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GreenHP – Air to Water Heat Pump Prototype for Retrofitting Buildings in Urban Areas

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Abstract

The Green Heat Pump project (www.greenhp.eu) aimed at developing a highly efficient heat pump concept for large renovated existing buildings in urban areas. Innovative approaches were pursued at component, unit and system level. The design of the GreenHP was strongly influenced by developments on the components level. The main factors determining the design were found in the areas of reducing refrigerant, using the natural refrigerant propane and optimizing the evaporator unit on the refrigerant and air side.
Based on a life cycle analysis, R290 emerged as the optimal refrigerant. To achieve high efficiency over the entire operating range and a limited refrigerant charge, the compressor included EVI and an optimized oil sump. Through the use of aluminum multi-port extrusion tubes in the condenser and evaporator the refrigerant charge was further minimized. An optimized design of the fan ensured a more efficient operation at icing conditions. On the refrigerant side, a newly developed bionic refrigerant distributor took care of the distribution of the refrigerant to the MPE tubes of the evaporator.
The newly developed components were assembled into a prototype heat pump and tested at AITs accredited testing facility according to international standards. The studies concentrated on the unit design and analysis of the performance and efficiency of the GreenHP prototype. The GreenHP reached a coefficient of performance (COP) of 3.0 at rating conditions A7/W55 and a seasonal coefficient performance (SCOP) for the average climate for high temperature applications of 3.3.

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1. Introduction

The renovation and retrofitting of urban buildings with modern heating systems that are highly efficient and capable of utilizing renewable energy sources is of utmost importance to achieve the EU 20-20-20 goals and represents an interesting future market segment for heat pumps. In this context, multi-family houses and commercial buildings are of high relevance as these buildings types are dominant in densely populated urban areas. Developing a heat pump system for the retrofitting market in cities has its particular challenges. First, space available for heat sources is often limited in existing urban areas. This asks for a flexible, preferably non-invasive heat source which can easily be integrated into existing building stock. Air is a promising candidate heat source in this respect. Second, the heat pump must be compatible with existing heat distribution systems often requiring higher supply temperatures. In this respect, the use of air as heat source is challenging due to potentially large temperature lifts in winter time. This issue needs to be tackled by means of intelligent integration of the heat pump as part of an overall heating concept. Last but not least, the heat pump system should be attractive in terms of installation and operational cost, should have a clear benefit in terms of primary energy savings compared to alternative solutions and should also have a minimum environmental impact in

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terms of the working fluids used. In addition, very low noise levels are of essential importance for the application in urban areas.
At EU level, air-based heat pump systems dominate the current market trend. The highest sales figures have been achieved in Germany, France, Sweden, UK and Switzerland. Air to water heat pumps (A/W-HP) are thus an established heat pump technology [1]. A/W-HPs are easier to install than ground source heat pumps because they do not require boreholes or space for horizontal collectors. These properties make them attractive for applications in which the use of the heat source ground is difficult or even impossible. This is typically the case for existing buildings, especially in densely populated areas and towns. For these reasons, A/W-HP will be an important integral part of the energy infrastructure in cities in the future.

2. Green Heat Pump Concept and Design

The Green Heat Pump project aims to develop the next generation of heat pumps for the use in large renovated existing buildings in urban areas. The research results will be applicable to systems on a larger scale with heating capacities of >100 kW for multi-family and office buildings, which will play an important role in the infrastructure of smart cities of the future.

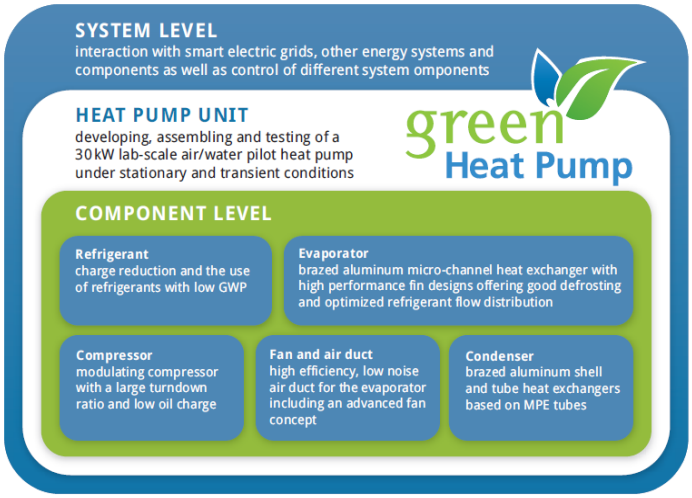


Fig. 1. GreenHP research approach [2]

Innovative approaches are being pursued at the component, unit and system level (Fig. 1) in order to achieve the following objectives [3]:

- Efficiency increase – development of new components and an integrated system-oriented setup of the refrigerant circuit and controls.
- Alternative refrigerants – investigation of low global warming potential (GWP) refrigerants, including hydrocarbons, as a basis for highly efficient heat pump and system design [4].
- Intelligent systems – development of interfaces to smart electric grids, other renewable energy systems and control of different system components [5].

The design of the GreenHP prototype is strongly influenced by the developments on the component level. The main factors influencing the design are to be found in the areas of refrigerant reduction, using the natural refrigerant propane and optimization of the evaporator unit on the refrigerant and on the air side. A detailed presentation of the development and research on components and system level can be found in [6]. 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. The newly developed components were assembled into a prototype with the following specifications:

- Refrigerant propane (R290)
- 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 evaporator
    - cooling capacity of approx. 18.4 kW
    - T_air_in – T_evaporation: 7 K
    - ΔT_air: 5 K

  - Air velocity: 1.3 m/s
- modulated scroll-compressor, with a speed range of 1800 to 7200 rpm
- Optimized design for part load condition A2/W42
  - cooling capacity of approx. 11.0 kW
  - heating capacity of approx. 15.4 kW

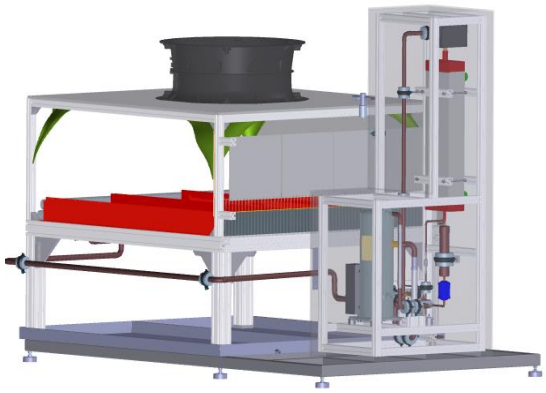




Fig. 2. (left) 3D-sketch of the GreenHP-Prototype [7]
Fig. 3. (right) Assembled GreenHP-Prototype

Fig. 2 and Fig. 3 show the 3D-sketch of the GreenHP-prototype construction drawing, and the assembled GreenHP prototype in front of the test rig.

3. Experimental studies and performance tests

To measure the performance of the prototype under stationary conditions, and to calculate both, the coefficient of performance (COP) and the seasonal coefficient of performance (SCOP), the GreenHP prototype was tested using the methods of EN14528 [8] and EN14511 [9]. The testing conditions and procedure are described in more detail in [10]. As the GreenHP is optimized for high temperature applications in “average” climate conditions, the corresponding testing conditions of EN14825 were selected for testing.
The tests were carried out for a variable outlet of the heat pump with a fixed mass flow. As stated in EN14825, the mass flow on the hydronic side of the heat pump was determined at the operating conditions A7/W55 according EN14511. Additionally to the standard measuring points required for the calculations according to standard, the whole refrigeration cycle and the unit were equipped with temperature and pressure sensors to get a deeper insight of both, the operational behaviour of the unit and of its single components.
Fig. 4 shows the GreenHP unit on the intermediate circuit; Fig. 5 depicts the unit at the AIT test rig. The AIT test rig used for testing the GreenHP unit is characterized as follows:

- air/water heat pumps up to 40kW (thermal capacity)
- Heat source temperature ranging from -25 °C to 45 °C (control precision of ±0,15 K)
- Heat sink temperature ranging from 20 °C to 65 °C (control precision of ±0,15 K)

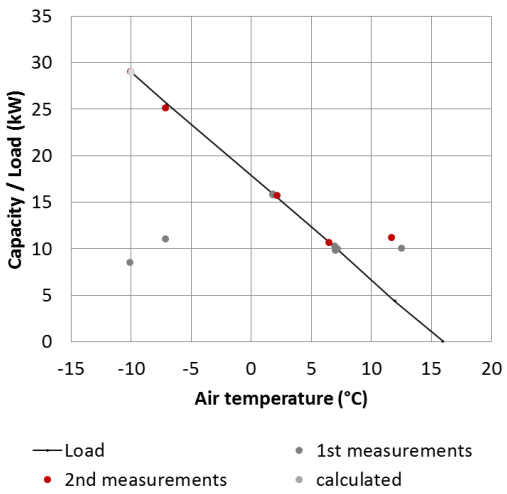




Fig. 4. (left) measurement set up
Fig. 5. (right) GreenHP on the test rig

3.1. Measurements based on EN14825

The start-up of the unit was performed at the rating condition A7/W55 according to EN14511 with a heating capacity of 20 kW in order to set the needed mass flow rate of 0.6 kg/s to reach a delta T of 8 K at the condenser outlet. The refrigerant charge required for maintaining a stable operating behaviour was 1900 g R290, which corresponds with 65 g per kW thermal capacity. This was more than calculated in the simulation, but still much less than conventional systems use (>200 g per kW thermal capacity). 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.
First tests showed that the condensing water remained in the evaporator heat exchanger after defrosting leading to blockage of the air passages (Fig. 17), therefore the measurements based on EN14825 for SCOP calculations were performed under “dry conditions”. This way, icing of the heat exchanger was avoided.
Fig. 6 shows the load curve according EN14825 for the GreenHP prototype selected for the design point A-10/W55 heat load of 29 kW. It can be seen that the heat pump can be operated at part load conditions fitting the load curve down 10.66 kW at the operating conditions A7/W36. At A12/W32 the needed heat load according the EN standard is 4.35 kW. Even when operating the heat pump with the lowest possible compressor speed of 1800 rpm, the heat pump still reaches a heating capacity of 11.15 kW due to the high temperature level of the heat source. Therefore the unit has to be operated in ON/OFF mode to reach the needed heat load.



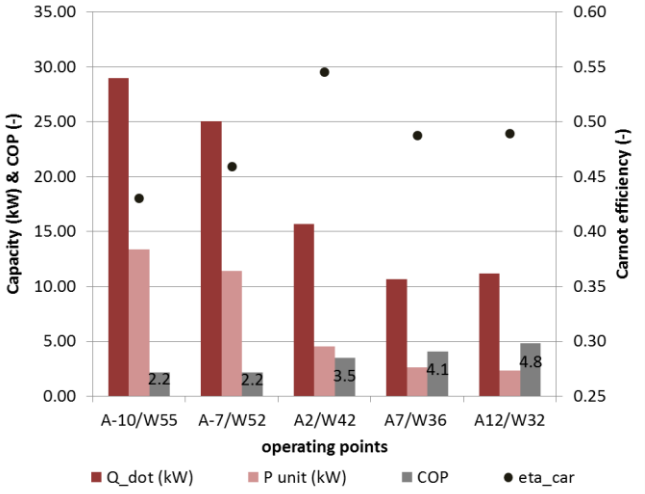


Fig. 6. (left) GreenHP capacity and load according EN14825 at different operating conditions
Fig. 7. (right) Test results at EN14825 test conditions

In order to compare the efficiency of the prototype at different temperature levels, the carnot efficiency (ε_{Carnot}) was calculated during the measurements for the different operating points according to Equation 1 and Equation 2.

$$COP_{Carnot} = \frac{T_1}{T_1 - T_2} \qquad \text{Equation 1}$$

$$\varepsilon_{Carnot} = \frac{COP_{real}}{COP_{Carnot}} \qquad \text{Equation 2}$$

T_1... condensing temperature during heat pump operation in Kelvin
T_2... evaporating temperature during heat pump operation in Kelvin

The maximum ε_{Carnot} was reached at an air temperature of +2 °C, which correlates to the design point of the compressor to reach the maximum efficiency at A2/W42. This behaviour is confirmed by **Fout! Verwijzingsbron niet gevonden.**, where the results needed for the SCOP calculation according to EN14825 are summarized.
Table 1 shows the input data for the SCOP calculation. In the reference condition the calculation methodology is set by choosing the application temperature, the water flow, the type of heat pump, the capacity control, the design capacity and if needed the bivalent point temperature. For the performance data, the measurement results capacity, water outlet temperature and COP are used.

Table 1. Overview of input data for the SCOP calculation according EN14825

Reference conditions

Climate	average		Temperature application		55°C	Water flow		fixed	
Type of heat pump	air-to-water		Capacity control		variable	Water outlet temperature		variable	
Tdesignh	-10	°C	Tbivalent	-10	°C	Qhe	59914	kWh	Manufacturer Data
Pdesignh	29	kW	TOL	-10.1	°C	Hhe	2066	hours	Test Data

Performance data

Condition	Outdoor air T°C	Part load ratio %	Part load (kW)	Water outlet temperature (testing)	Declared Capacity (kW)	COP at DC	C_{dh}	CR	COP at PL
A	-7	88	25.65	52	25.07	2.19	0.9	1.00	**2.19**
B	2	54	15.62	42	15.71	3.48	0.9	0.99	**3.48**
C	7	35	10.04	36	10.66	4.09	0.9	0.94	**4.06**
D	12	15	4.46	35	11.15	4.80	0.9	0.40	**4.17**
F (Tbiv)	-10	100	29.00	55	29.00	1.82	0.9	1.00	**1.82**
E (TOL)	-10.1	100	29.11	55	29.00	1.82	0.9	1.00	**1.82**

For calculating the SCOP, the energy demand of the GreenHP unit in the different operating modes including

thermostat off, off mode and crankcase heater operation was calculated (Table 2). Based on this input data, the calculated SCOP according EN14825 for the GreenHP prototype is 3.3. In comparison, state of the art A/W-HP units in a capacity range of 10 to 20 kW heating capacity reach SCOPs between 2.9 and 3.4 for the "average climate" for high temperature applications.

Table 2. Energy demand for thermostat off, off mode and crankcase heater

	Hours	Power input (W)	P * h (kWh)
Thermostat off	178	50	8.90
Stand by	0	50	0.00
Off mode	3672	50	183.60
Cranckase heater	3850	50	192.50

3.2. Verification of the Unit Design

Besides the performance tests using the method of EN14825 and EN14511, the prototype operating behaviour, and the unit and component design were further investigated during the experimental studies. At first, the right amount of refrigerant charge was verified by calculating the subcooling of the refrigerant after the condenser. Fig. 8 shows the subcooling, the evaporating-, condensing-, and discharge temperature at different operating conditions. At the operating conditions A2/W42, A7/W36 and A12/W32 the subcooling resulted in approximately 3 K. Only at the operating conditions A-7/W52 the subcooling is higher, which correlates to the findings in [11], that at low air temperature conditions the highest amount of refrigerant can be found in the refrigerant receiver. As the receiver of the prototype has been designed very small (0.6 litre) in order to reduce the overall refrigerant charge, a part of the condenser is filled with liquid refrigerant and therefore the subcooling rises at this certain operating point.

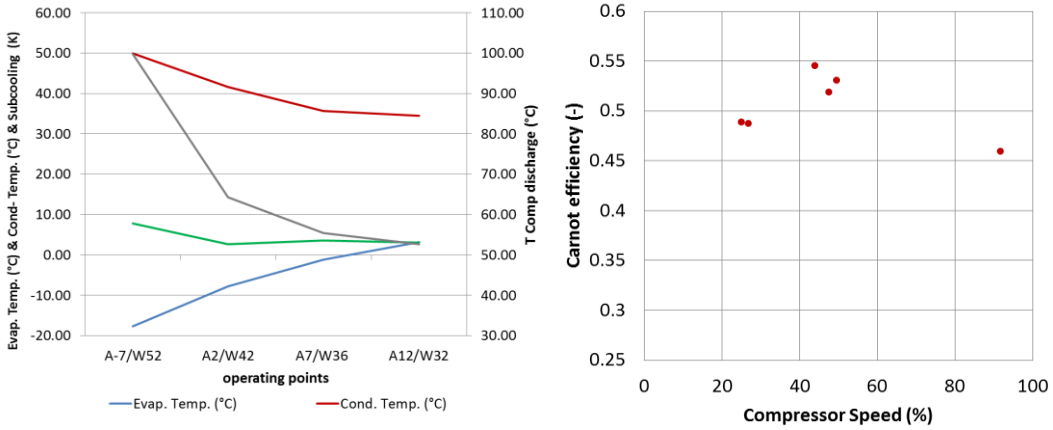
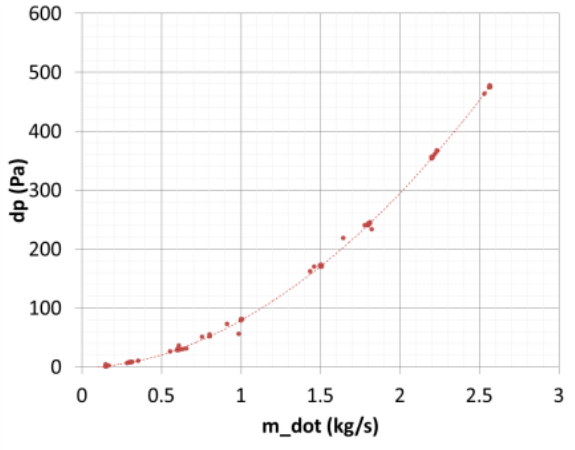


Fig. 8. (left) GreenHP refrigeration cycle behaviour at EN14825 test conditions
Fig. 9. (right) Carnot efficiency depending on compressor speed

To achieve a high SCOP a high efficiency of the heat pump at part load conditions is required. The GreenHP unit reached very good part load behaviour with the highest efficiency reached at 48 % of the compressor speed (Fig. 9). This confirms the optimized compressor design for part load conditions in the operating point A2/W42 at 15.4 kW heating capacity.

As the pressure drop on the water side of the condenser is very important for the implementation of the GreenHP unit into a heating system, Fig. 10 shows the pressure drop depending on the water mass flow. At the set mass flow of 0.6 kg/s a pressure drop of 38 mbar was reached. Additionally the condenser is well designed concerning heat transfer as the temperature difference at the pinch point was less than 1K.



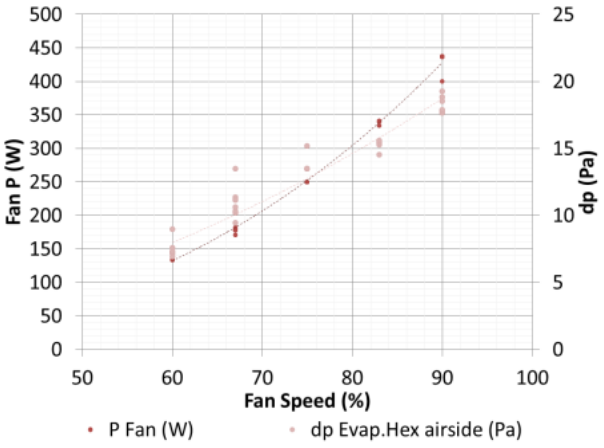


Fig. 10. (left) Pressure drop of the condenser on the water side
Fig. 11. (right) Fan power and evaporator pressure drop depending on the fan speed

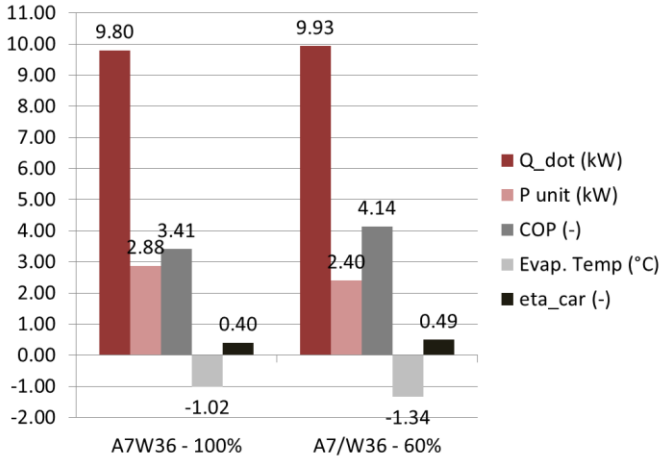


Fig. 12. COP depending on fan speed at A7/W36

The operation of the prototype at different fan speeds showed a large impact on the calculated COP, as the electric fan power is rising exponentially with fan step, as shown in Fig. 11. Increasing the fan speed from 60 % to 90 %, leads to a rise in power input from 130 W to 440 W. Fig. 12 depicts the effects on the units' efficiency when changing fan speed from 100% to 60% at the operating condition A7/W36. The lower fan speed and therefore reduced air volume flow through the evaporator, decreases the evaporating temperature from -1.02 °C to -1.34 °C, whereas the electric power input to the heat pump is reduced from 2.88 kW to 2.40 kW. Consequently, the COP is improved from 3.41 to 4.14, which is an efficiency improvement of 20% triggered only by a change of the fan speed. Therefore setting the right fan speed is of high importance to reach the highest possible efficiency.

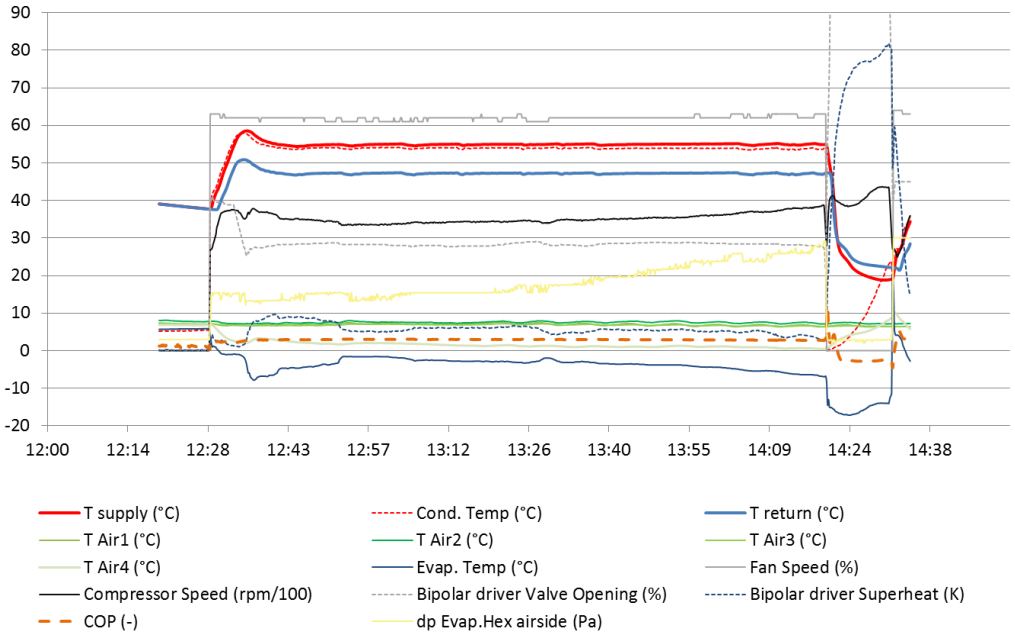


Fig. 13. Heating and defrost operation at A7/W55

Fig. 13 shows a complete heating and defrosting cycle at the operating condition A7/W55 with 20 kW heating capacity, which corresponds to a compressor speed of 48 %. The whole cycle from starting the unit until the end of the defrost cycle lasted approximately 2.5 h. During the measurement of the operating point A7/W35, the water in- and outlet temperature, the air temperature and the heating capacity of the prototype were kept constant, which is to be seen in the diagram between 12:40 h and 14:20 h.

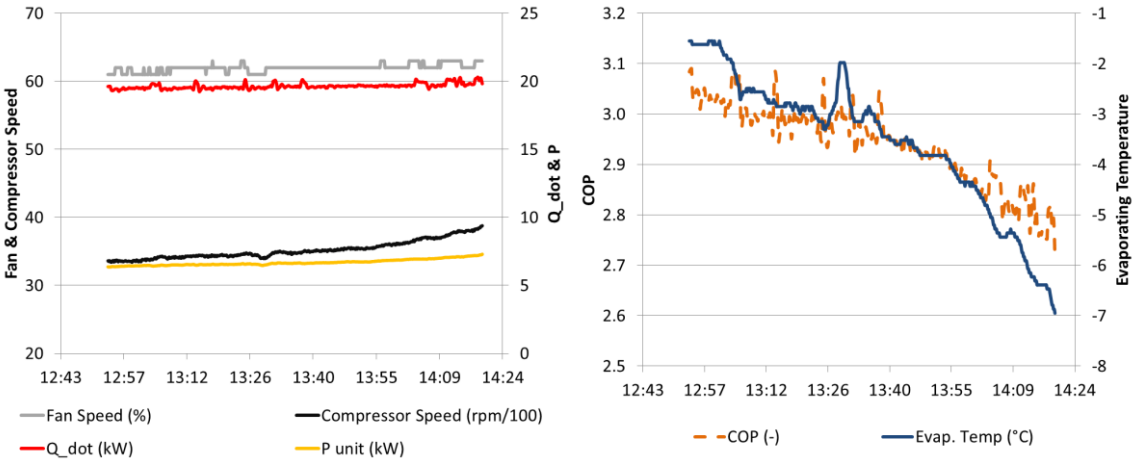


Fig. 14. (left) Heating capacity and electric power at A7/W55
Fig. 15. (right) COP and evaporating temperature at A7/W55

Due to icing of the evaporator during the heating operation, the evaporating temperature dropped from -1.6 °C to -7 °C (Fig. 13 and Fig. 15), whereas the pressure drop of the evaporator on the air side of the heat exchanger rose from 15 Pa to 30 Pa, which is comparable to state of the art heat exchangers.

In Fig. 14 it can be seen clearly that for maintaining the heating capacity of 20 kW, during heating mode under icing conditions of the evaporator, the compressor speed increased from 3300 rpm to 3800 rpm and thus the electrical power of the compressor rose from 6.4 kW to 7.2 kW. Additionally the COP decreased from 3.1 to 2.7 due to the decreasing evaporating temperature (Fig. 15). The temperature difference between in- and outlet at the air side of the evaporator ranged between 5 and 6.5 K, depending on the icing of the heat exchanger. During the heating operation approx. 6 litre condensing water remained in the heat exchanger.

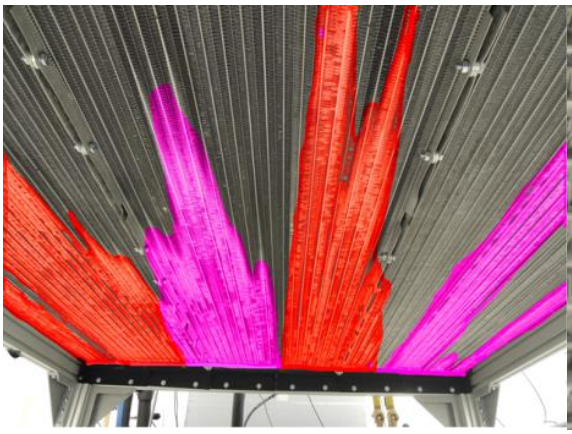
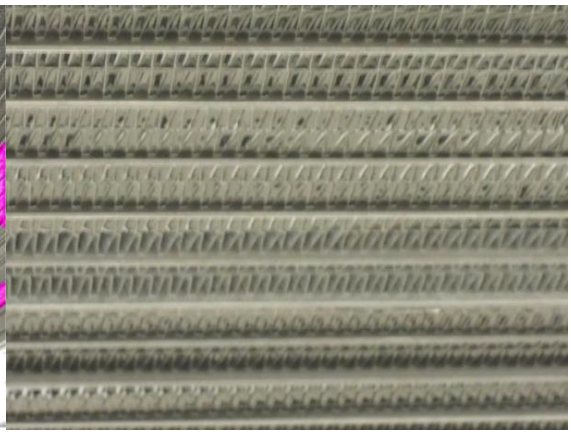



Fig. 16. (left) Ice formation on the airside inlet of the evaporator
Fig. 17. (right) condensing water in the heat exchanger

When observing the ice formation during heating operation (coloured red/pink in Fig. 16), it was evident that the icing on the evaporator did not occur uniform. Hence, the refrigerant distribution in the evaporator proved uneven.
The four different segments (shown in red/pink/red/pink in Fig. 16) indicate the four 16 port bionic distributors. It looks as if the inner ports of the distributors show higher refrigerant mass flow rates. It has been estimated that only 70 % of the heat exchanger surface is fully activated, and therefore the evaporating temperature is lower than expected. After defrosting, the condensing water stayed in the lamellas (Fig. 17). Even a reverse operation of the fan started to blow out the condensing water did not remove the condensing water completely from the heat exchanger. This part of the heat pump still needs improvements to harvest possible efficiency gains.

4. Summary and Conclusions

The GreenHP project aimed at developing a highly efficient heat pump concept for large renovated existing buildings in urban areas. Innovative approaches were pursued at component, unit and system level. The design of the Green Heat Pump was strongly influenced by developments on the components level. The main factors determining the unit design include the targeted refrigerant reduction, using propane as natural refrigerant and the optimization work on the evaporator unit on both, refrigerant and air side.
Based on a life cycle analysis, R290 emerged as the optimal refrigerant. With an optimum design of the refrigeration cycle and optimized components, the refrigerant charge for the 30 kW GreenHP prototype was reduced to less than 2 kg propane. To achieve high efficiency over the entire operating range and a limited refrigerant charge, the compressor included Enhanced Vapour Injection and an optimized oil sump. Through the use of aluminum multi-port extrusion tubes in the condenser and evaporator the refrigerant charge was further minimized. On the air side of the evaporator highly efficient fins less sensitive to icing were developed. An optimized design of the fan ensured more efficient operation at icing conditions. On the refrigerant side, a bionic refrigerant distributor was designed in order to distribute the refrigerant to the MPE tubes of the evaporator.
All newly developed components were integrated successfully and the overall operation of the GreenHP prototype showed good performance results. The GreenHP prototype performed very well in terms of its seasonal efficiency (SCOP). The calculated SCOP according EN14825 for “average climate” and high temperature applications resulted in a SCOP of 3.3 which is 0.2 higher than the SCOP aimed for at the beginning of the project and comparable with the best performing heat pump units in the capacity range of 10 to 20 kW heating capacity. The higher overall efficiency results from the optimized unit design for part load conditions, where 80 % of the heating demand in the average climate occurs.
The unit showed very good part load behaviour. From the test results it was evident, that the GreenHP 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 refrigerant charge required for maintaining a stable operating behaviour was 1900 g R290, which corresponds with 65 g per kW thermal capacity. This was more than calculated in the simulation, but still much less than conventional systems use (>200 g per kW thermal capacity).
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 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 evaporator 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 distribution was not possible. 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. This part of the heat pump still needs improvements to harvest possible efficiency gains.
Summing up, the GreenHP prototype showed very promising results on both, unit and component level to build up from in the upcoming research projects.

Acknowledgements

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