PROJECT FINAL REPORT

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Name, title and organisation of the scientific representative of the project's coordinator:
Andreas Zottl
AIT Austrian Institute of Technology
Tel: +43 50 550 6309
Fax: +43 50 550 6679
Email: Andreas.zottl@ait.ac.at

Project website: www.greenhp.eu

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1.1 Executive Summary

CO_2 emission levels from our rapidly expanding cities must be drastically reduced to meet the European environmental targets. One key aspect of turning cities into “smart cities” is renovating and retrofitting buildings with low emission heating systems. Heat pump technology plays a central role here, as it can use ambient energy for heating and hot water generation. Due to limited availability of space in cities, heat pumps need to be integrated into the existing buildings, and must be compatible with pre-existing heating systems. In addition, the installation and running costs must be attractive to make the changeover to heat pumps in urban areas both an economically and environmentally viable decision.

The GreenHP project addressed these challenges by developing an advanced 30kW heating system with minimum environmental impact using air/water heat pump technology for retrofitting multifamily and commercial buildings in densely populated areas and cities and the opportunity to integrate the system into future energy systems (e.g. Smart Grids).

The research undertaken was based on a comprehensive multi-level research approach ranging from developing innovative heat pump components such as new heat exchanger, fan and compressor prototypes, to advanced system integration concepts. In order to freely distribute the results and the GreenHP concept broadly, there was no heat pump manufacturer involved in the consortium but experienced research institutions and component manufacturers instead.

For integrating the GreenHP in the future smart energy systems, a concept for an enhanced three-layered controller platform based on a Model Predictive Control approach allowing load balancing at maximum efficiency of the overall heat pump system at the lowest costs was developed. In all investigated cases, the GreenHP controller saved between 4% and 22% of total cash flow when compared to the reference controller. The System Controller in charge of the overall heat management of the GreenHP system was designed on hardware and system level and developed to market maturity.

On component and unit level, the refrigeration cycle and several components were optimized for the use of propane leading to a reduced refrigerant charge for the 30 kW GreenHP prototype unit of less than 2 kg; this corresponds to 65 g per kW thermal capacity. The refrigerant charge was minimized by using aluminium multi-port extrusion tubes (MPE) in the condenser and evaporator. The air-side of the evaporator was optimized by developing both highly efficient fins and an advanced fan design, resulting in an energy demand reduction of the fan operation by 10%, at icing conditions. On the refrigerant-side, a novel bionic distributor was introduced to distribute the refrigerant to the MPE tubes of the evaporator. A variable speed scroll-compressor prototype making use of Enhanced Vapour Injection ensured high seasonal efficiency over the entire operating range of the GreenHP. To further decrease the refrigerant charge, the oil sump was optimized resulting in 1 litre oil, instead of 3.3 litres.

The GreenHP prototype unit accomplished nearly all performance indicators aimed at when tested in compliance with international standards. In particular, it performed very well in terms of its seasonal efficiency with an SCOP of 3.3, which is comparable to the best performing heat pump units in the smaller capacity range of 10-20kW.

Summing up, the GreenHP showed very promising results on unit, component and system level to build up from in the upcoming research projects. To harvest additional efficiency gains and further optimize the performance of the GreenHP unit, more research is required on component and system level including extensive field measurements.

1.2 Project Context and Objectives

The overall goal of the GreenHP project was to investigate and develop a new highly efficient heating system based on high-capacity air/water heat pumps for retrofitting multifamily houses and commercial buildings situated in cities. The reasoning for the projects' focus is explained in the following:

1.2.1 *Focus on retrofitting buildings*

With a share of approx. 40% on the final energy consumption, buildings are one of the key consumers of energy in Europe. In 2009, EU-28 households were responsible for 68% of the total final energy use in buildings mainly consumed for heating (78%). Approx. 40% of all residential buildings in Europe were built before 1960 and 45% between 1960 and 1990, and did not undergo major renovation since then[1]. In order to achieve the EUs ambitious goals on energy efficiency and use of renewable energy, it is essential to renovate and retrofit these buildings according to EU's standards for energy efficiency in buildings. Most of these buildings are currently equipped with old and inefficient heating systems based on fossil fuels that need to be replaced by **modern heating systems that are highly efficient and capable of utilizing renewable energy sources**.

1.2.2 *Focus on urban areas / cities*

At EU level, about 49% of all residential buildings are situated in densely populated, urban areas. By 2050, 66% of the world's population will live in urban areas[2], more than 75% of the total CO_2 emissions will be emitted in cities, and more than 73% of the total primary energy demand will be consumed by cities mostly for heating and cooling. **Urban areas must therefore play a key role in the future of renewable and efficient energy systems.** To deal with these challenges, cities will need to become 'Smart Cities'. They will have to focus on alternative and decentralized ways for energy production based on renewable energy sources as well as on energy conservation and efficient energy usage. For cities to become 'smart,' it is important to develop intelligent energy infrastructures including smart electrical and thermal grids. It is thus essential to develop heating systems that are tailored for application in urban areas and capable of interacting with future intelligent energy infrastructures of 'Smart Cities'.

1.2.3 *Focus on air/water heat pumps*

Air/water heat pumps have experienced significant growth rates during the last decade and dominate the current market trend. The total number of air/water heat pumps sold in Europe in 2014 totals 201.749 units. The highest sales numbers were achieved in Germany, Italy, France and UK.[3] Air/water heat pumps are successful as they are considerably cheaper and much easier to install than ground-coupled heat pumps, as they do not need boreholes or surface areas for ground-coupled horizontal collectors. These features make them attractive candidates for applications where installation of ground-coupled heat sources is difficult or not possible. **This is typically the case for existing buildings, especially when situated in densely populated areas and cities**. For these reasons, large scale air/water heat pumps are seen as an important, integrative component of the energy infrastructure in cities in the future

[1] BPIE, Europe's Buildings under the microscope: a country-by-country review of the energy performance of buildings, October 2011
[2] UN, World Urbanization Prospects, 2014
[3] EHPA, European Heat European Heat Pump Market and Statistics Report 2015, 2015

1.2.4 *Challenges to overcome*

The challenges addressed by the GreenHP project comprise:

- To achieve high efficiency, heat pumps have to be operated with the lowest possible temperature lift which is of high relevance for air-source heat pumps in particular. Old buildings, on the other hand, are mostly equipped with heat distribution systems that usually require higher temperatures. The desired efficiencies, in terms of primary energy savings and yearly seasonal performance, can therefore often not be reached. To solve this problem, air-source heat pump units need to become more flexible and function well over a vast range of temperature lifts and capacities.
- The integration of air/water heat pumps into existing buildings requires improved system layouts including thermal storages and improved system controls. Heat pumps must be integrated in order to cover a larger part of the heating demand at the lowest possible temperature lift for highest possible efficiency.
- The air/water heat pump must be designed to be smart in that it can respond to its environment and grid issues, and become an integrative component of a future 'Smart Grid' in a 'Smart City'.
- The environmental impact of air/water heat pumps should be enhanced by combination with other renewables like photovoltaic or solar thermal systems.
- Finally, there is a great potential to reduce the environmental impact of heat pumps by using alternative low Global Warming Potential (GWP) refrigerants. Characteristics like flammability and toxicity have so far limited the use of these promising refrigerants, and steps need to be taken in order to leverage them safely.

1.2.5 *Concept and goals of the GreenHP project*

Two key aspects were considered essential to develop a next generation heat pump fully capable to meet the aforementioned challenges:

- **Holistic approach** including system integration aspects, as it enables the possibility of reaching an overall optimum solution for a considered application. This approach is in contrast to the common engineering practice of optimizing different components (e.g. compressor, fan, etc.) independently achieving only local optima in a final technical application.
- **Harmonized research on the component level** (including the heat pump unit and its sub-components) based on requirements defined for an overall application scenario (holistic approach).

1.2.6 *Objectives*

The GreenHP project pursued the following objectives on system, unit and component level.

System Level
• Development of a holistic system concept for high-capacity air/water heat pumps for retrofitted multifamily and commercial buildings in urban areas. • Investigation and definition of building integration concepts for high capacity air/water heat pumps including control strategies. • Definition and testing of use-cases for integration of high capacity air/water heat pumps into smart electric grids. • Validate the overall control concept and smart gird integration concept by means of hardware-in-the-loop (HIL) testing on lab scale.
Unit Level
• Evaluation of alternative refrigerants for use in high-capacity air/water heat pumps • Development of design methods for high efficiency air/water heat pump cycles with substantially reduced refrigerant charge using an alternative refrigerant • Validate the developed concepts for the heat pump unit and the components on lab scale by means of testing of pilots.

Component Level
• Development of a control platform for air/water heat pumps integrating other renewable energy sources including thermal storages and interfacing the smart grid. • Design and investigation of new compressor concepts for an alternative refrigerant enabling high efficiency and low refrigerant charge. • Investigation of new concepts for high performance aluminium microchannel heat exchangers for evaporators and condensers enabling low refrigerant charges for high capacity air/water heat pumps • Investigation of new concepts for high performance axial fans with optimised airflow system for air/water heat pumps enabling low energy demand and low noise emissions. • Development and investigation of a novel evaporator unit based on advanced anti-icing and defrosting methods.

The **GreenHP research approach** is depicted in Figure 1.

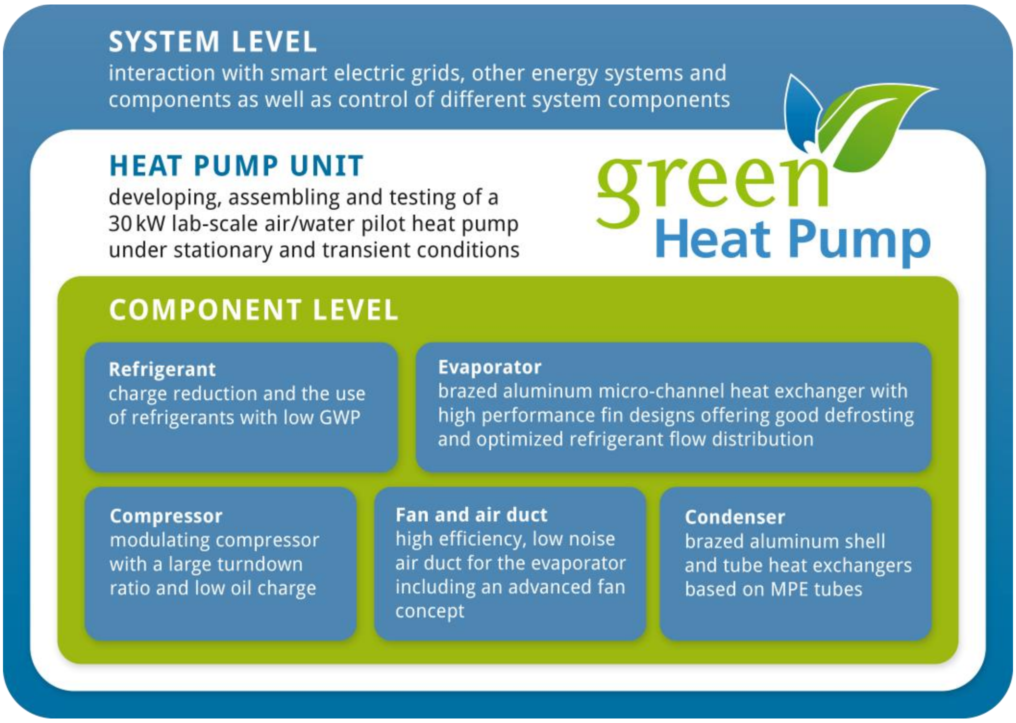


Figure 1: GreenHP Research Approach

1.3 Key Scientific and Technical Results

The GreenHP project achieved the following **key scientific and technical (S&T) results**:

- Development of an innovative **GreenHP concept based** on a holistic research approach geared at reaching maximum system efficiency on component, unit and system level.
- Development of an **enhanced 3-layered controller platform** based on Model Predictive Control to allow load balancing at maximum efficiency of the overall heat pump system including the newly developed **System Controller,** which allows amongst others the optimized integration of other renewables.
- Development of a **30 kW lab-scale air/water heat pump prototype unit** for the use in multifamily houses using propane as natural refrigerant, equipped with the following innovative components:
  - a novel alumium **evaporator prototype** constructed with multi-port extruded tubes and equipped with a novel bionic refrigerant distributor aiming to significantly minimize the refrigerant charge and increase the efficiency of the heat exchangers;
  - an enhanced very well performing aluminium **condenser prototype**;
  - **a novel fan** designed to be operated with low energy consumption and noise emissions;
  - an **optimized variable speed scroll-compressor** using Enhanced Vapour Injection over-achieving especially the target on oil charge reduction.

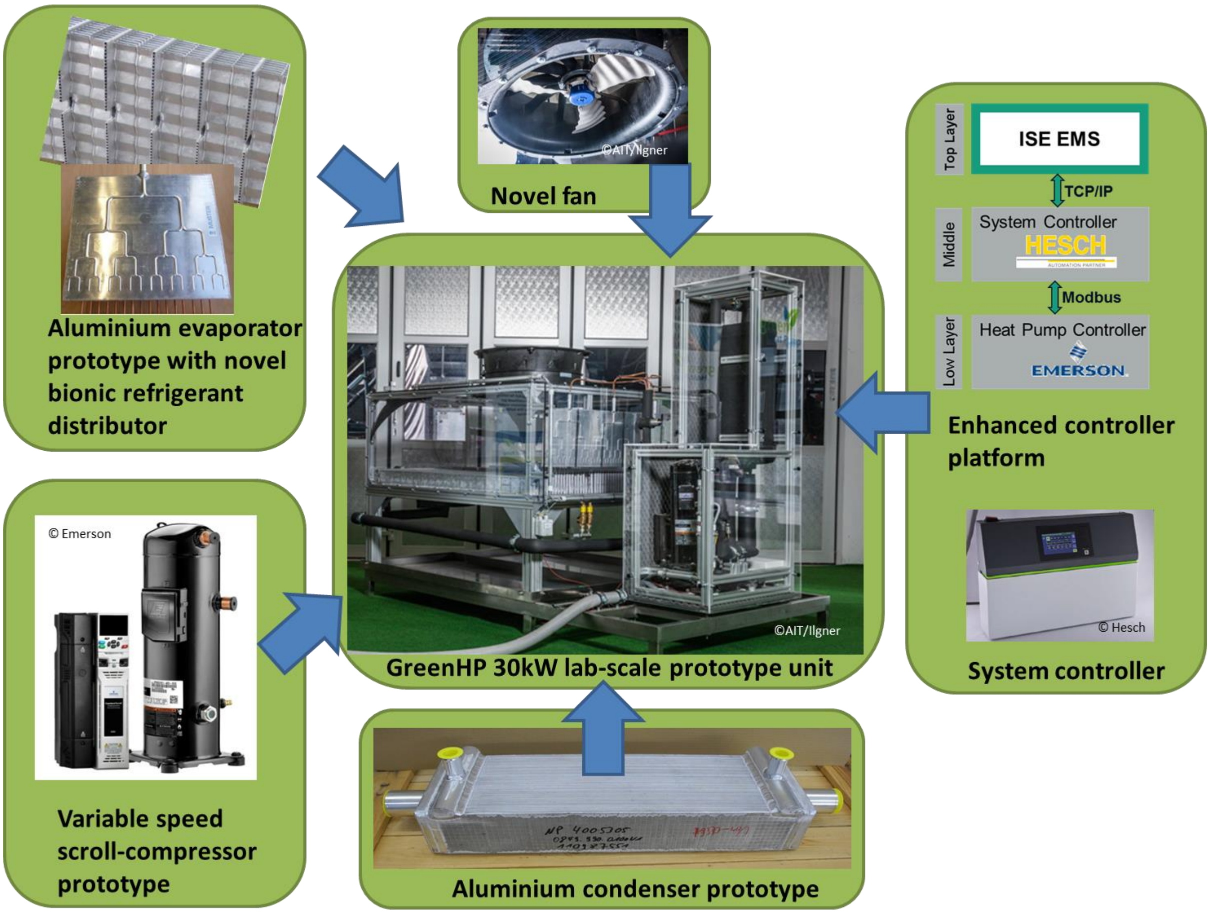


Figure 2: Overview of main S/T results of the GreenHP project

As evident from Table 1, which compares the performance indicators of state-of-the art heat pumps at the time of project start, to the targeted indicators for the GreenHP unit, and finally to the performance indicators achieved during testing, the **GreenHP prototype accomplished nearly all performance indicators aimed at**. In particular it performed very well in terms of its seasonal efficiency with an SCOP[4] of 3.3, which is 0.2 higher than the SCOP aimed for and is comparable with the best performing air/water heat pump units in the smaller capacity range of 10-20 kW heating capacity.[5]

The system showed higher overall efficiency results than expected with a primary energy ratio (PER) of 1.32 and CO_2-emissions of 187 g CO_2/kWh usable energy. In addition, the system uses with propane (R290) a natural refrigerant with a very low global warming potential (GWP) of only 3. The highest Carnot efficiency reached was 54 % at part load conditions. Out of the performance indicators, only three proved to be over-ambitious: firstly, the COP of 3.5 could not be reached. Secondly the refrigerant charge required for maintaining a stable operating behaviour was 1900 g, which corresponds with 65 g per kW thermal capacity. This is more than aimed for and calculated in the simulation, but still much less than conventional systems use. Finally, the energy used for defrosting could be decreased less than anticipated.

Table 1: GreenHP test results compared to Target performance indicators

Specifications	State-of-the-art*	GreenHP unit planned	GreenHP prototype
Carnot efficiency factor	40-45%	55-60 % (increased by 15%)	54 %
COP (A7/W55)	2.5	3.5	3.0
SCOP ** (based on EN14825)	2.1	3.1	3.3
Primary Energy Ratio (PER)* (kWh useful energy/kWh primary energy)**	0.84	1.24	1.32
CO_2 Emissions * (g CO2/kWh usable energy)**	294	200	187
Refrigerants used	primarily HFC	natural refrigerant or HFO	natural refrigerant
GWP of refrigerants used	>1300	<150	3
Refrigerant charge range (g refrigerant / kW heating)	200 – 500	30 (hydrocarbons) 60 (HFOs)	65
Defrost Energy Used	about 10 %	<5%	<10 %
Smart grid integration	N	Y	Y

* State-of-the-art relates to most advanced air/water heat pumps available on the market 2012

** Seasonal Coefficient of Performance (SCOP) calculated based on EN 14825

*** Based on EU UCTE-Mix; PEF=2.5; EN 15603: CO2-UCTE-Mix = 617g CO_2/kWh final energy

In the following chapters the **scientific and technical work and results achieved** are described in more detail on system, unit and component levels.

[4] The SCOP was calculated according EN14825 for “average climate” and high temperature applications.
[5] Heat pump units in the capacity range of 10 – 20 kW reach SCOPs between 2.9 and 3.4.

1.3.1 *System level*

1.3.1.1 *System specification (Work Package 1)*

The operating conditions of an efficient heat pump system interacting with a smart electric grid are mainly influenced by **climate** and **location, electric infrastructure** and **building type** (Zottl 2013). To define the general requirements for the technical specification of the GreenHP on system, unit and component level, supplementary considerations of existing buildings (BPIE 2011), retrofitting options for buildings (IWU 2011) and hot water demand of buildings (Knight 2007) were considered. For the detailed system design of the GreenHP unit the following had to be specified:

- Reference climate zone
- Reference location
- Building characteristics
- Heat load and load profiles of buildings.

Reference climate zone

The GreenHP reference climate zone was selected based on market potential, state of the existing air/water heat pump market, substitution (replacement) potential, size of cities, their population density, and electric infrastructure, as depicted in Figure 3. To keep in line with EC-regulations (ErP, ECO-labelling), the three ErP climate zones – cold (C), average (A) and hot (H) – were used (Figure 4).

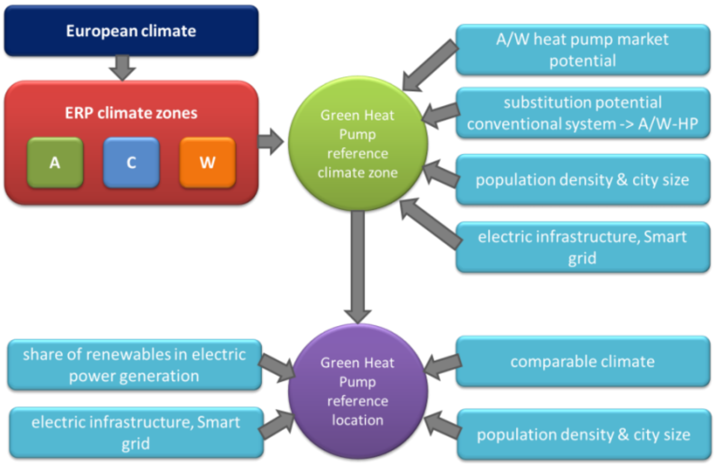


Figure 3: Procedure for selecting climate zone and location of the GreenHP system

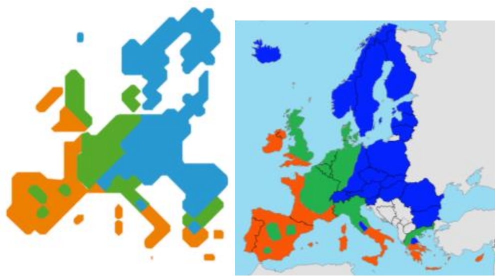


Figure 4: ErP Directive Climate Zones

France, Germany and UK show high population densities, numerous major cities (Eurostat, 2012), a large market shares of air/water-heat pumps (EHPA, 2012); hence, a high level of acceptance and therefore market potential for air/water-heat pumps is expected. In addition, these countries possess also the highest “Absolute Load Shifting Potential” in Europe (see Figure 5). As the countries considered are mainly located in the average climate zone, the **average climate zone** was defined as the **GreenHP reference climate zone** (green coloured areas in Figure 4).

Reference location

To ensure high level of interacting of the GreenHP unit with the electric grid and realize a maximum of load shifting potential, one important aspect for defining the reference location for the GreenHP unit was their smart grid potential. Other important parameters include comparable climate conditions to the ErP reference locations, share of renewables in electric power generation and population density (see Figure 3).

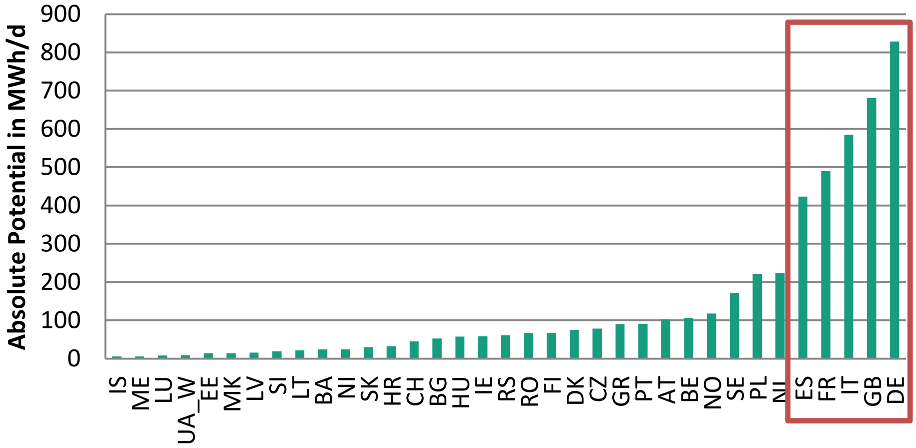


Figure 5: Absolute Load Shifting Potential based on total grid load (Fraunhofer ISE)

Based on the criteria mentioned above, Dusseldorf (average climate zone), Stockholm (cold climate zone) and Barcelona (warm climate zone) were selected as GreenHP reference locations.

Specification of buildings and load profiles

Finally, the building types and load profiles were investigated in more detail as input for the detailed system design and layout.

The main application focus was put on retrofitting multifamily houses constructed before 1995 due to the following facts (BPIE 2011):

- 38% of all residential buildings were built before 1960; 45% between 1960 and 1995.
- 49% of all residential buildings in Europe are situated in densely populated areas.
- 27% of the total floor space is situated in apartment blocks compared to only 12.8% for commercial buildings.

As Dusseldorf came up as the best location fulfilling all parameters, the **system design** and **detailed system layout** of the GreenHP unit was carried out for Dusseldorf. The average heat load of multifamily houses (TABULA 2013) in Dusseldorf and their domestic hot water demand was assessed as follows: the peak heat load for non-refurbished multifamily houses in Dusseldorf was estimated at 33 kW to 37 kW; refurbishment of the building envelope including external wall, roof and windows is expected to decrease the peak heat loads to 16 kW to 23 kW. The heating capacity of the unit will be therefore 30 kW.

Summarizing, the GreenHP system was laid out and optimized for providing space heating and domestic hot water for a **multifamily house** at the **reference location Dusseldorf** with a **living area of about 600m².** A multifamily house is defined as house with 6 dwellings. The domestic hot water demand was set at **100 to 120 litres per day** for a dwelling with a 100 m^2 floor area based on an **average temperature rise of 45°C**.

1.3.1.2 *System Integration and hydraulic layout (Work Packages 2)*

The integration of a heat pump such as the GreenHP unit into an energy system of a multifamily house in Dusseldorf geared at reaching maximum system efficiency on component, unit and system level is a complex task, as there is a variety of integration options considerably influencing the operation conditions and therefore the seasonal efficiency of the heat pump unit. To find the best solution from a systematic point of view, the following questions had to be answered:

1. How to best integrate the heat pump into the hydronic system of the building?
2. How to best integrate renewable energy sources into the energy system of the building?
3. How to best integrate the building energy system into the local and regional power grid?

Configuration, sizing and control approach are closely interconnected and strongly influence each other. Each decision also influences the conditions under which the heat pump is operated. The following research questions had to be solved within the GreenHP project on the issue of optimal system integration and hydraulic layout:

1. What is the most promising configuration layout for the integration of the GreenHP unit in a refurbished multifamily house?
2. What is the optimal sizing of the components for the configuration selected?
3. How can the GreenHP system be operated in a future energy system and how will changes in the operation affect the seasonal performance of the heat pump system?

1.3.1.2.1 System and hydraulic layout

To come up with a final design of the GreenHP system layout, various possibilities of integrating the heat pump unit into the thermohydraulic system of the building, of using solar sources (PV, solar thermal) and a thermal energy storage were considered. Several system layouts were investigated in more detail and simulated, to identify the most energy efficient layout from a systemic point of view.

Before installing an air/water heat pump into an existing building, refurbishment of the building is advisable to keep the heating supply temperatures low and minimize energy losses over the building envelope. In the described case for the GreenHP, an **exchange of the radiator system was not taken into account, mainly due to economic reasons, but the heat demand of the building was lowered by means of better insulation so that a design temperature for the heating system of 55/45°C** was possible. To manage several heat generators (heat pump and solar thermal) and consumers (space heating, domestic hot water), a **stratified thermal storage tank** was used.

As the mode of domestic hot water (DHW) preparation highly impacts on the operating temperatures of the heat pump system, two approaches were analysed in more detail: a centralized and a decentralized domestic hot water system. According to legionella legislation, the water temperature of DHW storages has to be maintained at 60°C, which is higher than the heating supply temperature. Hence, in the case of a central DHW production, the temperature of the upper part of the thermal storage tank has to be kept above 60°C, while the medium part can be maintained at the heating supply temperature determined by the heating curve of the building.

Alternatively, decentralized heat exchangers can be placed in each dwelling to heat fresh hot water when needed, thus avoiding the legionella restrictions on thermal storages. Therefore, the supply temperature of the thermal storage can be kept considerably lower (at 45 °C to 50 °C) than in the central case.

Since most of the heat generated by a heat pump is used for space heating, it is advantageous to have **two separate supply temperature set points, one for DHW and one for space heating**. This way, the flexibility of the heat pump operation is improved and the operation of the heat pump towards the highest supply temperature at all hours can be avoided.

1.3.1.2.2 Optimal configuration and sizing of components

To reach the full thermodynamic potential of the GreenHP system, all system components were dimensioned in a proper way and appropriate system configurations were selected. Furthermore, the complex dynamic interaction among the heat pump systems' components were considered in the system design. Since changing one parameter in any single component was likely to affect several other components, a comprehensive system model[6] was developed to investigate four commonly used system configurations and compare them regarding their annual performances.

The four different system layouts depicted in Figure 6 to 9 were geared to find out the optimized connection/integration of the GreenHP unit to the stratified thermal storage tank and the multifamily building.

[6] Including several sub-models such as building, heat pump, storage tank, climatic conditions, etc.

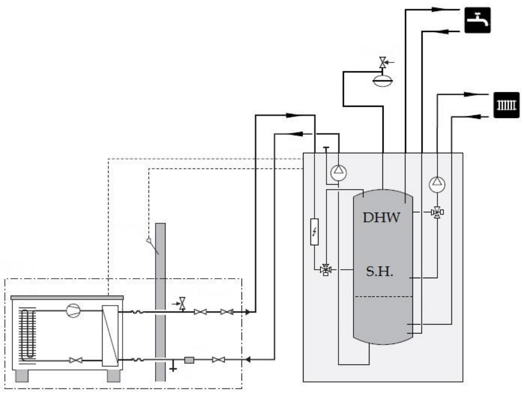


Figure 6: Indirect 3-pipe system

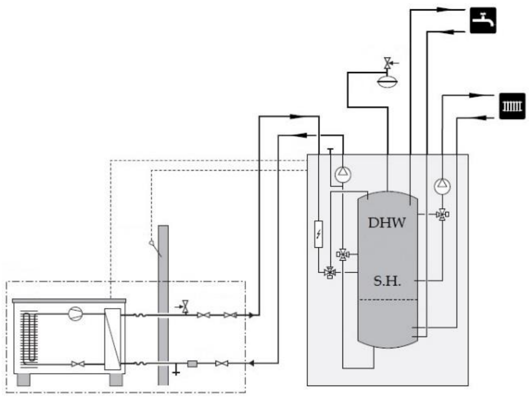


Figure 7: Indirect 4-pipe system

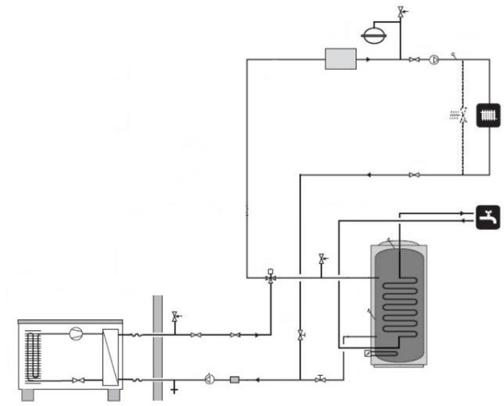


Figure 8: Direct one-tank system

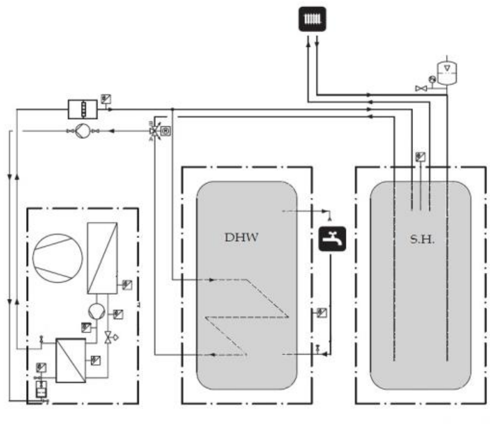


Figure 9: Direct two-tank system

The two indirect systems[7] showed the best performance regarding efficiency and operating conditions for the variable speed compressor. A comparison of the results from annual system modeling showed, that the **seasonal efficiency can be improved by 18 % when using the indirect DHW system.** It became evident, that the 3-pipe system configuration outperforms during summer, when only DHW is needed, and underperforms during winter when used to provide both, DHW and space heating. As a conclusion, it was decided **to design the GreenHP system** in a way to **adapt the system configuration to the season and operating condition to reach the highest seasonal performance.** The two three-way-valves in the 4-pipe **heat pump system layout** allow the **use** of the **4-pipe configuration during winter,** and the **3-pipe configuration** during **summer.**

This change in the system layout, led to an improved annual system performance by 5 %. The advantages resulting from this system layout are expected to be higher than its costs, as it enables a more flexible operation of the heating system at lower system temperatures by introducing solar thermal, heat storage and using the thermal mass of the building for storing energy. Moreover, the results from annual system modelling showed that the **storage tank** with a volume of approximately **2.5 m³** can meet almost 100 % of the DHW demand over the year. The main findings from the component sizing are summarized in Table 2.

[7] In the indirect 3-pipe system, the heat pump is connected with one return and two supply lines to the tank for DHW and space heating operation. Based on the demand and control strategy, a three-way-valve directs the supply water from the heat pump either to the space heating or to the DHW section of the storage tank. On winter days, the 3-pipe configuration leads to higher temperatures than needed in the storage section intended to be used for space heating due to the higher temperatures needed for DHW. Consequently, the heat pump works on higher frequencies to provide high temperature that has detrimental effects on the heat pump efficiency. In the indirect 4-pipe system, an additional three-way-valve connects the heat pump also with two return lines to operate the heat pump for DHW (upper connections of supply / return line) and for space heating (bottom connections of supply / return line). Thus, compared to the 3-pipe system, the 4 pipe system offers more flexibility on the temperature levels for DHW and space heating.

Table 2: Key technical features for the GreenHP layout

Technical component	Description
Hydraulic system layout	four-tubed system with a separate heat exchanger for hot tap water in each dwelling
Solar thermal system layout	Collector area: 40-50 m^2 Thermal storage: 2-3 m^3 Annual useful heat production: approx. 7000 kWh
PV	10 kWp (not optimised)

1.3.1.2.3 GreenHP Control Strategy and Smart Grid Integration

Considering the conditions under which the GreenHP will be operating in the future energy system, three system configurations (*Baseline*: heat pump and thermal storage; *High renewable energy (RE)*: heat pump, thermal storage, PV, solar thermal (ST), and *PV only*: heat pump, thermal storage and PV) and two different electricity price conditions (constant electricity price, EEX), were selected to represent four use-cases. In order to analyse the impact of different operation strategies on the effiency of the GreenHP while at the same time aiming at maximising financial revenues (or minimizing operation costs, each use-case (marked in blue in Figure 10) was simulated and tested with both, a conventional controller (reference controller[8]) and the ISE-EMS controller, which is an enhanced model predictive controller (MPC) making use of weather forecast, predictions of thermal loads, renewable electricity production, electricity prices, etc.[9]

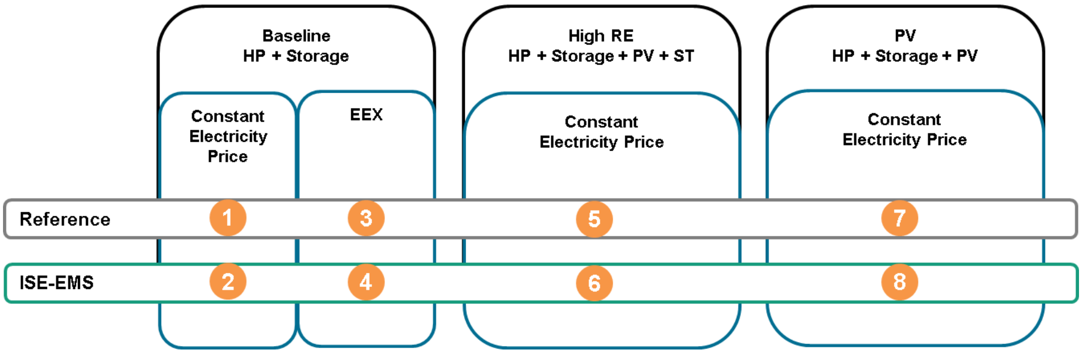


Figure 10 Use cases investigated

A comparison of the control approaches showed, that the **ISE-EMS controller increased** the **seasonal performance factor (SPF)** of the heat pump **in three of four use cases.** In case of “Baseline with constant electricity price”, the SPF improvements reached **up to 6%** leading to **6 % lower annual electricity costs**. When using day-ahead electricity spot prices (Baseline EEX[10]) the SPF was reduced, but the use case was still providing financial benefits compared to the reference controller. In all cases, the **ISE-EMS controller saved between 4% and 22% of total cash flow** when compared to the reference controller. The financial benefits increased when local renewable energy production was taken into account (“PV only” and “High RE” use cases).

Additionally to optimizing the control strategy, the effect of variable electricity prices and increased use of PV in the residential sector on the system sizing was investigated. According to the results, **today's heat pump and back-up heater sizing is nearly fully optimised** and does hardly change when introducing PV and variable electricity prices. Only the s**torage size changes, depending on the scenario selected**. In the scenarios with an installed PV capacity below 30 kWp and a variable share of electricity price below 40% (+-3.8€ ct/kWh), a storage increase by a

[8] The reference controller kept storage and building temperatures at set points given without reacting to other signals such as PV/ST production or varying electricity prices

[9] Described in more detail in chapter 1.3.2

[10] The heat pump operates during night-time when both electricity price and heat demand is low, and stores it for later use when heat demand is high

maximum of 30% is sufficient. If the storage is allowed to be overheated, no change in sizing is needed, even though this might come at a cost of increased storage losses and decreased heat pump efficiency. More information on that topic is published in (Fischer 2015).

Summarising, the GreenHP **system layout** for a multifamily house in Dusseldorf with a **design temperature** of the heating system of **55/45°C** due to better thermal insulation was defined as follows: **four-tubed system** with a separate heat exchanger for hot tap water in each dwelling, use of **4-pipe configuration during winter** and **3-pipe configuration during summer**. Depending on the case scenario selected, the use of an **enhanced MPC controller saved** between **4 and 22% of total cash flow**. Financial benefits increased when local renewable energy production was taken into account. Specification for local renewable energy production: collector area: 40 to 50 m², thermal storage size: 2-3 m³ PV: 10kWp

1.3.2 *GreenHP controller platform (Work packages 2, 4 and 10)*

1.3.2.1 *Definition and specification of the GreenHP control platform*

An innovative hierarchical control platform, capable of locally integrating high shares of on-site renewable energy generation and operating under variable prices was developed.

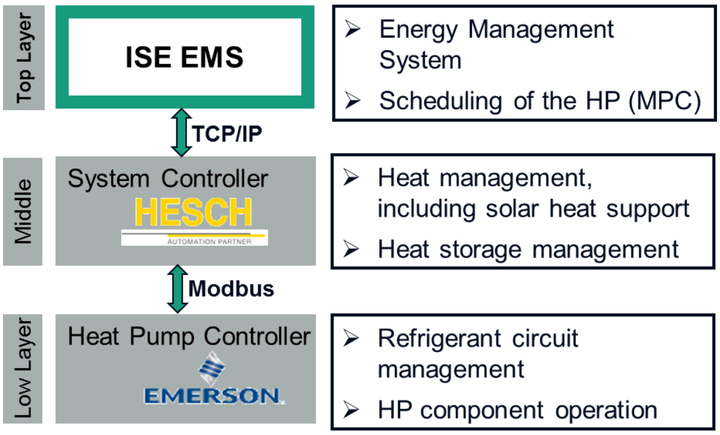


Figure 11 Controller hierarchy of the GreenHP project

The controller hierarchy depicted in Figure 11 consists of three levels (layers):

On the top level, the energy management system (EMS) calculates optimal heat pump operation based on the current system state, predictions of thermal loads and renewable electricity. The time variable electricity price is received as external control signal and used for the calculation of the operation schedule, which is sent to the system controller in the middle layer.

The system controller is in charge of the heat management keeping the temperatures in the thermal storage and building at the level desired by the user, while trying to fulfil the operation schedule received from the energy management layer. The heat pump itself is controlled by the heat pump controller (RCC) of Emerson, in charge of the refrigerant circuit, compressor, valves and the fan as it is presented in the low layer of the above figure. It was realized by a separate controller unit that communicates to the upper layers via Modbus-RTU.

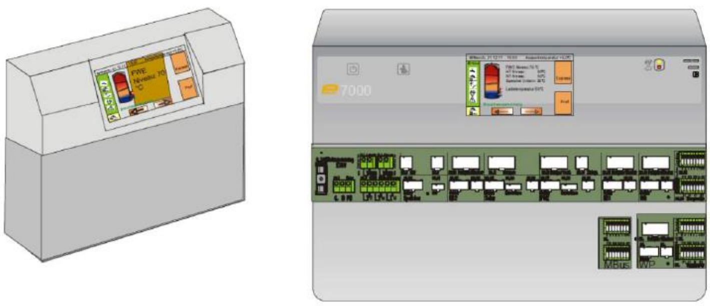


Figure 12: Top of the housing of the System Controller

The three controllers of the control platform were specified functionally including the definitions of their interfaces to the other layers. The system controller of HESCH was specified technically on hardware (see e.g. the housing of the system controller in Figure 12) and software-level, manufactured and tested successfully.

The research work undertaken resulted in a **market-ready CE marked System Controller** for heat pumps (Figure 13) to be integrated into the grid for load balancing. The system controller is already in operation in a heating application in Germany.

Figure 13: System Controller

1.3.2.2 *Comparison of controllers for system integration*

Heat pumps equipped with a thermal storage, a direct storage or using the thermal mass of the building offer the possibility for load shifting, as thermal demand can be decoupled from heat generation of the heat pump unit. This capability can be used for different purposes within the power system. Three main areas of interest were identified resulting in different operational targets for the heat pump:

1. Operation focussed on direct signal from the electric grid.
2. Operation along variable prices, which serve as a signal for various purposes.
3. Operation aiming to increase the use of on-site generated renewable electricity.

The work undertaken in the GreenHP project focussed on case 2 and 3. As variable prices can represent different sources and motivations, they offer the possibility to align heat pump operation to various objectives. When the price is a network tariff, case 1 can be demonstrated as well. Case 3 leads to a local tariff, as it offers a direct benefit on the level of the single house. Case 1 is partly included in case 2 and is seen as less crucial as the GreenHP is intended to be placed in inner-city areas where the distribution grid is often sufficiently interconnected.

An illustration of the system integration of the GreenHP unit is shown in Figure 14. A paper including a detailed study on heat pumps in smart grids was handed in for publication in “Renewable and Sustainable Energy Reviews” (Fischer 2016a).

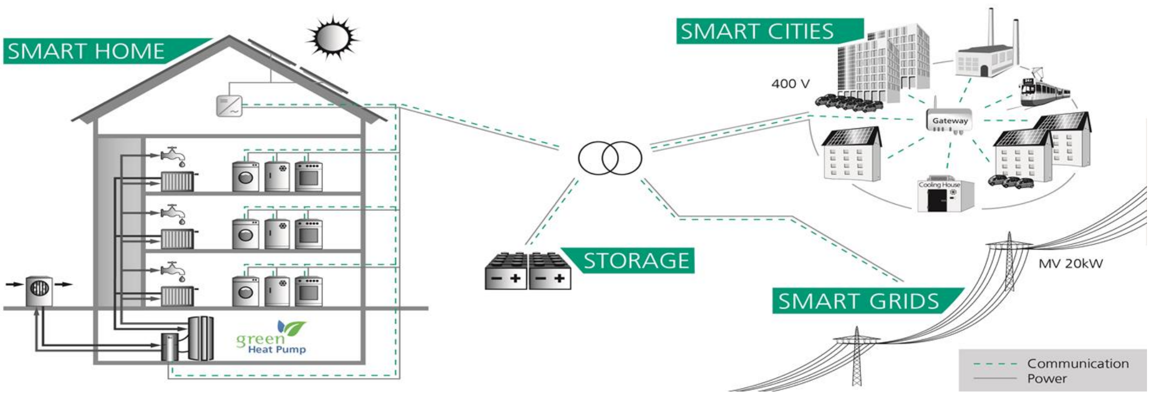


Figure 14: Integration of the GreenHP in the surrounding energy system.

A comprehensive comparison of control approaches of different complexity and differently capable of fulfilling the operational targets put upon heat pumps was conducted. Three use-cases were investigated for the climate of Dusseldorf, Stockholm and Barcelona aiming at the following targets for the controls: satisfying the thermal demand of the building, increasing the use of locally generated renewable electricity and optimising costs under time variable electricity prices. More information on the work and results is provided in the following:

Tested controllers

The control approaches tested reflect the main control approaches currently in use. Those are:

- Non-predictive rule based controls
- Predictive rule based controls
- Model predictive controls

The control approaches described in more detail in Table 3 differ regarding their use of external information, complexity, computational requirements and flexibility in use.

Table 3: Main Control approaches pursued

Control approach	Description
Rule-based thermal:	This is the baseline control approach. The controller is a standard hysteresis controller combined with a P-controller for the compressor speed. The storage temperature at the DHW and the space heating level is kept within a tolerance band. The heat pump speed is increased proportionally to the temperature deviation from the set-point.
PV-follow (rule based smart):	This is an improved version of the thermal rule-based controller for PV applications. If the excess power from on-site generation in the building is higher than the minimum required electricity for switching on the heat pump, it is switched on. The compressor speed is controlled using a PID controller targeting to put the net household electric demand as measured at the smart meter to zero.
Time controlled:	The time based controller is a further modification of the rule-base controller. The heat pump is blocked at hours of high electric prices and switched on at hours of low prices or negative residual load. When switched on by the clock program the heat pump heats up the storage to 55 °C
Model predictive controls:	In addition to the rule-based approaches, a model predictive controller was used. Based on the current temperatures in the storage and predictions of the heat demand, the DHW demand, the ambient temperature and the prices, an optimal control problem is solved for each half hour 24h in advance. At each time step the current solution of this optimisation problem is applied to the system and a new optimal control problem based on the system state is solved. Convex quadratic optimization is used to solve the problem. Therefore the heat pump COP is partly linearized and the system is modelled as linear time invariant system. For DHW and space heating operation as well as for the use of PV a new HP instance is introduced. The sum of heat provided by the individual HP instances is limited to the maximum heat output of the heat pump. The piecewise linearization of the COP results in a quadratic objective function and linear constraints.

For the GreenHP system, a set of different tailored controls was designed and implemented to compare the individual approaches. For all control approaches the minimum needed temperature at the top storage level for DHW is 50°C. The required temperature level for the water used for space heating is calculated by using an ambient temperature depending on the heating curve.

Tested use-cases

To test the controllers, three use cases were simulated for each climate zone and controller (see scenario space in Figure 15). The smart grid environment is implemented via a dynamic electricity pricing scheme linked to the day-ahead electricity spot price.

The three resulting use-cases are:

1. **Thermal:** A constant electricity price of 19ct/kWh and no PV plant is installed.
2. **PV self-consumption:** A 10 kWp PV plant southwards oriented and 35° inclined is installed on the building. A feed-in tariff of 12.34 ct/kWh is paid and the electricity price is constant.
3. **Variable electricity prices:** The time variable electricity price based on the day-ahead spot price and no PV plant is applied.

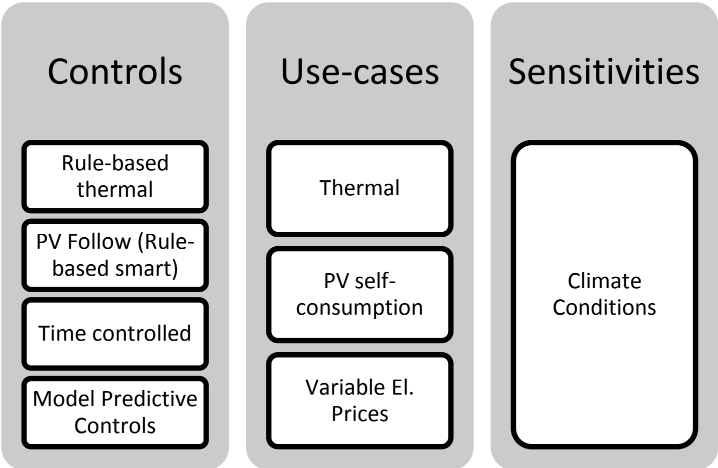


Figure 15: Simulated scenarios

In all cases, control should keep the temperatures in the heating circuit, the building and the DHW part of the storage within the allowed limits. Within these limitations, the different control strategies were tailored to fulfil the thermal demand in the most economical way.

The results showed that the required electric energy and consequently electricity costs for the GreenHP operation can be decreased by using a more sophisticated control. Within all tested scenarios, the **Model predictive controller is able to reduce the energy demand between 1 % and 15 % compared to the default rule-based controller achieving an absolute saving between 118 €/a to 306 €/a depending on the used scenario and the climate region.**

The PV follow control was adapted especially to the PV scenario and showed good results for this case. **The PV-plant operation costs could be further decreased to 61 to 93 €/a depending on the climate zone** of the system**.**

More information on the controls used and the comprehensive results are already published (Fischer 2014a) respectively handed in for publication (Fischer 2016b).

1.3.2.3 *Integration of the GreenHP in the HIL environment*

To test the control concept under realistic conditions and to evaluate the smart gird interoperability of the GreenHP platform, various tests were performed in a HIL environment at the so called SmarEnergyLab of Fraunhofer ISE. The components used for the HIL tests included:

1. ISE Energy management system
2. The system controller, developed within the project by HESCH
3. The refrigerant cycle controller (RCC) boards of the GreenHP unit, provided by Emerson
4. The expansion valves connected to the RCC
5. Sensor & actor emulation including pumps, temperature sensors and pressure sensors
6. Dynamic system simulation of the building, the storage and heat distribution system

To test the controllers, a **new approach of deriving a small number of reference days** in order to represent a whole year **was developed** (Fischer 2015b). This approach allows running a real time HIL simulation within three testing days while still representing the properties of a full year simulation within accuracy of 90%. Using the type day sequence, the test of the GreenHP control platform was performed successfully in the HIL environment.

Figure 16: Setup of the GreenHP control framework in the SmartEnergyLab

Further ramping and switching signals were used to identify the dynamic properties of the GreenHP with respect to the requirements of smart grid operation. It was found, that the heat pump can dynamically adjust compressor speed, which can be used to react to switching and ramping requirements. However, the dynamics are limited. In the switching case, it took 5 minutes for the heat pump from zero to full load conditions. This is not sufficient for prequalification in primary reserve power markets. In the case of ramping, the maximum load gradient achieved is 3000 W/min. The bottlenecks identified concerning reaction speed of the GreenHP unit are:

- Limitations in the allowed ramp rate of the compressor.
- The time at minimum compressor speed (rpm) after a switch on and before a switch off.
- Minimum runtimes

In terms of grid integration, it would be favourable to set the electricity consumption directly instead of setting the heat request. However, this would possibly lead to challenges on the side of thermal system management.

Summarizing, an **advanced**, three layered, **controller platform** was designed and tested for several use cases to allow load balancing at maximum efficiency and lowest costs of the overall heat pump system. Depending on the use scenario and climate zone selected, it was shown, that the **Model predictive controller reduces the energy demand** of the GreenHP system between **1 % and 15 %** achieving an **absolute saving** between **118 €/a to 306 €/a**. Following the PV follow approach, the **PV-plant operation costs** could be further **decreased to 61 to 93 €/a** depending on climate zone of the system.

The **System Controller** of HESCH, in charge of the overall heat management, was **developed** successfully on hardware and software level **to market maturity** and is offered as **CE-certified product for all heating solutions with storage capabilities**.

1.3.3 *GreenHP Prototype*

1.3.3.1 *Unit Design (Work package 3)*

The design of the GreenHP was strongly influenced by the anticipated developments on the component level. The main factors influencing the design include: refrigerant reduction, using propane as refrigerant, and optimization of the evaporator unit on both, refrigerant and air side.

1.3.3.1.1 Refrigerant selection

To determine the best suited refrigerant from environmental and technical aspects, a thorough life cycle climate performance (LCCP) analysis was performed, as well as an analysis of the future prospects for the new HFO refrigerants, an analysis of the safety requirements using different types of refrigerants and an analysis of the implications for the different components of the system using different types of refrigerants.

As the GreenHP was designed for low environmental impact, only refrigerants with low global warming potential (GWP) were considered. For the analysis, four refrigerant options were selected: 2 hydrocarbons (R290, R1270) and 2 low GWP hydrofluorocarbons (R152a, R1234yf). The refrigerant R410A was also included in the analysis as a reference as this is a common refrigerant today, known to give good performance.

The results, summarized in the Figure 17, demonstrate that the indirect emissions are dominant in the total lifetime emissions. The share of indirect emissions is, for these refrigerants, exceeding 99%. Consequently, the selection of the most environmentally friendly refrigerant is also a selection of the most efficient one.

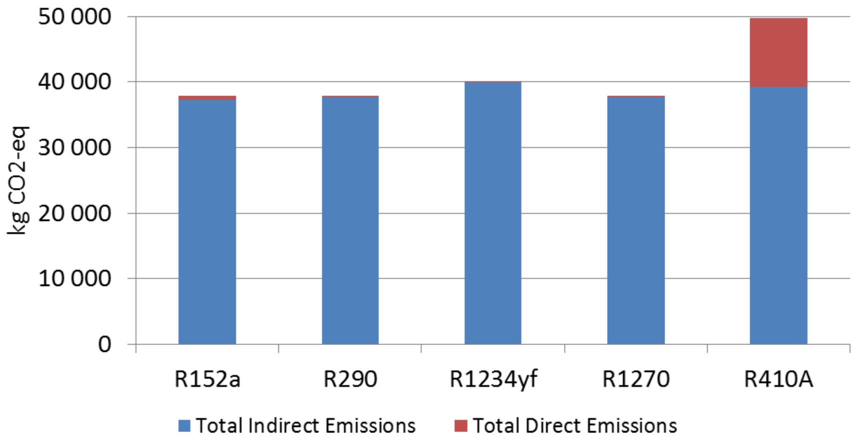


Figure 17: Result of LCCP analysis. Total direct and indirect emissions using different refrigerants

All pre-selected low GWP refrigerants exhibit lower lifetime CO_2-eq emissions when compared to the R410A baseline. Among the low GWP refrigerants, R1234yf has the highest lifetime emissions, whereas R152a, R290 and R1270 are almost equal in their lifetime CO_2-equivalented (CO_2-eq.) emissions, with hydrocarbons showing the best environmental performance. Thus, from the environmental point of view, R290, R1270 and R152a are more or less equally good, and the selection out of these three for the GreenHP project had to be made based on other selection criteria. Considering the possible limitations on the use of HFC refrigerants, as well as other aspects of the refrigerant use, preference was given to hydrocarbons, of which the refrigerant R290 (propane) was selected as the refrigerant to base the heat pump design on.

1.3.3.1.2 Development and Simulation of the Refrigerant Circuit of the Heat Pump Unit

Within the GreenHP project, different options for the refrigerant circuit were considered: separate desuperheater, internal heat exchanger between the liquid line and the suction line, and two stage expansion with Vapor Injection (VI) at an intermediate pressure at the compressor

High efficiency (high SPF) was a prime criterion in the selection process. Another important criterion was low refrigerant charge of the system. The rationale between this was safety: propane is highly flammable, and the risk of accidents is directly related to the amount of refrigerant released in case of leakage. The system is designed to be placed outside, e.g. on the top of a roof. In this case there is no limit to the charge of hydrocarbon refrigerants according to the standards.

However, it was considered important to show that a low charge is achievable, both for public acceptance of the technology, and for possible location inside. Additionally, for the case of HFC refrigerants, low charge is important for environmental reasons as this would imply that leakages are detected already after small losses of refrigerant, and that the release of refrigerant would be low in the case of loss of the whole charge, e.g. due to a valve failure.

For the analysis, two simulation programs were developed, to test the performance under different running conditions, with different selections of components. Additionally, the charge of refrigerant for the different options was estimated. Part of the simulation results were disseminated through a conference paper presented at the Gustav Lorentzen conference in Hangzhou (Palm, 2014).

Based on the simulations, it was decided to design the heat pump with two stage expansion and vapour injection (VI), but not to use a desuperheater or internal heat exchanger.

From the theoretical calculations, it was found that the refrigerant charge could be as low as about 500g. To reach this low charge, it was assumed that:

- The condenser was designed with minichannels on the refrigerant side, no subcooling in the condenser.
- The evaporator was designed with minichannels, that the volume of the headers was minimized and designed to avoid any uphold of liquid.
- The oil charge of the compressor was minimized.
- All connecting tubing was selected with lowest possible length and diameter while maintaining acceptable pressure drops under all running conditions.

However, for the system as it was finally designed, the total necessary charge was estimated to be 960g. During the final tests, it was found that a charge of 1900g was required for the system to run well under all conditions. The difference between the theoretical calculations and the tests may be attributed to the following:

- The required charge during defrost (1900g) was higher than during normal operation (1600g). Defrost operation was not simulated beforehand as the performance during defrost was considered of minor importance.
- The headers of the evaporator were not drained perfectly in the actual design as they were manufactured by hand. Some refrigerant may be trapped there, which was not considered in the calculations.
- The estimate of refrigerant absorbed by the oil is difficult to do in an exact way, particularly as oil is not located only in the oil sump of the compressor, but also distributed in other parts of the compressor and the system as a whole.

Although the **charge** was higher than expected, the result is still very good, corresponding to **just over 65 g of refrigerant per kW of heating capacity**. To the knowledge of the experts within the consortium, **there has not been an air/water heat pump designed before with such a low refrigerant charge.**

1.3.3.1.3 Preparation of drawings and documentation of the design

The drawings of the GreenHP were prepared in stages, from initial sketches to the final construction drawings. Additionally, lists of components were set up during the initial design phase. Based on the specifications of the components, a detailed construction drawing of the prototype unit was made including an optimal arrangement of the single components and with a special focus on the aspect of refrigerant reduction, which is only achievable with the piping length kept as short as possible. Figure 18 shows the 3D-sketch of the GreenHP-prototype construction drawing.

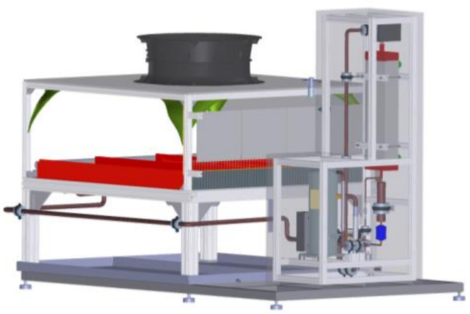


Figure 18: 3D-sketch of the GreenHP-Prototype

The **final main specifications of the GreenHP prototype unit design** are as follows:

- Using propane as refrigerant
- Heating capacity of 30 kW at the design point A-10/W55 (T_0-18/T_c+58).
- A -10 / W 55 (design point for the heat exchanger):
    - evaporator capacity of approx. 18.4 kW
    - dT: T_air_in – T_evaporation: 7 K
    - T_air: 5 K
- Modulating scroll compressor with a speed range of 1800 to 7200 rpm
- Optimized design for part load condition at the design point A2/W42
    - Heating capacity approx. 15.5 kW
    - Cooling capacity approx. 11 kW

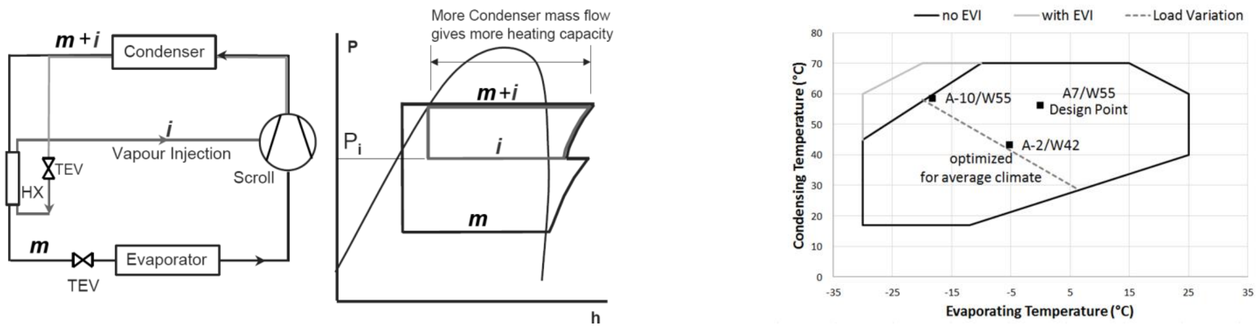
1.3.3.2 *GreenHP components*

1.3.3.2.1 Compressor (Work package 5)

The main aspects of utmost relevance for the compressor design were the use of propane as a refrigerant, refrigerant charge reduction and a large operating range as needed with air/water heat pumps. For the refrigerant charge reduction, the amount of oil and the oil discharge needed had to be reduced too, as the compressor oil dissolves the refrigerant when the heat pump is not in operation. Therefore, a new concept for the oil sump in the compressor had to be developed to reduce the amount of oil significantly.

In order to achieve these goals, various options were investigated: utilization of new types of variable speed motors, wet vapour injection and economizer ports and the extension of the variable speed range of the compressor to frequencies down to a level around 15 Hertz.

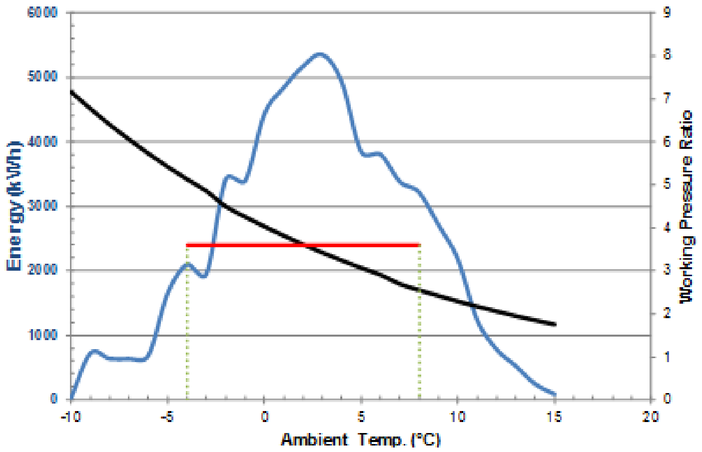
For the use of the compressor in air/water heat pumps for refurbished (and non-refurbished buildings), a large operating and performance range with optimized part-load performance is required. To meet these requirements, the compressor enhanced within the GreenHP project includes Enhanced Vapour Injection (EVI). Figure 19 a shows the scheme of the EVI circuit and the process in the log p, h - diagram. **By using an EVI in combination with the refrigerant propane, the compressor outlet temperature is reduced, a higher COP (+ 11 %) and an increased heating capacity (up to 48 %) are achievable at large temperature lifts between heat source and sink** (Bella 2015). Figure 19 b shows the compressors' operating range with and without EVI. It can be seen that especially for A/W-HPs the operating range at low evaporation temperatures <-15°C and high condensing temperatures +70°C can be extended with EVI.



a) b)

Figure 19: a) Scheme of the EVI-Cycle (Bella 2015). b) Operating range with/without EVI (Emerson 2015).

For the optimum design of the scroll compressor to be used in the average climate zone, the main operating range and the typical pressure ratio for the design of the scroll was based on the climate data and the heat demand depending on the outdoor temperature. 80 % of the heating demand occurs in the temperature range between -4°C and +8°C outside temperature (Figure 20 a). With this information, the compressor was designed for a heating capacity of 30 kW at the operating point evaporating temperature -18°C / condensing temperature + 58°C and optimized for the operating point evaporating temperature -5°C / condensing temperature + 43°C concerning the scroll design (Bella 2015). With the optimization of the compressor and the control at the design point for a part load of 18 kW heating capacity, the highest efficiency for the main operating area with the most hours of operation was achieved. Figure 20 b depicts one of the compressor prototypes developed with sight glass for the investigations of oil management during lab measurements. First compressor tests resulted in a COP of approximately 3.6 at the standard rating point A7/W55 which was promising regarding a high overall efficiency.



a)

b)

Figure 20: a) Pressure Ratio Optimization for an Average Climate (Bella 2015). b) R290 EVI-Prototype Compressor with Oil Sight Glass (right) (Emerson 2015).

Concluding, the extensive work undertaken on the compressor resulted finally in a prototype of an optimised variable speed scroll compressor using EVI, not only meeting all performance targets set, but **over-achieving especially the target on oil charge reduction**. Within the GreenHP project, the **oil used inside the compressor was decreased from initially 3.3 litres in the standard compressor design to finally 1.0 litre in the second optimised design variant.**

1.3.3.2.2 Condenser (Work package 8)

The condenser was designed to reach the lowest possible charge of refrigerant while maintaining at the same time very high demands on performance. For that purpose, extruded multiport aluminium (MPE) tubes were used as the refrigerant channels, see Figure 21.

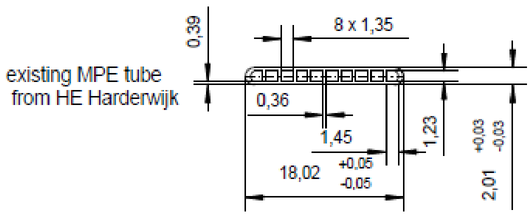


Figure 21: Drawing of single MPE tube, (B. Nitsch, AKG)

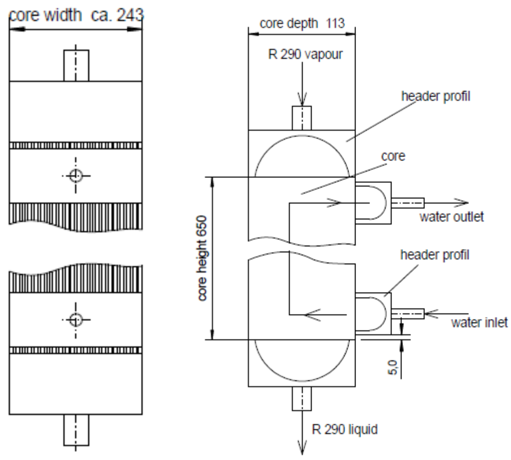


Figure 22: Drawing of complete condenser (B. Nitsch, AKG)

Such tubes were arranged side by side to form layers of refrigerant channels. These layers were then stacked in between offset strip fins, forming the water channels. Finally, the pile was inserted into a shell, sealed at the MPE tube ends, and headers attached to the shell. The complete heat exchanger is shown in Figure 22. Figure 23 shows a photo of the condenser and Figure 24 displays photos through the ports on the refrigerant and water sides.

The design of the condenser was inspired by a similar prototype condenser used at KTH in a previous project on propane heat pumps, and also by a design commercially used by Thermia in their heat pumps for many years. The whole condenser was made of aluminium, designed and assembled at the project partner AKG.

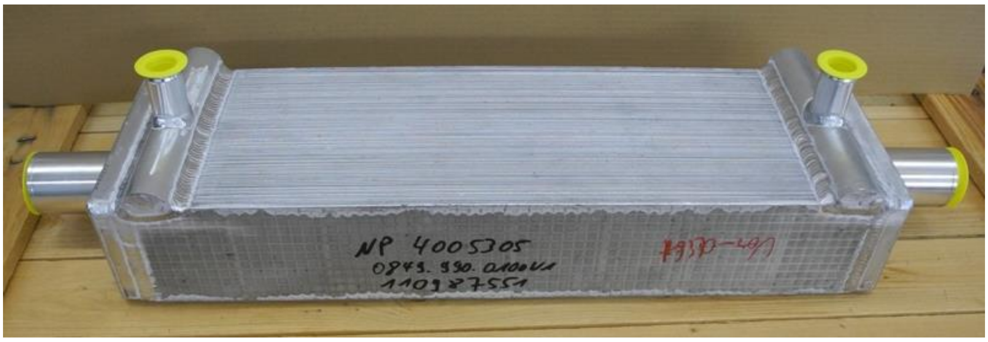


Figure 23: Photo of condenser

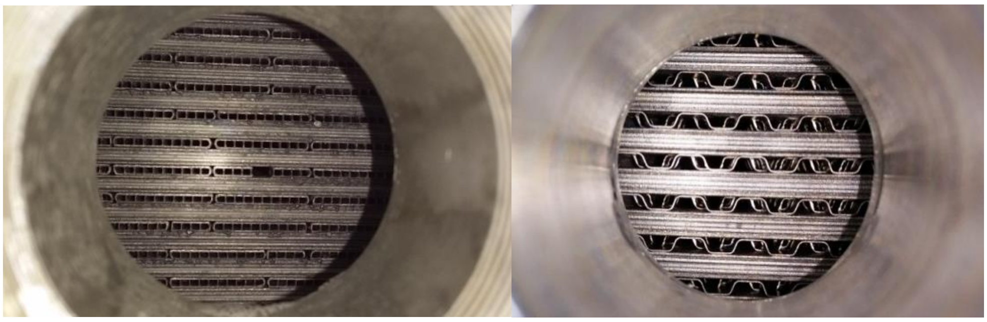


Figure 24: a) Photos through refrigerant port b) and water port

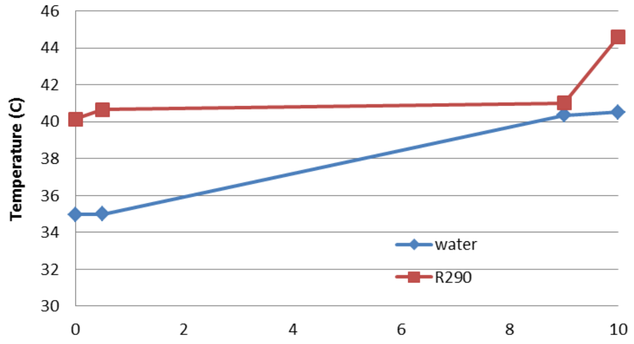
Pressure tests undertaken showed that the burst pressure of the condenser was 136 bar. As the initial tests of the condenser’s thermal and hydraulic performance indicated that the performance was as expected, it was decided to keep the initial design of the condenser.

After a temporary break in the performance tests, during which the condenser was not connected to the system, it was found that severe corrosion had occurred inside the condenser, leading to leakage between the water and the refrigerant side. It is believed that there may have been water left inside the complex structure of the heat exchanger, and that this water, in combination with the oxygen in the atmosphere led to the corrosion. A detailed analysis of the condenser showed that almost all corrosion was in the form of pitting corrosion, and that the corrosion products found inside consisted mainly of aluminium and oxygen, but also of sulphur, calcium, iron and copper. These substances probably originate from the water and from other components in the system. It is also important to note that all pitting corrosion attacks were on the plates and turbulators made from AA3003 alloy, while the MPE tubes made of AA3110 were not attacked. Selection of material therefore proved to be very important.

A new condenser was built and to avoid further corrosion attacks the condenser was connected to the water heat sink system through an intermediate heat exchanger. The small loop in between the two heat exchangers was filled with de-ionized water with corrosion inhibitor added.

The condenser was tested first (before the leakage) in water to water tests, so called Wilson plot-tests, in order to determine the heat transfer coefficients on the shell side of the heat exchanger under different flow conditions. These results could later be used together with the measurements of the overall heat transfer coefficients determined from the final tests when used as a condenser.

The tests as condenser was done both with and without a desuperheater connected in between the compressor and the condenser. Tests were also performed in the same system with a standard brazed plate heat exchanger with approximately the same heat transfer surface area in order to compare the performance between the two types. Results from the tests can be displayed in several different ways. Figure 25 shows that the temperature difference between the two fluids is very small in the pinch point, showing that with the water flow rate used even an ideal, infinitely large condenser would not have reached much lower condensing temperature.



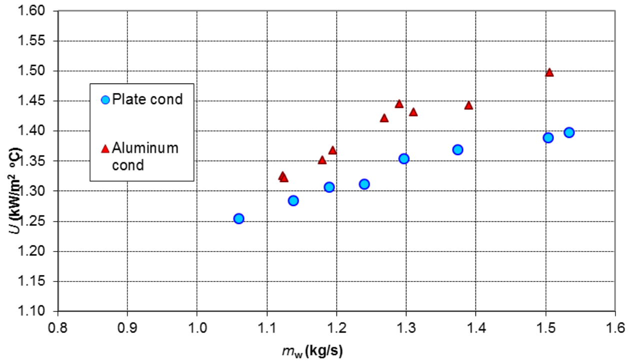


Figure 25: Temperatures in the condenser connected with desuperheater at about 30 kW.

Figure 26: U-value of aluminium condenser and of plate heat exchanger condenser

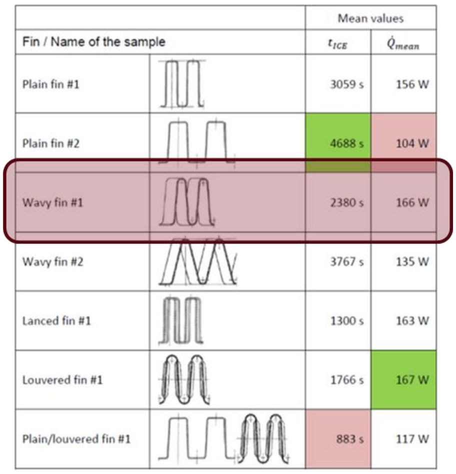
Figure 26 shows the results in terms of the U-value (referred to the heat transfer area on the refrigerant side) of both the aluminium heat exchanger and of the plate heat exchanger. As shown, the **performance for the aluminium heat exchanger is much better**, and the difference is largest at high water flow rates. Pressure drop on the water side was also measured, and found to be only slightly higher for the aluminium condenser.

Summarizing, the work undertaken resulted in a **novel aluminium condenser prototype** using extruded multiport aluminium (MPE) tubes as refrigerant channels, which **showed very good performance** when compared to state-of-the art condensers. Before commercializing a product based on the design developed, the material selection must be considered carefully in order to assure that corrosion problems can be avoided.

1.3.3.2.3 Evaporator (Work packages 6 and 7)

As the work on the evaporator was split into a) air side and b) refrigerant side, the research results are described in the following separately for the air and refrigerant side, before the final design of the complete evaporator unit and its testing results are presented.

<u>Air-side of the evaporator:</u> To energetically optimize the air side of the evaporator, icing and defrosting behaviour of the evaporator were closely investigated on component and unit level. Different fin geometries, structures (Figure 27a) and coatings for heat exchangers were examined experimentally and numerically with respect to the change in the pressure drop and heat transfer due to icing.



Fin / Name of the sample		Mean values t_{ICE}	$\dot{Q}_{mean}$
Plain fin #1		3059 s	156 W
Plain fin #2		4688 s	104 W
Wavy fin #1		2380 s	166 W
Wavy fin #2		3767 s	135 W
Lanced fin #1		1300 s	163 W
Louvered fin #1		1766 s	167 W
Plain/louvered fin #1		883 s	117 W

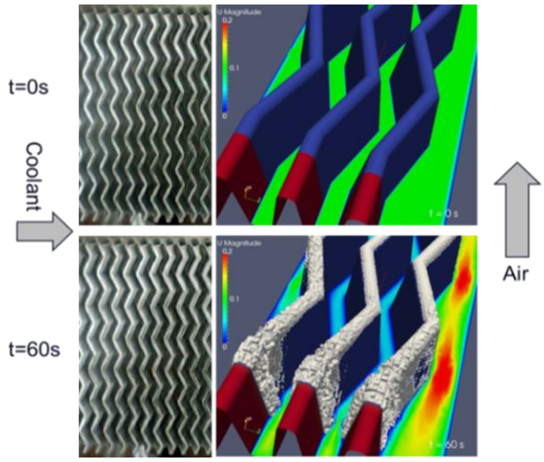


Figure 27: a) Icing Times and Heat Transfer for Various Fin Samples (Reichl 2015). Comparison Experimental / Numerical Studies (right) (Popovac 2015).

A comparison of experimental tests and numerical calculations showed that the **strongest icing takes place at the entry of the evaporator heat exchanger** (Figure 27b). Based on the results, **wave fins sized 10.4 mm in height and** with **4.5 mm fin pitch** were selected for the heat exchanger as they showed the best anti-icing, defrosting and heat transfer characteristic (Popovac 2015).

The overall area of the evaporator was determined by the requirements of the air side to provide the heat at design air speed. The thermodynamic sizing of the evaporator was performed using the known fin performance and the calculations showed that a core with 1500x1507x95 mm was sufficient for the requested thermal performance (Table 4 and Figure 28).

Table 4: Specification of the evaporator core

Core length	1500	mm
Core width	1507	mm
Core depth	95	mm
Number of passages	64	
Number of tubes side by side	2	
Number of passes refrigerant	1	
Tube design:		
Tube depth	45	mm
Tube height	1,8	mm
channel type	rectangular	
channel height (internal)	1,4	mm
channel width (internal)	2	mm
outer channel width (internal)	2	mm
amount of channels per tube	21	

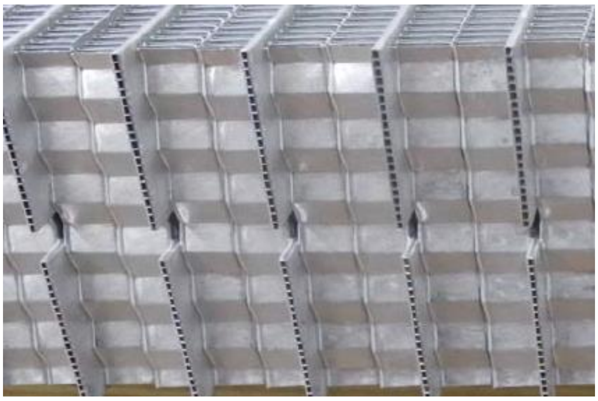


Figure 28: Detail of evaporator core with pairs of MPE-tubes and two layers of fins between

Refrigerant side of the evaporator: On the refrigerant side of the MPE-tube evaporator, the major challenge to overcome was to generate an even distribution of the refrigerant. For that purpose, a new distribution concept based on "bionic" distribution was investigated, manufactured and tested.

The refrigerant side design pressure drop was limited to be equivalent to 1 K of temperature drop. To meet this requirement, the pressure drop was simulated with different models for one- and two-pass for full load conditions. Only in one-pass configuration all models showed a sufficiently low pressure drop, therefore this configuration was chosen.

The usage of a bionic refrigerant distributor limits the amount of parallel passages to 2^n where n is an integer. To distribute the refrigerant to 128 MPE-tubes in 64 passages, four 16 port bionic distributors connected to 64 distribution headers, each connected to two MPE-tubes, were used. Therefore the final design for the refrigerant distributor consisted of a 1-to-4 venturi distributor in cascade with four 1-to-16 bionic distributors. The channels of the bionic distributor were stamped into an aluminium sheet. This sheet was then brazed onto an aluminium plate by vacuum brazing.

Figure 29 a shows the bionic distributor with single 1-to-16 elements, Figure 29 b depicts all elements connected to the venturi distributor and the evaporator core, which was determined by the requirements of the air side.

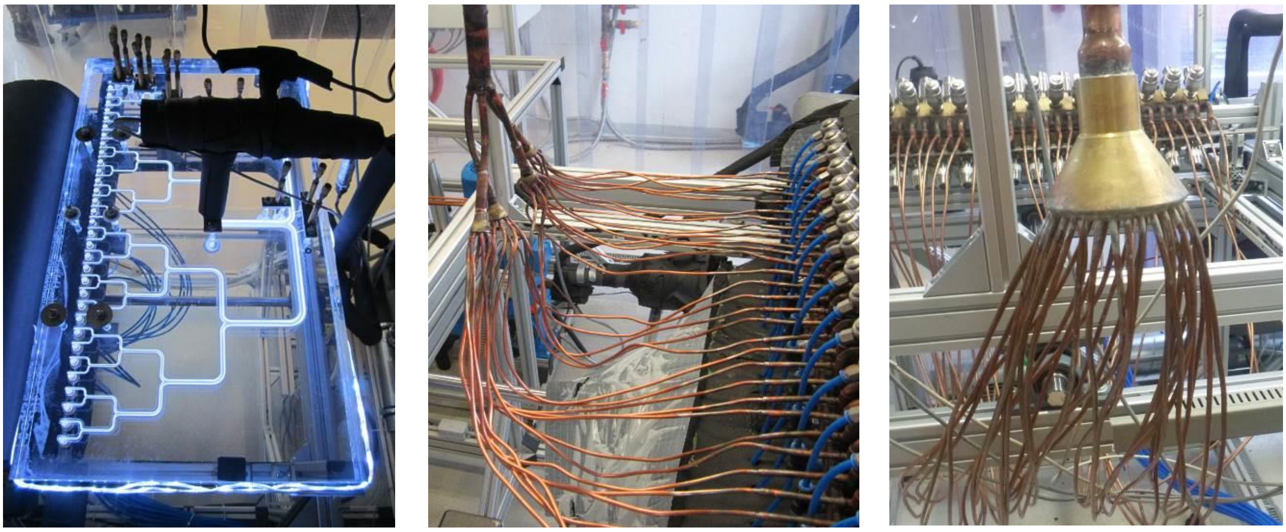
a) stamped single 1-to 16 element distributor

b) all elements connected to the venturi distributor and evaporator core

Figure 29: Bionic distributor

In order to compare the distribution quality of the new fluid distributor device, several fluid distributors were tested on a test rig at Fraunhofer which allowed conditioning refrigerant R290 to defined pressure, temperature, vapour quality and mass flow and then pass it through a fluid distributor. The tested distributors included:

- Bionic plexiglass: A prototype of the bionic distributor made of plexiglass to optically access the flow. Its distribution ratio is 1 to 32, it has round flow channels (see Figure 30 a)
- Bionic aluminum: A sample distributor as it is installed in the GreenHP unit. It has a distribution ratio of 1 to 16 and semi-circular flow channels' section
- Cascade of venturi distributors: In order to get a distribution ratio of 1 to 32, a cascade of one 1 to 4 venturi distributor and then four parallel 1 to 8 venturi distributors was mounted. (see Figure 30 b)
- Nozzle distributor: a commercially available 1 to 32 nozzle distributor by Sporlan (see Figure 30 c)



a) bionic plexiglass distributor **b) cascade of venturi distributors** **c) nozzle distributor**

Figure 30: Bionic distributors investigated

To compare the distribution quality of the different distributors, the relative mass flow difference between the exit with the highest and the one with the lowest mass flow was taken as characteristic number. Figure 31 depicts a comparison of the four distributors at four different load conditions. Ideal distribution would be zero. The **bionic aluminium distributor showed comparable results to the nozzle distributor and a better distribution behaviour to the cascade of venturi distributors.**

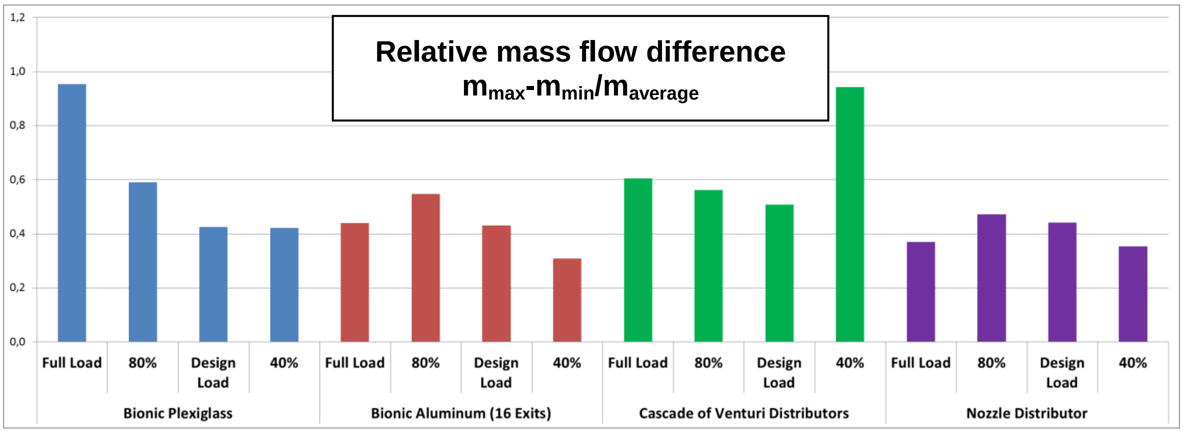


Figure 31: Comparison of distribution qualities of the tested distributors at different load conditions

Final design of the evaporator unit, manufacturing and testing

For defining the layout of the evaporator heat exchanger, its form factor, and the arrangement, a number of fans have been investigated by means of CFD-simulations, which resulted in the favourable configuration of **using one fan on the suction side of the quadratic heat exchanger**. The inflow/outflow surfaces of the heat exchanger are square shaped with a length of L=1.5m and a width of B=1.5m. The depth in airflow direction of the heat exchanger is 0.1m. The distance fan inlet plate to heat exchanger is 0.5m; the heat exchanger is elevated about 0.5m above the floor.

The geometrical framework conditions of the evaporator unit were fixed as outlined in Figure 32. The heat exchanger is positioned horizontally, and the main airflow direction through the heat pump is from bottom to top. The main reasons for this arrangement came from icing and defrosting considerations.

Figure 33 shows the final evaporator unit design in CAD. The basic construction was done by using standard aluminium profiles incorporating a high flexibility. For good optical access to the interior components, acrylic glass panels were used massively for the evaporator housing. The fan system (in black colour in the CAD drawing) can easily be exchanged[11]. In the top corners of the evaporator housing, air guiding elements leading to a more homogenous air velocity distribution on the heat exchanger were applied.

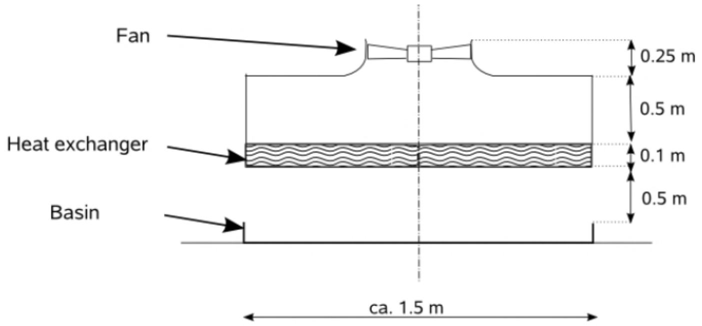


Figure 32: Geometrical outline of the evaporator

Figure 33: Final design of the evaporator

To determine the optimal design of the evaporator unit, the air flow below, within and above the evaporator unit was analysed experimentally and numerically. The experimental methods used for characterizing the flow field comprise laser optical (particle image velocimetry, PIV) and probe techniques (constant temperature anemometry, CTA); CFD was applied for calculating air flow distributions inside the evaporator unit using boundary conditions below the heat exchanger and above the fan. The obtained results of the PIV measurements showed that the flow distribution within the evaporator unit in its unblocked state (see Figure 34), can be considered as symmetric and equally distributed over the whole range of the evaporator. The symmetric flow distribution was confirmed by the numerical results (Figure 36). However, the icing and blocking of the evaporator led to a strong flow asymmetry (Figure 35).

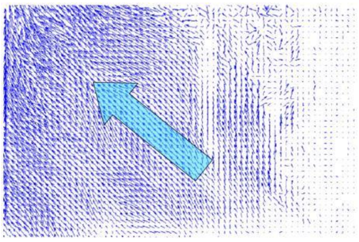


Figure 34: Flow distribution within the evaporator unit in its unblocked state (PIV measurements)

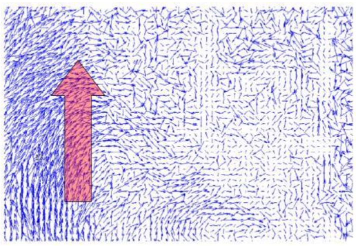


Figure 35: Flow distribution within the evaporator unit in its blocked state (PIV measurements)

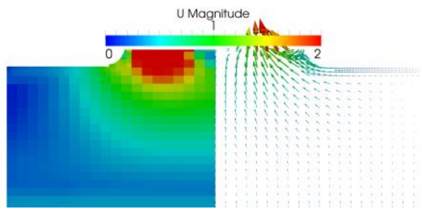


Figure 36: Numerical analysis of the flow distribution in vertical plane, when heat exchanger is not blocked

[11] Important, as two different fans were developed within the project

The CTA technique was used for the punctual investigation of the horizontal flow distribution of the evaporator unit. The experimental results of the CTA measurements are presented both in two dimensional box views (along the x and y axis) and linear graphs (along the x-axis). The main observation was that the **flow features qualitatively do not change much between 60%** (˜550 rpm) **and 90%** (˜1250 rpm) **of the fan control voltage; only the magnitude levels are different** (compare Figure 41: 60% left, and 90% right).

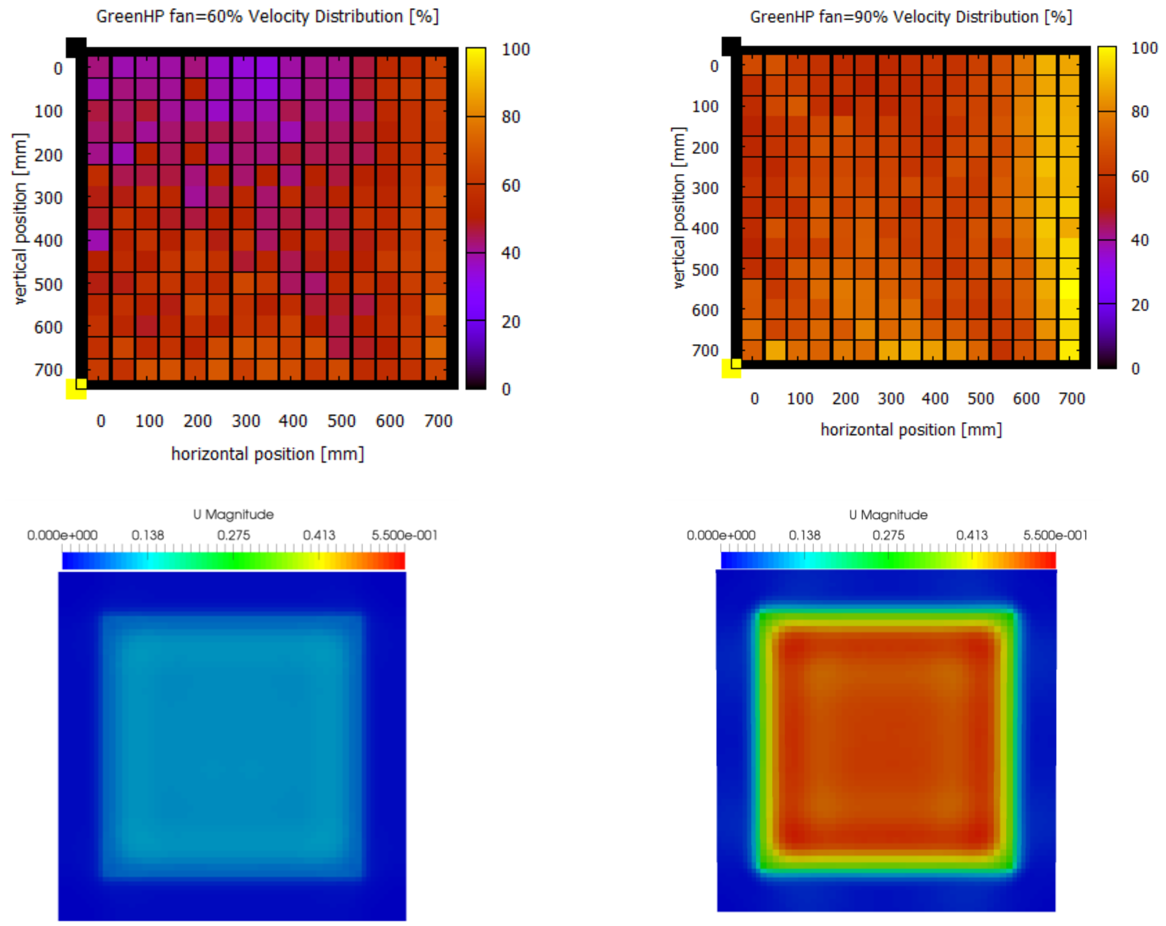
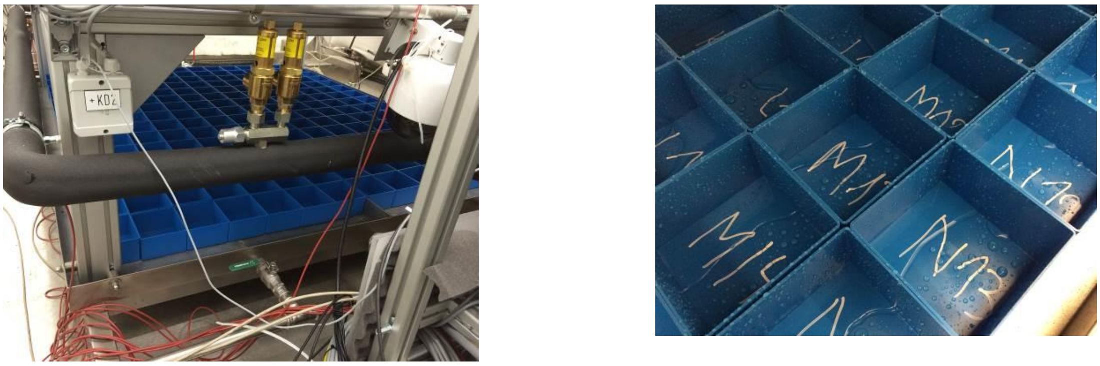


Figure 37: a) Velocity distribution [m/s] under heat exchanger, fan 60% b) and 90%, experiments (above) and simulations (below) in the suction mode (the velocity scale for the two modes kept the same).

From the measurements 10 cm below the heat exchanger it became evident that **in the suction mode the flow is rather uniform over its entire area:** only some small uniformity disturbance can be observed at the edges of the heat exchanger. [12]

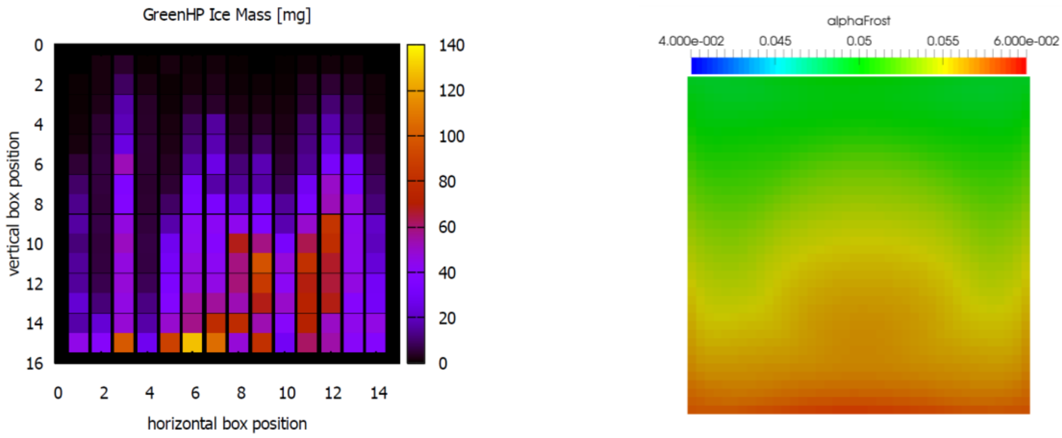
Furthermore, the frost mass distribution of the complete evaporator was measured using scale techniques and analysed numerically. The spatial distribution of the frost mass was determined from the grid of holders which were collecting the water melted from the frost created at the respective location (see Figure 38). The total collected condensing water amounted to 6 litres.

[12] This can be attributed to the fact, that in the suction mode, the main flow orientation at the edges will deviate significantly to a normal to heat exchanger configuration.



a) b)

Figure 38: a) Box holders placed below the heat exchanger. b) After defrosting, water is collected in the boxes and the weight increase can be measured using a scale.



a) **b)**

Figure 39: a) Spatial distribution of the accumulated frost mass, measurements and b) simulations.

The obtained **frost mass distribution** (see Figure 39) **featured a pronounced non-uniformity**, with the **maximum frost accumulated near the coldest heat exchanger section**. Furthermore, from the flow measurements presented above it is clear that the **air flow around the heat exchanger cannot be the cause of this non-uniform distribution**. Obviously, the **main cause of the non-uniformity is the temperature distribution of the coolant within the heat exchanger** (in fact, the obtained frost mass distribution reflects also the characteristics of the coolant distribution).

Summarising, extensive research work was undertaken to optimize the evaporator on both air and refrigerant side applying a novel refrigerant distribution concept. The wave fins of the heat exchanger were **sized 10.4 mm in height using 4.5 mm fin pitch**. The evaporator design was fixed **using one fan on the suction side** of the **quadratic heat exchanger sized 1500x1507x95** mm. The **bionic aluminium distributor** showed comparable results to the nozzle distributor and a **better distribution behaviour** to the **cascade of venturi distributors.**

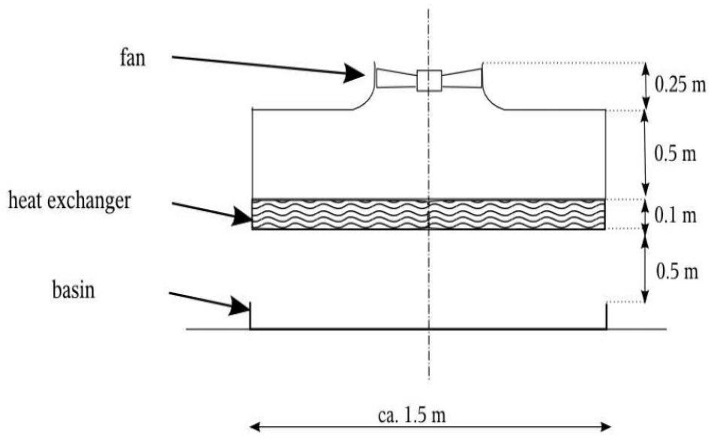
The flow distribution within the evaporator unit proved symmetric in an unblocked, “ice-free” state. However, **icing** of the evaporator during testing **led to asymmetries** in the **flow distribution and** hence to efficiency losses.

The open research questions concern in particular the fin geometry of the heat exchanger, the arranging of the components within the evaporator to avoid condensing water remaining after the defrost cycle, and a refrigerant distribution concept guaranteeing an even distribution of the refrigerant.

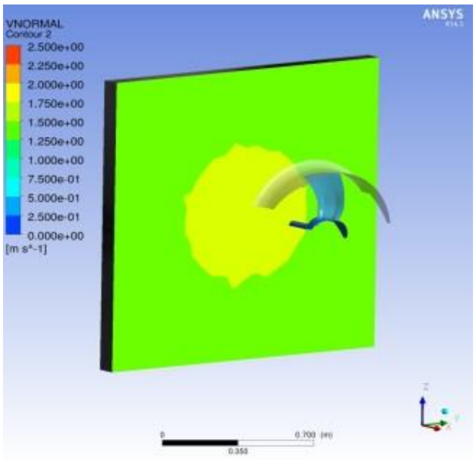
1.3.3.2.4 Fan (Work package 7)

For an efficient operation of an air/water heat pump such as the GreenHP, a proper technical design and an optimal installation of the fan are of great importance, as in further consequence, the efficiency (power consumption of the fan) and noise emissions are directly influenced by the integrated fan (Dietle 2014). To meet the requirements, tests and developments in the areas of fan blade design, fan diameters, drive motor, distance between evaporator and fan, velocity distribution and noise emissions were performed.

The optimum positioning of the fan was determined by the square shape of the evaporator (see Figure 40 a). The fan is centrally positioned on top of the evaporator unit with a distance of 500 mm from the evaporator heat exchanger (Lörcher 2015). With the configuration selected, a uniform velocity distribution, between 1.0 and 1.2 m/s was achieved across the entire evaporator surface (Figure 40 b).



a)



b)

Figure 40: a) Geometric Outline of the Evaporator Unit (Lörcher 2015). b) Velocity Distribution across the Evaporator Surface (Ziehl-Abegg 2014).

For the given evaporator framework, energetically and acoustically optimized fans were developed. This was done independently for two different fan drives, a smaller one available at a lower cost and a bigger one available at higher cost. The benefit of the bigger motor is the possibility to achieve the required power at lower rotational speed leading to advantages in regard to noise emissions.

The optimum fan diameter was specified for both fan types by means of annual simulations. Figure 41 shows the best fan geometry found for the small EC90 motor drive with a diameter of 0.63 m and 3 blades, which are slightly forward swept. Its annual energy consumption can be found in Table 5:

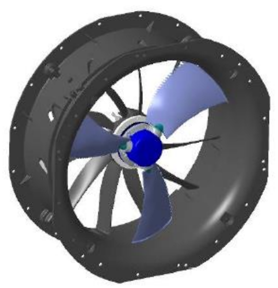


Figure 41: Best fan geometry in CAD view (Ziehl-Abegg)

Table 5: Comparison of best performing fans (Ziehl-Abegg)

annual cons. [kWh]	**SUM**	**OP 1**	**OP 2**	**OP 3**	**OP 4**
best fan	*2061*	*570*	*524*	*670*	*297*
best OP 1	*2256*	*544*	*522*	*890*	*290*
best OP 2	*2234*	*564*	*518*	*850*	*302*
best OP 3	*2087*	*599*	*525*	*655*	*308*
best OP 4	*2269*	*551*	*537*	*850*	*288*
best max. efficiency	*2282*	*555*	*520*	*915*	*292*

OP… Operating Point

Both fan types were built as prototypes and tested experimentally on component and unit level. To further decrease the sound level, three additional prototypes incorporating a novel sound reduction technique were built and analysed on the test rig and in the evaporator unit.

For the fan design it was shown, that the annual energy consumption can be reduced considerably compared to methods, where fans are selected just considering their best efficiency point or their efficiency at design duty point (Lörcher 2015). Especially the high pressure drops under icing conditions showed an important impact, even if they occur only in a smaller part of the operating time. These operating points lead to the necessity of a significantly forward swept blade. The experimental and numerical methods which were adapted and enhanced during the project, allow for qualitative and quantitative analysis of the flow and frosting phenomena expected in an air based heat pump unit.

The fans developed within the GreenHP project show **minimized annual energy consumption** and **low noise emissions** compared to state of the art fans. The **tonal noise reductions** achieved within the project are **up to 4dB** in octave bands**.**

In addition, a **CFD-based tool** supporting the design of an optimized air duct and the selection of the optimal fan for various applications was developed.

1.3.4 *Performance of the GreenHP prototype (Work package 9)*

All components developed were finally integrated into a heat pump prototype unit equipped with several plexiglass casings as depicted in Figure 42 to allow better monitoring during testing.

Figure 42: Prototype of the GreenHP unit

Various test runs were carried out at the AITs accredited heat pump testing facilities based on international standards EN14825[13] and EN14511[14] to measure the performance of the GreenHP heat pump prototype unit and compare the results with the performance indicators aimed at.

The measurements for the SCOP calculation were performed under “dry conditions” to avoid icing of the evaporator. The testing conditions and procedure are described in more detail in D9.1 “Test

[13] EN 14825:2016, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling. Testing and rating at part load conditions and calculation of seasonal performance.

[14] EN14511-1 to 4: Air conditioners, liquid chilling packages and heat pumps for space heating and cooling and process chillers using electrically driven compressors

report of the GreenHP unit at specified test conditions" available for download at http://www.greenhp.eu/deliverables/public-deliverables.

As described earlier on, and depicted in Table 6, the **GreenHP prototype accomplished nearly all performance indicators aimed at.** In particular, it performed very well in terms of its seasonal efficiency with an SCOP[15] of 3.3, which is 0.2 higher than anticipated and comparable with the best performing heat pump units in the smaller capacity range of 10-20 kW heating capacity.[16] Furthermore, the system showed higher overall efficiency results than expected, which is due to the optimization of the unit system for part load conditions. The higher SCOP resulted also in an improved primary energy ratio (PER) of 1.32 (instead of 1.24) and lower CO_2-emissions than anticipated (187 g CO_2/kWh usable energy instead of 200g). In addition, the system uses with propane a natural refrigerant with a very low global warming potential (GWP) of only 3.

Out of the performance indicators, only three proved to be over-ambitious: firstly, the COP of 3.5 could not be reached; instead the measured COP was 3.0. Secondly the refrigerant charge required for maintaining a stable operating behaviour was 1900 g, which corresponds with 65 g per kW thermal capacity. This is more than aimed for and calculated[17] in the simulation, but still much less than conventional systems use. It is doubtful if an air/water heat pump has ever been designed with so low charge. Finally, the energy used for defrosting was decreased less than anticipated.

Table 6: GreenHP results compared to target performance indicators

Specifications	State-of-the-art*	GreenHP unit planned	GreenHP prototype
Carnot efficiency factor	40-45%	55-60 % (increased by 15%)	54 %
COP (A7/W55)	2.5	3.5	3.0
SCOP ** (based on EN14825)	2.1	3.1	3.3
Primary Energy Ratio (PER)* (kWh useful energy/kWh primary energy)**	0.84	1.24	1.32
CO_2 Emissions * (g CO2/kWh usable energy)**	294	200	187
Refrigerants used	primarily HFC	natural refrigerant or HFO	natural refrigerant
GWP of refrigerants used	>1300	<150	3
Refrigerant charge range (g refrigerant / kW heating)	200 – 500	30 (hydrocarbons) 60 (HFOs)	65
Defrost Energy Used	about 10 %	<5%	<10 %
Smart grid integration	N	Y	Y

* State-of-the-art relates to most advanced air/water heat pumps available on the market 2012
** Seasonal Coefficient of Performance (SCOP) calculated based on EN 14825
*** Based on EU UCTE-Mix; PEF=2,5; EN 15603: CO2-UCTE-Mix = 617g CO2/kWh final energy

[15] The SCOP was calculated according EN14825 for "average climate" and high temperature applications.
[16] Heat pump units in the capacity range of 10 – 20 kW reach SCOPs between 2.9 and 3.4.
[17] The deviation in the calculation is probably caused by parts in the unit where liquid refrigerant is collected and which were not assumed for in the calculations. Additionally, it is difficult to estimate the amount of refrigerant in the compressor, as this component contains oil which is miscible with propane. The amount of propane in the oil is difficult to estimate due to multiple interdependencies. There are further investigations required on the level of both, calculation methodology and on real setup of the prototype, to tap additional optimization potentials such as the piping length and the arrangement of the components within the unit.

From the very good test results on the part load behaviour with the highest efficiency reached at 48% of the compressor speed, it was evident, that the unit was designed for the part load conditions at A2/W42 with the highest Carnot efficiency of 54 % at this operating point. The **fan control settings turned out to have a big impact on the overall efficiency, with an increase of COP by 20% by reducing the fan speed from 100 % to 60 % at the operating point A7/W36.**

The condenser showed a very small temperature difference at the pinch point of less than 1K. Also the pressure drop on the water side, 38 mbar at 0.6 kg/s water mass flow rate, was rather low. Therefore **less auxiliary drive energy for circulating pumps is needed**, when integrating the GreenHP unit into a hydronic system. On the air side of the evaporator, the temperature difference between in- and outlet was between 5 and 6.5 K as expected during the design phase. Depending on the ice formation, the pressure drop on the evaporator ranging between 15 and 30 Pa is comparable to state of the art heat exchangers.

Due to the **selected fin geometry**, the **condensing water remained in the heat exchanger after the defrost cycle and blocked the air passages**. Even a reverse operation of the fan could not remove the condensing water. **Further investigations in arranging the heat exchanger in the evaporator unit and the fin geometry selection have to be undertaken to avoid this in future**.

A complex system was needed to distribute the refrigerant to 128 MPE-tubes in 64 passages. Therefore four 16 port bionic distributors connected to 64 distribution headers, each connected to two MPE-tubes, were used. It turned out that a **uniform refrigerant distribution was not possible by using this concept**. Thus only 70 % of the heat exchanger surface was fully active during the operation of the heat pump, resulting in a lower evaporating temperature than expected. **Hence the evaporator still needs improvements to harvest further possible efficiency gains**.

Summing up, the **GreenHP prototype showed very promising test results** on both, unit and component level **to build up from in the upcoming research projects**. The prototype performed particularly very well in terms of its seasonal efficiency with an SCOP of 3.3 leading also to an improved primary energy ratio of 1.32 and CO_2-emissions of only 87 g CO_2/kWh usable energy. In addition, the system uses with propane a natural refrigerant with a very low global warming potential of only 3. To harvest additional efficiency gains and further optimize the performance of the GreenHP unit, more research is required on component and system level including extensive field measurements.

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1.4 Potential impact – Socio-Economic and Wider Societal Implications

Evaluating the impact of the project at the moment of completion is challenging as it involves also assumptions on the future. The development of new, innovative concepts on unit and system level, and advanced research on heat pump components, and their broad dissemination to various target groups, especially heat pump and component manufacturers and the HVAC research community, do not guarantee a widespread uptake.

The feedback of the European heat pump manufacturer and component suppliers participating at the three industrial GreenHP workshops and at more than 50 industry-related fairs / conferences / workshops where the GreenHP project was presented was very positive.

The largest interest was gained by the following topics:

- Use of propane (R290) as refrigerant: for most companies only acceptable when used outdoors due to safety concerns; demand within industry on better understanding safety regulations.
- Refrigerant charge reduction: is seen as a relevant topic; measures have been implemented already to some extent by most companies respectively several companies are planning to take measures on that issue.
- Low-energy consumption and low noise level are considered very important when selecting the fan of air source heat pumps. Most crucial for noise reductions are frequency ranges of up to 125 Hz.
- Use of micro-channel aluminium heat exchangers for condenser and evaporator: questions on corrosion and manufacturability.
- Bionic concept for refrigerant distribution: even distribution of the refrigerant as challenge.
- Variable speed compressors, pumps and fans: are seen as important measures to increase the system efficiency.
- Smart grid integration and operation of heat pumps: most companies currently don't use an approach to smart controls.

The GreenHP concept was widely distributed within the European heat pump sector at in total more than 50 events and numerous publications. It is not known to the consortium, how many European heat pump manufacturers took up parts or the full concept as it was provided for free.

It is expected that the exhibition of the GreenHP prototype and the presentation of the final test results at the Chillventa 2016, one of the largest exhibitions worldwide for refrigeration, heat pumps, AC and ventilation taking place in Nürnberg, and at the 12th IEA Heat Pump Conference 2017 in Rotterdam, will further increase the interest of the European heat pump industry in the project and its results. The final measurement results of the heat pump unit are seen as the most relevant project result for heat pump and component manufacturers.

In the following, the economic, environmental and social impacts generated by the GreenHP project are presented.

1.4.1 *Economic impacts*

1.4.1.1 *Enhancing the application range of heat pumps*

Focusing on a medium-sized 30kW heat pump driven by a natural refrigerant for applications demanding supply temperatures of up to 55°C, the GreenHP project brought about substantial improvements on component and system level (as described in chapter 1.3) for market segments which are still in rather early stages of development: the use of heat pumps in refurbished MFH and in small commercial applications.

Another enhancement concerns the **cost gains** achievable by the use of the GreenHP components developed which are as follows:

- Reduction of the refrigerant charge by at least 70%[18] to a minimum of 65 g/kW heating, which results in about 2 kg refrigerant. Cost advantages are achievable by the lower refrigerant charge, and the lower price for propane (R290)[19]. Further advantages derive from the F-Gas regulation. Hermetic closed systems with less than 3 kg refrigerant are subject to less periodic maintenance work.
- Reduction of the oil charge of the compressor by 70%[20] leading not only to cost gains, but also to a lesser soil pollution in case of leakages.
- Reduction of the energy costs of the new fan design up to 10%.

1.4.1.2 *Sales figures generated by project partners*

The GreenHP system controller developed to market readiness by HESCH, an SME partner, is already being implemented successfully in a heating and cooling application in Germany. Further projects are in the acquisition pipeline of the partner.

All other components still need further development work (see chapter 1.3) before introducing them in larger numbers onto the market for HVAC solutions in general and heat pumps in particular.

1.4.1.3 *Other economic impacts of project partners*

The applied research institutions AIT and Fraunhofer ISE developed some tools/methods within the project, which can be either used in further research projects, but also sold in the framework of research services directly to component and heat pump manufacturers. In case of AIT, this includes a methodology for selecting the most energy efficient fin design of heat exchanger candidates prior to manufacturing and an improved test rig for performing testing of large-sized heat pumps. Fraunhofer TAG developed a distributor test bench for the characterisation of new and existing flow distribution technologies. Fraunhofer ISE further improved their Smart Grid and MPC control for local integration of renewables and improved on their HIL testing for system controllers.

Several industrial partners participated the first time in a European co-funded research project. Especially the smaller ones experienced an improved visibility / recognition and the opening up of new market perspectives for the company.

1.4.2 *Environmental impacts*

In the following, the potential environmental impacts of the GreenHP project to be harvest in the medium to long term, after the heat pump unit was demonstrated successfully in relevant operational environment, and implemented in large numbers, are described.

1.4.2.1 *Enhancing energy efficiency*

The GreenHP unit has the potential, to significantly impact on primary energy and CO_2 savings in the long run. Considering a refurbishment rate of 3% and taking into account the performance measurement results of the GreenHP prototype, the primary energy savings add up to 500 TWh per year, while the CO_2 emission savings total 60.33 Mtoe per year for eight relevant European reference countries, France, Italy, Belgium, Germany, UK, Austria, Sweden, and Spain.

This is equivalent to a reduction of primary energy by 48.2% per year and a reduction of CO_2 emissions by 41.31% per year compared to the considered building stock in the current situation.

The table below summarizes the savings to be achieved till 2050 if a refurbishment rate of 3% per year is considered.

[18] The refrigerant charge of state-of-the art heat pumps is between 200 and 500 g / kW heating capacity.
[19] Refrigerant whole sale prices / pro kg refrigerant: R134a = 7,70€, R290=2,44€
[20] Standard compressor needs 3.3 liter oil, with the improved oil sump 1.0 liter is necessary.

Table 7: CO_2 and Primary energy saving potential

	CO_2 savings in Mtoe		Primary energy savings in TWh	
	Worst case 30%	*Best Case 60%*	*Worst case*	*Best Case*
Cumulated savings based on 3% refurbishment p.a. (2018 - 2030)	6,2	11,8	51,4	97,9
Cumulated savings based on 3% refurbishment p.a. (2018 - 2050)	16,0	27,8	132,2	230,2

1.4.2.2 *Rationalizing use and storage of energy*

An increased share of fluctuating renewable energy generation (such as e.g. wind, PV) in the power system has led to an increased need for capacity reserves throughout Europe. Heat pumps such as the GreenHP unit can facilitate the integration of renewable energy sources into the power system by providing flexibility on the demand side. Space heating offers the largest flexibility potential within the average (and cold) climate zone; however this flexibility is available only in winter. In contrast, the flexibility of the domestic hot water is almost constant during the whole year. In general, the annual flexibility of residential buildings is higher than the one from office buildings as they are larger in numbers. However, the flexibility provided by office buildings is more evenly distributed over the year, when taking into account also the cooling demand in summer.

In the case of **Germany**, which is one of the biggest markets for fluctuating power generation, the annual flexibility offered by residential buildings is **5.1 TWh/a,** which is about ten times higher as the one offered by offices. One **third of the flexibility**, which is 1.7 TWh/a, is provided by **domestic hot water (DHW) demand**. 65% of the average daily electricity demand for DHW preparation can be shifted throughout the year. For space heating, this figure is between 20% and 30% in wintertime (Fischer 2014 b).

1.4.3 *Social impact*

1.4.3.1 *Impact on employment*

Within the GreenHP project, approx. 430 person months were worked, which corresponds to 35 persons working full-time on the project for one year. Considering that the project duration was 45 months, 9.33 employees worked on the GreenHP project at an average project year.

1.4.3.2 *Impact on knowledge production and sharing*

- Production of scientific publications

During the project duration **20 scientific publications** were produced and diffused within the scientific community.

They include:

- 3 published papers in peer-reviewed Journals (2 more were submitted, additional 3 are close to submission at the project end).
- 17 published papers in (peer-reviewed) proceedings of Conferences / workshops (3 more were accepted to be published in 2016 after the project end).
- 9 of them are publicly available either on www.researchgate.net or https://journals.rtu.lv/index.php/AHNGT/article/view/rehvaconf.2015.015.

- Research activities induced by the GreenHP project

The GreenHP project initiated seven research projects at the applied research institutions involved. In the near future, further research proposals are planned involving both various GreenHP consortium partners but also new project partners in particular heat pump manufacturers.

In case of the coordinator AIT, three research proposals aiming at improving the noise emissions of air-based heat pumps which are mainly caused by fan, compressor and evaporator, and one project dealing with the removal of other market barriers for heat pumps in multifamily houses were started respectively submitted successfully during the project duration. One important project to mention in this context is the IEA Heat Pump Annex on “Acoustic Signatures of Heat Pumps” developed and proposed by Dr. Reichl as Operating Agent in 2015 with an anticipated start in 2017.[21] One large-sized Austrian heat pump manufacturer shows great interest in participating in a subsequent experimental research project demonstrating several innovative heat pump components developed within the project and the GreenHP concept on unit and system level, most likely in a nationally co-funded research project. Two European heat pump manufacturers with a business location in Austria contacted AIT in regard to undertake research projects to improve especially on the noise emissions of their air-based heat pump systems.

Within Fraunhofer, the GreenHP initiated four research projects during project duration, one internally, two nationally and one internationally co-funded research project with a focus on improved controls, heat pump and component design. In the near future, projects shall be submitted on European level on the topics of smart controls and heat pumps and heat pump pools, and improved control of the GreenHP.

1.4.3.3 *Impact on social capital*

In the course of the GreenHP project, 5 PhD and 8 Master students were involved in conducting research tasks at the two research institutions Fraunhofer ISE and KTH.

[21] http://www.heatpumpcentre.org/en/projects/proposals/Sidor/default.aspx

1.5 Main dissemination activities

The dissemination activities organised within the GreenHP project were mainly geared at the widest possible dissemination of project results in particular among the heat pump sector, the R&D community and EU policy makers / influencers.

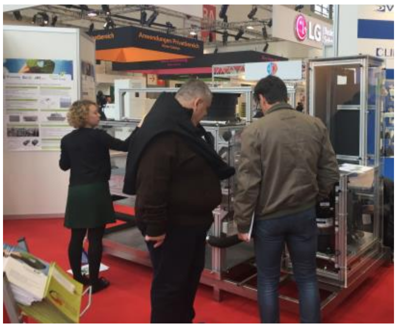
Beside the website (www.greenhp.eu), which went online in project month 3, the project consortium has undertaken, amongst others, the following **main dissemination activities** for the specified target group:

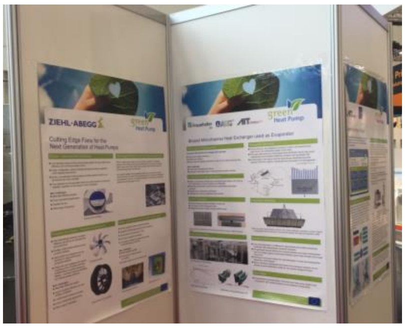
Table 8: Overview of main dissemination activities

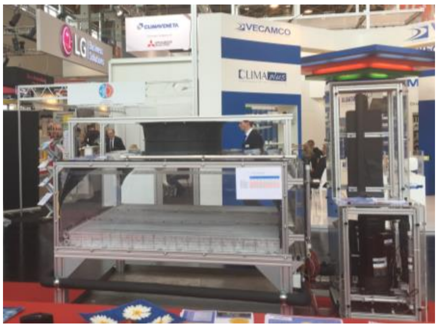
Target group	Dissemination activities
Manufacturers of heat pumps and components for heat pumps	• Organisation of **three technical workshops** for the European Heat Pump Industry (manufacturers of heat pumps and components) in 2013 and 2016 with in total more than 150 participants. • **27 oral presentations at 19 conferences** / workshops attended by relevant industry and R&D such as e.g. 11th IEA Heat Pump Conference Montreal 2014, 11th Forum Wärmepumpe, OTTI Seminar on Air Source Heat Pumps in Retrofitting Applications, 4th International Congress and Exhibition on Aluminium Heat Exchanger Technologies for HVAC&R, etc. • Presentation of the GreenHP project at **5 relevant industry fairs** (ISH 2013/2015, Chillventa 2014/2016 and 40th Mostra Convegno Expocomfort at the EHPA booth • **4 EHPA Newsletter contributions** on the project content and results • **2 Articles in IEA Heat Pump Newsletters**
R&D community	• **3 peer reviewed paper** publications during project duration. • **16 paper** publications in **proceedings of conferences** and workshops including amongst others: 11th IIR Gustav Lorentzen Conference on Natural Refrigerants 2014 (1 paper), 24th International Congress of Refrigeration 2015 (4 papers), FAN 2015 (1 paper), OTTI Seminar on Air Source Heat Pumps in Retrofitted Buildings (2 papers).
Political stakeholders, green NGOs	• Presentation of the GreenHP project at the **European Sustainable Energy Weeks** 2013/2014/2015/2016 via leaflet, roll-up and posters • **Meetings with 5 relevant stakeholders** from relevant organisations such as Covenant of Mayors, Architects of Europe and Energy Cities • Articles in European Energy Innovation and Renewable Energy Projects Catalogue of EUREC • Presentation of the project in the framework of EHPA press cocktails, EHPA heat pump forum 2013/2014/2015 & 2016
End-users	• Exhibition of a GreenHP cube in Brussels Parc • Website

All dissemination activities undertaken are compiled in a comprehensive report available for download at http://www.greenhp.eu/deliverables/public-deliverables/.

Table 9: GreenHP prototype at Chillventa 2016







1.6 Further project data

1.6.1 *Project contact*

Andreas Zottl, Engineer and Project Manager
AIT Austrian Institute of Technology
Tel: +43 50 550 6309, Email: Andreas.zottl@ait.ac.at

1.6.2 *Project partners*

Partner	Contact person	eMail
AIT Austrian Institute of Technology	Andreas Zottl	Andreas.Zottl@ait.ac.at
AKG Group	Andreas Strehlow	andreas.strehlow@akg-gruppe.de
EHPA European Heat Pump Association (BE	Eirini Litina	Eirini.Litina@ehpa.com
Emerson Climate Technologies GmbH	Bella Bachir	Bella.Bachir@emerson.com
Fraunhofer ISE	Thore Oltersdorf	thore.oltersdorf@ise.fraunhofer.de
Gränges SE	Claudi Martin Callizo	claudi.martin-callizo@granges.com
Hesch Industrie-Elektronik GmbH	Werner Brandis	w.brandis@hesch.de
KTH Kungliga Tekniska högskolan	Björn Palm	bjorn.palm@energy.kth.se
Ziehl-Abegg	Frieder Lörcher	Frieder.Loercher@ziehl-abegg.com

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