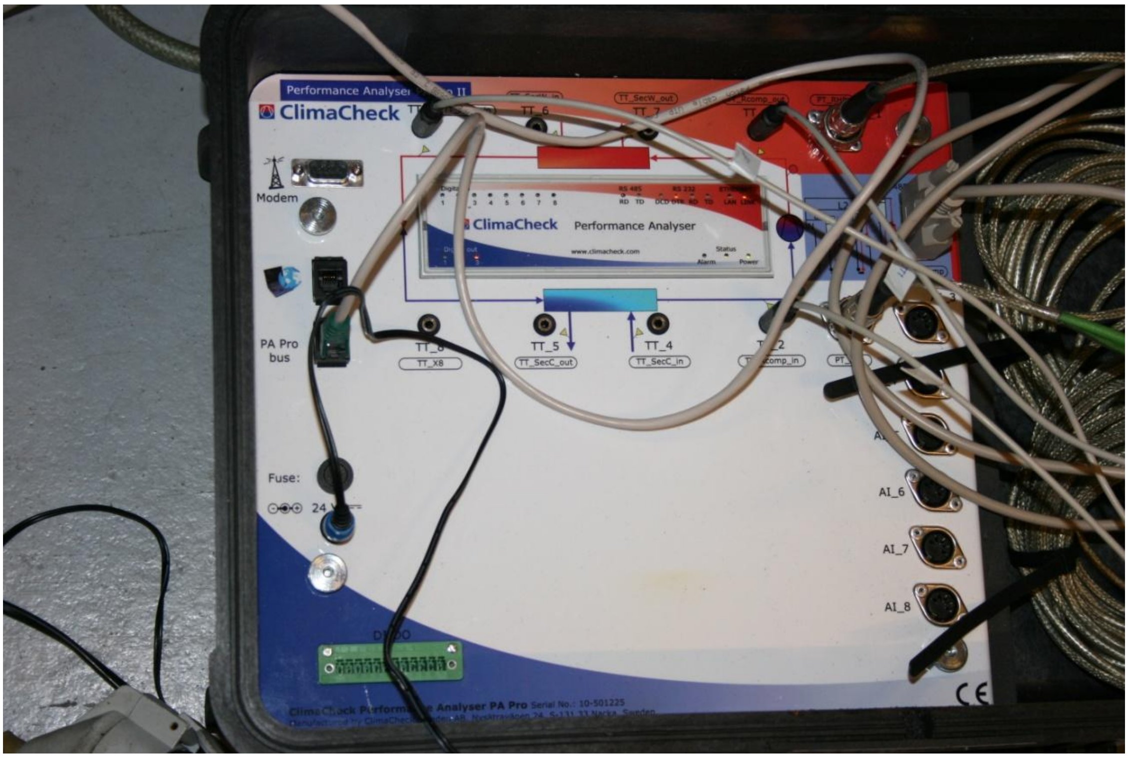

Figure 3.10: Climacheck data logger with the sensors inserted

Based on the measurement, all the points shown in Figure 3.8 could be specified in the p-h diagram, as illustrated in Figure 3.11.

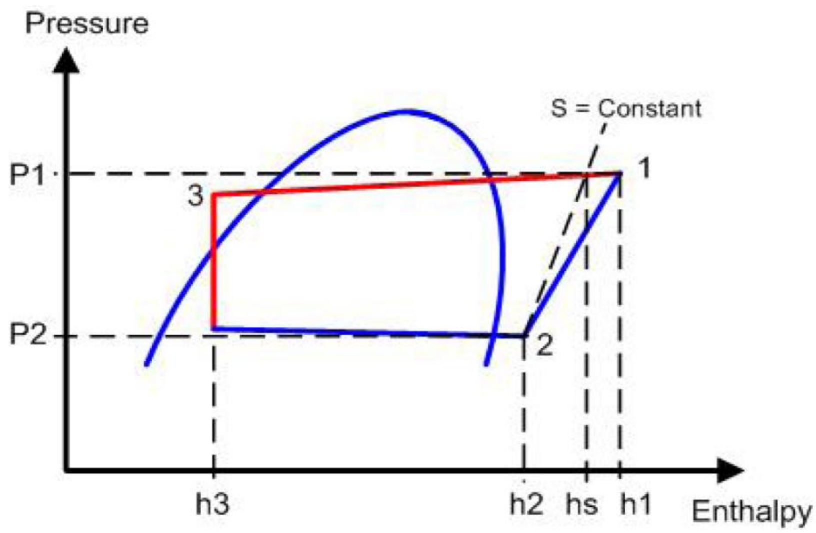

Figure 3.11: Refrigerant state with the corresponding points in p-h diagram

As could be noticed, there is a pressure loss through condenser and evaporator, while there is no pressure sensor at point 3, outlet of condenser. Nevertheless in the liquid region the temperature curve is almost vertical so it would not influence the enthalpy so much if the same pressure level as at point 1 is considered.

Another parameter required is the mass flow rate of the refrigerant. This is achieved in an innovative way with the Climacheck Method. Different refrigerants could be selected in the defaulted template of Climacheck since refrigerant properties are needed for enthalpy calculation. Meanwhile the heat loss factor *f* through the compressor is assumed to be 7 %[2], as demonstrated in Figure 3.12, and then the mass flow rate can be easily determined by thermodynamic calculation using equation (3.20).

$$\dot{W}_{comp} = \frac{\dot{M}_r \cdot (h_1 - h_2)}{(1 - f)} \qquad (3.20)$$

[2] The value of the heat loss factor has been verified for small hermetic compressors for residential heat pumps, for more information, see [19]

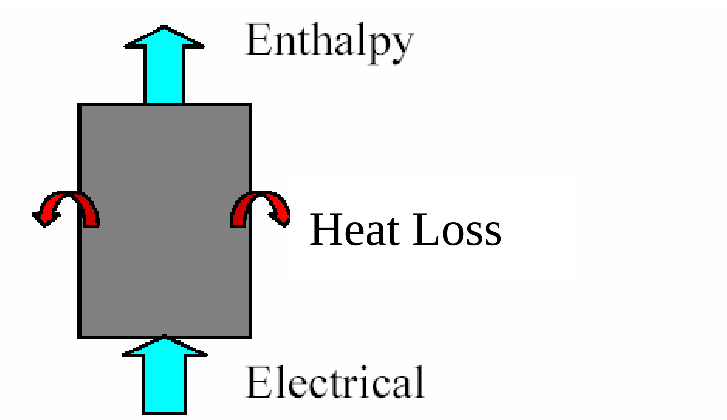

Figure 3.12: Energy flows through the compressor

After $\dot{M}_r$ is determined, equation (3.19) can be applied to calculate the heating capacity. With the Climacheck data logger itself, it is only possible to calculate COP based on the compressor power input, but an additional power meter could also be connected to the heat pump to measure the total electricity consumption so that the reference system boundaries can be created while comparing with the other testing methods.

The Climacheck Method is available neither for testing during defrosting, nor in mixture region due to its limitations to trace the state of point 3. However in normal heating mode, the accuracy is expected to be within 5 % when the conditions are good, this was verified over a wide range of operating conditions of different heat pumps, according to Fahlén and Johansson [20].

The Refrigerant Enthalpy Method gives good solutions to the problems which the Climacheck Method could not manage. This method also focuses on the refrigerant cycle and was first validated by testing an air-to-water heat pump in laboratory with water enthalpy as the reference, and the data from experiment shows that the method is reliable even during defrosting. The method could further be developed to act as a reference to the external methods for air-to-air heat pumps .

The laboratory instrument setup is illustrated as in Figure 3.13. Instead of resorting to the enthalpy change through the compressor to get the mass flow rate; here it is directly measured with an intrusive Coriolis flow meter. An additional role that Coriolis flow meter plays is that it could be applied at the same time to observe the refrigerant density at the specific location. The enthalpies before and after the condenser are determined by measuring the temperature and pressure of the corresponding points. In addition, heating capacity is also achieved from the water cycle, since it is easier and more accurate to test the flow rate of water.

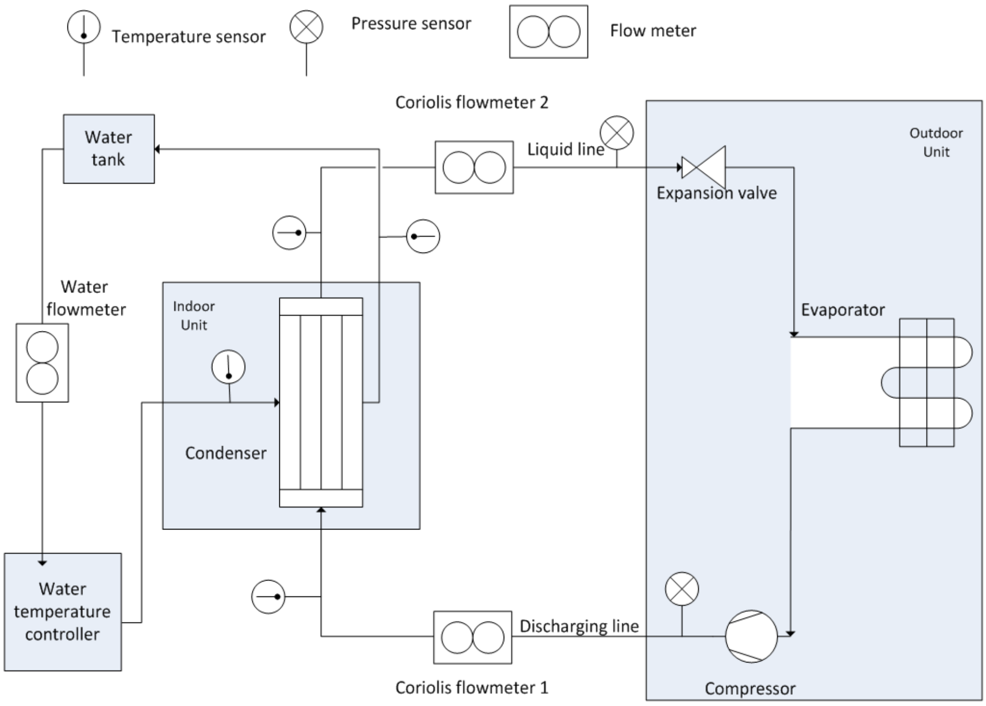

Figure 3.13: Instrument setup for refrigerant enthalpy method

This method works smartly to get the void fraction α and the vapour quality x_r of the refrigerant in the discharging line during defrosting, since it is not sufficient to derive the enthalpy just from the temperature and pressure when the state lies in the liquid vapour mixture region. Together with the x_r-value, the biphasic refrigerant enthalpy could be defined. Detailed calculations are shown in .

The results from the Refrigerant Enthalpy Method also concerns the influence of the oil concentration in the refrigerant flow, here it is assumed to be 2 % and it is the sensible heat that is transferred by the lubrication oil. So finally the heating power is achieved by equation (3.21).

$$\dot{Q}_{heat} = \dot{M}_r \cdot \left[\left(1 - C_g\right) \cdot \left(h_{r\_cond\_in} - h_{r\_cond\_out}\right) + C_g \cdot \left(h_{oil\_cond\_in} - h_{oil\_cond\_out}\right)\right] \quad (3.21)$$

Under laboratory conditions, the heating capacity calculated in the refrigerant cycle could be compared with what is achieved in the water cycle, by using the equation (3.22), the results from the paper suggest that the average relative difference between the water side and the refrigerant side is 1.82 %.

$$\varepsilon = \frac{\sum \dot{Q}_{heat\_r}}{\sum \dot{Q}_{heat\_w}} - 1 \quad (3.22)$$

In which the sum is taken over a period of time.

The main drawback of Refrigerant Enthalpy Method is the difficulty of installing the Coriolis flow meter in situ due to its big size as well as the fact it is intrusive. Therefore, it is not a feasible field test method for both practical and economical reasons.

3.3.3 Calorimeter method

Empirical data with in situ measurement suggests that the deviations between the internal and external methods are within 5 % . Nevertheless, it is not suitable to have either of the testing methods introduced above to play the role of the reference. Regarding the evaluation problem, the European standard EN 14511 applies for “Air conditioners, liquid chilling package, and heat pumps with electrically driven compressors for space heating and cooling” [19]. It specifies relevant test conditions and methods for those devices in order to determine their COP (Coefficient of Performance). This method is carried out in a climate chamber under certain lab conditions. Energy balance serves as the theoretical fundaments and it could exactly follow the heat pump performance particularly in steady state. This method, named as Calorimeter Method, could be fully accepted as the reference while having other methods tested simultaneously.

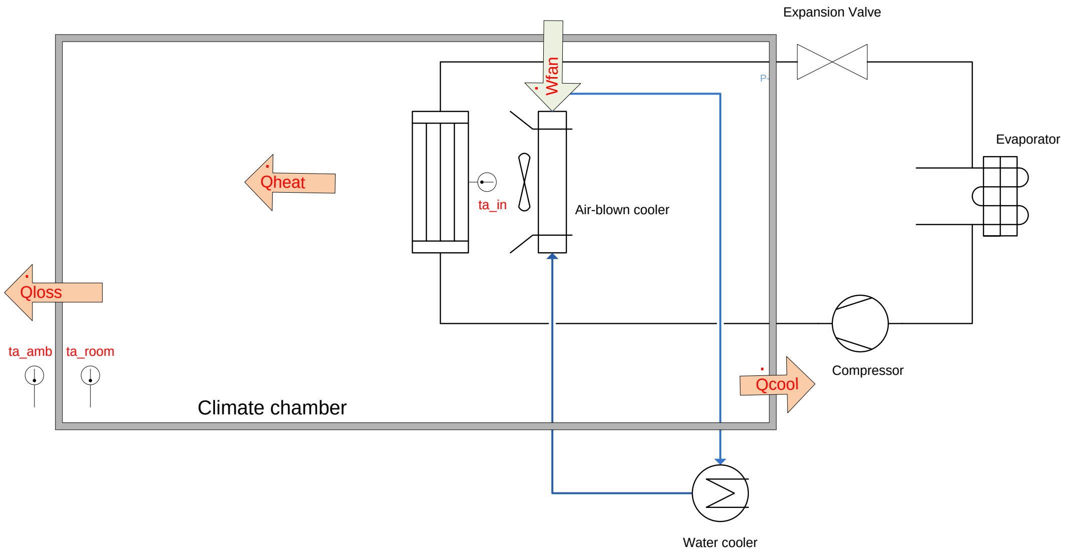

Figure 3.14: Equipment setup and energy flow of indoor compartment for the Calorimeter Method

For lab testing with the Calorimeter Method, certain conditions have to be satisfied according to the standard; for heating capacity measurement, louvers and fan speed shall be set for maximum air flow while the inlet air to the heat pump should be kept at 20°C. There are specific requirements for the deviation of temperature during steady state and defrosting period. It is also available to take into consideration the effects from dynamic process, but a testing period at least as long as 260 minutes is needed, normally divided into 3 stages, named as preconditioning period, equilibrium period and data collection period. The heat content in the climate chamber should return to the original value at the beginning of the test, so that heat change during dynamic process can be covered in the whole testing period (heat storage is constant in other words). More requirements can be found in [19]. The instrument setup and energy flow are shown in Figure 3.14. Based on the energy balance assuming the heat storage change is 0 J, the heating capacity is achieved from equation (3.23), with the absolute value considering the energy flow directions.

$$\dot{Q}_{heat} = Q_{cool} + \dot{Q}_{loss} - \dot{W}_{fans} \tag{3.23}$$

Here room temperature and ambient temperature outside of the chamber is measured. Heat loss through the chamber is then determined together with the UA value of the chamber, which could be easily gotten from a separate experiment . Power input to the cooling fan is measured with a power meter and the cooling capacity from the cooling device is calculated with equation (3.24), by measuring the inlet and outlet temperature of cooling water, besides, the volume flow rate of cooling water is also measured.

$$\dot{Q}_{cool} = \rho_w \cdot C_{p\_w} \cdot \dot{V}_w \cdot \left(t_{w\_out} - t_{w\_in}\right) \tag{3.24}$$

$$Q_{cool} = \int_0^{\tau} \dot{Q}_{cool} d\tau \tag{3.25}$$

4 Methodologies

The aim of this thesis work is first to review the existing test methods and standards, then make in situ setups to practically measure the COP of the target heat pump with different testing methods mentioned above, Mainly three methods were used in the work, the Calorimeter Method, the SP Method as well as the Climacheck Method. The testing methods are tested in parallel under different outdoor air conditions. Results are compared and further analyzed in order to figure out the advantages and disadvantages of each method, while the operation characteristic is also studied at the same time, in order to suggest an optimal testing method and theories which is suitable for practical field testing of HSPF.

4.1 General set up

The heat pump tested is an air-to-air heat pump Bosch 6.0AA. It is a split system (outside and inside units shown in Figure 4.1). The refrigerant cycle could be seen from Figure 4.2.

Figure 4.1: Outdoor unit (above) and indoor unit (below) of BOSCH 6.0AA heat pump

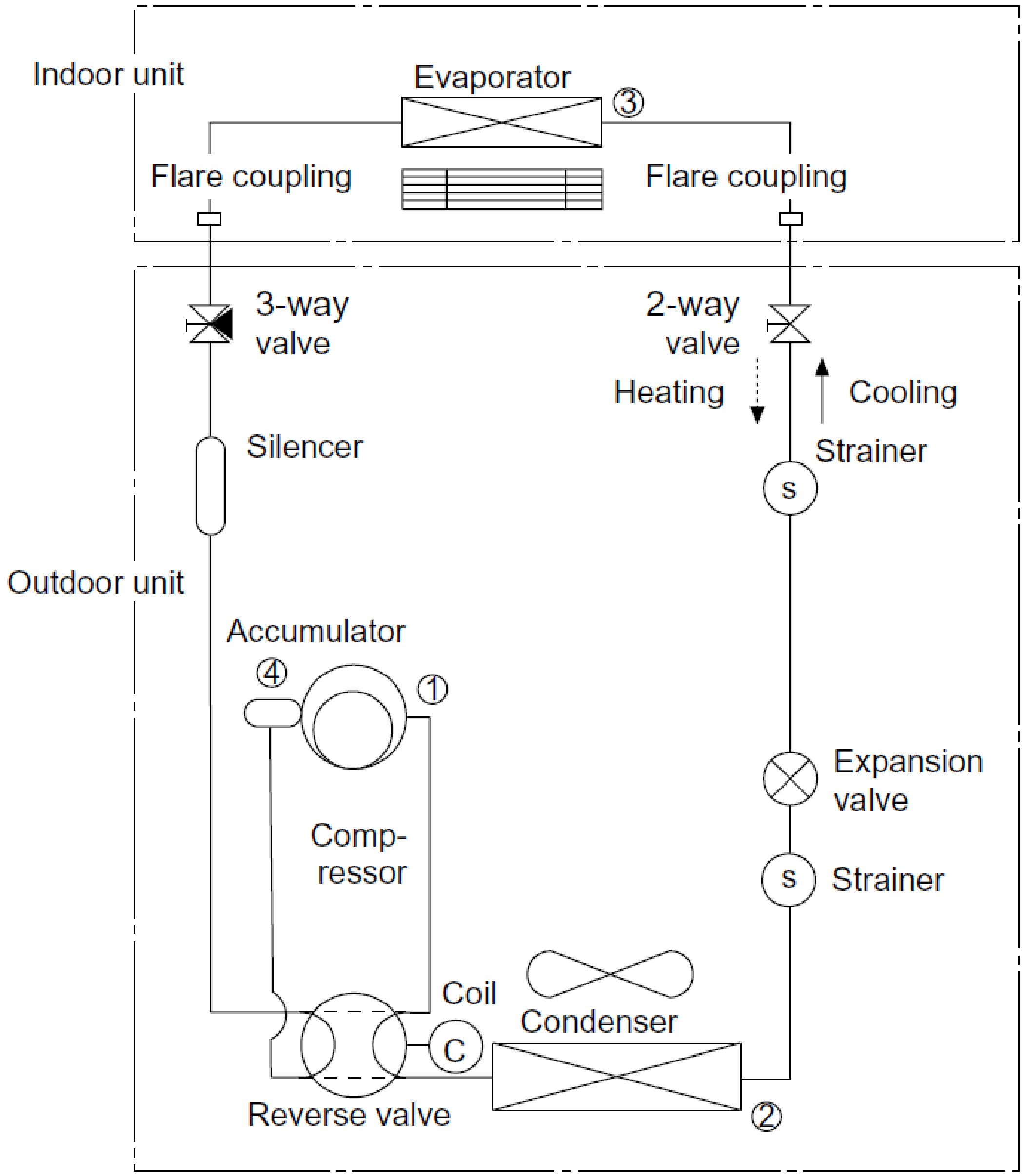

Figure 4.2: Refrigerant cycle schematic diagram of the tested unit

The indoor unit (including mainly evaporator and indoor fan) is wall mounted inside the climate chamber. The outdoor unit is placed on the roof, so real outdoor conditions will be taken into consideration during testing. The units are joined together by pipes carrying the refrigerant R410A. The heat pump uses reverse cycle (thanks to a 4-way valve) to handle both heating and cooling, nevertheless only the heating mode will be investigated in the experiments.

The manufacturer’s specification is 6.0 kW as the maximum heating capacity and 4.0 kW as the rated capacity, at which the heat pump is designed to work. . As required for the Calorimeter Method, the indoor temperature will be set at the maximum which can be set up (32°C) on the heat pump controller. This is to ascertain that the heat pump is working constantly at maximum capacity. The manufacturer’s data from the outdoor unit label are shown in Figure 4.3. The detailed specification of the testing unit is further described in Appendix I.

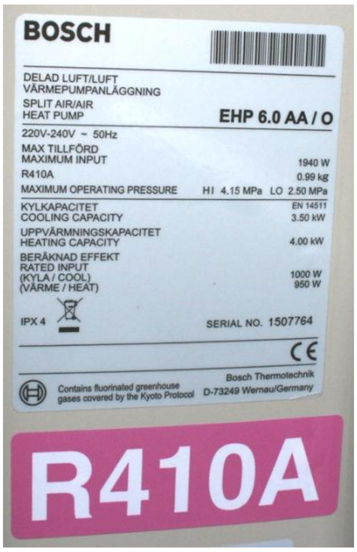

Figure 4.3: Heat pump properties as listed on the outdoor unit label

The manufacturer’s data include the performance curve of the heat pump in heating mode from a theoretical point of view. The standard conditions used are given in Table 4.1, and Figure 4.4 shows the performance curve when outdoor temperature is varied outside the standard conditions,

Table 4.1: Standard rating condition from the manufacturer, according to EN14511

Heating mode	*Indoor unit*	*Outdoor unit*	
	Dry bulb temperature	Dry bulb temperature	Humidity
	20°C	7°C	84 %

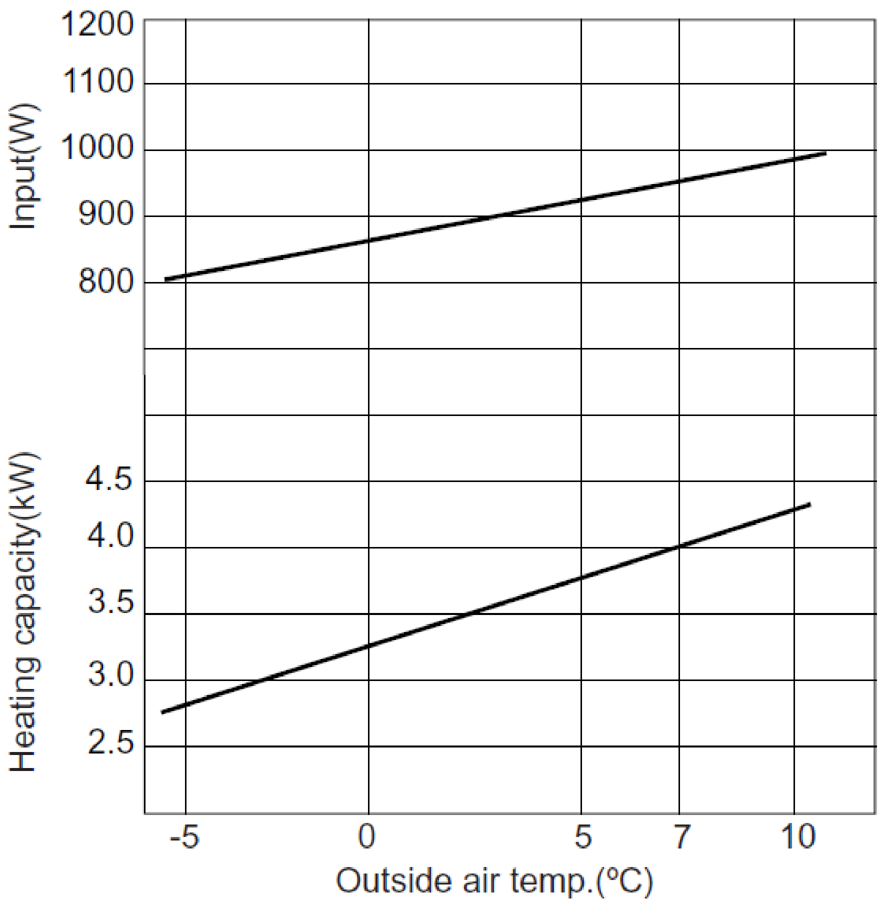

Figure 4.4: Performance of BOSCH 6.0AA Heat Pump in heating mode with 20°C indoor temperature and variable outdoor temperature

The Calorimeter Method will be carried out in a climate chamber (indoor compartment). On the contrary to the requirements, no outdoor compartment will be used. Instead the outdoor unit is located outside, and thus it is subjected to real outdoor conditions.

Since for heating the relative humidity does not play a big role, the change of relative humidity in the indoor room will be neglected (so no humidifier will be installed). The cooling device uses a water system to cool down the air and an air fan as shown in Figure 4.5 and Figure 4.6. The inlet water temperature can be adjusted as well as the water volume flow rate giving a certain cooling capacity to the device. The goal is to keep the unit inlet air temperature at 20°C (listed as a requirement for the EN14511 rating Method). The inlet temperature will never reach the set temperature (32°C) so the heat pump will be constantly running.

As said before, the heat pump behaviour depends on the outdoor conditions (mostly temperature and humidity) and since it is hard to obtain the same outdoor conditions at different times it has been decided to carry out the three methods at the same time. Since the Climacheck method is an internal method, it does not have any influence on the Calorimeter Method. However, the SP Method requires the use of an electric fan so the input to that device should be added to the energy balance carried out in the Calorimeter Method.

The combination of all the three methods as it was tested is shown in Figure 4.7.

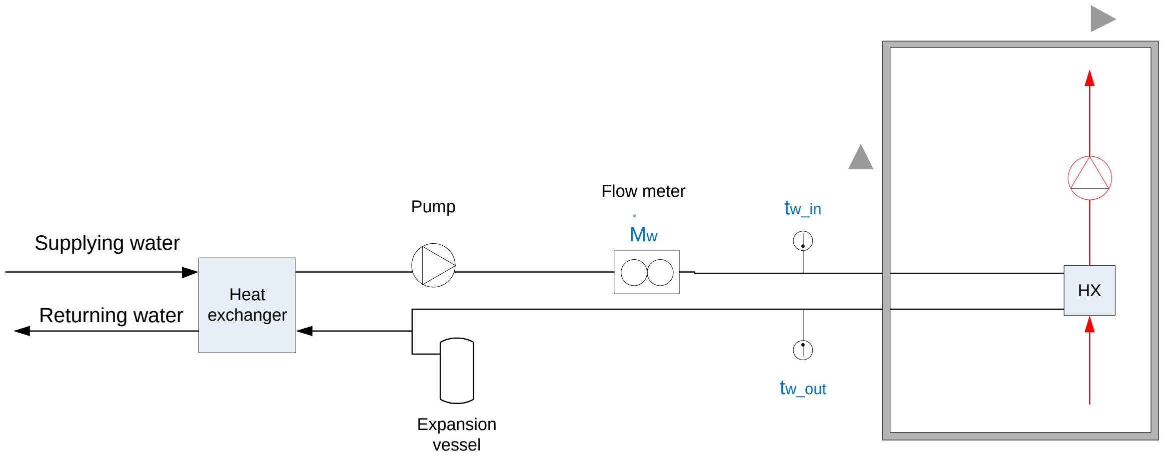

Figure 4.5: Cooling device sketch

Figure 4.6: Cooling device used for testing

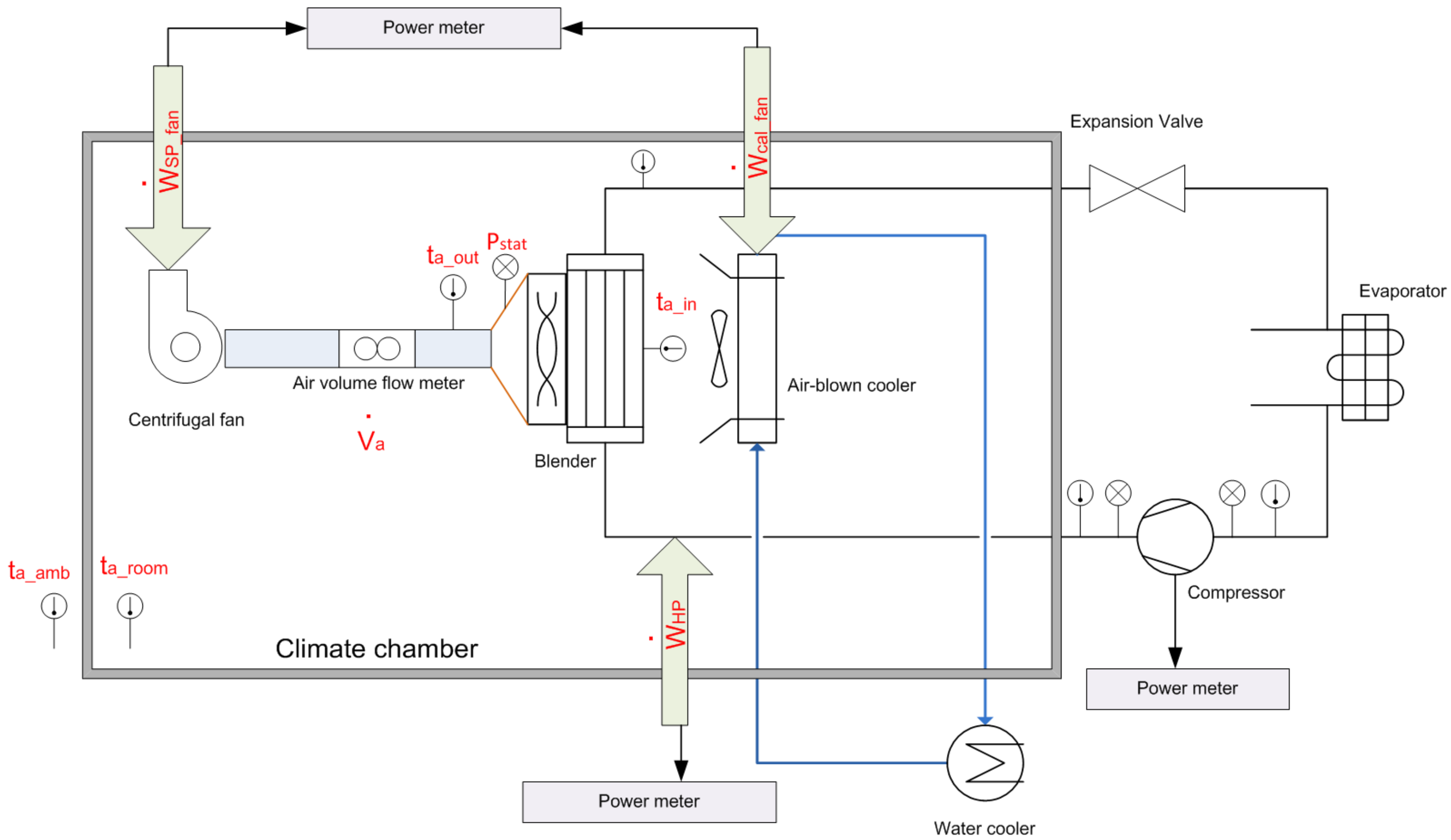

Figure 4.1: Actual set up for the testing with the combination of Calorimeter, Climacheck and SP methods

4.2 Equipment/instrumentation used for testing

The equipment needed (excluding instrumentation) for the test is summed up in Table 4.2 and some pictures of it can be observed in Figure 4.8, Figure 4.9 and Figure 4.10.

The instrumentation needed for each method is described in Table 4.3, Table 4.4 and Table 4.5. When several temperature sensors were used, the average was taken as the measured value for calculation. Figure 4.11, Figure 4.12, Figure 4.13, Figure 4.14, Figure 4.15 and Figure 4.16 illustrate that instrumentation.

The purpose of using a power meter for SP centrifugal fan is related to the combination of that method with the calorimeter room method. Since that energy flow based method just aims at calculating the electric power input into the chamber, the measurement of $\dot{W}_{SP\_fan}$ has been combined with the measurement of $\dot{W}_{cal\_fan}$. Consequently, this set up enables to limit the number of required power meters for testing.

Furthermore, in the case where several methods need to measure a certain temperature, only one set of temperature sensors were used of course.

Table 4.2: List of equipment needed for the experiments (excluding sensors)

Method	Component	Model
Calorimeter	Cooling fan	
	Air-water heat exchanger	
	Water pump	Grundfos UPS 25-80
SP	Collector	
	1 circulation fan (centrifugal)	Nederman Fan N29
Climacheck	PA Pro II (main box for temperature and pressure sensors)	
	EP Pro (power box)	

Figure 4.8: The Calorimeter Method set up with cooling fan above the indoor unit on the left-side of the picture and the air-water heat exchanger on the right-side

Figure 4.9: SP collector

Figure 4.10: SP circulation fan

Table 4.3: List of sensors needed for the Calorimeter Method measurements

Instrument	**Model**	***Location* / Description**	**Monitored parameter**
1 power meter		• *Connected to the heat pump power input*	$\dot{W}_{HP}$
4 temperature sensors	Pt100	Testing the ambient air temperature outside of climate chamber;	t_{a_amb}
2 temperature sensors	Pt100	To evaluate the average indoor temperature (placed up and down and far away from the disturbing equipment i.e. heat pump and fans)	t_{a_room}
Cooling device			
1 power meter		• *Connected to cooling circulation fan in the indoor compartment;*	$\dot{W}_{cal\_fan}$
Water volume flow meter	Enermet MP115	Testing the flow rate of cooling water	$\dot{M}_{w}$
2 temperature sensors	Pt100	• *Inlet of cooling water* • *Outlet of cooling water*	t_{w_in} t_{w_out}

Table 4.4: List of sensors needed for the SP Method measurements

Instrument	**Model**	***Location*** **/ Description**	**Monitored parameter**
2 power meter		• *Connected to the heat pump power input* • *Connected to centrifugal fan*	$\dot{W}_{HP}$ $\dot{W}_{SP\_fan}$
1 air flow collector		Outlet of the heat pump indoor unit	
1 volume flow meter		Testing the flow rate of air	$\dot{V}_{a}$
1 manometer	Furness FCO12	To check that the static pressure inside the collector is 0kPa (±0.5 kPa)	p_{stat}
1 manometer	Furness FCO12	To read volume flow meter data	p_{diff}
1 manometer		In the lab	p_{atm}
4 temperature sensors	Pt100	To measure T at the indoor unit outlet • *Connected to the air collector, before the circulation fan and after the pressure sensor*	t_{a_out}

Figure 4.11: Temperature sensors at the air inlet of the indoor unit (the Calorimeter Method and the SP Method)

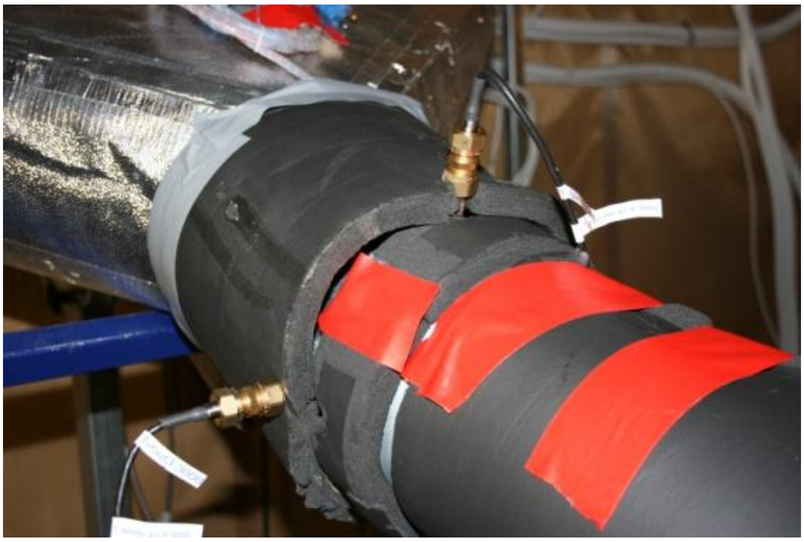

Figure 4.12: Temperatures sensors at the air outlet of the indoor unit (the SP Method)

Figure 4.13: Manometer Furness FCO12 used for the SP method

Figure 4.14: Air flow meter used for the SP Method

Table 4.5: List of sensors needed for the Climacheck Method measurements

Instrument	**Model**	***Location* / Description**	**Monitored parameter**
3 current clamps	Toleka M1 100A	• *Compressor electrical wires*	$\dot{W}_{comp}$
3 voltage clamps	Test probe V	• *Compressor electrical wires*	
1 temperature sensor	Pt1000	• *Inlet of compressor*	TT_2
1 temperature sensor	Pt1000	• *Outlet of compressor*	TT_1
1 temperature sensor	Pt1000	• *Outlet of condenser*	TT_3
1 High pressure sensor	Climacheck Pie-zoresistive Pressure Transmitter 35bars	• *Outlet of compressor*	p_1
1 Low pressure sensor	Climacheck Pie-zoresistive Pressure Transmitter 10bars	• *Inlet of compressor*	p_2

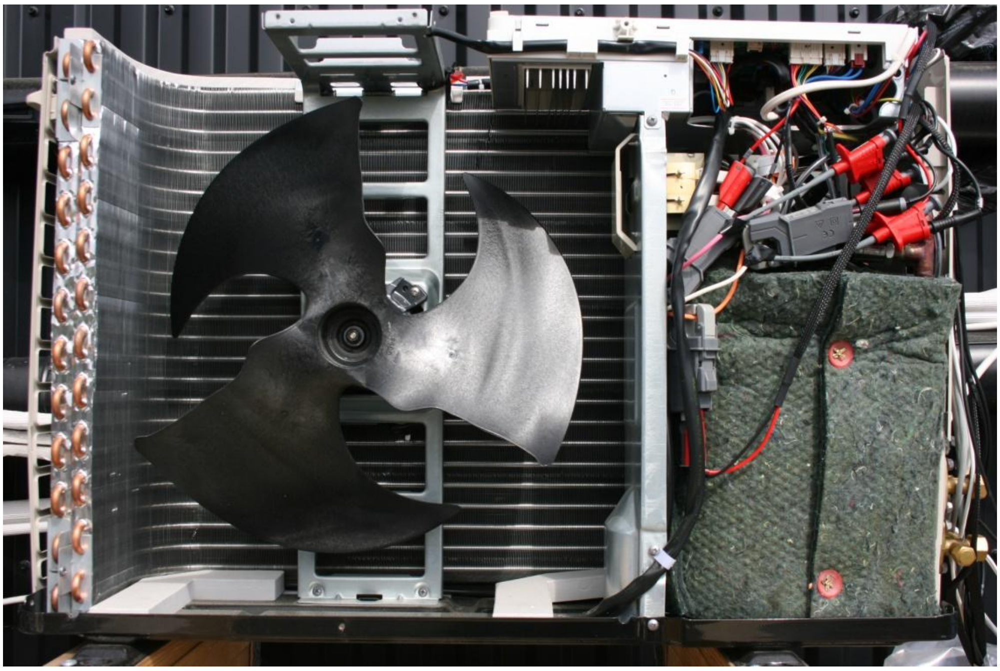

Figure 4.15: Current and voltage clamps in the outdoor unit (the Climacheck Method)

Figure 4.16: On the bottom-left of the picture temperature sensors under the insulation (the Climacheck Method)

4.3 Practical testing process

Considering the requirements of all the three methods, a certain set up and process should be followed in practice, as described below.

1. Make good installations of Climacheck sensors.
2. Check insulations of pipes, ducts and the climate chamber to make sure there are no leakage and misconnection.
3. Turn on the testing unit, set the temperature at 32 °C and fan speed at high mode, which represents peak load, from the remote controller.
4. Turn on the cooling device, adjusting the cooling fan speed as well as the cooling water inlet temperature, in order to achieve the standard required for the Calorimeter Method (t_{a_in}=20 ±1°C, individual value, or 20 ±0.3°C, arithmetic mean value). This process might last up to more than half an hour to ensure stability, due to the dynamic performance of the heat pump.
5. Adjusting the circulation fan speed, so that the static pressure could maintain as 0 Pa during the steady state.
6. Make suitable configurations and start the data logger to record all the parameters, set the data collecting sampling rate for all the testing method at 10 s.
7. When the heat pump changes to partial load or defrosts during the operation, necessary adjustment is required to maintain the testing standard (during defrosting, t_{a_in} =20 ±2.5°C, static pressure should be kept as 0 Pa during all the time).Those adjustments are mainly referring to reducing the cooling fan speed and the circulation fan speed, which could be carried out with automatic controlling system or manually.
8. The testing period should last for at least 250 minutes so as to meet the transient measurements requirements of the Calorimeter Method, while data from the SP Method and the Climacheck Method could be compared except the defrosting process.

Parameters to be recorded:

1. Power input to the testing unit, $\dot{W}_{hp}$.
2. Air temperature at the inlet and the outlet of the indoor unit, t_{a_in} & t_{a_out}.
3. Atmosphere pressure, Pp_{atm}.
4. Static pressure before the manometer, p_{stat}
5. Pressure difference through the manometer, p_{diff}.
6. Power input to both the cooling fan and the circulation fan,$\dot{W}_{fans}$.
7. Cooling unit supplying water temperature and returning water temperature, t_{w_in} & t_{w_out}.
8. Water volume flow rate in the cooling device, $\dot{V}_w$.
9. Indoor temperature of the climate chamber, t_{a_room}.
10. Ambient temperature outside of the climate chamber, t_{a_amb}.
11. Power input to the compressor of the heat pump, $\dot{W}_{comp}$.

12. Refrigerant temperature and pressure at the inlet of compressor, TT_1 & p_1.
13. Refrigerant temperature and pressure at the outlet of compressor, TT_2 & p_2.
14. Refrigerant temperature after the condenser, TT_3
15. Outdoor air temperature and relative humidity, t_{a_out} & RH% (data from the weather station owned by SP and located 4 km away)

4.4 COP calculation for all the testing methods

4.4.1 Steady state

Here are the detailed calculation steps from equation (4.1) to equation (4.17) to determine COP by the different methods for steady-state.

4.4.1.1 The Calorimeter Method

$$\dot{Q}_{heat} = \dot{Q}_{cool} + \dot{Q}_{loss} - \dot{W}_{fans} \qquad \text{(W)} \qquad (4.1)$$

$$\dot{Q}_{cool} = \rho_w * c_{p\_w} * \dot{V}_w * (t_{w\_out} - t_{w\_in}) \qquad \text{(W)} \qquad (4.2)$$

$$\dot{Q}_{loss} = UA * (t_{a\_room} - t_{a\_amb}) \qquad \text{(W)} \qquad (4.3)$$

$$COP = \frac{\dot{Q}_{heat}}{\dot{W}_{HP}} \qquad (4.4)$$

4.4.1.2 The SP Method

$$\dot{Q}_{heat} = \rho_a * c_{p\_a} * \dot{V}_a * (t_{a\_out} - t_{a\_in}) \qquad \text{(W)} \qquad (4.5)$$

$$\rho_a = \frac{p_{atm}}{R} * (273 + t_{a\_out}) \qquad (\text{kg/m}^3) \qquad (4.6)$$

$$c_{p\_a} = 1.007 \qquad \text{(kJ/kg/K)} \qquad (4.7)$$

A specific calibration [3] provides the air flow rate equation,

$$\dot{V}_a = 9.71613 * \frac{P_{diff}}{\rho_a}^{0.48335} \qquad \text{(L/s)} \qquad (4.8)$$

$$COP = \frac{\dot{Q}_{heat}}{\dot{W}_{HP}} \qquad (4.9)$$

[3] See calibration certificate of the manometer used.

4.4.1.3 The Climacheck Method

$$\dot{Q}_{heat} = \dot{M}_r * (h_1 - h_3) \qquad \text{(W)} \qquad (4.10)$$

$$\dot{M}_r = \frac{\dot{W}_{comp} * 0.93}{h_1 - h_2} \qquad \text{(kg/s)} \qquad (4.11)$$

$$h_1 = h(TT\_1,\ p_1) \qquad \text{(kJ/kg)} \qquad (4.12)$$

$$h_2 = h(TT\_2,\ p_2) \qquad \text{(kJ/kg)} \qquad (4.13)$$

$$h_3 = h(TT\_3,\ p_1) \qquad \text{(kJ/kg)} \qquad (4.14)$$

4.4.2 Cycles with defrosting

Neither the Climacheck Method nor the SP Method could take correct value and further calculate the removed heat during defrosting, so only the Calorimeter Method can be applied to determine the COP of the heat pump by recording data from several complete cycles with defrosting.

$$Q_1 = \int_{\tau 1}^{\tau 2} \dot{Q}_{heat} \qquad \text{(J)} \qquad (4.15)$$

$$W_{HP} = \int_{\tau 1}^{\tau 2} \dot{W}_{HP} \qquad \text{(W)} \qquad (4.16)$$

$$COP = \frac{Q_{heat}}{W_{HP}} \qquad (4.17)$$

5 Case Studies

Two main types of case studies were carried out: long term and short term testing. The former includes defrosting of the heat pump while the other one encompasses the three methods when the heat pump is working in steady-state.

5.1 Long term testing case

In Figure 5.1, Figure 5.3 and Figure 5.4 are the testing results during three days continuous measurement with the Calorimeter Method. The measurement was carried out in late March, when it is still possible to have ice formed on the surface of the evaporator due to the cold and humid weather condition in Borås (see Figure 5.2). That method was used as a reference and enables observation of the general behaviour of the heat pump.

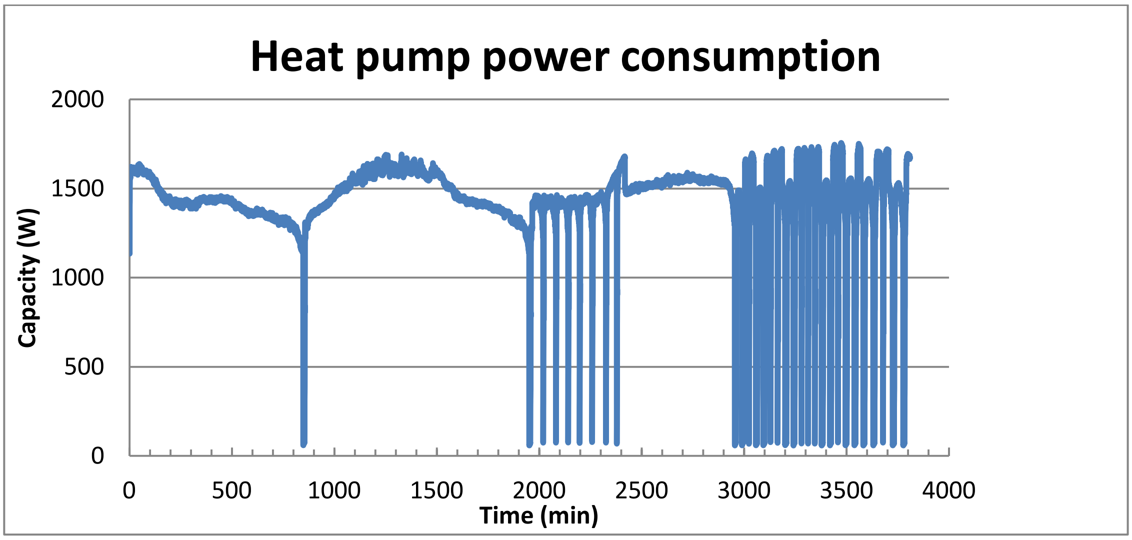

Figure 5.1: Heat pump power consumption

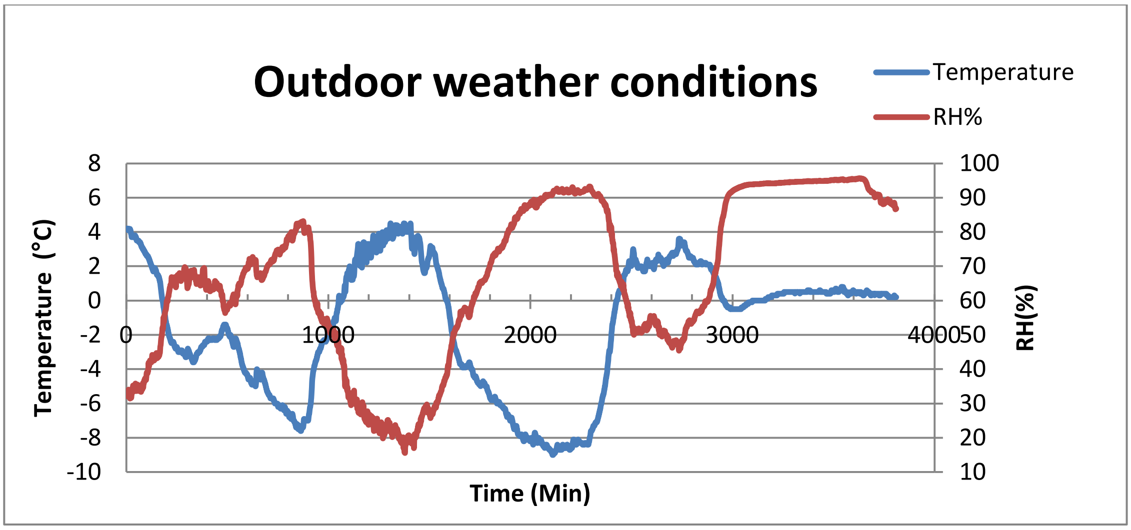

Figure 5.2: Outdoor weather conditions March 25th -28th

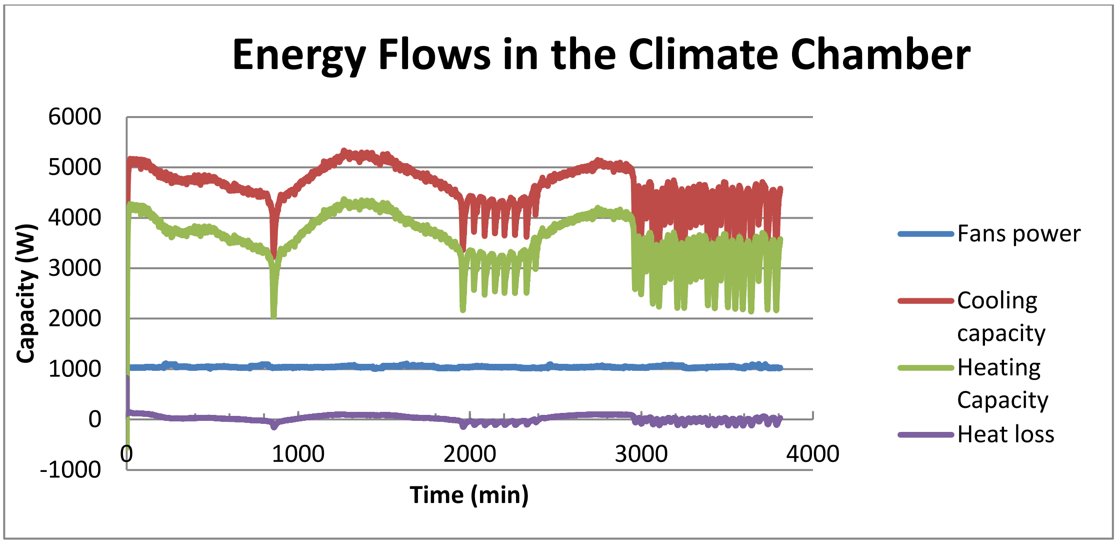

Figure 5.3: Energy Flows in the Climate Chamber

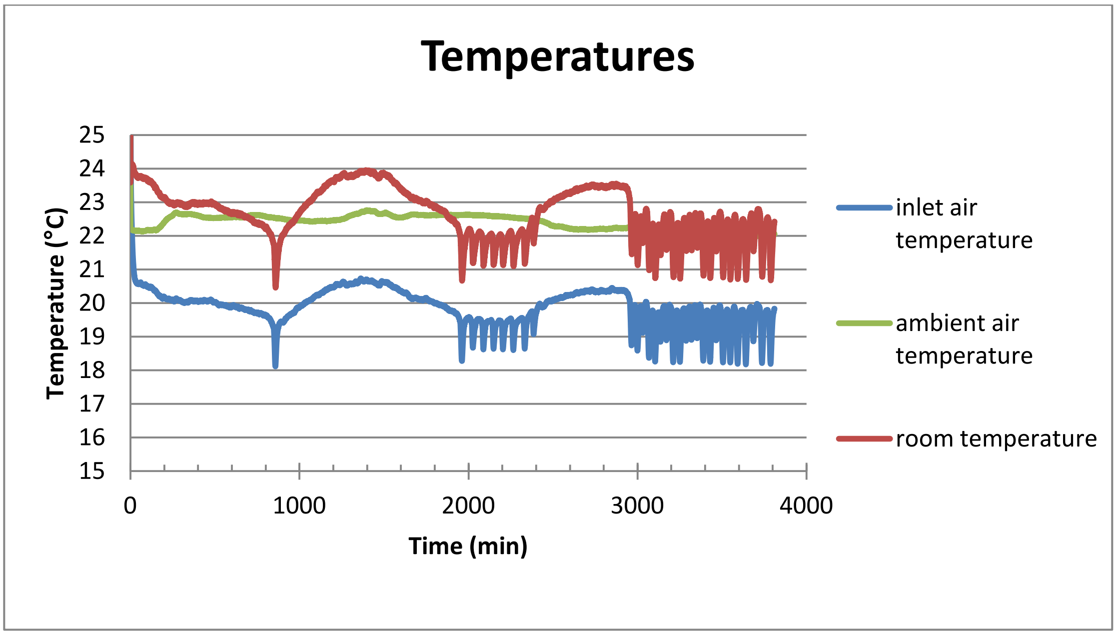

Figure 5.4: Air temperature data during testing

The data were taken from 04:05pm 25th March to 08:35am 28th March, with the sampling time interval set as 10 seconds. It could be indicated from the inlet air temperature scenarios that steady state is not achieved during the whole testing process; however, the dynamic behaviour of the heat pump seems to be able to meet the requirements of the testing standards of the Calorimeter Method, the inlet air temperature is within the permitted variation even during defrosting. (More information about the temperature requirement can be read in the test standard EN14511 [19].

It could also be noticed two different types of cycles with defrosting occurred around 2000 to 2500 minutes and around 3000 to 3500 minutes separately, which are shown in Figure 5.5 and Figure 5.6. Since both the room temperature and the ambient temperature after several complete cycles with defrosting could come back to the initial value, the heat storage of the chamber can be assumed to be unchanged. Therefore, net heat input from the heat pump can be calculated with the energy balance theory for both of the testing periods.

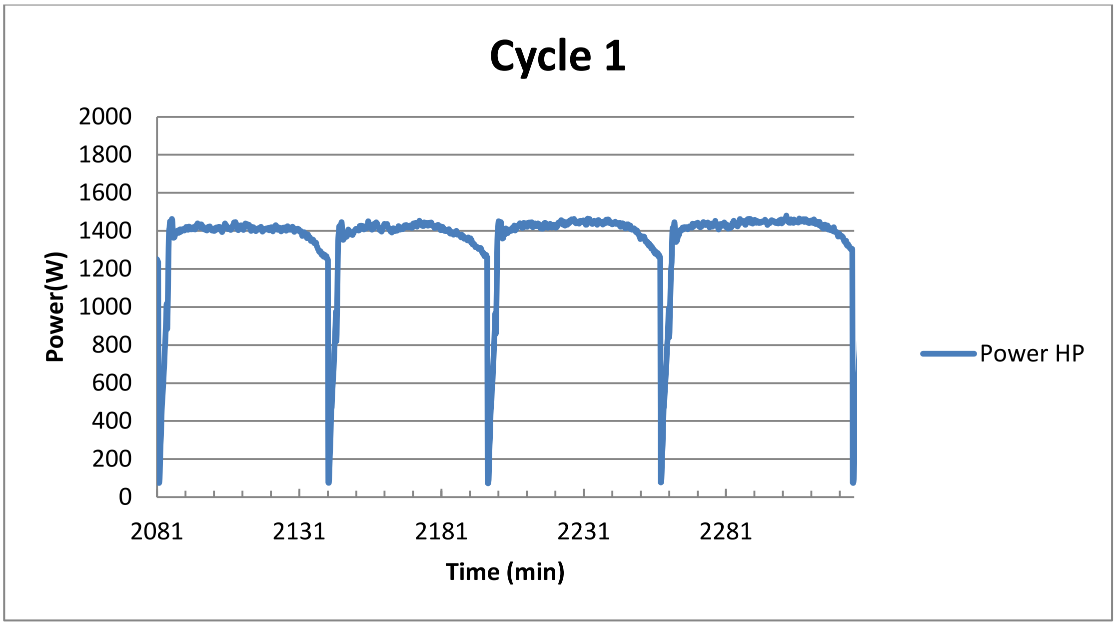

Figure 5.5: Cycle 1

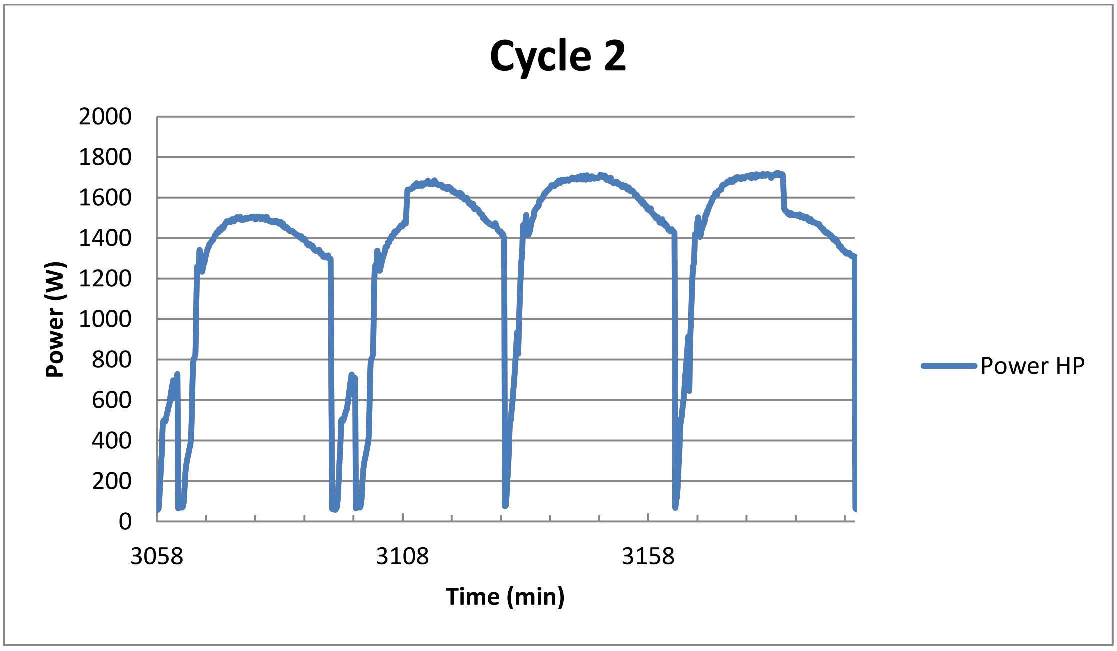

Figure 5.6: Cycle 2

Here four complete cycles with defrosting are taken for each test period, which is reflected in the power consumption of the heat pump. The COP value for heating is then the ratio of the integration of heating capacity to the total electricity energy input to the heat pump during each calculation period, as shown in equation (5.1).

$$\mathrm{COP} = \frac{\sum \dot{Q}_{\mathrm{heat}}}{\sum \dot{W}_{\mathrm{HP}}} \qquad (5.1)$$

And COP is 2.27 for cycle 1 while COP is 2.36 for cycle 2 from the calculation result.

The possible factors leading to defrost are explored in Section 6.2.

5.2 Short term testing case

The results shown below mainly concern the steady-state testing, since except the Calorimeter Method, the SP Method and the Climacheck Method are supposed to work properly only under steady state. Theoretically, all the three methods have the possibility to be carried out within one testing unit simultaneously without influencing each other, as explained in Chapter 4.

These data were taken from the measurements of the early April because of the relatively moderate outdoor conditions, which would lead to less frost while a considerably long steady state could be reached.

The results from the Calorimeter Method, the SP Method and the Climacheck Method can be respectively observed in Figure 5.7, Figure 5.8 and Figure 5.9. The static pressure fluctuates between -0.1 Pa and +0.2 Pa, and the inlet air temperature varies around 19.6°C which fulfill the requirements for the Calorimeter Method ($18°C \leq t_{a\_in} \leq 22°C$) and SP Method ($-0.5Pa \leq P_{stat} \leq 0.5Pa$).

While the SP Method and the Calorimeter Method have similar results, heating capacity calculated with the Climacheck Method is much higher, resulting in a big change in COP (see Table 5.1). The reason for the large deviation for the Climacheck method is not a fault of the method as such and will be discussed in 7.2

Table 5.1: COP values for the 3 methods during the steady-state sampling time span

	The Calorimeter Method	The SP Method	The Climacheck Method
COP	3.7	3.7	4.8

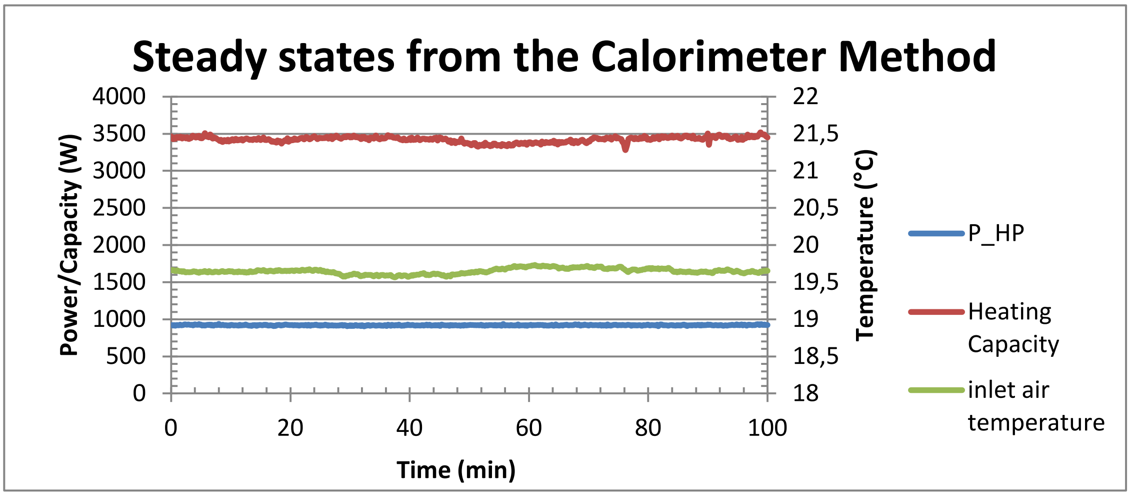

Figure 5.7: Steady state parameters from the Calorimeter Method

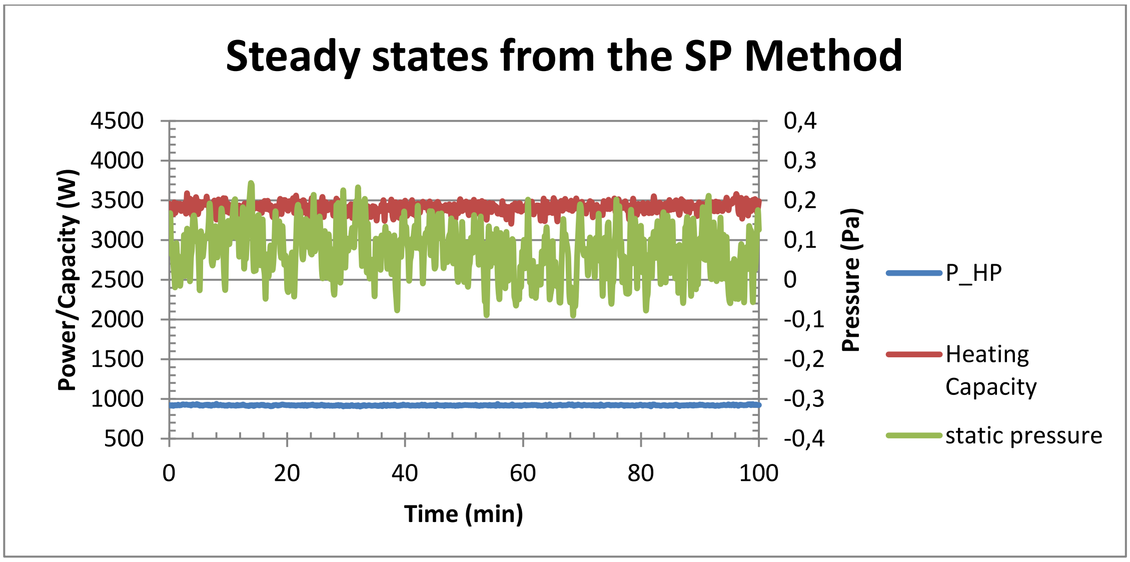

Figure 5.8: Steady state parameters from the SP Method

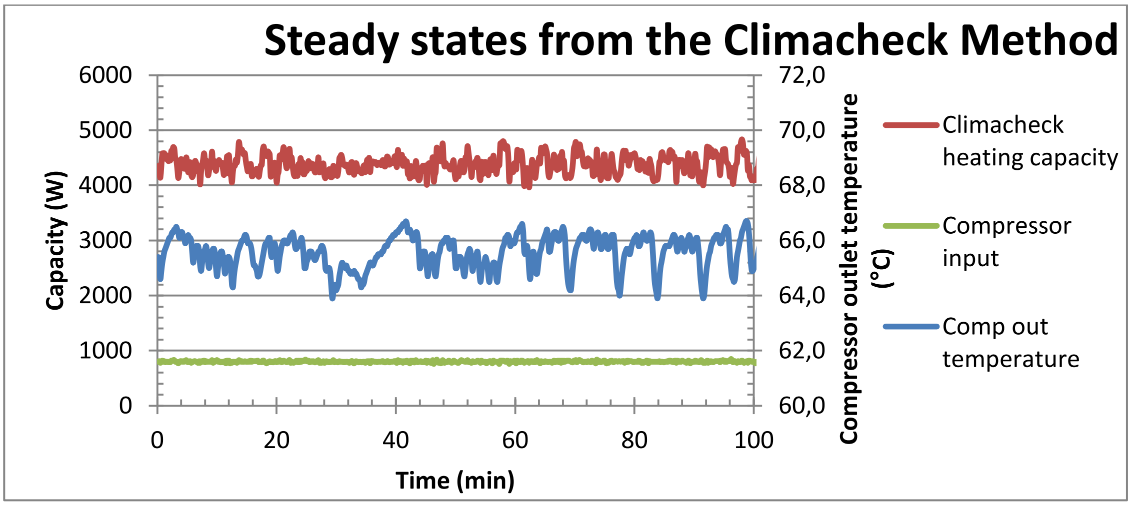

Figure 5.9: Steady state parameters from the Climacheck Method

6 Results and discussion

6.1 Influence of different sampling time intervals to the long term testing

It is highly recommended to have a testing period long enough to get a reliable HSPF, especially for the Calorimeter Method. Meanwhile, the sampling time interval shall be as short as possible in order to follow the performance of the heat pump dynamic process more accurately, especially during defrosting. It is recognized that consequently the information recorded would be enormous, so the question of how important the sampling time interval is was raised. Then in the case where a larger sampling time is selected, how much will it affect the testing results? Here this issue is studied and discussed by looking into four complete cycles (with heating period followed by defrost period) observed by the Calorimeter Method, see Figure 6.1.

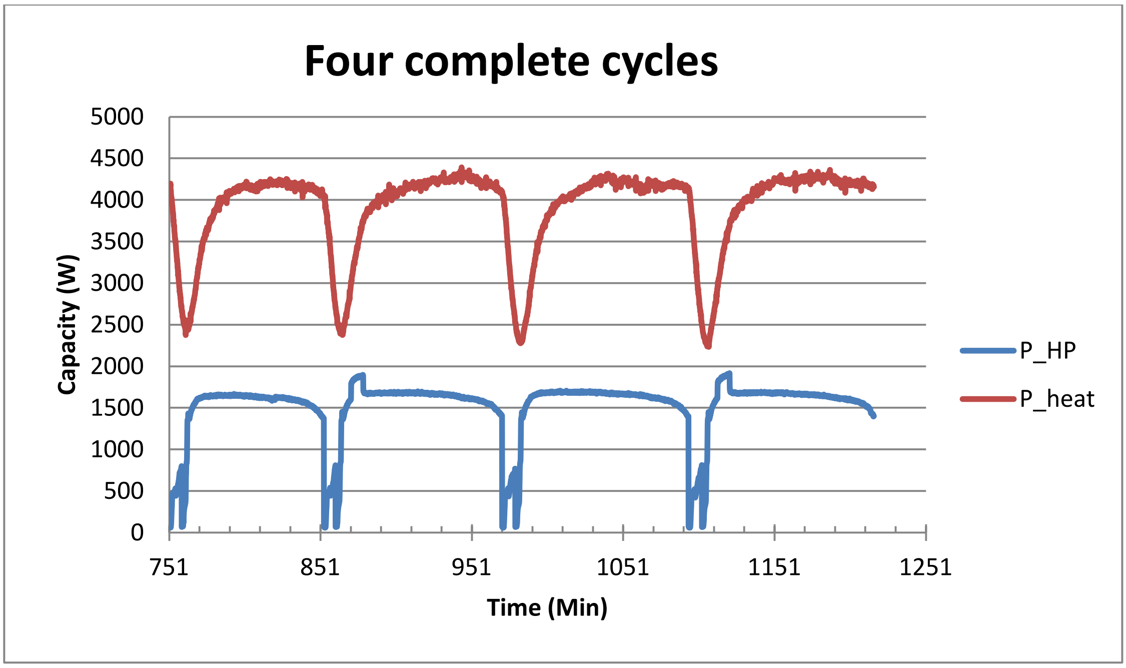

Figure 6.1: Four complete cycles with defrosting resulting from measurements by the Calorimeter Method

All the data needed were recorded with 5 seconds as a time interval and the COP value was calculated to be 2.582 (three decimals were kept here in order to see the change). Then, to investigate the influence of time interval the data sample is kept but information is deleted so as to get a corresponding sampling time interval of 10 seconds. In a similar way a time interval of 30 seconds was also probed. Integration is made to compare the total heat energy output and total electricity energy input with the original case. For the 10 seconds case, the COP value remains to be 2.582 while for the 30 seconds case, the COP value becomes 2.581 and thus only changes by 0.001.

That result can be extended to the whole result testing period since the experienced cycles with defrosting for the whole testing days were similar. Thus, sampling time interval would not influence the testing result much. A 10 seconds interval works perfectly to follow the dynamic process and even 30 second interval is acceptable.

Since the defrosting process will last for at most 10 minutes, an interval longer than 1 minute is not reliable while both 10 seconds and 30 seconds interval could be optimal options depending on the capability of the data recording and processing system.

6.2 Influence from outdoor air conditions to the performance of heat pump

Defrosting is a common phenomenon for air-to-air heat pump in heating mode. It could be empirically concluded that low temperature and high relative humidity lead to the formation of frost on the heat exchanger surface of the outdoor unit during heating operation. Normally, outdoor air dry-bulb temperature and relative humidity are recorded to evaluate how heat pump performs in different climate circumstances, as in the paper .

In this project, air temperature and relative humidity were also recorded by the SP weather station, as shown in Figure 5.2. However, when it is related to the heat pump performance during the same testing process, as shown in Figure 5.1, the weather condition seems not to follow the tendency reasonably. For example, the temperature is higher around 3500 minutes comparing to what happened around 2000 minutes, while the relative humidity is similar to each other, but the defrosting occurs more frequently in the former case. Fahlén has pointed out that the worst frosting situation is around 2°C and that is why the heating rating condition is normally set to be 2 to 20°C in Sweden. Regarding this problem, another outdoor air parameter is preferably to be created in order to explain the influence to the heat pump performance.

Here vapour ratio could play the role of indicator for cases concerning frosting process, since it interprets how much latent heating capacity could potentially be removed from the air. In addition, it could be easily determined from the dry-bulb air temperature and relative humidity, either by applying a Mollier diagram or by calculation see .

Figure 6.2 and Figure 6.3 give the information concerning the relationship between outdoor air condition and power consumption of heat pump, by means of air vapor ratio and air temperature separately. It is more intuitive to find that a higher vapour ratio could lead to a more frequent defrosting process. Here some conclusions could be drawn from the observed result. For the condition that the air temperature is far below 0°C (so far below the saturated temperature for vapour under atmosphere pressure) vapour ratio seems not to be such a significant factor to trigger defrost while it actually acts as the dominating factor for ice forming when the temperature is around 0°C (from 2920 minutes of test). By contrast, when the air temperature is sufficiently high, defrosting process hardly happens although the air remains wet. Nevertheless, further information is still required to study the exact effect from the outdoor air condition to the performance of air-to-air heat pump, since internal heat load as well as controlling system of the heat pump plays important parts simultaneously, and these factors are also supposed to be dynamic and complex. The issue of frosting and defrosting has been thoroughly analyzed by Fahlén .

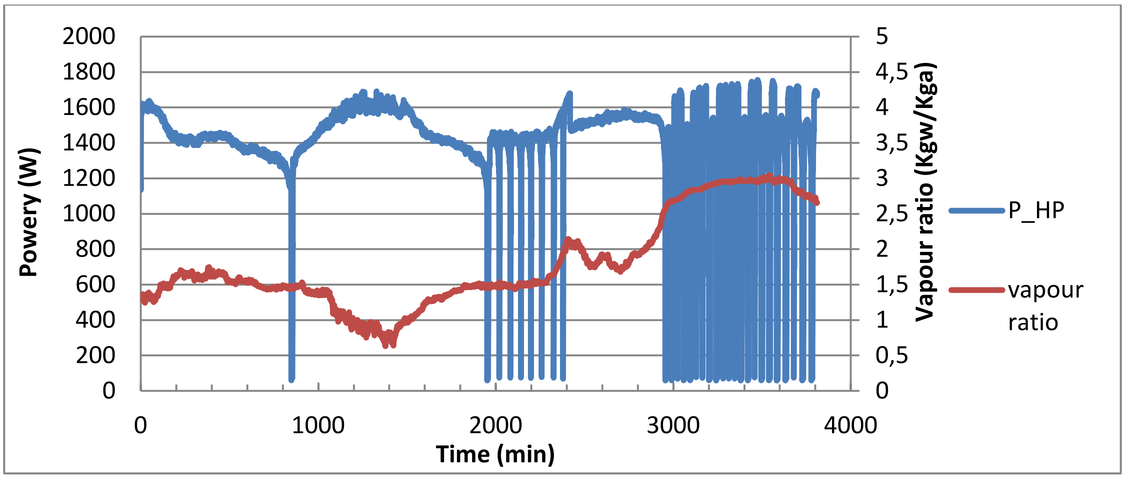

Figure 6.2: Influence of vapour ratio to heat pump performance

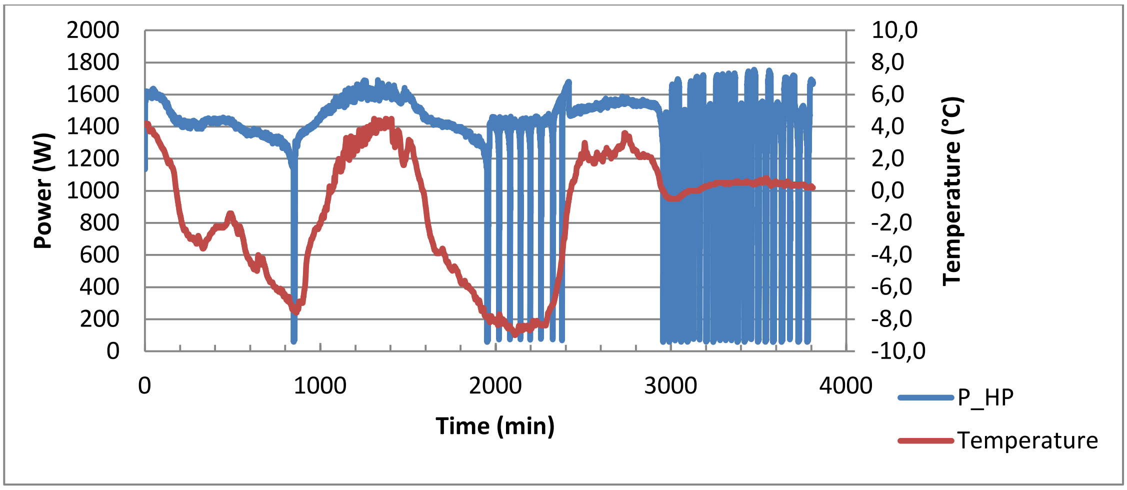

Figure 6.3: Influence of air temperature to heat pump performance

6.3 Heat balance for the Calorimeter Method

It is essential to keep the heat balance within the climate chamber during testing of COP with the Calorimeter Method, but there are also high requirements for the controlling system to maintain the balanced state during dynamic performance of the heat pump, defrosting process in particular. The heat balance achieved during the former steady state is seriously damaged by defrosting process; recovering the resulted heat loss would imply the use of a heater within that specific disturbing period of time, nevertheless it is particularly hard to set up that heat compensation process in practice.

Even though the heat balance analysis could not be carried out within only one cycle with defrosting, the Calorimeter Method is still applicable to get the COP value for a data collecting period containing several complete cycles, unless it is against the requirement of the testing standard.

Here in order to make an evaluation of how the Calorimeter Method testing results will react against different controlling environments during defrosting process, two experiments were carried out within consecutive days, on March 28th and March 29th, and similar outdoor weather conditions are chosen to make comparisons, which can be seen from Figure 6.4 and Figure 6.5. One took no controlling actions during de-frosting while the other one used automatic control of the circulation fan within the cooling device. The controlling system is based on the air temperature difference at the outlet and inlet of the indoor unit and the fan will stop when the air temperature difference goes below 3K and will restart when it returns to 7K.

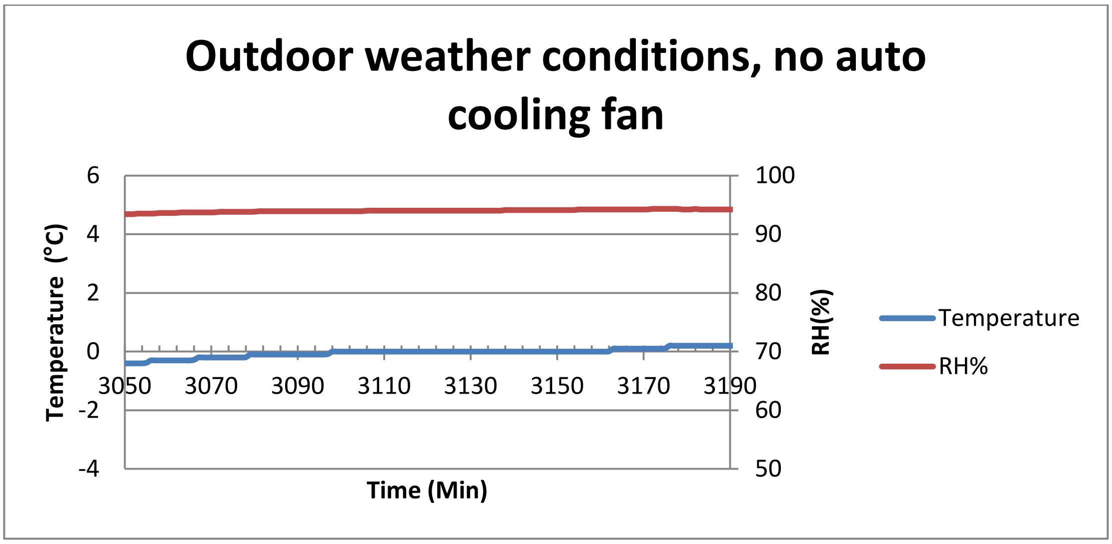

Figure 6.4: Outdoor weather conditions, no auto cooling fan

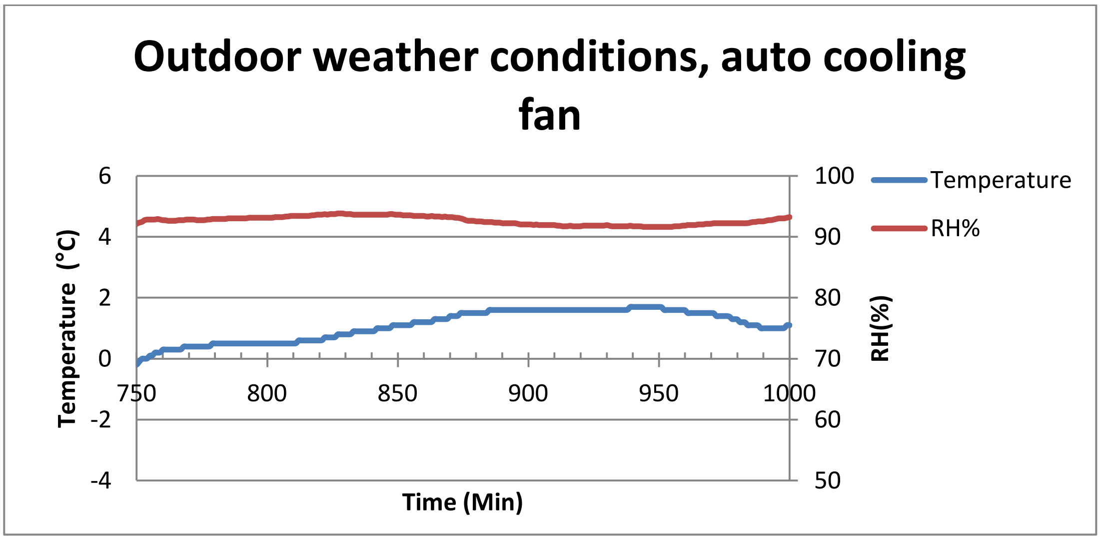

Figure 6.5: Outdoor weather conditions, with auto cooling fan

For both of the experiments, inlet air temperature were recorded to check whether the deviation was beyond the testing standard; meanwhile, ambient air temperature and room temperature were also taken before and after the data collection periods to make sure the dynamic cycles do not affect the heat content within the testing chamber. The data measured are illustrated in Figure 6.6 to Figure 6.9.

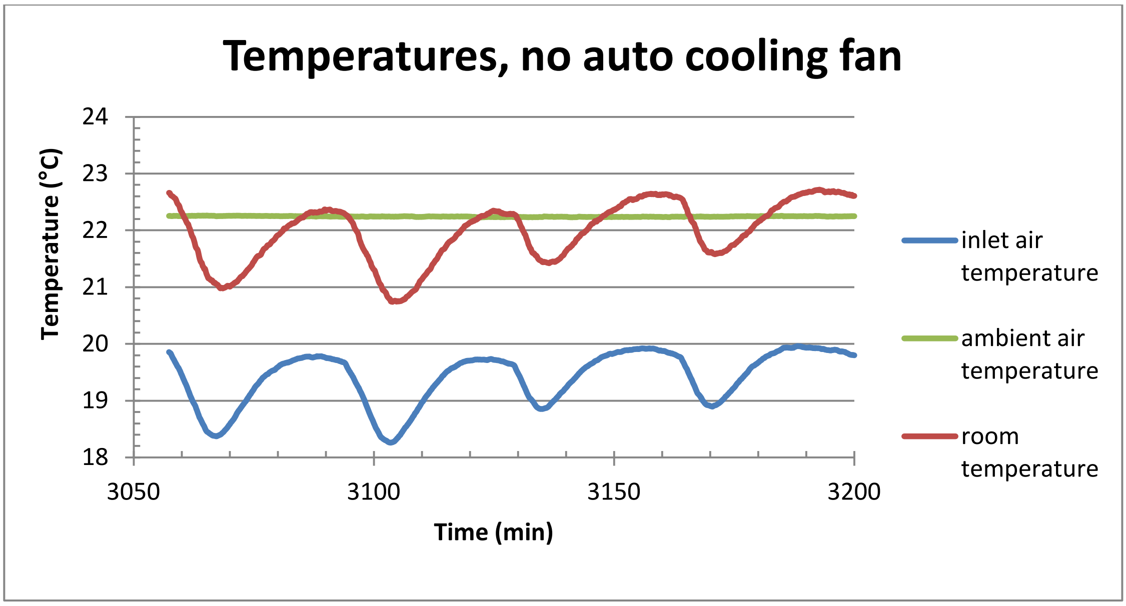

Figure 6.6: Measured temperatures, no auto cooling fan

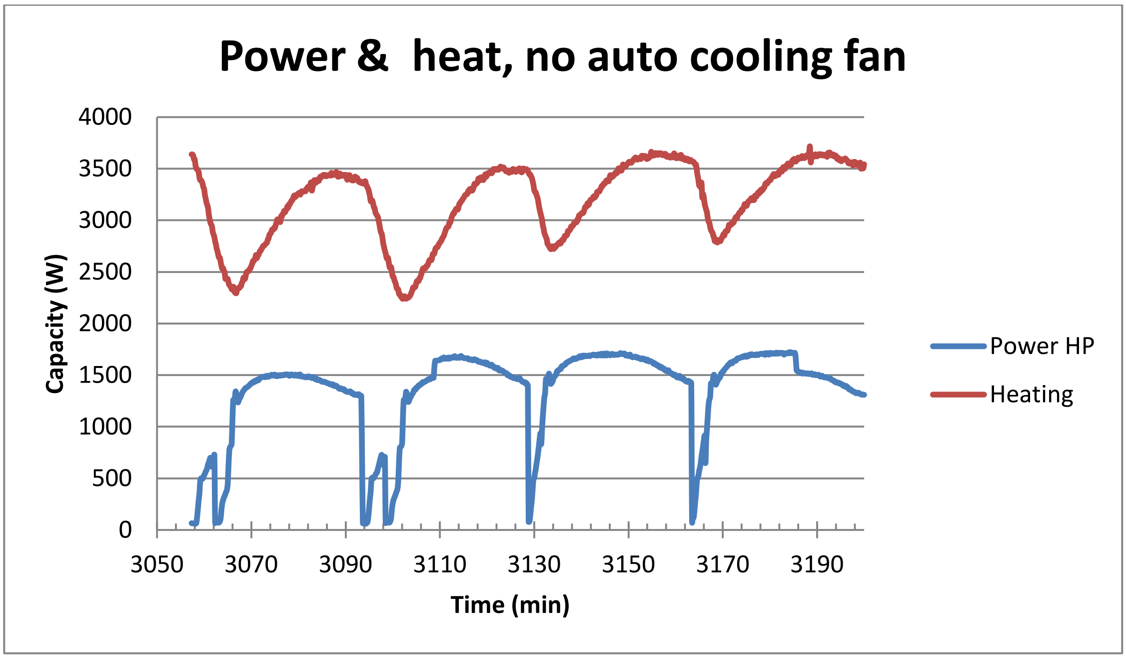

Figure 6.7: Power & heat, no auto cooling fan

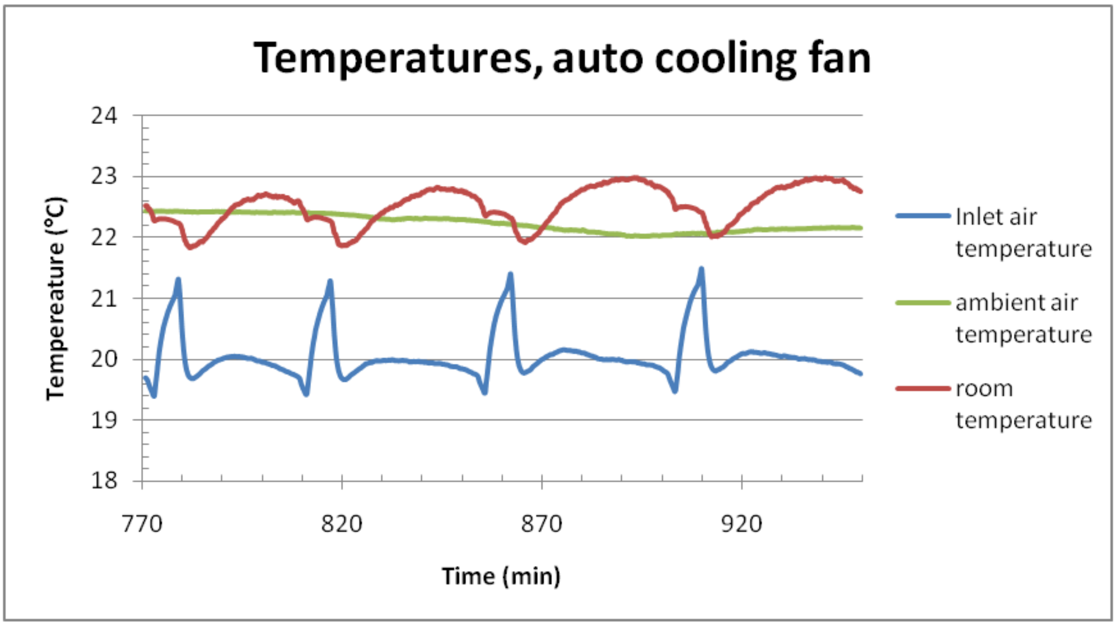

Figure 6.8: Measured temperatures, with auto cooling fan

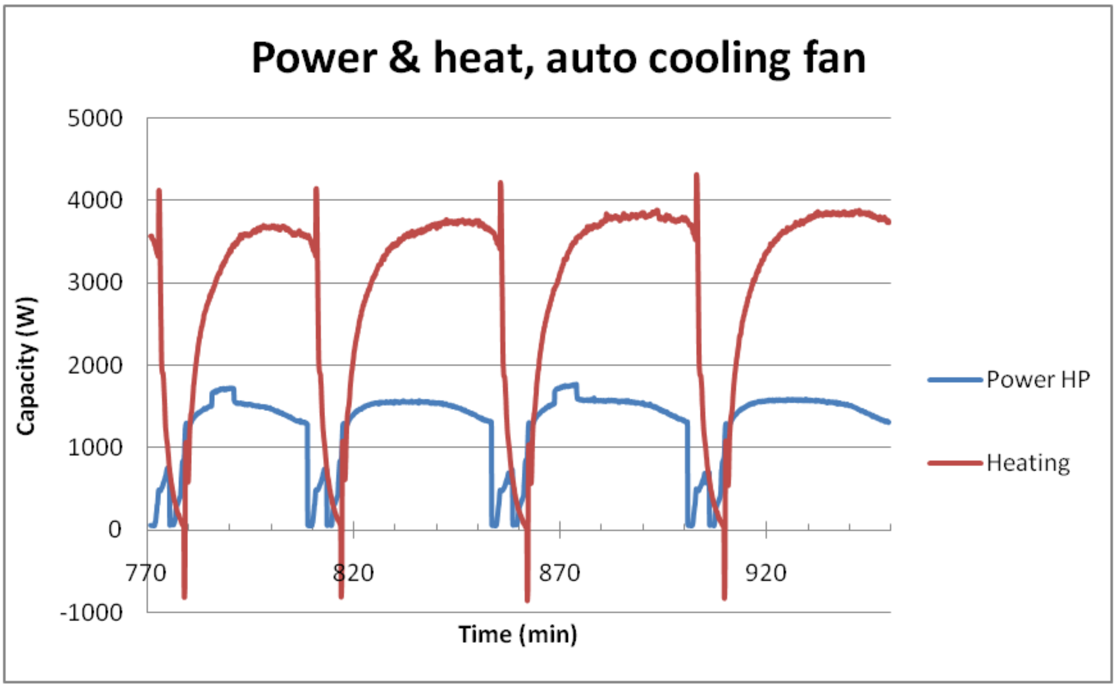

Figure 6.9: Power & heat, with auto cooling fan

As can be seen in Figure 6.6 and Figure 6.8, both the cases without and with cooling fan automatic controlled, the deviation of inlet air temperature could meet the requirement in the testing standard, which is ±2.5K during defrosting. The recorded room temperature and ambient air temperature suggest that the thermal states before and after data collection could be in accord with each other. However, the temperature fluctuates more seriously when no auto cooling fan is applied, while there are some small peaks existing in the case with auto cooling fan used. From the empirical analysis, the former one damages the heat balance more during defrosting, but the later could only be a compromised solution by using inlet and outlet air temperature difference as the controlling parameter, which contains some delays when defrosting happens. Furthermore, from the power and heat diagram achieved with auto cooling fan in usage, some negative heating capacity could be noticed during defrosting. That is due to the dramatic decrease of cooling capacity input to the chamber when defrosting happens. However, it cannot precisely reflect the heat removed from the chamber during defrosting process since no heat compensation is made simultaneously, and it is really hard considering the short duration of its dynamic performance.

COP value for both of the cases is then calculated and compared, for each case; four complete cycles with defrosting were taken into account. For the case without auto cooling fan, COP is 2.362 while for the case with auto cooling fan, COP is 2.318. The result gives no big gap, which suggests that the controlling system for heat balance maintaining during dynamic process is not seriously significant when up to several cycles are considered. While it shall also be noticed that the outdoor weather conditions for these two experiments were not identically the same, and it would influence the performance of the air-to-air heat pump to some extent.

6.4 Changing of static pressure at the outlet of indoor unit

For long term testing with the SP Method, there is great possibility that the static pressure at the outlet of indoor unit changes, since the circulation fan keeps running at the same speed while the heat pump is in dynamic performance, defrosting or partial load for instance. Therefore, it would be interesting to investigate how it will influence the testing result consequently.

The experiment was carried out together with the Calorimeter Method. The heat pump was set at 32°C and strong fan mode, the static pressure was adjusted, by setting the circulation fan speed, from -3 Pa to 3 Pa. It is obvious to notice from Figure 6.10 that the power consumption of heat pump has the same changing trend as the static pressure, which indicates the change of static pressure will affect the performance of heat pump.

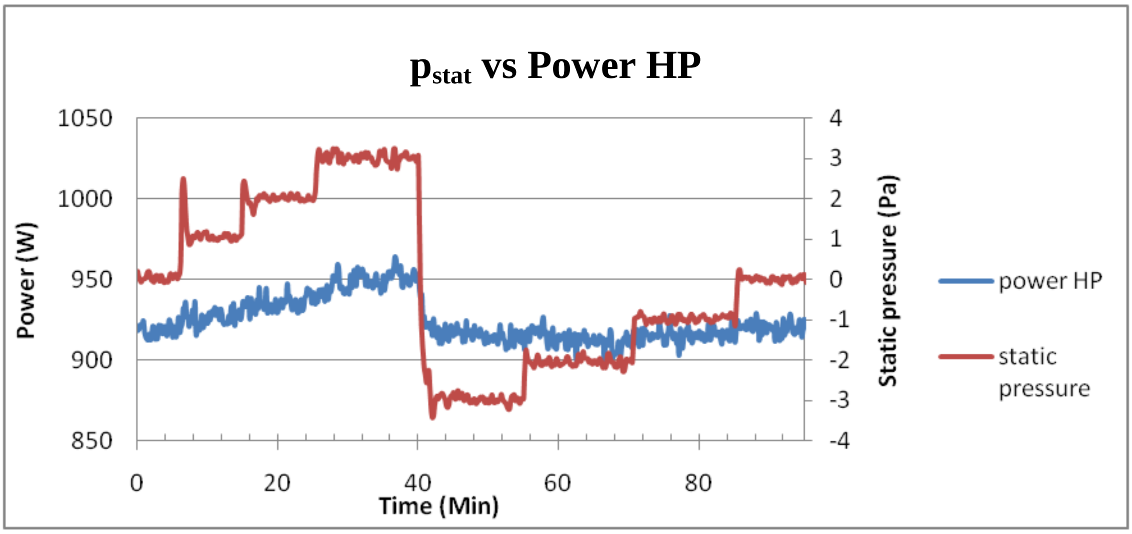

Figure 6.10: Static pressure vs Power consumption of heat pump

This phenomenon could be further explained by looking into the correlation between static pressure and air temperature at the outlet of the indoor unit as shown in Figure 6.11.

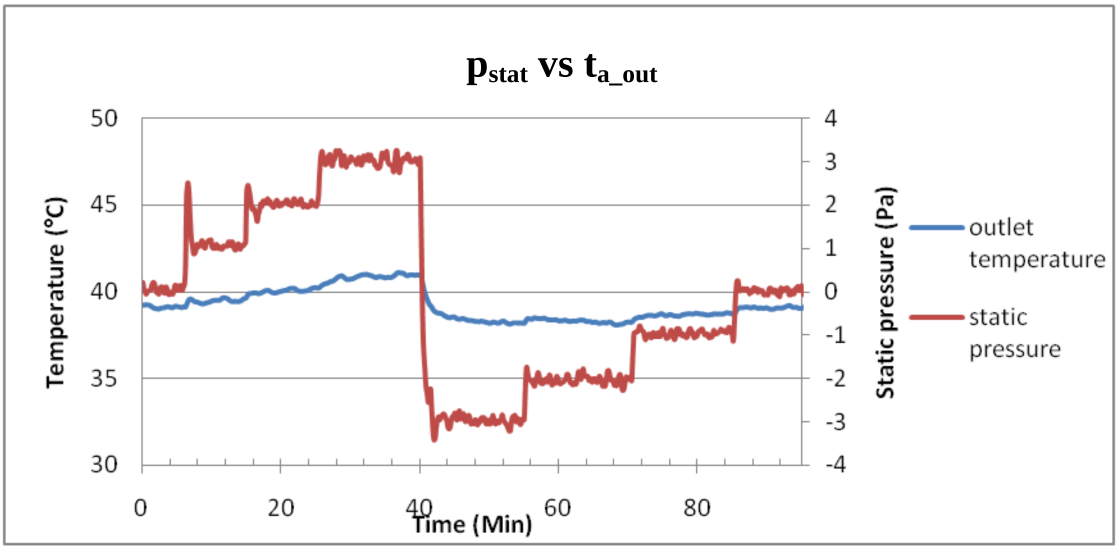

Figure 6.11: Static pressure vs t_{a_out}

When the circulation fan is turned down to get a larger static pressure, the hot air is accumulated around the outlet of the condenser and makes the air temperature increase. As illustrated in Figure 6.12, the refrigerant temperature at the discharging point of indoor unit would increase due to the reduction of heat exchanging capability through the condenser, the controlling system will then increase the load of the compressor thus to have more heating capacity available.

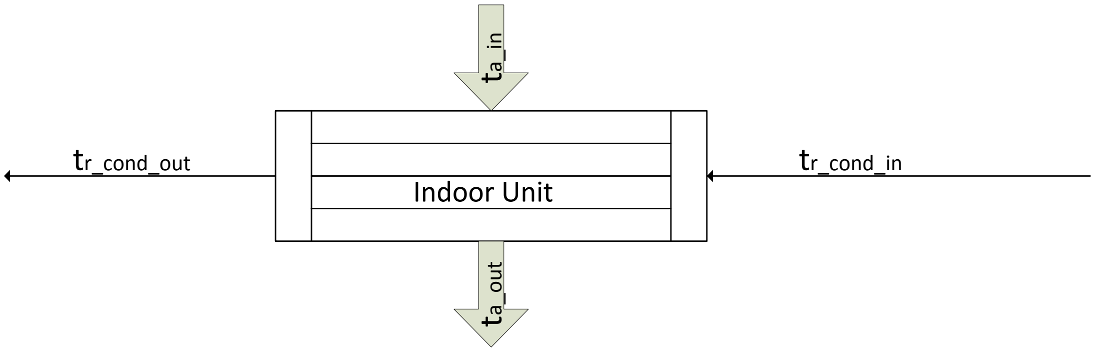

Figure 6.12: Temperature change through the indoor unit

The total testing period could be considered as a dynamic process for the Calorimeter Method. However, from Figure 6.13, the temperature changing tendency suggests that thermal capacity is fully recovered to the original state, so that COP could be calculated by energy balance considering the whole data taking period. Heating capacity changing is as shown in Figure 6.14.

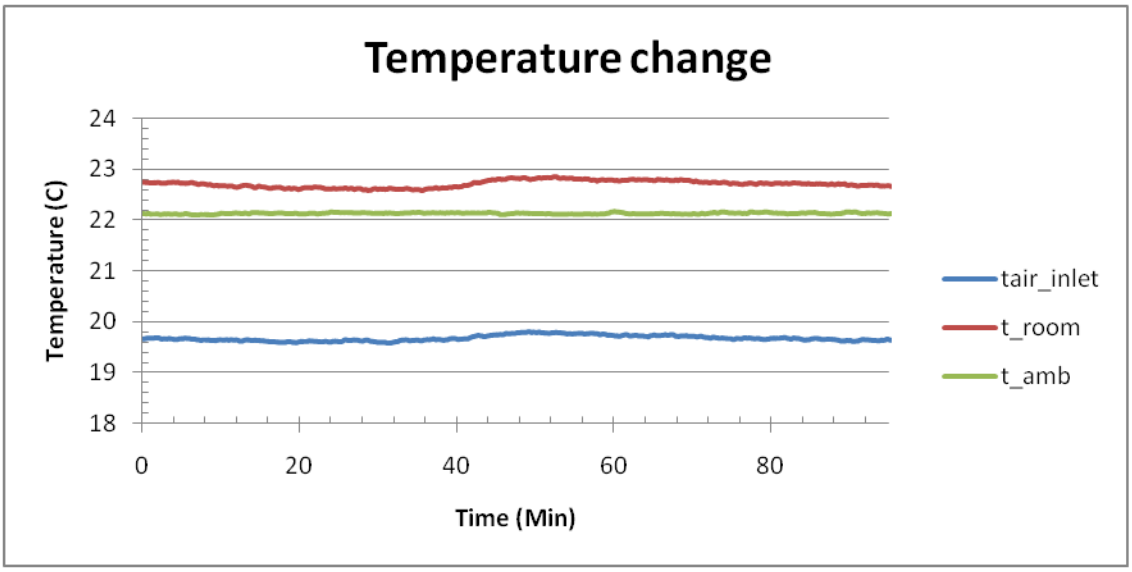

Figure 6.13: Temperature changing tendency during the testing period

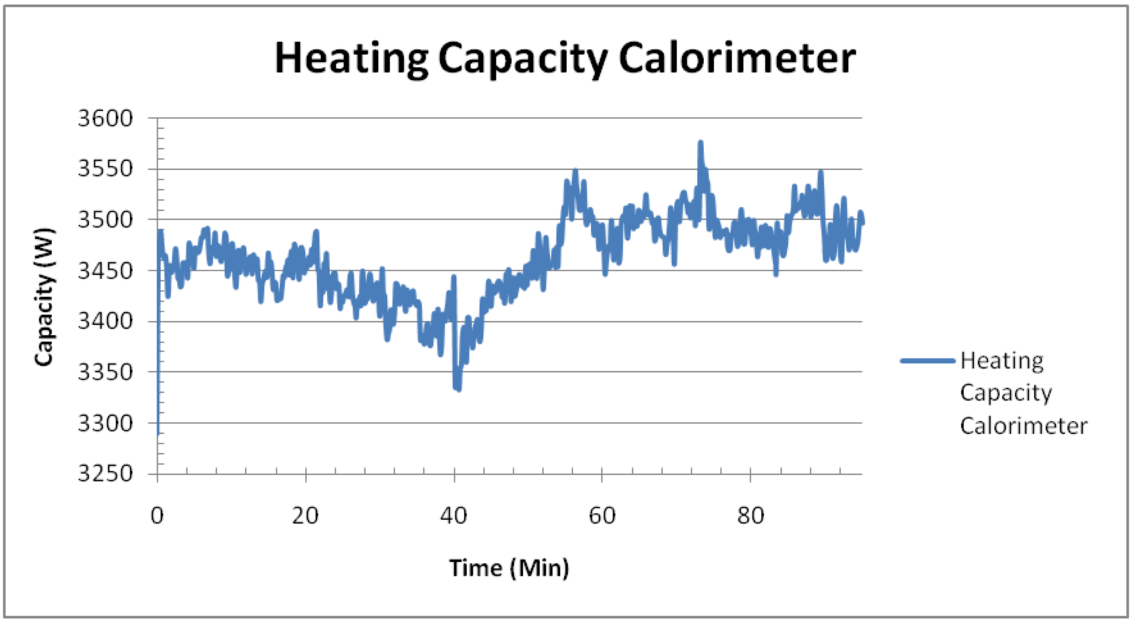

Figure 6.14: Heating capacity change

COP calculated with the Calorimeter Method comes out to be 3.745. While COP is also calculated as 3.715 with the SP Method during the same testing period, though the volume flow rate is not trustable due to the fluctuation of static pressure. It gives not big deviation just concerning the COP value measured; nevertheless, the way to achieve the air volume flow rate failed to meet the theoretical condition of zero pressure difference. In addition, the heat pump would work differently if the static pressure is artificially changed away from 0 Pa, which does not apply in practical field testing of COP. Suggestions of improvement could be installing an automatic controller onto the variable speed SP fan, so that the outlet air pressure difference could be reduced to a minimum degree.

6.5 Positions of condenser inlet temperature sensor on the discharging line

By default, the Climacheck software takes into account for calculation three points even though the box allows the connection of other sensors. Those positions, called 1, 2 and 3, are respectively described as compressor outlet, compressor inlet and expansion valve inlet. As a reminder from Section 3.3.2, pressures at 1 and 2 are measured while pressure at point 3 is assumed equal to the one at point 2 (no pressure drop), Then, the program looked into the thermo physical properties of R410A (following IIR settings) to determine the 3 enthalpies h_1, h_2 and h_3. Then the software calculates the heat delivered by the heat pump with equation (6.1).

$$\dot{Q}_{HP} = \frac{(1-f).\dot{W}_{comp}}{(h_1 - h_2)} * (h_1 - h_3) \qquad (6.1)$$

The first factor is actually a trick to determine the mass flow rate and focuses on the compressor, so it seems indeed on purpose to locate the (temperature and pressure) sensors 1 and 2 next to the compressor, meanwhile the second factor is related to the change of enthalpy due to the heat exchange within the indoor room, so the points 1 and 3 should be placed next to the condenser. Thus the equation which reflects the reality is equation (6.2).

$$\dot{Q}_{HP} = \frac{(1-f).\dot{W}_{comp}}{(h_{comp\_out} - h_{comp\_in})} * \left(h_{cond\_in} - h_{cond\_out}\right) \qquad (6.2)$$

In practice there is some relative distance between the indoor and outdoor units even if the installer tries to limit it to avoid unnecessary expenses. In the present case, the two units were about 15 meters from each other. If one considers a significant refrigerant parameter such as temperature, its value changes between the outlet of the compressor and the inlet of condenser due mainly to some heat losses.

During steady state on April the 13th, the temperatures at the compressor inlet and condenser inlet have been registered for comparison in Figure 6.15. The two curves fluctuations are very similar in the shape except that the condenser inlet temperature has a smaller deviation compared to the compressor outlet temperature. Moreover, there is a small shift (about 30 seconds) due to the time delay for the heat transfer.

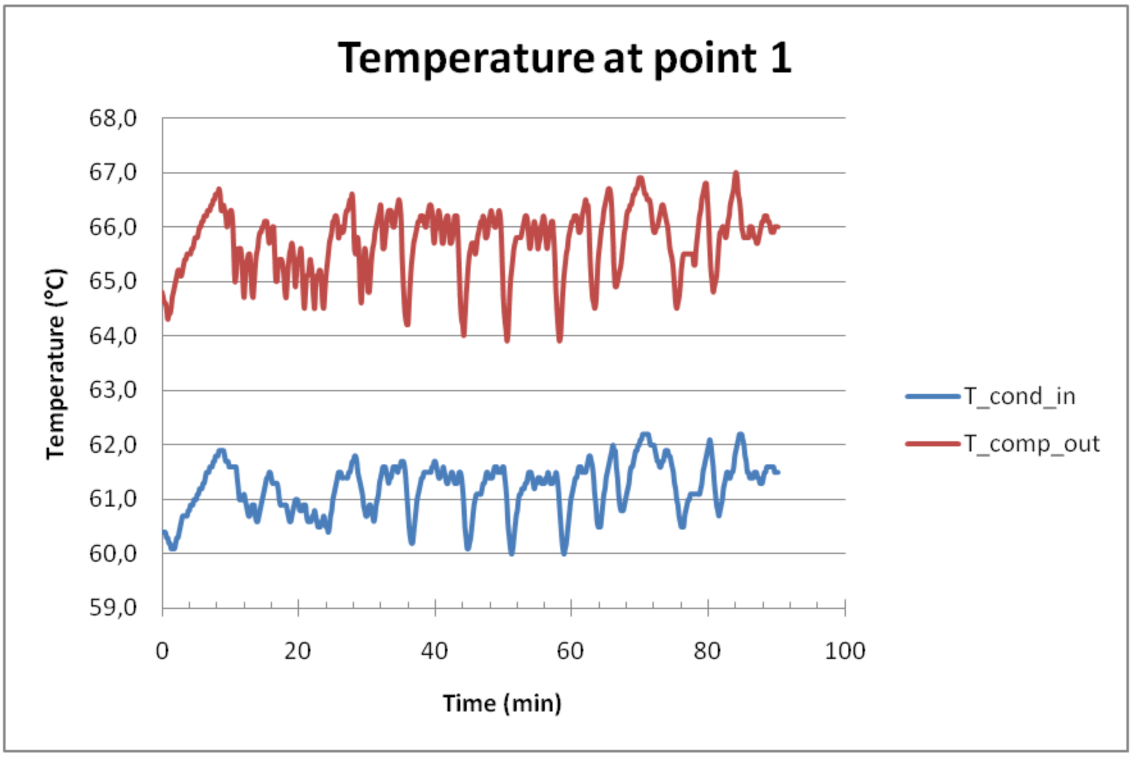

Figure 6.15: Temperatures measured at the condenser inlet and the compressor outlet

Table 6.1: Average values of different parameters for both studied steady-states

	$\dot{W}_{comp}$ (kW)		comp_out	cond_in	cond_out	comp_in	$\frac{h_{cond\_in} - h_{comp}}{h_{comp\_out} - h_{comp}}$ Change of the 1st factor (%)	$\frac{h_{cond\_in} - h_{cond}}{h_{comp\_out} - h_{cond}}$ Change of the 2nd factor (%)
13th April	0.80	t (°C)	81.2	76.5	47.3	4.8	**84.8**	**96.6**
		h (kJ/kg)	468.4	462.0	279.4	426.1		
15th April	1.43	t (°C)	65.7	61.2	38.4	4.3	**81.8**	**96.9**
		h (kJ/kg)	458.2	452.2	263.0	425.2		

* The 1st factor refers to $h_{comp\_out} - h_{comp\_in}$ in equation (6.2) while the 2nd factor refers to $h_{cond\_in} - h_{cond\_out}$

Table 6.1 compares the average values of both low capacity (13th April) and high capacity (15th April). For instance, on April the 18th, there is 4.7°C of difference in average between t_{comp_out} and t_{cond_in} which leads to a decrease in the 2nd factor of equation (6.2) of 3.4 %, while the 1st factor will change by 15.2 % if t_{cond_in} is taken instead of t_{comp_out}. Calculations have also been made for the high capacity testing from April 15th and similar results are achieved. By comparing the data from Table 6.1, suggestions could be made that TT_1 is preferably to be installed close to the outlet of compressor to get a relatively high accuracy for the Climacheck Method.

Nevertheless, when the distance between condenser and compressor is not so close, it is recommended to connect an additional temperature sensor at the condenser inlet to curb the uncertainty of the Climacheck heating capacity result.

6.6 Overestimation of isentropic efficiency

From the results of Climacheck, a much higher capacity has been observed than from the reference method as can be seen from Figure 6.16.

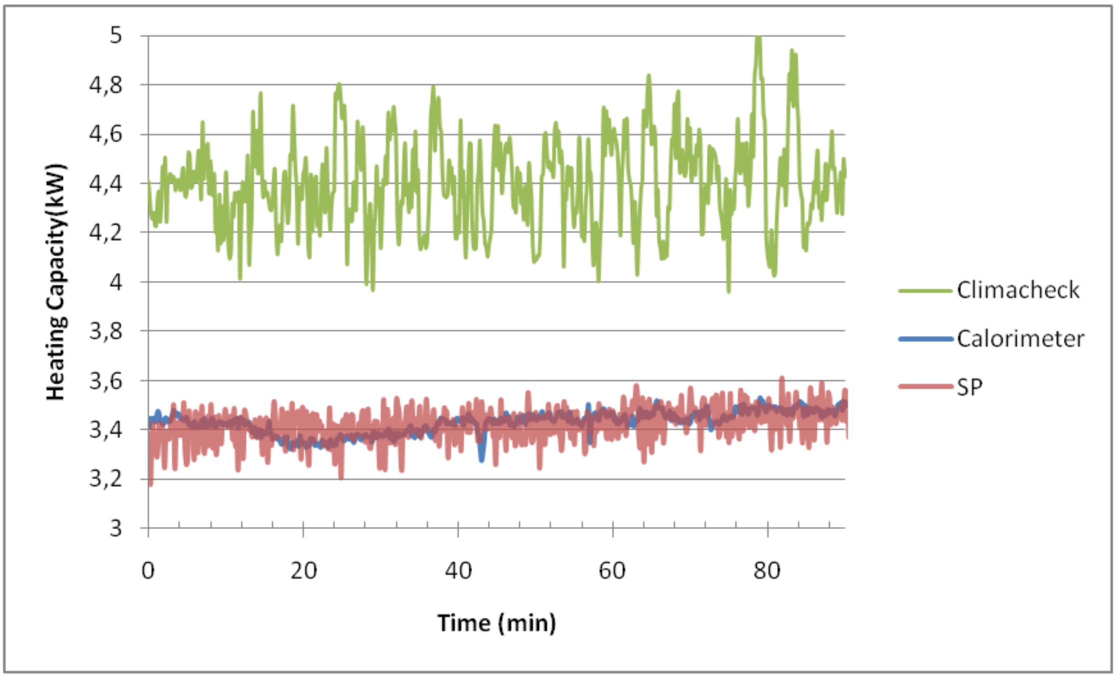

Figure 6.16: Heating Capacity calculated from Calorimeter, SP and Climacheck Method during the same testing period

This observation may indicate a malfunctioning in the Climacheck sensor settings or in the way of calculation of the method. The problem in the Climacheck final results seems related to a very high and fluctuating compressor isentropic efficiency that has been observed, plotted in Figure 6.17. However, it was subsequently determined that the compressor was running “wet” due to a faulty adjustment of the expansion valve. Too low superheat will induce oscillations in the refrigerant flow control and a “wet” (suction gas including liquid droplets) operation of the compressor will yield a faulty estimate of the isentropic efficiency (the Climacheck equipment will automatically indicate this situation).

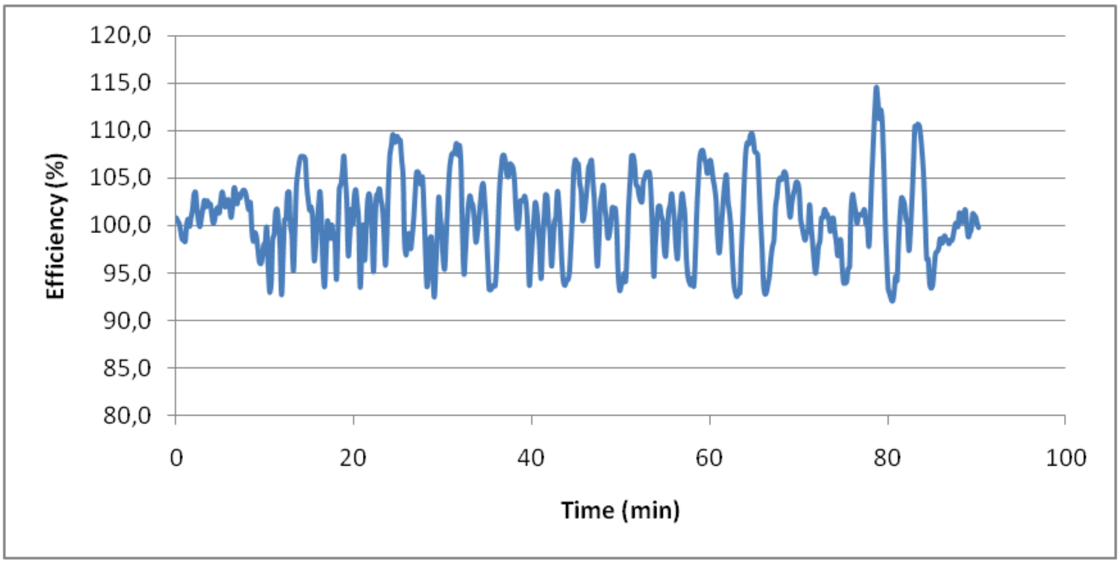

Figure 6.17 : Isentropic efficiency observed by Climacheck

By definition, isentropic efficiency is the ratio of ideal gas compression power to actual absorbed power. Note that this presumes superheat vapour (ideal gas) with no liquid droplets in the suction gas. The main losses occurring in compressors consist of heat losses, friction losses, flow losses and electrical motor losses. Those vary from one type of compressor to another.

The actual case uses a rotary compressor. From the literature the common values of isentropic efficiency for that type of compressor are between 75 and 85 % .

To investigate further, the way of calculation of isentropic efficiency needs to be checked. In Figure 6.18, the compression process for a general case is shown and the difference between actual and isentropic process is highlighted. The equation used for calculating the isentropic efficiency is equation (4.3).

$$\eta_{is} = \frac{h_{comp\_out\_is} - h_{comp\_in}}{h_{comp\_out} - h_{comp\_in}} = \frac{h_{1\_is} - h_2}{h_1 - h_2} \qquad (4.3)$$

Thus, the compressor inlet and outlet enthalpies should be checked to identify the source of the abnormal efficiency.

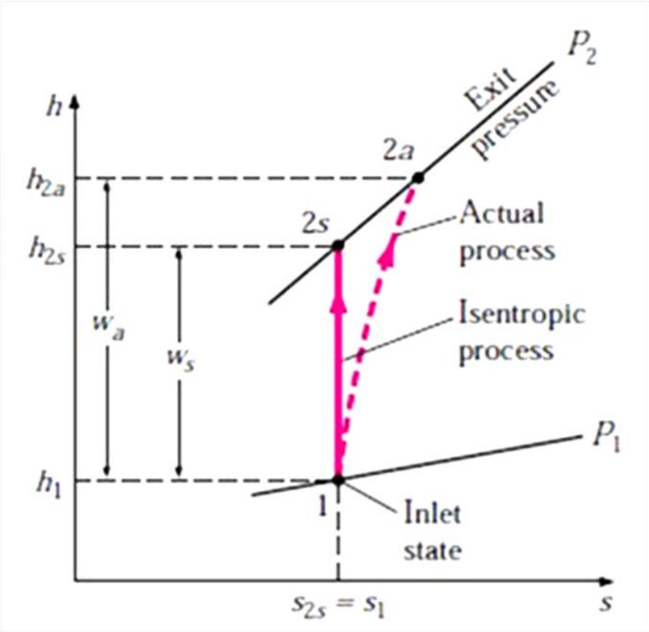

Figure 6.18 : Diagram h-s showing the difference between the actual compression process and the isentropic one (General case)

The data collection period is from April the 13th between 11:59.17 and 13.29:27.

For the further investigation a 90 minutes data sample has been chosen (time span of 10 seconds). In Figure 6.19 an abnormal fluctuation in the values of h_{comp_in} can be observed. That phenomenon is connected to an oscillation of t_{comp_in} sensed by TT_2 (Figure 6.20) whereas the pressures p_1 and p_2 seem stable (Figure 6.21).

The average (180 values in total) for the different data sensors as well as the standard deviation is presented in Table 6.2. It shows for t_{comp_in}value a high fluctuation (standard deviation of 1.569°C) whereas the other data seem more stable.

Table 6.2 : Statistic data for the 90 minutes selected sample.

	p_2 kPa	p_1 kPa	t_{comp_in} (TT_2) °C	t_{comp_out} (TT_1) °C	t_{exp_in} (TT_3) °C
Average value	800.3	2614.7	4.3	65.7	38.4
Standard deviation	7.59	15.732	1.569	0.61	0.395
Minimum value	778.6	2584.7	0.4	63.9	37.6
Maximum value	818.8	2650.4	8.0	67.0	38.9

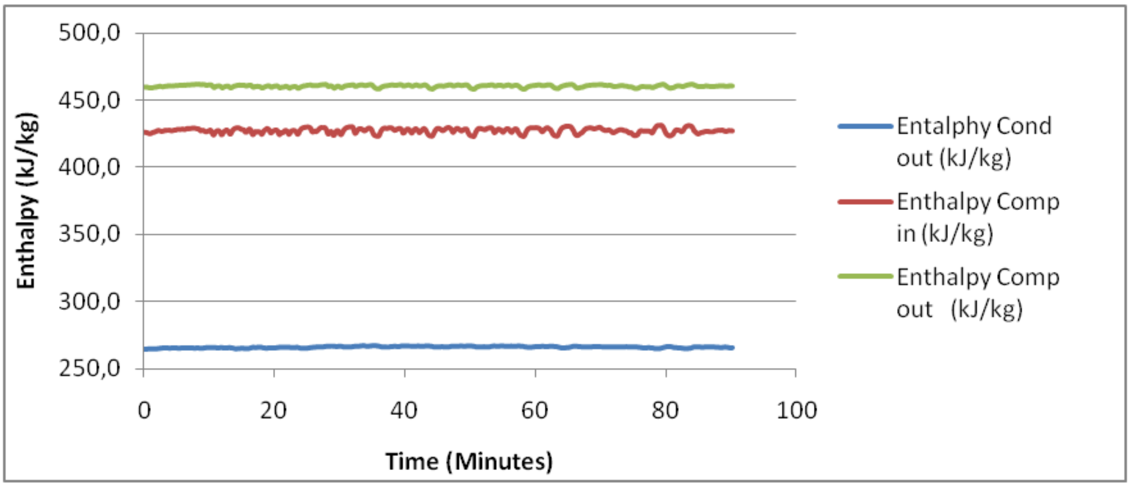

Figure 6.19: Enthalpies obtained with Climacheck sensors and software for the 90 minutes selected sample

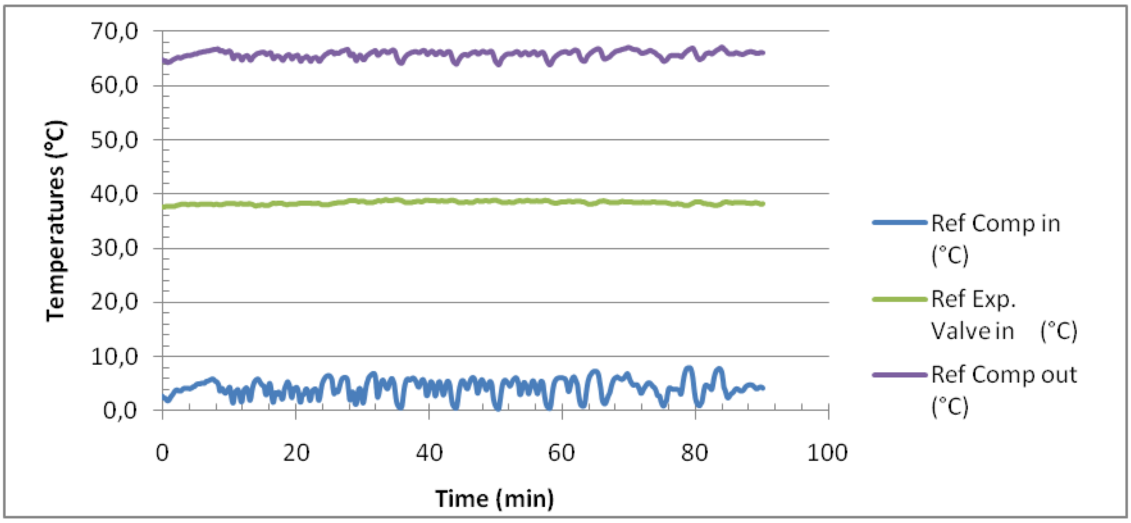

Figure 6.20: Temperatures given by Climacheck sensors for the 90 minutes selected sample

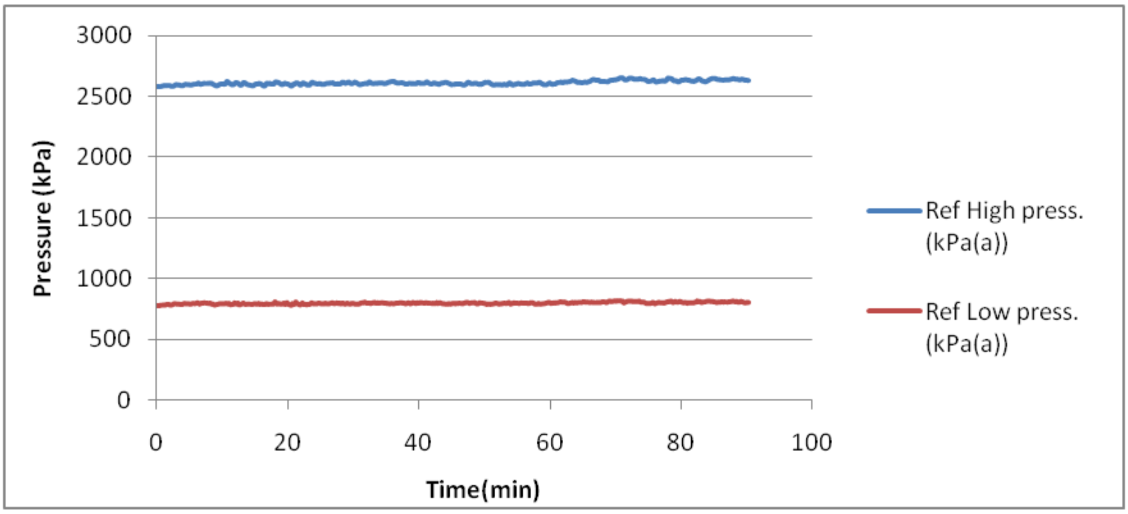

Figure 6.21: Pressures given by Climacheck sensors for the 90 minutes selected sample

Furthermore, according to the manufacturer data (Climacheck has itself developed those sensors), the present measured temperature is within the approved range and the deviation of the temperature sensor is 0.15°C. Then such fluctuation cannot be explained by a deficient sensor.

Consequently, the possibility of liquid refrigerant sucked into the compressor was raised. It is actually indicated in the Field Manual that liquid carry over or excessive oil transport would lead to the low super heat investigated, and when heat pump reacts under the circumstance mentioned, the value from Climacheck is not as satisfying as expected . Indeed that assumption implies that a 2-phase refrigerant (liquid droplets into the gas phase) enters the compressor. And since the temperature sensor takes the surface temperature instead of the inner temperature, Climacheck calculation assumes an evenly distributed temperature inside the tube. Therefore the case of 2-phase refrigerant cannot be handled by that method. The main reason, however, that the Climacheck method does not work with liquid carryover is that the assumptions used in calculation of refrigerant specific enthalpy are no longer valid.

The fluctuations observed point out the phenomenon of mixture between gas and liquid droplets. To confirm that assumption, calculations can be made for one specific "point" in the sampling time. The sensor data are summed up in Table 6.3 and the cycle is plotted in Figure 6.22.

Table 6.3: Sensor data at the studied point

	comp_in (2)	comp_out (1)	evap_in (3)
Temperature (°C)	4.3	65.4	38.2
Pressure (MPa)	0.802	2.62	2.62

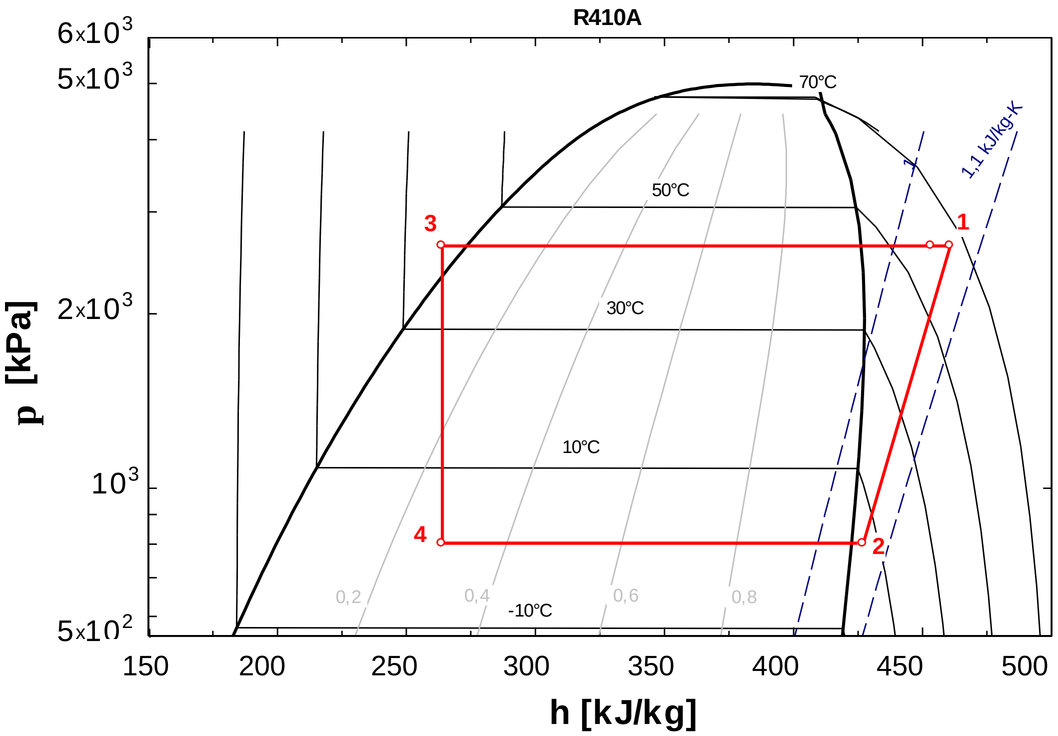

Figure 6.22: p-h diagram of the cycle at the studied point with Climacheck data plotted with the software 'Engineering Equation Solver'

Using the properties of compressor inlet (in Table 6.4), new calculations can be done assuming the low pressure from the sensor being correct. A more reasonable value of isentropic efficiency, 80 %, is assumed, and the corresponding vapour quality was determined as 0.968. As shown in Table 6.5, the heating capacity is around 17 % lower, at 3869 W, which is much closer to the average value observed for the two other methods as (around 3.45 kW for the Calorimeter Method). The new cycle is plotted in Figure 6.23. The new isentropic efficiency is about 80.5 % which is a relatively reasonable value.

Table 6.4: General properties for the compressor inlet

$t_{2,sat}$ (dew point)	0.15 °C
$h_{2,sat}$	426 kJ/kg
$h_{2,liquid}$	200 kJ/kg
P_2	0.802 MPa

Table 6.5: Input and calculated data from the Climacheck box and with an assumption of biphasic refrigerant entering the compressor

	From Climacheck results	**From assumption**
x_2	1	0.97
t_2 (°C)	4.3	0.15
p_2 (MPa)	0.802	0.802
h_2 (kJ/kg)	426.0	419
$\dot{Q}_{heat}$ (W)	4658	3869

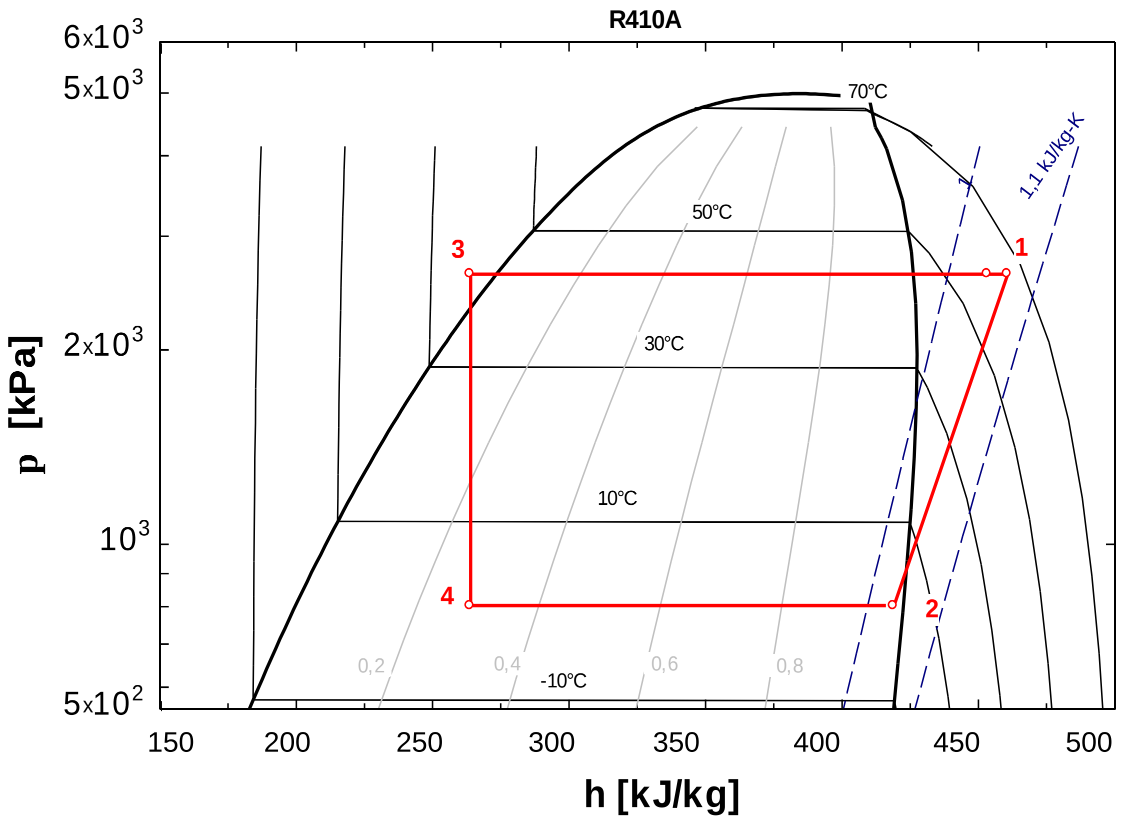

Figure 6.23 P-h diagram of the cycle at the studied point with assumption of Table 6.5 plotted with the software 'Engineering Equation Solver'

Another possibility to explain the high isentropic efficiency as suggested in the field manual is a large amount of oil mixing in the refrigerant, leading to a significant change in the actual fluid properties. Therefore, the refrigerant should be considered as a mixture of R410A and oil for the calculation. The oil fraction can be assumed to be 2 % and even higher next to the compressor which can be a possible but not likely reason for the fluctuations in enthalpy which has been observed.

7 Evaluation

7.1 Uncertainties

In this paper, measurement uncertainty calculation process is based on BIPM (Bureau International des Poids et Mesures), which is widely accepted internationally. This evaluation standard takes two different types of uncertainties into consideration, named as type A uncertainty and type B uncertainty, which are supposed to be independent of each other. Type A uncertainty is also called estimated uncertainty and it is determined by statistical methods, while type B uncertainty, also called expected uncertainty, is determined by more or less subjective methods. Further explanation could be achieved from . The combined uncertainty for each parameter measured is then computed by equation (7.1)

$$u_x = \sqrt{{s_x}^2 + {w_x}^2} \qquad (7.1)$$

In addition, a numerical factor k is introduced to achieve a satisfying confidence level, normally referring to 95 %. This factor k is known as coverage factor and it is in the range of 2-3.

Then the overall uncertainty is given by equation (7.2)

$$U_x = k \cdot u_x \qquad (7.2)$$

The overall uncertainty of the COP value tested in this project is highlighted and evaluated, as expressed in equation (7.3). The confidence level of 95 % is chosen with the coverage factor k=2 .

$$\frac{U\_COP}{COP} = k \cdot \frac{u\_COP}{COP} = k \cdot \sqrt{\left(\frac{u\_\dot{W}_{HP}}{\dot{W}_{HP}}\right)^2 + \left(\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}}\right)^2} \qquad (7.3)$$

Prior to collecting data for uncertainty analysis, it is essential to ensure that the heat pump is operated under stable condition for a minimum period of 30 minutes. For this case, dynamic behaviour such as defrosting process shall not be recommended for measurement uncertainty analysis . In this project, a steady period of 100 minutes is selected from the 13th April testing. The calculation is further explained as below.

The power supplied to the heat pump has a combined uncertainty $u\_\dot{W}_{HP}$ of 1.2 % during the testing period. And it could be applied for all the three testing methods. Uncertainty of heating capacity is calculated separately due to the different testing principles.

For the SP Method, uncertainty of heating capacity is calculated based on equation (7.4).

$$\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}} = \sqrt{\left(\frac{u\_c_{p\_a}}{c_{p\_a}}\right)^2 + \left(\frac{u\_\rho_a}{\rho_a}\right)^2 + \left(\frac{u\_\dot{V}_a}{\dot{V}_a}\right)^2 + \left(\frac{u\_(t_{a\_out} - t_{a\_in})}{(t_{a\_out} - t_{a\_in})}\right)^2} \qquad (7.4)$$

The calculated result is 4.7 % according to the recorded data. The value with 95 % confidence level is 9.4 %, which is higher compared with the measurement tolerance for the standard, which is 8.7 % given in , but it is still under 10 %. The uncertainty of the COP measurement is then calculated to be 9.7 %, with 95 % confidence level.

For the Climacheck Method, uncertainty of heating capacity is calculated based on equation (7.5), as suggested by NT standard

$$\frac{\Delta\dot{Q}_{heat}}{\dot{Q}_{heat}} = \frac{\Delta\dot{W}_{comp}}{\dot{W}_{comp}} + \frac{(h_3 - h_2)\cdot\Delta h_1}{(h_1 - h_2)\cdot(h_1 - h_3)} + \frac{\Delta h_2}{(h_1 - h_2)} - \frac{\Delta h_3}{(h_1 - h_2)\cdot(h_1 - h_3)} - \frac{\Delta f}{(1-f)} \quad (7.5)$$

and $\frac{u_{\dot{Q}_{heat}}}{\dot{Q}_{heat}} =$

$$\left[\left(\frac{u_{\dot{W}_{comp}}}{\dot{W}_{comp}}\right)^2 + \left(\frac{(h_3 - h_2)\cdot u_{h_1}}{(h_1 - h_2)\cdot(h_1 - h_3)}\right)^2 + \left(\frac{u_{h_2}}{(h_1 - h_2)}\right)^2 + \left(\frac{u_{h_3}}{(h_1 - h_2)\cdot(h_1 - h_3)}\right) + \left(\frac{u_f}{(1-f)}\right)^2\right]^{\frac{1}{2}}$$

The result is 3.4 % when the enthalpy uncertainty analysis is not based on the condition of temperature and pressure tested, as the method described in NT standard, due to the lack of information concerning the refrigerant R410A. Instead, enthalpy and electricity input to the compressor are taken directly from the Climacheck data file. The COP uncertainty is then able to be determined as 7.1 % at 95 % confidence level.

For the Calorimeter Method, uncertainty of heating capacity is calculated based on equation (7.6).

$$\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}} = \sqrt{\left(\frac{u\_\dot{Q}_{cool}}{\dot{Q}_{heat}}\right)^2 + \left(\frac{u\_\dot{Q}_{loss}}{\dot{Q}_{heat}}\right)^2 + \left(\frac{u\_\dot{W}_{fans}}{\dot{Q}_{heat}}\right)^2} \qquad (7.6)$$

The result is 2.8 % and the COP uncertainty is 6.0 % at 95 % confidence level accordingly.

Appendix B shows all the parameters' related to the COP uncertainty calculation, type A and type B uncertainties for each parameter are also attached.

As could be noticed from the uncertainty budgets in the appendix, the uncertainty of the refrigerant temperature at the inlet of compressor is extraordinarily high, and even up to more than 50 %. This phenomenon suggests that Climacheck Method is not carried out under suitable conditions, as studied in Section 6.7. This could have been remedied by proper adjustment of the expansion valve according to Fahlén to achieve sufficient super heat. There was, however, not sufficient time for this during the current thesis work. Considering the measurement uncertainties, as summarized in Table 7.1, the methods tested are all within acceptable range during steady stated. While comparing the result achieved, the Climacheck Method has a large variation from the testing results from the other two methods, which indicates that further improvements shall be applied either to the heat pump operating condition or to the measurement method itself.

Table 7.1: Measurement Uncertainties of all the tested methods

	SP Method	Climacheck Method	Calorimeter Method
$\frac{u\_\dot{Q}}{\dot{Q}}$	4.7 %	3.4 %	2.8 %
$\frac{u\_COP}{COP}$	4.9 %	3.6 %	3.0 %
$\frac{U\_COP}{COP}$	**9.7 %**	**7.1 %**	**6.0 %**

7.2 Limitations of the methods tested

Due to the lack of anemometer and Coriolis flow meter, only three test methods, namely the SP Method, the Climacheck Method and the Calorimeter Method, were carried out in the lab during this study. Here the Calorimeter Method is used as the reference to be compared with the other two methods. As discussed in Chapter 4, these three methods could be carried out at the same time for the reason that they are all based on different principles which are supposed to not be influencing to each other. And it is also the only solution that makes sense when different testing methods have to be run with the same heat pump under the same outdoor air conditions, if the outdoor air condition is not controllable. However, the results from Section 6.4 suggested that the performance of the air-to-air heat pump would be affected if the air flow or the temperature scenario is artificially changed, which indicates that the position of air collector might, to a certain extent, influence the performance of the heat pump. In particular, back mixing effect from the indoor unit outlet air is alleviated. The extent of back mixing could be estimated by carrying out a parallel measurement only with the heat pump in a separated chamber, temperatures at the inlet and outlet of the indoor unit as well as the room temperature are recorded, thus energy balance could be created to determine the share of outlet air back mixed. Even though the result follows pretty well what is achieved from the Calorimeter Method, it cannot reflect the true value when heat pump is working without such disturbances.

The Calorimeter Method could achieve the high accuracy when running in long term to test the COP, but it is the standard which can only be used for testing heat pump in the lab, besides a good quality Climate Chamber is always needed. However, the Calorimeter Method is not good at precisely following the dynamic heat process, either, though it would not be of any significance for long duration testing. A model was developed for the purpose of determining the heat removed from defrosting process .

For the measurement in field, both internal and external methods could be applied. The Climacheck Method has good user interface and professional analyzing software, the size of the equipment is considered to be not so bulky, except the current clamps when the power input to the compressor is measured. As can be noticed in Figure 7.1, the space inside the outdoor unit is limited. It is extremely hard to have all the clamps and cables within the box. Furthermore, the insulations of temperature sensors were not explicitly required, especially for the ones outside and exposed to the harsh outdoor environment, installations should be as good as possible in order to make sure the surface temperature sensors give trustable data. Climacheck can only give reasonable results under ideal working conditions, to be exact, no liquid droplet is allowed in the suction pipe, which is very hard to be guaranteed even though the heat pump has good quality. Otherwise, the surface temperature sensor would give the value of the refrigerant at the mixed state; an unreasonably high isentropic efficiency is achieved, which has already been explained in Section 6.6 in detail. This is actually a major advantage of the Climacheck Method. It will provide a result but also warn when the heat pump is not working properly. In this particular case, the Climacheck information would suggest adjustment of the expansion valve not only to provide a relevant measuring situation, but primarily to make the heat pump operate as it should.

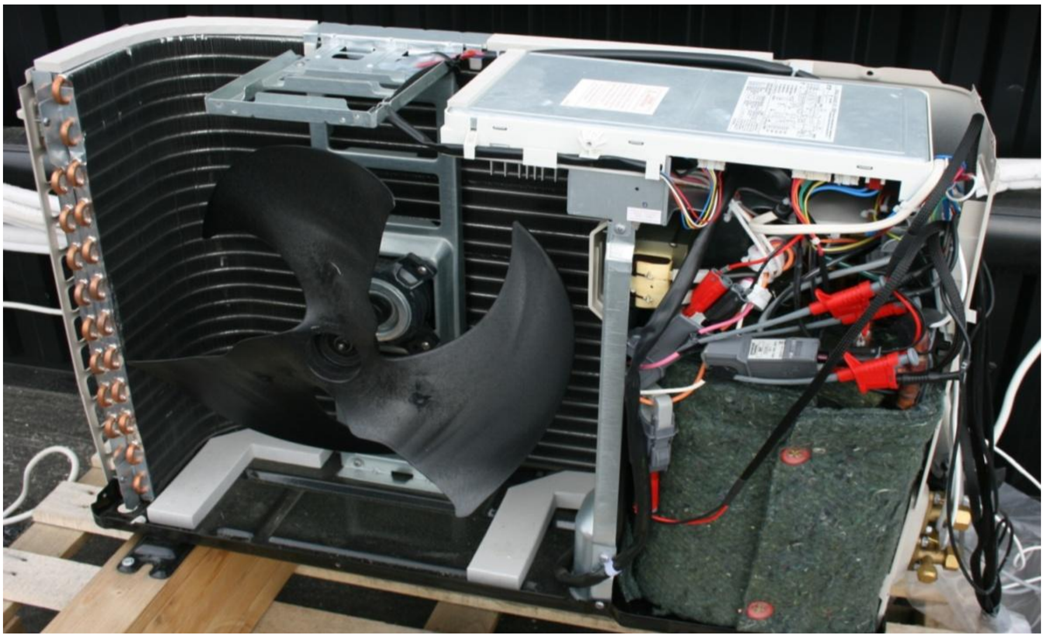

Figure 7.1: Outlook of how Climacheck current clamps are installed

The SP Method seems to be working well concerning the relatively accurate results compared with the Calorimeter Method. However, it might not be the normal working state of the heat pump if the air collector is installed; some further testing is still needed though. In addition, the instruments needed are bulky so that it is not the optimal choice for testing in field under long term tests.

Neither the Climacheck Method nor the SP Method is feasible for measurement during defrosting, which suggests that they are not suitable for long term testing for HSPF where defrosting plays an important role in determining the result. For the case when there is no reverse cycle, almost the total electricity consumption of the heat pump is contributed to heating operation, while for the case with a normal reverse cycle, integration of electricity input could be made concerning both heating period and defrosting period to make further comparisons. For instance, evaluation was made based on the first complete cycle with defrosting in Figure 6.1, and the electricity consumed during defrosting only accounts for 2.6 %, and it takes even less share when the heating operation is longer. However, for the SP method, the defrosting causes an offset of the static pressure, which results in inaccurate measurements in the following heating cycles. The size of the offset has not been found to be generic in this diploma work, see Section 6.4.

Tokyo Gas Method and Refrigerant Enthalpy Method were not carried out in practical experiment for this study due to the shortage of equipment; however, theoretical analysis has been made regarding both of them. Tokyo Gas Method does not need such a bulky equipment setup for field testing, but the accuracy of air flow volume rate is doubtful and the method has no solution for defrosting issues, either. Refrigerant Enthalpy Method is supposed to be accurate according to the results from an air-to-water heat pump in the paper . What is also noticeable that this method takes into consideration of vapour quality at some interesting locations in the refrigerant pipe. Therefore it is possible to determine the enthalpy together with the corresponding temperature and pressure even during defrosting. For the heating capacity achieved on the refrigerant side, the average deviation from the water side lies between -2.5 % and 7.5 % from the report , which is quite acceptable. But the difficulty of installing the Coriolis flow meter acts as an obstacle for this method to be carried out in field measurement.

8 Forward study for a potential new external method

The analysis of the already existing methods resulted in the willingness of developing a new method accurate enough but less bulky. Since the collector is used principally for measuring the volume flow rate of the air blowing out from the indoor unit, the idea of determining that value from the indoor fan consumption was raised up.

In theory, the relationship between flow rate and electric power is shown in equation (8.1) .

$$\dot{W}_{in\_fan} = \alpha \, \dot{V}_a^{\,n} \quad (8.1) \qquad \text{with } 2< n < 3$$

Here α is a constant and it could be determined by a preliminary test with SP collector set up.

That method has to be probed to check whether it is feasible. Here some reflections about whether in theory that method could be achieved are given.

For the actual indoor unit, no easy way to take directly the indoor fan consumption was found (through clamps for instance such as the one used for the compressor). Instead, the indoor fan consumption could be calculated by using two power meters which monitor the outdoor unit consumption as well as the total power consumption. However the difference between those two values is very low (below 100W) while the calibration is not accurate enough for the measured power resulting in a very likely high uncertainty. An alternative method would be to measure the indoor fan consumption on the circuit board; an investigation about the feasibility of that method is suggested as a further work.

For the tested heat pump, the fan speed can be set manually to low, medium and high speed. There is also an automatic mode, which, through the controlling system, estimates which fan speed is required depending on the outdoor conditions and the heat demand. The average values (when conditions on static pressure are verified) are compared to the values from the manufacturer Bosch and the supplier IVT (cooling mode) in Table 8.1.

Table 8.1: Volume flow rate in m^3/s from different sources compared to the test results

		Low speed	Medium speed	High speed
Cooling mode	From testing	0.1041	0.1284	0.1565
	From Bosch[4]	0.1150	0.1450	0.1767
	From IVT[5]	0.1000	0.1283	0.1650
Fan-only mode[6]		0.0968	0.1241	0.1505
Heating mode	From testing	No data[7]	No data[5]	0.1571
	From IVT	0.1217	0.1533	0.1967

Even if the values from the different sources are quite close from each other, it seems difficult to associate one fan speed mode with a certain volume flow rate, especially since the values from the supplier differ between cooling and heating modes.

Furthermore, the actual air volume flow rate depends on how clean the filter is, and it is very hard to make sure that the filter will be kept clean by the user on the field.

Thus air volume flow rate remains the trickiest parameter to measure for external method. From the preliminary analysis, the proposed method shows a few limitations related to the uncertainty of the indoor fan power measurement and whether the coefficient α is constant over time. Nevertheless further investigation is needed to estimate how accurate the volume flow rate calculation would be.

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[4] From [22]
[5] From own tests performed by IVT
[6] The heat pump tested had the possibility to be used on fan-only mode through the Plasmacluster® function (air purification process).
[7] No data are available for that speed because the outdoor weather temperature became too high resulting in malfunctioning of the heat pump.

9 Conclusions

In this project, three methods, namely SP Method, Climacheck Method and Calorimeter Method, were practically carried out with testing results evaluated. Tokyo Gas Method and Refrigerant Enthalpy Method were only theoretically studied due to the lack of necessary testing equipment. Lab setups were organized to satisfy the requirements for testing these three methods at the same time.

For long term testing, the SP Method can not accurately represent the HSPF value for the reason that the defrosting process could not be properly recorded. The method could however be used to sample the COP value for certain test points. The Calorimeter Method is not able to determine the heat removed during defrosting in the setup used in this diploma work, but it is accurate if several cycles with defrosting are taken into account to calculate the COP value over the corresponding period of time. The calorimeter method is not intended for field measurement however.

Different sampling time intervals are also selected to see the influence to the COP value calculation, in particular for cycles with defrosting. An interval larger than one minute is not recommended, since it is too rough to follow the whole defrosting process, which is no longer than 10 minutes. Results from Section 6.1 suggest that having 30 seconds as sampling time is quite acceptable with the difference of only 0.001 in COP calculation. Furthermore, experiment of heat balance maintenance for the Calorimeter Method was also carried out, by having the cooling input to the indoor compartment minimized during defrosting. However, the COP calculation result gave no big difference from the case without cooling capacity controlled, which indicates that in the long term, testing with the Calorimeter Method does not have to be bothered with heat balance maintenance, unless the condition’s deviation is beyond the testing standard. Conclusion could also be drawn that heat pump performance changes sensitively regarding different outdoor air conditions; outdoor relative humidity plays an important part concerning the frequency and form of defrosting process at around zero to ten degrees centigrade.

For short term testing, the combination testing of all the three methods during steady state makes it possible to evaluate each method. The measurement uncertainties for COP calculation are acceptable; they are 9.7 %, 7.1 % and 6.0 % for the SP Method, the Climacheck Method and the Calorimeter Method separately. Heating capacity tested from the SP Method and the Calorimeter Method makes no big difference, but the one from the Climacheck Method is higher. However, the accuracy of the Climacheck Method testing is considered to be very low, due to the difficulty of accessing good value of refrigerant properties when the heat pump is running with relatively low superheating. In addition, a second temperature sensor other than the one close to the outlet of compressor is suggested on the discharge line, if the distance between compressor and condenser is relatively long. As illustrated in Section 6.5, in this way, the accuracy of the Climacheck Method could be improved considerably. For the SP Method, the influence from changing static pressure at the outlet of indoor unit has also been studied, by setting the SP circulation fan at different speed. The results prove that the air collector attached could affect the performance of the testing unit during dynamic process, and it is essential to keep the static pressure at 0 Pa and minimise the variation.

New proposals of testing method were also made and practically tested in the lab, which aim at testing the volume flow rate of air related to different indoor unit fan speed. The results show that it might be possible to establish a relationship between electrical input to the indoor fan and fan speed through preliminary testing including SP equipment. Then, by placing temperature sensors to the air input and output to the indoor unit, it might be possible to estimate the heating capacity of the heat pump. Nevertheless the accuracy of this method needs to be tested in the lab.

10 Perspective

Accurate measurement of air-to-air heat pump COP value in field testing makes great sense in energy saving as well as alleviation of greenhouse gas emission, and more work is still needed to be done in this field. The limitation of this report is the challenge to carry out all the testing methods strictly under the same conditions, therefore a controllable outdoor climate chamber is preferably to be applied in the future testing. It is also interesting to have the thermal expansion valve adjusted to achieve a higher superheat, thus to have the Climacheck Method tested properly in a more stable condition. Furthermore, it is also a big challenge to have the refrigerant properties tested accurately even during dynamic process with less bulky instruments, and it could be an interesting field for further development of the Climacheck Method. For the SP Method, it is critical to have a good evaluation of to what extent heat pump performance is affected by applying the air volume flow rate measuring system, and further to develop an optimal way to get the air volume flow rate when the operating conditions change

11 References

[1] Harper, Carl (2011). “The history of heat pump” [Online]. *EHow*. Available at http://www.ehow.com/about_6682179_history-heat-pump.html. Accessed May 12, 2011.

[2] Lindholm, Torbjörn (2009). “The vapour compression process (Part 1)” in *Air-condition, refrigeration and heat pump technology*. Compendium K2009:1, pp3-30 (Chalmers University of Technology) Gothenburg. ISSN 1652-6007.

[3] Forsén, Martin (2010). “Focus on selected European markets: Sweden” in European Heat Pump Association, *European heat pump Statistics, Outlook 2009*, pp. 46-51. Brussels.

[4] Polyzou, Olympia (2011). “The SEPEMO-Build project” in *SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector*. Contract for the European Communities - Contract No.: IEE/08/776/SI2.529222, Pikermi Attiki (Greece).

[5] Ding, Yanjing; Ma, Guoyuan; Chai, Qinhu, and Jiang, Yi (2004). "Experiment investigation of reverse cycle defrosting methods on air source heat pump with TXV as the throttle regulator," *International Journal of Refrigeration*, vol. 27, no. 6, pp. 671-678, September 2004.

[6] Fahlén, Per. (1996). *Frosting and defrosting of air-coils*. Building Services Engineering, Thesis for Ph.D., D36:1996, pp. 240 (Chalmers University of Technology) Göteborg.

[7] University of Plymouth (1999). “Refrigeration (& Heat pumps)” [Online]. *University of Plymouth, School of Marine Science and Engineering.* Lecture Notes from Module Applied Thermodynamics THER205. Available at http://www.tech.plym.ac.uk/sme/THER205-web/refrig.pdf. Accessed May 1, 2011.

[8] Frazer, Stephen (2003) “Refrigerants” [Online]. *Student Notes for Buiding Services.* Available at http://www.arca53.dsl.pipex.com/index_files/frig4.htm. Accessed May 20, 2011.

[9] CIPCO (2009). “HSPF” [Online]. *CIPCO Central Iowa Power Cooperative*. Available at: http://cipco.apogee.net/res/rehhspf.asp. Accessed May 5, 2011.

[10] Rivière, Philippe; Coevoet, Michel; Tran, Christophe and Nordman, Roger (2011). *D4.1 Guideline for heat pump field measurements*”. Version 1.0 – Draft, pp. 4-5. Contract for the European Communities - Contract No.: IEE/08/776/SI2.529222 (SEPEMO Heat Pump SEasonal PErformance Monitoring*)* Borås.

[11] Fahlén, Per (2005). “Performance audits of heat pumps - procedures and uncertainties”. *8th IEA Heat Pump Conference*. Las Vegas, 2005-05-30--06-02. vol. CD - proceedings.

[12] Fahlén, Per (1996). *Field testing of refrigeration and heat pump equipment -*

Performance data inferred from measurements in the refrigerant system. Report SP-AR 1996:23, pp. 1-12 plus 1-29 (SP Swedish National Testing and Research Institute) Borås.

[13] Fahlén, Per and Kjellgren, Christer (1995). *Performance testing of air-to-air heat pumps* (in Swedish). Report SP-arbetsrapport 1994:51, pp. 16. (SP Swedish National Testing and Research Institute) Borås.

[14] Won, Anna; Ichikawa, Toru; Yoshida, Satoshi; Sadohara, Satoru (2009). "*Study on Running Performance of a Split-type Air Conditioning System Installed in the National University Campus in Japan"*, Journal of Asian Architecture and Building Engineering, vol. 8, no. 2, pp. 579-583.

[15] Fahlén, Per (2004). "Methods for commissioning and performance checking of heat pumps and refrigeration equipment". *IIR conference compressors 2004.* Papiernicka, Slovakia, 2004-09-29--10-01 (International Institute of Refrigeration)

[16] Berglöf, Klas (2005). "Method and Potential for on-site performance validation of air conditioning, refrigeration, and heat pump systems". *8th IEA Heat pump Conference*, Las Vegas, 2005-05-30--06-02.

[17] Tran, Christophe and Rivière, Philippe (2011). "On side refrigerant measurement of heat pump seasonal performance". *10th IEA Heat pump Conference*. Tokyo, 2011-06-27--08-01.

[18] Climacheck (2010). Specifications for Climacheck Pt1000 Temperature Sensors. Version 2010-09-30 (Climacheck Sweden AB) Nacka.

[19] European Committee for Standardization (2007). "Part2: Testing conditions," in *EN14511 , Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors*, EN 14511-2:2008, pp. 5-32 (CEN).

[20] Fahlén, Per; Johansson, Kent (1989*). Capacity measurement on heat pumps - A simplified measuring method (in Swedish)*. Report R4:1989, pp. 1-8 (Statens Råd för Byggnadsforskning) Stockholm.

[21] Zou, Lei (2011). *Dynamic heat process in a climate chamber.* Building Services Engineering, Master Thesis Report (unfinished when referred here) (Chalmers University of Technology) Göteborg.

[22] BOSCH (2008). *Service Manual for Room Air conditioner EHP6.0A. V*ersion 2008-08 *(BOSCH)*

[23] *Fahlén*, Per (2008). "Part 3: Air coolers" in *Air-conditioning, refrigeration and heat pump technology* . Compendium K2003:03rev3, pp. 62 (Chalmers University of Technology) Göteborg.

[24] Air Conditioning and Refrigeration News (2006). "Compressors Part 5" [Online]. *ACR-News*. Published by Faversham House Group Ltd (South Croydon, UK) on May 1, 2006. Available at http://www.acr-news.com/masterclass/ news.asp?id=657&title=Masterclass%3A+Compressors+-+Part+5. Accessed May 10, 2011.

[25] Vanka, S. Pratap. (2008) "Isentropic Efficiencies of Turbines, Compressors and Nozzles" [Online]. *University of Illinois - Mechanical Science and Engineering*. Notes from Lecture 28 on November 3, 2008, pp 14-15. Available

at http://mechse.illinois.edu/media/uploads/course_websites/lecture___28.2008 1103.490f20b0965045.32096808.ppt. Accessed April 30, 2011.

[26] BETP (2011). “Isentropic efficiencies of Compressors and Pumps” [Online]. *BETP Best Place for Engineering and Technology*. Published on March 26, 2011. Available at http://www.betp.net/2011/03/isentropic-efficiencies-of-compressors-and-pumps/. Accessed May 3, 2011.

[27] Climacheck (2010). *Climacheck Field Manual Performance Analyzer PA Pro v3.1 - Measuring and Analysing System for Refrigeration, Air-conditioning and Heat Pumps*. Version 2010_05-17 (Climacheck Sweden AB) Nacka.

[28] Fahlén, Per (1994). *Performance tests of air source heat pumps under frosting conditions - Quality of Results*. SP Report 1994:01, pp. 1-338 (SP Swedish National Testing and Research Institute.) Borås.

[29] Fahlén, Per (1996). *Field testing of refrigeration and heat pump equipment - General conditions*. Report SP-AR 1996:22, pp. 1-6 plus 1-44 (SP Swedish National Testing and Research Institute.) Borås.

[30] Kadribegovic Ragib (2009). “HVAC Applied Flow Theory” [Online]. *Chalmers University f Technology - Department of Energy and Environment -Building Services*, Lecture Notes from Heating, Ventilation, Air-Conditioning Systems ENM045 on September 29,2009. Available at https://www.student.chalmers.se/sys/FileHandling/downloadFile?folderName=6415hout&path=Lectures/4_Hvac_Systems&fileId=Applied_flow_theory_lecture_01.pdf. Accessed 2011-05-20.

Appendices

Appendix I: Testing unit properties

Testing Unit			Bosch room air conditioner		Unit
Indoor Unit/Outdoor Unit			EHP6.0AA/I	EHP 6.0AA/O	
Rated heating capacity (Min-Max)			4.0 (0.9-6.0)		kW
Rated input electricity in heating mode (Min-Max)			950 (200-1700)		W
Power factor in heating mode			94 %		
Maximum operating current			8.7		A
Compressor	Type		Hermetically sealed rotary type		
	Model		DA111A1F22F		
	Oil charge		450cc (Ester Oil VG74)		
Refrigerant system	Evaporator		Louver Fin and Grooved tube type		
	Condenser		Corrugate Fin and Grooved tube type		
	Control		Expansion valve		
	Refrigerant (R410A)		990		g
	De-ice system		Micro computer controlled reversed systems		
Fan system	Direct driven		Cross flow fan	Propeller fan	
Air flow quantity (at cooling)		High	10.6	30.2	m^3/min
		Low	8.7	-	m^3/min
		Soft	6.9	-	m^3/min
Refrigerant coupling			Flare type		
Refrigerant tube size Gas. Liquid			3/8’’.1/4’’		
Grain pipe			O.D 18		mm

Appendix II: Measurement Uncertainties

Table 1: Measurement Uncertainty Budget of Power input to the Heat Pump

	$\dot{W}_{HP}$
	W
$\bar{x}$	920.4607
s_x	4.999073
$s_{\bar{x}}$	0.204086
$w_{\bar{x}}$	10.702111
$u_{\bar{x}}$	10.704056
$\frac{u_{\bar{x}}}{\bar{x}}$	0.0116

Table 2: Parameters recorded for the SP Method measurement uncertainty calculation

	t_{a_in}	t_{a_out}	p_{diff}
	°C	°C	Pa
$\bar{x}$	19.7	39.1	350.8
s_x	0.0374	0.109	13.5
$s_{\bar{x}}$	0.0015	0.004	0.552
$w_{\bar{x}}$	0.0487	0.186	23.280
$u_{\bar{x}}$	0.0487	0.186	23.286
$\frac{u_{\bar{x}}}{\bar{x}}$	0.0025	0.0047	0.066

Table 3: Uncertainty calculation for the SP Method

$\frac{U\_COP}{COP}$	$\frac{u\_COP}{COP}$	$\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}}$	$\left(\frac{u\_c_{p\_a}}{c_{p\_a}}\right)^2$	$\left(\frac{u\_\rho_a}{\rho_a}\right)^2$	$\left(\frac{u\_\dot{V}_a}{\dot{V}_a}\right)^2$	$\left(\frac{u\_(t_{a\_out} - t_{a\_in})}{(t_{a\_out} - t_{a\_in})}\right)^2$
0.097	0.0486	0.047	0	0	0.002130	0.000094

Tables 4a and 4b: Parameters recorded for the Climacheck Method measurement uncertainty calculation

	p_2	p_1	TT_2	TT_1	TT_3
	Bar	Bar	°C	°C	°C
$\bar{x}$	6.9509	25.0622	4.0714	65.5820	38.3022
s_x	0.059380	0.088579	1.478224	0.594567	0.275824
$s_{\bar{x}}$	0.002424	0.003616	0.060348	0.024273	0.011260
$w_{\bar{x}}$	0.102705	0.180591	2.119673	0.971120	0.405394
$u_{\bar{x}}$	0.102734	0.180627	2.120532	0.971424	0.405550
$\frac{u_{\bar{x}}}{\bar{x}}$	0.014780	0.007207	0.520838	0.014812	0.010588

	h_2	h_1	h_3	$\dot{W}_{comp}$
	kJ/Kg	kJ/Kg	kJ/Kg	kW
$\bar{x}$	427.07	460.22	264.09	0.801
s_x	1.559	0.799	0.514	0.014
$s_{\bar{x}}$	0.063	0.032	0.021	0.0006
$w_{\bar{x}}$	2.305	1.391	0.746	0.028
$u_{\bar{x}}$	2.306	1.391	0.747	0.028
$\frac{u_{\bar{x}}}{\bar{x}}$	0.0054	0.0030	0.0028	0.0350

To be noticed, enthalpy uncertainties in Tables 4a and 4b are not calculated based on the uncertainties of temperature and pressure at the corresponding points due to the lack of information concerning refrigerant R410A. Instead, they are directly read from the Climacheck data file.

Table 5: Uncertainty calculation for the Climacheck Method

$\frac{U\_COP}{COP}$	$\frac{u\_COP}{COP}$	$\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}}$	A1	A2	A3	A4	A5
0.071	0.036	0.034	0.035	− 0.035	0.070	0.004	0.032

Here A1-A5 represents separated parts which comprise the uncertainty calculation equation for heating capacity, shown as equation (1) to equation (5) (ref. Per Fahlèn. SP AR 1996:23. NT standard)

$$A1 = \frac{u\_\dot{W}_{comp}}{\dot{W}_{comp}} \qquad (1)$$

$$A2 = \frac{(h_3 - h_2) \cdot u\_h_1}{(h_1 - h_2) \cdot (h_1 - h_3)} \qquad (2)$$

$$A3 = \frac{u\_h_2}{(h_1 - h_2)} \qquad (3)$$

$$A4 = \frac{u\_h_3}{(h_1 - h_3)} \qquad (4)$$

$$A5 = \frac{u\_f}{(1 - f)} \qquad (5)$$

Table 6: Parameters recorded for the Calorimeter Method measurement uncertainty calculation

	t_{w_in}	t_{w_out}	$\dot{V}_w$	$\dot{W}_{fans}$	t_{a_amb}	t_{a_room}
	°C	°C	m³/h	W	°C	°C
$\bar{x}$	13.19	16.25	1.2360	1009.66	22.1	22.7
s_x	0.0695	0.052607	0.001913	8.934	0.0154	0.0304
$s_{\bar{x}}$	0.00284	0.002148	0.000078	0.365	0.0006	0.0012
$w_{\bar{x}}$	0.09227	0.073949	0.011381	12.88	0.0278	0.0415
$u_{\bar{x}}$	0.09231	0.073980	0.011382	12.88	0.0278	0.0416
$\frac{u_{\bar{x}}}{\bar{x}}$	0.00700	0.00455	0.00921	0.0128	0.001256	0.001833

Table 7: Uncertainty calculation for the Calorimeter Method

$\frac{U\_COP}{COP}$	$\frac{u\_COP}{COP}$	$\frac{u\_\dot{Q}_{heat}}{\dot{Q}_{heat}}$	$\left(\frac{u\_\dot{Q}_{cooling}}{\dot{Q}_{heat}}\right)^2$	$\left(\frac{u\_\dot{Q}_{loss}}{\dot{Q}_{heat}}\right)^2$	$\left(\frac{u\_\dot{W}_{fans}}{\dot{Q}_{heat}}\right)^2$
0.06	0.03	0.028	0.00076	0.0000014	0.0000142

Here $\left(\frac{u\_\dot{Q}_{cooling}}{\dot{Q}_{heat}}\right)^2$ and $\left(\frac{u\_\dot{Q}_{loss}}{\dot{Q}_{heat}}\right)^2$ are predetermined as shown in Table 8 below.

Table 8: Uncertainty calculation for cooling capacity and heat loss

$u\_\dot{Q}_{cooling}$	$\frac{u\_\dot{Q}_{cooling}}{\dot{Q}_{cooling}}$	$\left(\frac{u\_C_{p\_w}}{C_{p\_w}}\right)^2$	$\left(\frac{u\_\rho_w}{\rho_w}\right)^2$	$\left(\frac{u\_\dot{V}_w}{\dot{V}_w}\right)^2$	$\left(\frac{u\_(t_{w\_out} - t_{w\_in})}{t_{w\_out} - t_{w\_in}}\right)^2$
94.5	0.02	0	0	0.0000848	0.00038
$u\_\dot{Q}_{loss}$	$\frac{u\_\dot{Q}_{loss}}{\dot{Q}_{loss}}$	$\left(\frac{u\_UA}{UA}\right)^2$	$\left(\frac{u\_(t_{a\_room} - t_{a\_amb})}{t_{a\_room} - t_{a\_amb}}\right)^2$		
4.1	0.1	0.000021	0.01		

$\frac{u\_UA}{UA} = 0.004587$, according to .