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HP-LAUNCH

Heat Pump Design

To:
HP-Launch consortium

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1 INTRODUCTION

1.1 Document revision history

Report release date	*Author*	*Report name and version*	*Remark / document change*
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1.2 General

The objective of the HP-Launch project is to develop a novel air to water heat pump that will accelerate the introduction of air source heat pumps to the Dutch market. Work package 2 (WP2) of the HP-Launch project focusses on the design of the heat pump.

In WP1 it was concluded that a compact monoblock type heat pump with a placement "through the roof" and near the existing boiler would be a good candidate for a large portion of the Dutch housing market.

The initial design requirements of the heat pump were defined in WP1, and are:

- Nominal heating capacity of 3 to 3.5 kW at A7/W35 (NEN-EN14511),
- Energy efficiency comparable to best in class appliances on the market,
- Suitable for hybrid as well as full electric heating operation,
- Total system weight as low as possible, preferably below 20 kg,
- Use a natural refrigerant and apply a small charge,
- When using a hydrocarbon as the refrigerant, comply with the safety design limit of 150 grams for indoor units (EN-378, IEC 60335-4-2:2018),
- Use commercially available mass-produced components.

The heat pump will be installed partially indoors and will also be serviceable from within the house. For indoor systems a charge limitation of 150 g of flammable refrigerant applies. Due to this and the general desire to have a compact low refrigerant charge system, the development of the heat pump design focussed on both compactness and charge limitation.

This report presents the design of the HP-launch heat pump prototype and discusses the design decisions made. Next to this also a summary of the main results of the practical evaluations of the prototype are presented. For more detailed measurement data, reference is made to Re/genT report 20202/RO04/V1.

1.3 Content

Chapter 2 presents the selection of the natural refrigerant. This is followed by the selection of the heat pump configuration in Chapter 3. In Chapter 4 the heat pump design is presented, and the components used are discussed. Hereafter, in Chapter 5 the results of charge evaluations are given, and the final selection of the heat exchanger sizes is made. In Chapter 6 a summary of measurement results of the prototype is given, and the main observations made during the practical evaluation are discussed in Chapter 7. This is followed by Chapter 8 were an estimation of system performance for using the alternative heat exchanger sizes is presented. Finally, in Chapter 9 the main conclusions and recommendations are given.

2 REFRIGERANT SELECTION

Initially the following natural refrigerants were considered:

- R-717 (Ammonia, NH_3), cycle COP[1] = 5.18
- R-600a (Isobutane C_4H_{10}), cycle COP = 5.25
- R-290 (Propane, C_3H_8), cycle COP = 5.09
- R-290 / R-600a Mixture, cycle COP[2] = 5.38
- R-744 (Carbon dioxide, CO_2), cycle COP[3] =3.69

R-717 has relatively high cycle COP and moderate operating pressure and vapour density. The main disadvantage of R-717 is that it has a safety classification B2 (toxic and moderate flammability). Being toxic makes it unsuitable to be applied inside a domestic property. Next to this, for R-717 no standard, low cost, compressors are available for a heating capacity of 3.5 kW.

R-600a is the natural working fluid resulting in the highest calculated cycle COP. is a refrigerant with a safety classification A3 (nontoxic, highly flammable), which requires charge limitation and / or additional safety measures for indoor applications. For a refrigerant charge below 150 g, the safety measures are quite straightforward (i.e. like a domestic refrigerator). The disadvantages of using R-600a are the low operating pressures and low density of the vapour. For a heating capacity of 3.5 kW, using R-600a requires a compressor with a relatively large displacement in combination with refrigerant circuits with large cross-sectional area to avoid unacceptable pressure drop. Currently R-600a is not being applied in domestic heat pumps and no compressors of the required capacity are available on the market.

R-290 has also safety classification A3 (nontoxic, highly flammable). Therefore, regarding safety the same measures as for using R-600a apply. R-290 has a slightly lower (3%) calculated cycle COP than R-600a. Compared to R-600a, R-290 has a much higher operating pressure and vapour density. Due to this the required compressor displacement will be much smaller and the pressure drop will be acceptable when standard refrigerant tubing and heat exchangers are applied. Currently R-290 is being used in domestic heat pumps and refrigeration components, including compressors, are available.

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[1] For R-717, R-600a, and R-290 the cycle COP is calculated based on a theoretical heat pump cycle with a condensing temperature of 36 °C, a subcooling of 2 K, an evaporating temperature of 2 °C, a superheating of 4 K, a suction gas temperature of 10 °C, and an isentropic efficiency of the compressor of 0.6 and assuming isenthalpic expansion of the subcooled liquid refrigerant. The refrigerant properties are calculated using Refprop 10.0.

[2] For the R-290 / R-600a mixture the cycle calculation is based on optimal utilisation of the temperature glide within the heat exchangers. For the mixtures the cycle calculation is started based on the average phase change temperature, the other parameters are identical to the pure fluids. Hereafter, the saturation pressure is corrected towards equal heat exchange within the condenser and evaporator as obtained with the pure fluids. This correction is based on assuming equal heat exchanger performance and thereby fitting towards equal logarithmic temperature difference. The heat exchanger performance is based on a water inlet temperature of 30 °C, a water outlet temperature of 35 °C, an air inlet temperature of 7 °C and an air outlet temperature of 2.5 °C without superheating or subcooling in the heat exchangers.

[3] For R-744 the approach temperature is set to 2 K assuming a water inlet temperature of 30 °C. The discharge pressure is optimized towards maximum cycle COP, resulting in a discharge pressure of 79 bar. The other parameters were kept equal to the subcritical cycles (i.e. R-600a, R-717 and R-290).

R-290 / R-600a mixture. Mixing R-290 and R-600a results in a refrigerant with a relatively high cycle COP. The highest calculated COP results for a mixture existing out of 15% Propane and 85% R-600a (mass percentage). The main issue with such a mixture is that the phase change shows a temperature glide (e.g. 4.4 K during condensing and of 3.1 K during evaporating for a mixture of 85% R-600a and 15% propane). The temperature glide can be used effectively in the condenser, i.e. reducing the entropic losses by creating a more uniform temperature difference between the refrigerant and the water. For the evaporator, however, a quite special design is required to benefit from the glide. Next to this, due to the temperature glide frosting of the evaporator will be much more severe and local, which significantly increases the defrosting events. A final issue with a such a mixture is the uncertainty of having the correct composition throughout the refrigeration system. Especially at low refrigerant charges, the solubility of the oil and the two-phase sections within the heat exchangers have a large impact on refrigerant composition. Capacity wise R-290 compressors can be used for a mixture with relatively high content of R-290.

R-744 has a critical temperature of 31 °C. Due to this a transcritical operating cycle results for using R-744 in an air to water heat pump. Transcritical operation is characterised by relatively low system COP. Another disadvantage of R-744 in a transcritical cycle is the high operating pressure, up to above 100 bar when warm water temperatures are required. To cope with such high operating pressures, relatively thick walls result. Especially for the compressor, this results in a large increase in component mass and therefore costs. The advantage of a transcritical R-744 cycle is that a large quantity of heat is available at high temperature, which makes such a system quite suitable for hot water production. The commercial availability of suitable compressors is limited.

For the project R-290 is selected as the refrigerant. This decision is based on the good cycle performance of R-290 in combination with the commercial availability of refrigeration components. The main downside of using R-290 is its high flammability. Within the project this is covered by:

A) Designing the heat pump for small refrigerant charge (< 150 grams),
B) Integration of the heat pump inside a shell within the “through the roof design” This shell is open towards the ambient air by means of ventilation and is hermetically sealed towards the inside of the building. Therefore, when a leakage occurs, the refrigerant cannot enter the occupied area and will be dispersed into the ambient air. Note: In principle the heat pump is an outdoor unit.

3 SYSTEM CONFIGURTAION

Within the project three design concepts were evaluated, namely:
- split unit (split in the refrigerant system),
- split unit using a secondary fluid between the evaporator and an outdoor heat exchanger,
- monobloc.

Split-unit

The basic idea of a split unit is to have a small indoor and a larger outdoor unit in which only the connection tubing is passing through the outer shell of the house. Within the consortium it is decided that splitting the refrigeration system is not desired as it increases the complexity of installation, introduces additional risks regarding refrigerant leakage and increases the refrigerant charge.

Split-unit using secondary fluid

The idea of this concept is to create a very compact heat pump with low refrigerant charge existing out of two fluid to refrigerant heat exchangers (i.e. two brazed plate heat exchangers). Where the condenser is directly connected to the water loop of the central heating system and the evaporator is absorbing heat from a secondary fluid. This secondary fluid is then absorbing the heat from the ambient via an outdoor air to liquid heat exchanger, while the main heat pump can be installed indoors. The main advantage of the system was seen in the connection of liquid (e.g. water/glycol mixture) instead of refrigerant lines and in the reduced refrigerant charge compared to the standard split unit. Disadvantages are the increased temperature lift on the evaporator side due the additional heat transfer loop, the additional costs and energy consumption of peripherals (e.g. pump for secondary fluid), and issues with defrosting. After validating the pro’s and con’s this concept was rejected within the consortium.

Monobloc

The main advantage of a monobloc is it simplicity in installation, i.e. only connecting the water and electricity connections, and the advantage of having a single unit. The main disadvantage of the monobloc was initially seen in the placement of the unit. Monobloc heat pumps are typically a rectangular box that is placed in the garden, on a flat roof, or against a wall using brackets. Which are all good solutions when space is available. For this project, which focusses on the typical Dutch houses “single family terraced housing” this is not the case. Next to this, for a hybrid solution, ideally the heat pump is installed close to the current gas boiler, which in the Netherlands is typically installed on the attic.

Within the project, it was decided to develop a “through the roof” solution. This finally resulted in a design that is integrated in the roof and can be serviced from the inside of the house. For this design, the distance between the air side and water side heat exchanger is small and a large volume is available to install the complete heat pump unit. Based on this and the disadvantages of both split unit options the monobloc solution is selected.

4 HP-LAUNCH HEAT PUMP

Figure 1 presents a schematic of the HP-Launch heat pump. Pictures of the final prototype are shown in Figure 2. The various components and their selection are discussed in the following subsections.

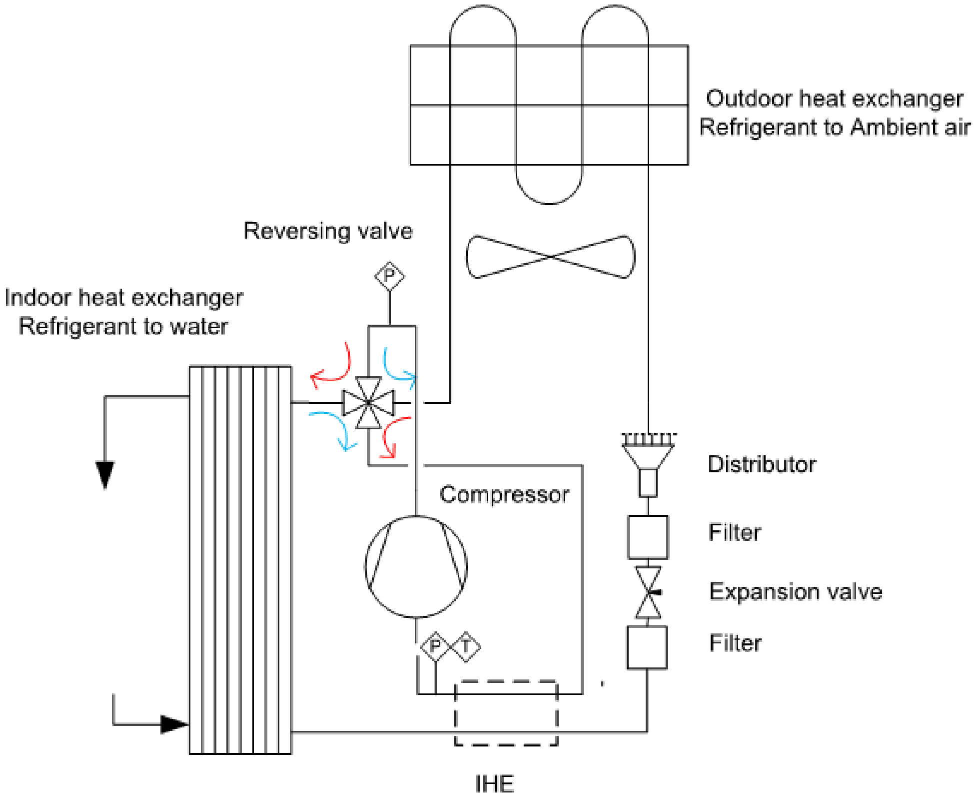

Figure 1: Schematic of the HP-Launch prototype

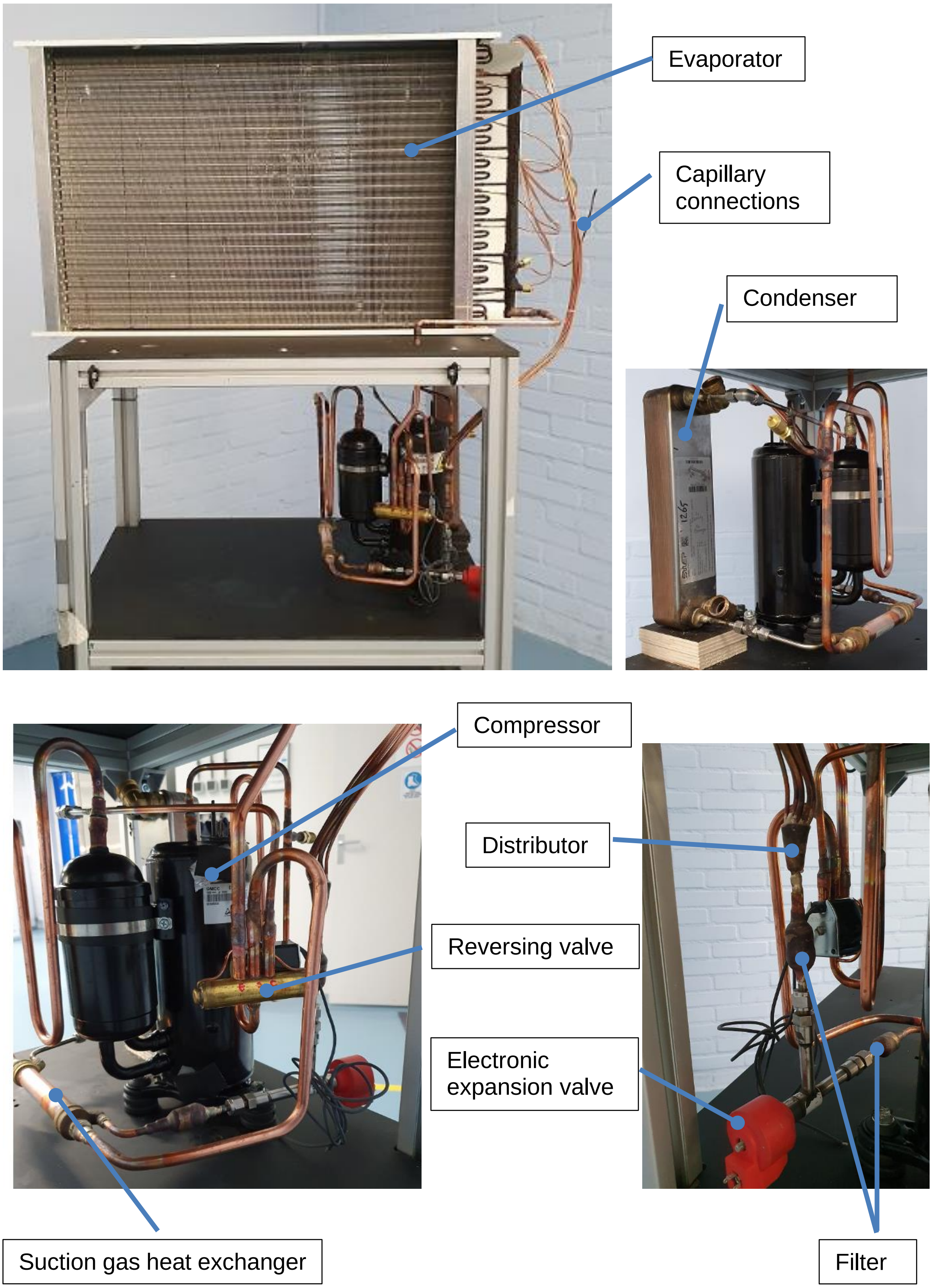

Figure 2: Pictures of the prototype before applying insulation. "During the testing, the compressor shell (18 mm Armaflex), condenser (18 mm Armaflex) and refrigerant tubing (9 mm Armaflex) was insulated"

4.1 Compressor

After searching for suitable R-290 compressors, based on capacity, performance and dimensions the options presented in Table 1 were selected. It must be noted that, although increasing, the availability of R-290 compressors models is limited.

Model	Type	Displacement	Speed [rpm]	Nominal heating capacity[4] [W]	Isentropic efficiency[5] [-]	Volumetric efficiency[5] [-]	Weight [kg]
Tecumseh RKA5518UFZ	Rotary	24.4 cm^3	~3000	4.0	0.59	0.82	14
Sanhua Aweco QXD-C223 A030A	Rotary	22.3 cm^3	~3000	3.8	0.67	0.88	14.6
Sanhua Aweco QXD-C184 A030A	Rotary	18.4 cm^3	~3000	3.1	0.69	0.86	13.8
GMCC DSM180D19UDZ	Rotary	17.9 cm^3	600 to 7200	3.7	0.67	0.90	9.7
GMCC DTN180D32UFZ	Rotary (Twin cylinder)	18.1 cm^3	480 to 7200	3.8	0.68	0.90	8.5
Copeland ZH04KCU-PFZN	Scroll	5.8 m^3h^{-1}	~3000	5.5	0.63	-	23

Table 1: Selected compressor options, data taken from manufacturer data

Table 1 shows that most of the options are rotary compressors. The table also shows that based on manufacturer data high isentropic efficiencies result (i.e. almost up to 0.7). One critical aspect of a rotary compressor is the noise and vibration level.

Based on performance, low weight and controllability the GMCC compressors were selected for the HP-Launch design. Note that these compressors have a relatively large speed range and the design capacity of 3.5 kW at A7/W35 (NEN-EN14511) is reached at approximately 3600 rpm. Initial validations were performed using the DSM180D19UDZ compressor. In the prototype, due to the smaller shell vibration of the twin cylinder design, the DTN180D32UFZ compressor is applied. Next to this the twin cylinder design also required a slightly lower refrigerant charge, see Section 5.1.

4.2 Inverter

For the HP-Launch prototype a commercially available general-purpose inverter was selected, namely the Altivar 320 (1.5 kW) from Schneider Electric (model: ATV320U15M2C)

Note: The use of a general-purpose inverter was recommended by the supplier of the compressors (AREACOOL) as no specific lower-cost manufacturer recommended inverter was available. Such a general-purpose inverter is a relatively expensive component. Component costs can be reduced by using a more dedicated inverter. The selection or development of such an inverter is outside the scope of this project.

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[4] Based on an evaporating temperature of 2 °C, a condensing temperature of 36 °C, a refrigerant subcooling of 2 K and a superheating of 10 K and the compressor efficiencies and swept volume presented in Table 1.
[5] Calculated from the manufacturer data at the reference condition specified by the manufacturer

4.3 Condenser

Refrigerant charge being a critical aspect in the design, the selection is based on using the condenser with the smallest internal refrigerant volume for acceptable performance. In academic literature, various very low charge micro channel configurations are discussed. However, none of these microchannel designs is commercially available. Therefore, it is decided to apply the typical heat pump solution of using a brazed plate type heat exchanger as the condenser.

After a detailed search for plate heat exchangers with small refrigerant volume, the SWEP B8LAS heat exchanger design is selected. This heat exchanger is characterised by its asymmetric channel design resulting in a smaller volume on the refrigerant side compared to the water side.

Using the selection software of SWEP (SSPG8) Performance estimations are made, for various numbers of plates used, based on a heat rejection of 4 kW, a refrigerant inlet temperature of 70 °C, a subcooling temperature of 3 K and a water inlet and outlet temperature of respectively, 30 and 35 °C, see Table 2.

Model	Number of plates [#]	Refrigerant volume [dm^3]	HEX Mass [kg]	Heating capacity [kW]	Refrigerant inlet temperature [°C]	Water inlet temperature [°C]	Water outlet temperature [°C]	Subcooling [K]	Condensing temperature [°C]	ΔT refrigerant (pressure losses)[K]	Net pumping power [W]	Estimated pumping power (pump efficiency 25%) [W]
B8LAS	42	0.408	3.32	4	70	30	35	3	34.6	0.26	1.4	5.6
B8LAS	40	0.388	3.20	4	70	30	35	3	34.6	0.28	1.5	6.1
B8LAS	35	0.347	2.96	4	70	30	35	3	34.8	0.33	1.8	7.4
B8LAS	30	0.286	2.61	4	70	30	35	3	35.1	0.46	2.5	10.1
B8LAS	28	0.265	2.49	4	70	30	35	3	35.2	0.52	2.9	11.4
B8LAS	22	0.204	2.14	4	70	30	35	3	35.8	0.79	4.4	17.5
B8LAS	16	0.143	1.78	4	70	30	35	3	37.0	1.38	7.8	31.2

Table 2: Condenser performance estimation: SWEP B8LAS with various numbers of plates used

Table 2 shows that high thermal efficiency results when more than 30 plates are applied (water outlet above the condensing temperature). The table also shows that reducing the number of plates from 42 to 28 results in an reduction in internal refrigerant volume from 0.408 to 0.265 dm^3, at the cost of 0.6 K increase in condensing temperature, and a 0.26 K increase in refrigerant flow losses due to the pressure drop, and an almost 6 W increase in the expected pumping power (water). Further reduction in plate number results in large required pumping powers.

For the project a 40 and 30 plates B8LAS condenser[6] were bought and evaluated. In the prototype the 30 plates version is applied to minimise refrigerant quantity. The selection of the condenser is further discussed in chapter 5.

[6] After consulting the supplier, it became clear that B8LAS condensers with 30 and 40 plates are on stock items and that using 42 or 28 plates required additional delivery time and costs.

4.4 Evaporator and fan

Within the HP-Launch project the design of the evaporator and the selection of the fan have their own work package, see WP3. Those activities resulted in an evaporator with 10 parallel refrigerant circuits existing out of 5 mm refrigerant tubing and a fin pitch of 1.8 mm and the selection of two 350 mm fans of Papst (model: A3G350-AN01-11). Within the project two evaporator configurations were considered, namely a configuration with two and a configuration with three tubes in airflow direction. The actual built of a prototype heat exchanger was outside the scope of the project. Alternatively, due to its similarity in tube diameter, fin pitch and estimated heat transfer, two mini channel LU-VE LMC3N condensers were bought. Namely, one sample with 2-tubes in airflow direction and a second sample with 3-tubes in airflow direction. The 3-tube design will result in largest system COP, albeit at a larger required refrigerant charge. For further details about the evaporator and fan selection, reference is made to work package 3 of the HP-Launch project.

To minimise refrigerant quantity, the 2-tube version of the LU-VE LMC condenser is applied as the evaporator in the prototype. Next to this, due to the design of the LU-VE LMC3N, only a single fan was fitted (Papst: A3G350-AN01-11). The charge-based selection of the evaporator is further discussed in Chapter 5.

4.5 Refrigerant distributor

No distributor fulfilling the design requirements was found. Due to this a specific distributor was designed and manufactured, see Figure 3 and Appendix A. To minimise system refrigerant content the distributor was fitted as close as possible to the expansion valve.

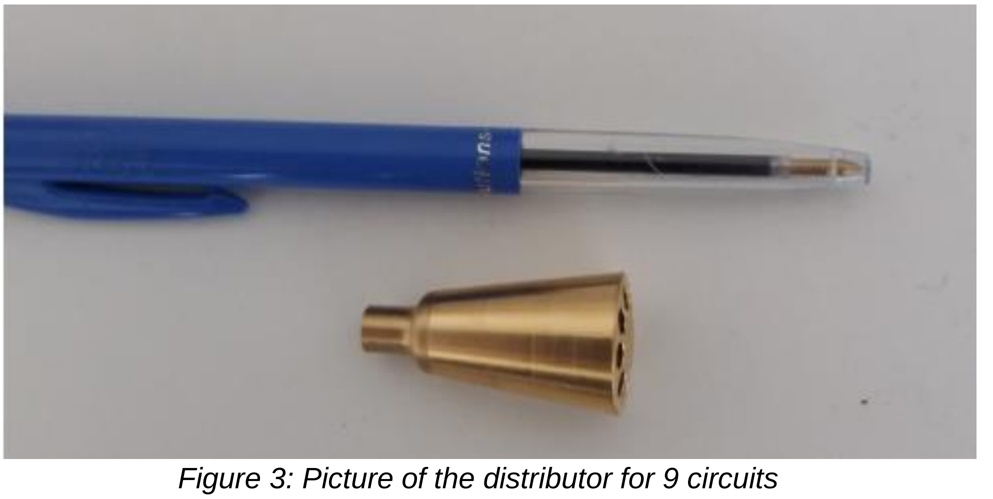

Figure 3: Picture of the distributor for 9 circuits

4.6 Capillary connections

Standard capillary tubing with a length of 1.2 m and an outer diameter of 3.0 mm and inner diameter of 1.8 mm (Refco TC70) was used to connect the distributor to the evaporator.

4.7 Tubing

To dampen tube vibrations and to avoid the risk of rupture, the discharge and suction tubing is fitted with additional flexibility (i.e. additional length, including a turn, in vertical orientation to the compressor), see *Figure 4*. To minimise the required system refrigerant charge, the other connections are made as short as possible. Especially for the liquid line this of importance. For the system the following tubing is applied:

	Material	Outer diameter	Wall thickness	Total tube length
Discharge line	Copper	6.35 mm (1/4”)	0.8 mm	1.25 m
Liquid line	Copper	6.35 mm (1/4”)	0.8 mm	0.23 m
Suction line	Copper	9.52 mm (3/8”)	0.8 mm	2.20 m

Figure 4: Additional length for flexibility in the tubing. Left: Discharge tubing. Right: Suction tubing.

4.8 Expansion device

Within the project two options were investigated: namely, first the standard and proven solution of using an electronic expansion valve and second the use of a capillary tube to further reduce refrigerant quantity.

A prototype heat pump system fitted with both a capillary tube and electronic expansion valve was built and evaluated with the test rig described in Re/genT report 20104/RO01/V1. This prototype used the GMCC DSM180D19UDZ compressor, the B8LAS condenser[7] and an off the shelf evaporator available within the Re/genT laboratories. In the prototype, the expansion device is selected using a manual 3-way valve. See Figure 5, were a schematic of the set-up is given and Figure 6 were a picture of the system is shown.

[7] Charge measurements were performed with both the 30 and 40 plates condenser.

Note: This prototype is also used for the detailed charge measurements of the complete system and at component level presented in Chapter 5. For this, the prototype is fitted with fast closing valves (V1 to V5) which after closing divide the system in multiple control volumes (CV1 to CV5) for which the refrigerant quantity is measured by PVT-superheat[8] method within a low measuring vessel (CV0). The results of these measurements are discussed in Chapter 5.

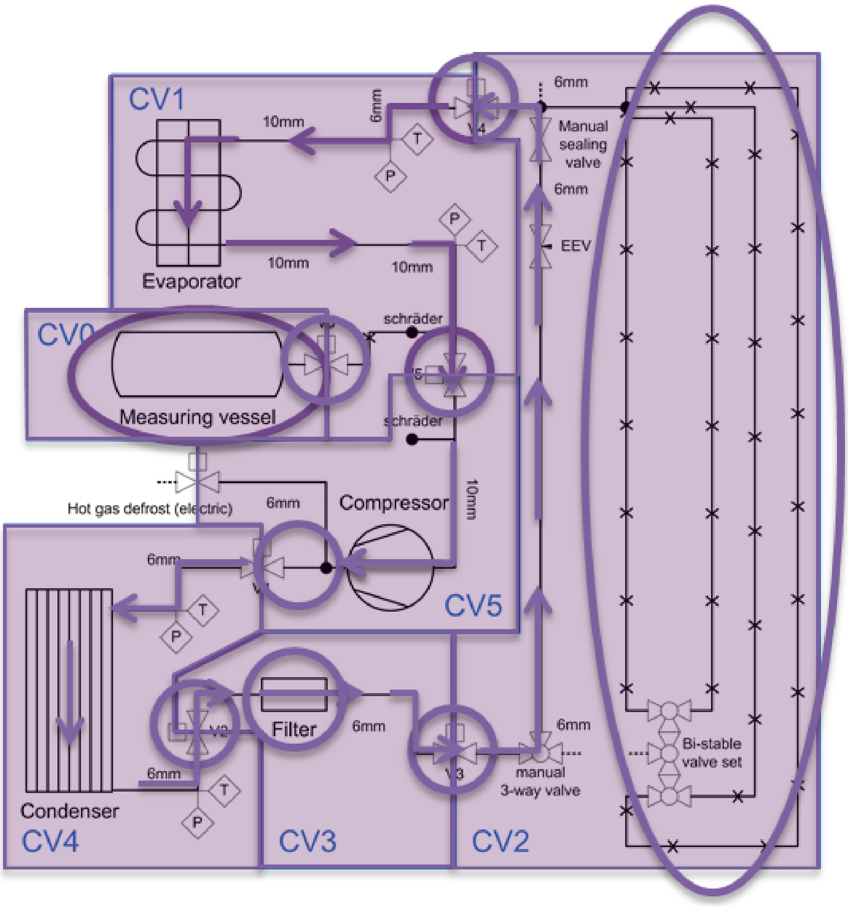

Figure 5: Principle sketch of the prototype heat pump used for charge measurement and comparison of capillary and electronic expansion valve.

[8] The measurement system used for the refrigerant mass measurement is available at the Re/genT laboratories and is not part of the HP-Launch project.

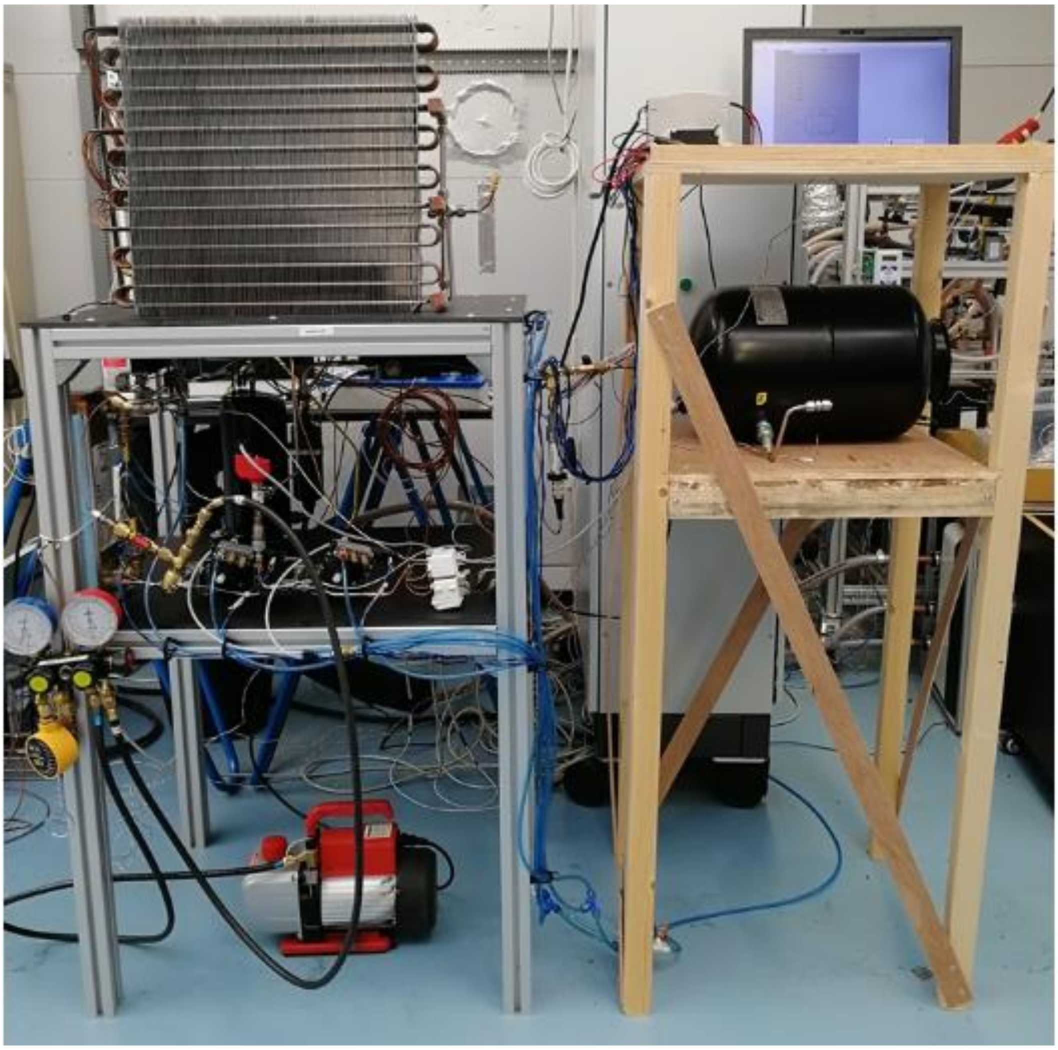

Figure 6: Picture of the prototype heat pump used for charge measurement and comparison of capillary and electronic expansion valve. Left: heat pump including closing valves and expansion devices. Right: Measurement vessel.

4.8.1 Measurement results

Testing at an air temperature of 7 °C and a water outlet temperature of 35 °C, showed highest COP for using the electronic expansion valve. The capillary tube, however, showed the highest system COP when operating at low refrigerant charge (e.g. suboptimal charge, below 155 gram). At a refrigerant charge of 155 gram, the capillary tube and electronic expansion valve show similar COP, see Figure 7.

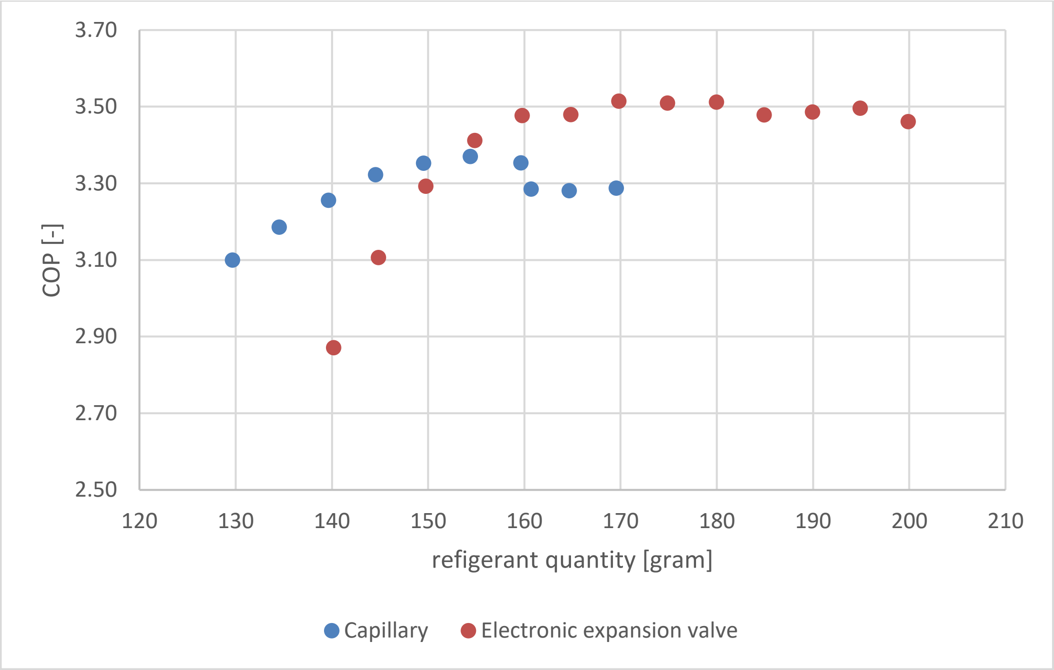

Figure 7: Prototype COP using a capillary tube or an electronic expansion valve as the expansion device.

The testing showed that a capillary makes it possible to operate the heat pump with less refrigerant charge, albeit at reduced efficiency. The testing also showed that with a charge of 155 grams, which is very close to the design limit of 150 gram, no difference resulted for using a capillary tube and electronic expansion valve. Based on this, and the desire to reverse the cycle for defrosting, a reversible electronic expansion valve of Sanhua was selected (model: DPF(T01)1.3C-07). Based on availability, however, a bi-flow electronic expansion valve of Carel (model: E2V14BSF00) was applied in the final prototype.

It needs to be noted that the prototype system was not optimised and was fitted with a relatively small evaporator, which resulted in low evaporating temperatures (approximately -7 °C at an ambient air temperature of 7 °C). Additional evaluation of heat pump controllability with a capillary tube is recommended for further study.

4.9 Reversing valve

For hot gas defrosting a 4-way reversing valve is selected. Model Sanhua SHF(L) 4H 23U 5. Due to availability, however, a reversing valve of Danfoss (valve model: STF0101G and coil model: STF 01AB500A1) was applied in the prototype. Note: the Danfoss valve is not specified for the use of R-290.

4.10 Suction gas heat exchanger

A suction gas heat exchanger was fitted to maximise the utilisation of the evaporator. The main idea here is to control the expansion valve opening based on the temperature of the superheated refrigerant downstream to the suction gas heat exchanger instead of using the evaporator superheat. It is expected that this control can lead to proper control with minimum superheat in the evaporator, and therefore results in maximum evaporator heat absorption.

Based on availability and small internal volume a Danfoss suction gas heat exchanger was selected (model: HE05). This heat exchanger has a liquid volume of 8.3 cm^3 and a vapour volume of 23 cm^3,

4.11 Filter

To reduce system refrigerant charge, filters with minimum internal volume were fitted up and downstream of the expansion valve. No suitable commercially available options were found. Therefore, the filters were self-made by modifying standard filter driers (Vulkan lokring, model DR10). The original filters/drier were cut, the absorbent was removed, and a standard pipe reducer coupling (5/8" to 1/4") was fitted directly after the internal filter-gauze, see Figure 8.

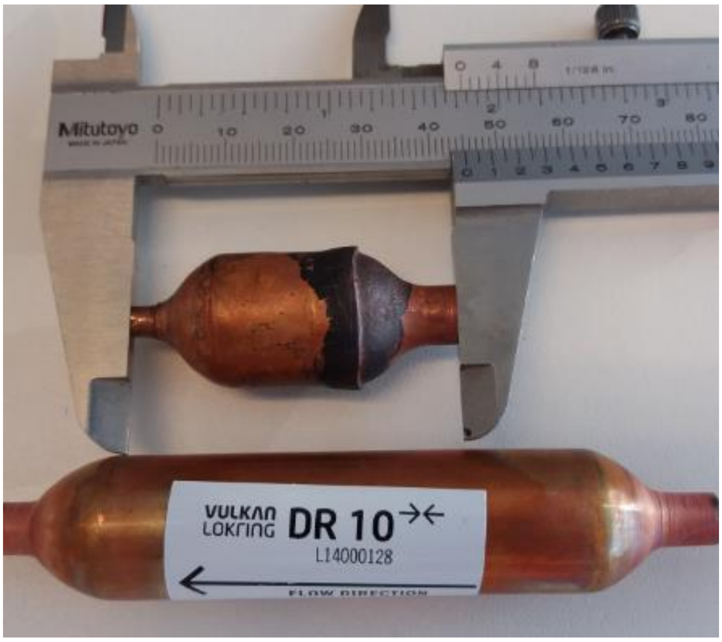

Figure 8: Picture of self-made filter. "The picture of the self-made filter is of a first trial in which a similar filter with a reduced outlet was modified. In the prototype the filters are of equal size but are made by modifying the Vulkan Lokring DR10 filter//drier"

5 CHARGE MEASUREMENT AND ESTIMATIONS

5.1 Charge measurement

For the selection of the number of plates for the condenser and the number of tube rows for the evaporator as well as to validate the feasibility of a heat pump with a refrigerant charge limit of 150 gram R-290, a prototype heat pump was built, see Figure 5 and Figure 6. The prototype uses the selected condenser and compressor and both these components can be easily replaced. The prototype is fitted with fast closing valves, which divide the system in 5 control volumes (CV1 to CV5) and is connected to a measurement vessel. Using the closing valves and the measurement vessel the refrigerant in each control volume can be measured using the PVT-superheat method.

The control volume includes the main component as well as the connection tubing between the component and the closing valves, see Figure 5. Measurements were performed with both the 30 and 40 plates condenser installed and using both the single and twin cylinder rotary compressor of GMCC. Next to changing the condenser and compressor, small changes were made in the tube routing between both configurations. Therefore, next to the expected change in condenser volume also a change in the control volume containing the compressor, the filter and the expansion devices resulted. The tests were performed at steady state operation at the reference condition of ambient air 7 °C and water outlet 35 °C after optimisation of the refrigerant charge[9]. The results of the measurements are presented in Table 3 and Table 4.

	Control volume	Internal volume	Charge R-290
		[ml]	[g]
CV5	Compressor + oil (single cylinder)	1660[10]	51.0
CV4	Condenser 40 plates	440	73.5
CV3	Filter	29	8.8
CV1	Evaporator	662	25.4
CV0	Expansion section	63	7.6
	Total system with 40 plates condenser	2854	166.3

Table 3: Measured control volume and refrigerant charge with 40 plates condenser and single cylinder rotary compressor

	Control volume	Internal volume	Charge R-290
		[ml]	[g]
CV5	Compressor + oil (twin cylinder)	1915[10]	46.0
CV4	Condenser 30 plates	324	47.0
CV3	Filter	21	9.0
CV1	Evaporator	662	26.0
CV0	Expansion section	57.1	8.0
	Total system with 30 plates condenser	2980.5	136.0

Table 4: Measured control volume and refrigerant charge with 30 plates condenser and twin cylinder rotary compressor

[9] Optimum charge is defined as the charge resulting in maximum system COP.
[10] Free gas volume with the oil being inside the shell

Table 3 and Table 4 show that most refrigerant is present within the condenser followed by the compressor and oil. The tables show that the 30 plates condenser (as expected) has approximately 25% less internal volume (refrigerant side) than the 40 plates condenser. Resulting in a 26 grams smaller refrigerant content for the condenser when using 30 plates. The tables also show that the twin cylinder compressor contains less refrigerant than the single cylinder compressor, albeit having a much larger internal volume. The larger internal volume results from the much larger suction muffler on the twin cylinder compressor. The muffler being at suction pressure (low density refrigerant), the impact on refrigerant charge of the increased suction volume is negated by the smaller shell volume of the twin cylinder compressor. Note: The shell volume is at discharge pressure (high density refrigerant) and both compressors use the same shell. The presence of the additional cylinder reduces the free gas volume of the compressor shell.

5.2 Charge calculation

Charge predictions were made using a numerical charge model developed by Re/genT. The model estimates the refrigerant charge for steady state operating conditions based on given system dimensions and operating conditions using 22 different void fraction correlations for the two-phase sections in the heat exchangers. The model is originally developed for bottle coolers[11] but is modified to include the charge prediction of plate heat exchangers and multi circuit evaporators. The model has been validated using the measurement results of Table 3 and Table 4. This validation showed best agreement for using the Hughmark void fraction correlation for the condenser and using the Armand and Massina void fraction correlation for the evaporator. Table 5 presents the results of estimating the refrigerant charge for the prototype fitted with the 40 plate condenser and compares the estimation with the measured charge of Table 3.

	Calculated charge [g]	Measured Charge [g]	Error [g]	Error [%]
Compressor shell (single cylinder)	35.1	-	-	-
Oil	22.5	-	-	-
Compressor shell + oil	57.6	51.0	6.6	12.9
Discharge tubing	0.5	-	-	
Condenser (40 plates)	76.2	73.5	2.7	3.7
Liquid line	4.6	-	-	
Filter	6.0	8.8	-2.8	31.8
Expansion valve to distributor	11.1	7.6	3.4	44.7
Evaporator	20.7	25.4	-4.7	18.5
Suction tube	0.6	-		
Total system	177.5	166.3	11.2	6.7

Table 5: Calculated refrigerant charge for configuration with 40 plates condenser

Table 5 shows that the estimation results in a 6.7% larger charge for the complete system than measured and that the error strongly differs between the various components. The table also shows that the charge prediction of the large contributors (e.g. condenser and the compressor + oil) are within 15%. Based on this it was concluded that the model can be used as a design tool to obtain a first indication of charge quantity.

[11] M. van Beek, T. van Gorp, (2018), Charge Equation For Low Charge Hydrocarbon Based Commercial refrigeration Appliances, in 17th International refrigeration and Air Conditioning Conference, Purdue. (https://docs.lib.purdue.edu/iracc/2046/)

5.2.1 Charge prediction HP-Launch prototype

Using the numerical charge model, the refrigerant quantity of the HP-Launch system, as presented in Chapter 2, was estimated for the following heat exchanger configurations.

- 3-row evaporator and 40 plates condenser
- 3-row evaporator and 30 plates condenser
- 2-row evaporator and 40 plates condenser
- 2-row evaporator and 30 plates condenser

The results of these estimations are presented in Table 6.

		3 row evaporator 30 plate condenser	2 row evaporator 30 plate condenser [g]	3 row evaporator 40 plate condenser	2 row evaporator 40 plate condenser [g]
Compressor shell	[g]	36.4	36.3	36.3	36.3
Oil	[g]	21.6	21.6	21.6	21.6
Discharge tubing	[g]	1.0	1.0	1.0	1.0
Condenser	[g]	57.2	57.1	80.4	80.3
Liquid line	[g]	4.9	4.9	4.9	4.9
Filter	[g]	6.0	6.0	6.0	6.0
Expansion valve to distributor	[g]	0.6	0.6	0.6	0.6
Evaporator	[g]	41.5	26.4	48.7	31.5
Suction tube	[g]	1.7	1.6	1.7	1.6
Total system	[g]	170.9	155.7	201.4	184.0

Table 6: Calculated refrigerant charge for HP-Launch options

Table 6 shows that only the option with the 2-row evaporator and 30 plates condenser results in an estimated refrigerant charge close to the charge limit of 150 grams. Therefore, this configuration was selected for the prototype.

Note: For the prototype also the twin cylinder compressor was selected. The compressor selection, however, was strongly driven by the reduced vibration of the twin-cylinder design and the approximately 5 gram smaller refrigerant charge (see Table 3 and Table 4) was seen as an additional benefit.

6 PERFORMANCE MEASUREMENTS

The HP-Launch prototype is evaluated using the measurement set-up described in Re/genT report 20104/RO01/V1. Heat pump system temperatures are measured using T-type thermocouples and system pressures (i.e. discharge, evaporator outlet and suction) were measured using Keller PA33X pressure transducers. The compressor speed was not controlled, and a fixed speed setting was applied for each specific test. Defrosting was performed by manually switching the compressor, reversing valve and evaporator fan. And the expansion valve opening was controlled using a Labview based PID controller. The control is based on the superheat which is determined from the temperature measurement on the suction line and the saturation temperature calculated from the measured suction pressure using Refprop 10.0. For further details, see Re/genT report 20202/RO04/V1 and report 20104/RO01/V1.

Different to the testing of the reference unit, see Re/genT report 20105/RO02/V1, for the HP-Launch prototype the impact of defrosting is included. This was possible as the start of the defrosting action was known. This made it possible to pre-adjust the conditioning of the climate room (e.g. deactivation of room heating and humidification prior to defrosting). This resulted in an ambient temperature increase of less than 4 K and a return to the controlled ambient conditions within 10 minutes after defrosting. During the testing of the reference unit an ambient temperature rise of 5 to 10 K resulted and it took more than 15 minutes to return to the controlled ambient conditions. See also Re/genT report 20202/RO04/V1 and 20105/RO02/V1.

Heating performance is measured at the NEN-EN 14511 standard rating condition for air to water heat pumps for low temperature applications. The heating performance is also measured at the part load conditions for both low and medium temperature applications at average climate conditions following NEN-EN 14825. Next to this the seasonal heating performance is calculated according NEN-EN 14825. In this report only the main measurement results are presented. For more details, reference is made to the measurement report of the HP-Launch prototype, see Re/genT report 20202/RO04/V1.

6.1 NEN-EN 14511 standard rating condition

Two measurements were conducted at the standard rating condition of NEN-EN 14511 for air to water heat pumps at low temperature applications. Namely, a measurement applying a charge of 150 g of R-290 and an additional test at the optimum[12] refrigerant charge of 170 g of R-290. See Table 7 for the measurement conditions and Table 8 for the main measurement results.

Air temperature (dry bulb)	[°C]	7.0
Humidity (wet bulb temperature)	[°C]	6.0
Water inlet temperature	[°C]	30.0
Water outlet temperature	[°C]	35.0
Compressor speed setting	[rpm]	3600
Fan speed setting	[V]	6[13]

Table 7: Measurement conditions.

[12] The optimum charge is the refrigerant charge resulting in the largest system COP.
[13] A fan speed setting of 6 V, corresponds to an airflow of approximately 1930 m^3h^{-1}

Refrigerant charge	[g]	150	170
Heating capacity	[W]	3414	3510
Input power compressor + fan	[W]	746	739
Water pressure drop	[bar]	0.11	0.10
Calculated pump power[14]	[W]	18.8	18.4
Total input power	[W]	764	758
COP	[W/W]	4.47	4.63

Table 8: Main measurement results at 150 and 170 gram R-290

Table 8 shows that at a refrigerant charge of 150 gram of R-290, the HP-launch prototype heat pump has a measured system COP of 4.47. At the optimum refrigerant charge of 170 gram, the COP of the system increases to 4.63, which is similar with the COP of 4.65 that was measured for the refence heat pump at this operation condition.

6.2 NEN-EN 14825

Heat pump performance was measured at part-load conditions for low and medium temperature application for the reference heating season “average” following NEN-EN 14825. The part load capacity was calculated from the heating demand of the HP-launch reference house, i.e. heating demand of 5.5 kW at -10 °C outdoor temperature and no heating demand at an ambient temperature of 15 °C, see Work Package 1 of the HP-Launch project. Based on the heat pump design this results in a bivalent temperature of Tj = 2 °C.

During the measurements, the heat output of the heat pump was varied by setting the compressor speed (1200 to 3600 rpm) to a fixed value for each test. The fan speed setting (6V) and water mass flow rate (8.6 kg(min)$^{-1}$) were not adjusted during or between the tests.

6.2.1 Low temperature application for average climate

The test conditions are presented in Table 9 and the main measurement results are depicted in Figure 9 and Figure 10. Note: the measurement results at an ambient below Tj = 2 °C, include defrosting.

[14] Calculation of the input power of the water circulation pump based on the measured water side pressure drop following NEN-EN 14511.

	Part load ratio [%]	Compressor speed [rpm]	Outdoor air temperature dry bulb [°C]	Outdoor air temperature wet bulb [°C]	Water temperature outlet [°C]	Calculated Part load capacity [W]
A	88	3600	-7	-8	34	4865
B/F[15]	54	3600	2	1	30	2962
C	35	1800	7	6	27	1904
D	15	1200	12	11	24	846
E[15]	100	3600	-10	-11	35	5500

Table 9: Test conditions and part load capacity of the heat pump for low temperature application for the reference heating season "average".

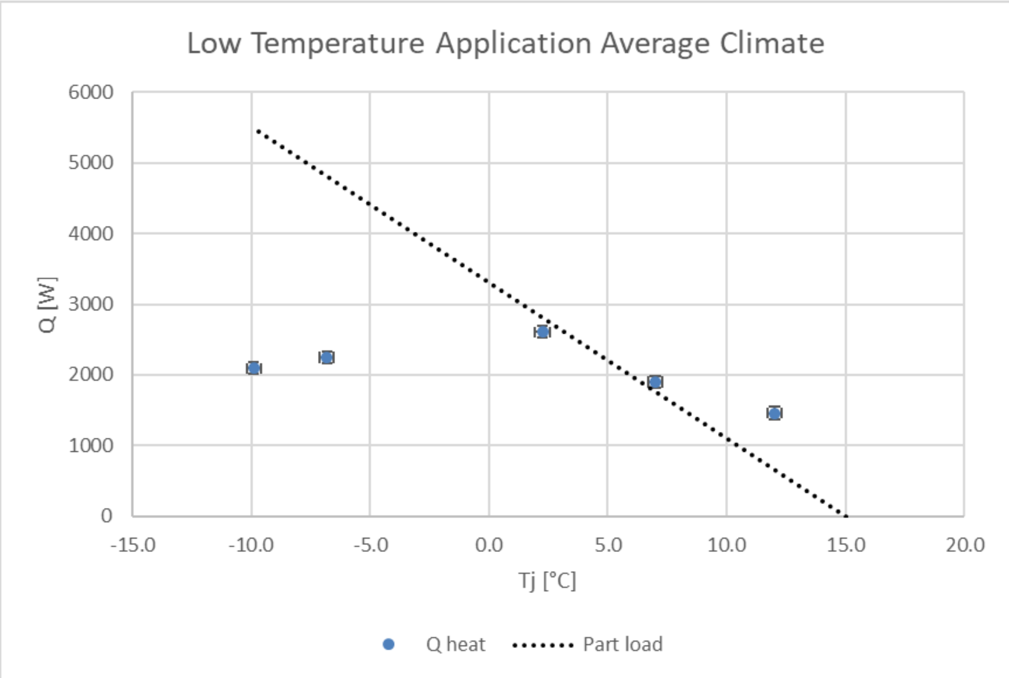

Figure 9: The part load heating capacity (dotted line), the heating capacity during measurement (blue dots) for low temperature application under average climate conditions. The uncertainty in measurement is presented by the error bars.

15 The heat pump has a bivalent temperature (T_{biv}) of 2 °C, and a heating capacity of Pdh = 3 kW at Tj = 2 °C. Following NEN-EN 14825, the reference design temperature conditions for heating (Tdesignh) for the "average" heating season is defined at -10 °C. NEN-EN 14825 also states that if the declared TOL is lower than the Tdesignh of the considered climate, then it may be assumed that TOL is equal to Tdesignh. The operation limit temperature of the heat pump is not yet defined, but TOL < -10 °C

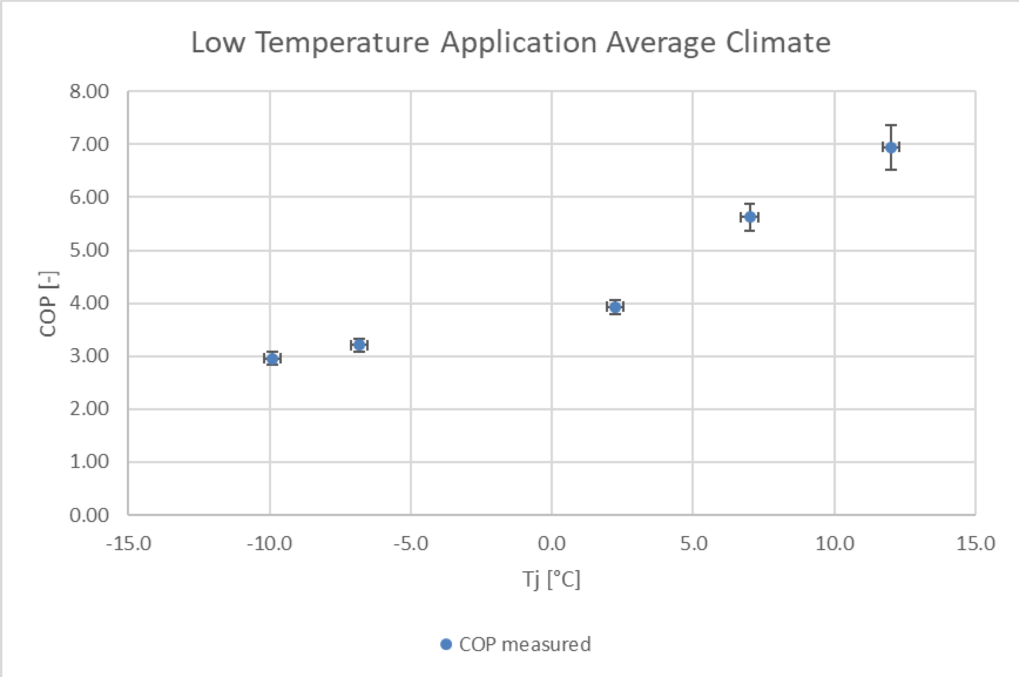

Figure 10: The measured part load COP (blue dots) for low temperature application under average climate conditions. The uncertainty in measurement is presented by the error bars.

Figure 9 and Figure 10 show that at an ambient of Tj = -10, the heat pump has a measured heating output of 2 kW with a COP of 3.0. And at the bivalent temperature (Tj = 2 °C) the heating output is measured at 2.6 kW with a COP of 3.9. Note that the capacity of the heat pump, albeit at the cost of efficiency, can be increased by increasing compressor speed. Testing is performed at a maximum speed of 3600 rpm, the maximum compressor speed is 7200 rpm.

6.2.2 Average temperature application for average climate

The test conditions are presented in Table 10 and the main measurement results are depicted in Figure 11 and Figure 12. Note: the measurement results at an ambient below Tj = 2 °C, include defrosting.

	Part load ratio	Comp. speed	Outdoor air temperature dry bulb	Outdoor air temperature wet bulb	Water temp. outlet	Calculated Part load capacity
	[%]	[rpm.]	[°C]	[°C]	[°C]	[W]
A	88	3600	-7	-8	43	4865
B/F[15]	54	3600	2	1	37	2962
C	35	1800	7	6	33	1904
D	15	1200	12	11	28	846
E[15]	100	3600	-10	-11	45	5500

Table 10: Test conditions and part load capacity of the heat pump for medium temperature application at average climate conditions

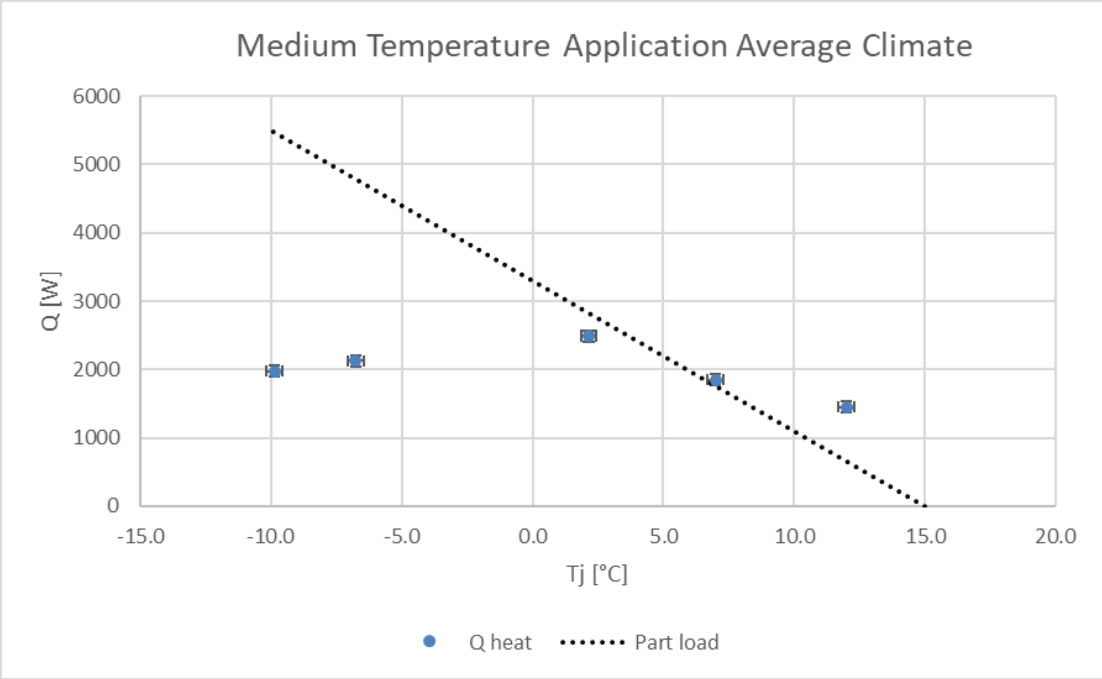

Figure 11: The part load heating capacity (dotted line), the heating capacity during measurement (blue dots) for medium temperature application under average climate conditions. The uncertainty in measurement is presented by the error bars.

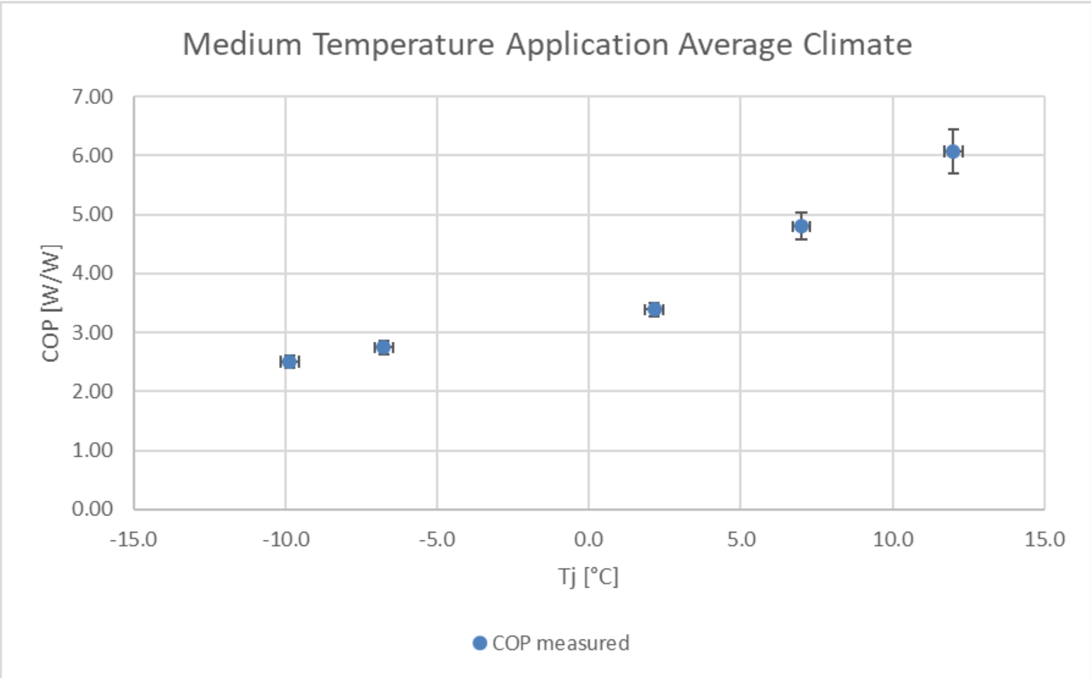

Figure 12: The measured part load COP (blue dots) for medium temperature application under average climate conditions. The uncertainty in measurement is presented by the error bars.

Figure 11 and Figure 12 show that at an ambient of Tj = -10, the heat pump has as a measured heating output of 2 kW with a COP of 2.5. And at the bivalent temperature (Tj = 2 °C) the heating output is measured at 2.5 kW with a COP of 3.4. Note that the capacity of the heat pump, albeit at the cost of efficiency, can be increased by increasing compressor speed. Testing is performed at a maximum speed of 3600 rpm, the maximum compressor speed is 7200 rpm.

6.3 SCOP and SPER

Using the measurement data and the heating requirements of the reference building The Seasonal Coefficient of Performance of the heat pump (SCOPnet) and the Seasonal Primary energy ratio of the heat pump only ($SPER_{HP}$) and of the heat pump gas boiler combination (SPER) are calculated, see Table 11. For more details, reference is made to Re/genT report 20202/RO04/V1.

	Low temperature application	Medium temperature application
$SCOP_{net}$	4.44	3.84
$SPER_{HP}$	1.63	1.46
SPER	1.53	1.37

Table 11: SCOP and SPER at average climate conditions

7 GENERAL OBSERVATIONS

7.1 Control

Testing showed that the initial idea of controlling the expansion valve opening based on the refrigerant superheating after the suction gas heat exchanger (SGHEX) does not work for the prototype. In fact, at this location, the superheating resulting in the highest COP strongly depends on the operating condition. The superheating after the evaporator, however, showed an optimum and has therefore been used to control the opening of the expansion valve. See Figure 13 and Figure 14, were respectively the superheating after the suction gas heat exchanger and the superheating at the evaporator outlet are presented. These tests are performed at various setpoints of evaporator superheat at the evaluated test conditions.

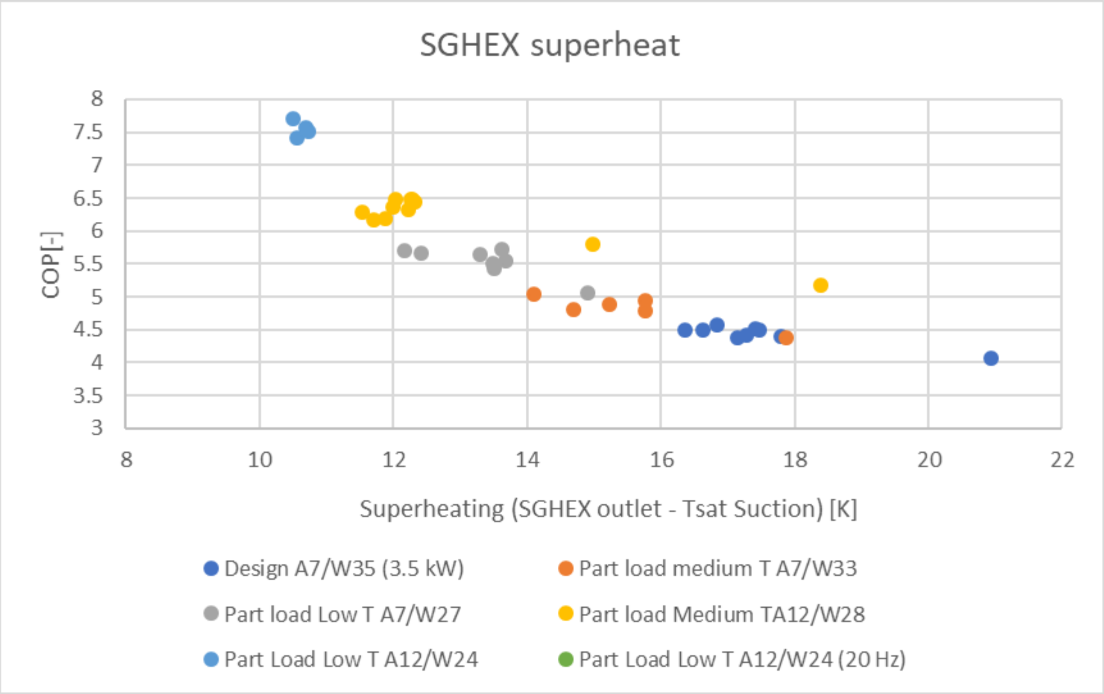

Figure 13: System COP against superheating measured after the suction gas heat exchanger. Temperature probe positioned just after the suction gas heat exchanger in combination with measuring the saturation pressure at the inlet of the compressor.

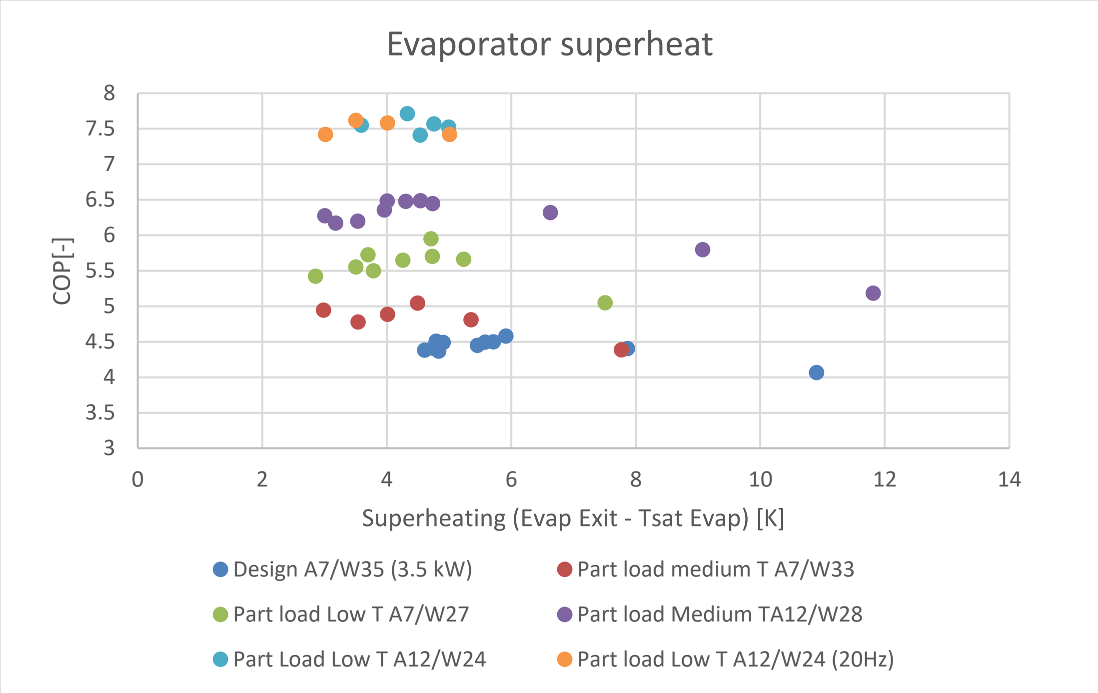

Figure 14: System COP against superheating measured after the evaporator outlet. "Note measurement points are equal to the points presented in Figure 13" Temperature probe positioned at the common outlet tube of the evaporator in combination with measuring the saturation pressure at the outlet of the evaporator.

Comparing Figure 13 and Figure 14 shows that the superheating based on the outlet of the SGHEX shows no optimum in COP while the superheating based on the evaporator outlet shows an optimum in COP for an evaporator superheating of approximately 5 K.

The reasons for the unexpected behaviour of the superheating after the SGHEX are:

1) The low refrigerant charge. With 150 grams there is insufficient refrigerant to create subcooling in combinations with substantial flooding of the evaporator (e.g. it is not possible to achieve small superheating at the SGHEX outlet.
2) The position is quite far from the evaporator and heat is exchanged in both the reversing valve and the SGHEX. Due to this the temperature after the SGHEX is much warmer than the evaporator outlet temperature and depends strongly on the operating condition of the system. For the reversing valve a temperature increase of the suction vapour of 1.5 to 3.5 K is measured and for the SGHEX a temperature increase between 2.5 and 9 K is measured between the various tests
3) The SGHEX has a strong impact on system behaviour, especially for operation at low refrigerant charge. When operating at low refrigerant charge (150 gram), i.e. small subcooling, the suction gas heat exchanger can become part of the condenser. In such situation, two phase flow enters the expansion valve, and due to the vapour at the expansion valve inlet, further opening of the valve does not reduce the superheating. In the prototype stability issues were also observed at larger refrigerant charge (180 gram). For these measurements when two-phase refrigerant enters the SGHEX from the evaporator, large heat transfer results, subcooling increases, and non-proportional increase in refrigerant flow results for further opening the valve.

The testing at increased charged, which is only performed at a single operating condition, shows an optimum in the superheating and can therefore be used for control, see Figure 15. However, it is expected, that based on the SGHEX the optimum superheating temperature will shift depending on the operating condition. Next to this, proper control parameters need to be defined to handle the additional nonlinearity caused by the increase in subcooling when two-phase flow enters the SGHEX from the evaporator.

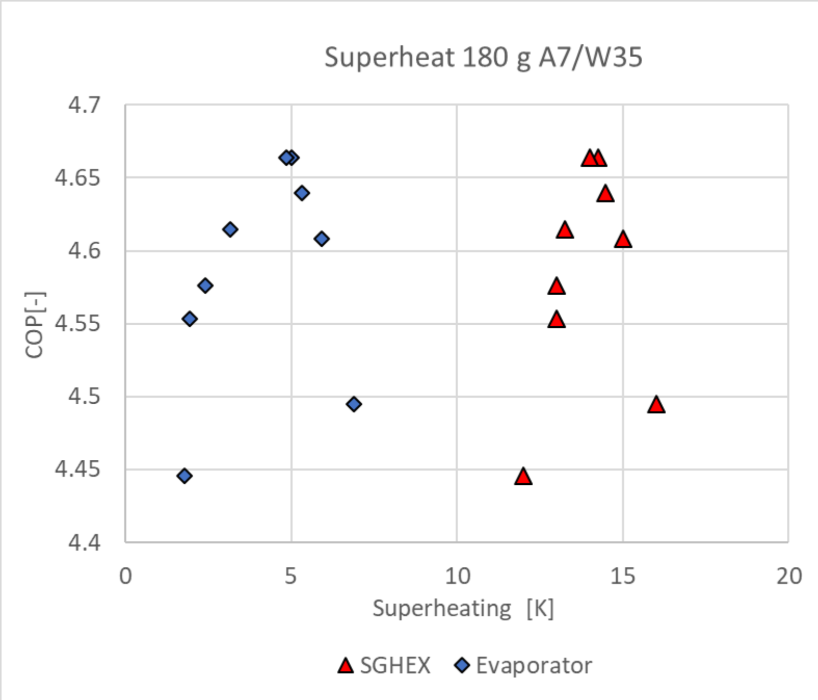

Figure 15: System COP against superheating measured after the evaporator outlet and after SGHEX. "Measurement with 180 g R-290 at A7/W35"

Note: No issues were observed by controlling the expansion valve based on 5 K evaporator superheat using PID control at a refrigerant charge of 150 grams.

7.2 Suction line heat exchanger

The suction line heat exchanger is an additional component, which increases system costs and introduces an additional flow resistance in the suction line. In the measurements the pressure drop over the suction line heat exchanger varied between 0.04 bar (A12/W28 @ 1200 rpm) to 0.15 bar (A7/W35 @ 3600 rpm), which corresponds to a drop in saturation temperature of 0.14 to 1.1 K. The testing did not show benefit of the SGHEX in system control. Therefore, it is recommended not to apply a SGHEX.

Note; Due to the reversing valve the superheating at the compressor inlet is already much larger than the superheating at the evaporator outlet. Thereby, avoiding possible issues with the viscosity of the oil returning to the compressor. Next to this, with low refrigerant charge, the risk of liquid slug strongly reduces.

7.3 Refrigerant charge

Designing towards minimum refrigerant charge while using standard available components showed that it is possible to design an air to water heat pump having a nominal heating capacity of 3.5 kW and a refrigerant charge of 150 grams of R-290. System charge optimisation showed that the prototype has highest system COP at a refrigerant charge of 170 gram of R-290. The efficiency difference between the suboptimal charge of 150 gram and the optimal charge of 170 grams, however, is within 3.5%, see Table 8.

It is expected that system refrigerant charge can be further reduced, through:

- Increasing the wall thickness of the evaporator tubing
    - ~ 3 g reduction in refrigerant charge by reducing the inner tube diameter from 4.85 to 4.60 mm.
- Removing the SGHEX
    - ~ 3 g reduction in refrigerant charge
- Further reduction of tubing length
    - ~ 1 g reduction in refrigerant charge based on 0.1 m reduction in liquid tube length
- Reducing the free gas volume within the compressor shell, see also Figure 16.
    - ~ 9 g reduction based on a reduction in free gas volume from 1.6 to 1.2 dm^3
- Reducing oil content.
    - ~ 2 g based on 10% reduction in oil charge
- Further optimisation of plate heat exchanger
    - ~ 3 g based on 5% reduction in condenser refrigerant volume

Doing all the above results in optimum system performance (i.e. maximum COP) at a refrigerant charge of 150 gram or less.

Figure 16:Large free volume exists above the motor within the shell of the GMCC DSM18019UDZ compressor.

7.4 Refrigerant distribution

The prototype evaporator included 9 parallel circuits. To distribute the refrigerant a distributor was designed, see Section 4.5 and capillary restrictions were fitted. However, this combination resulted in non-ideal distribution of the refrigerant throughout the evaporator. To improve refrigerant distribution, the capillary restrictions were manually adjusted. This was done by measuring the superheating at the outlet of each of the circuits and by pinching the capillary tubes until similar superheating resulted for all circuits. This was done during operation at A7/W35 with a compressor speed 3600 rpm with multiple additional thermocouples fitted on the evaporator tubing.

8 PERFORMANCE ESTIMATION ALTERNATIVE HEAT ECHANGER OPTIONS

Within the HP-launch project a heat pump simulation model has been developed. The model and the results of the validation are presented in Re/genT note 20201/R04/V1. Using the model, heat pump performance has been estimated for using the alternative heat exchanger options (i.e 3-row evaporator and 40-plates condenser) and for removing the SGHEX. Next, to this also the required refrigerant charge is estimated for these configurations. The main results of these calculations are presented in Figure 17, Figure 18 and Table 12.

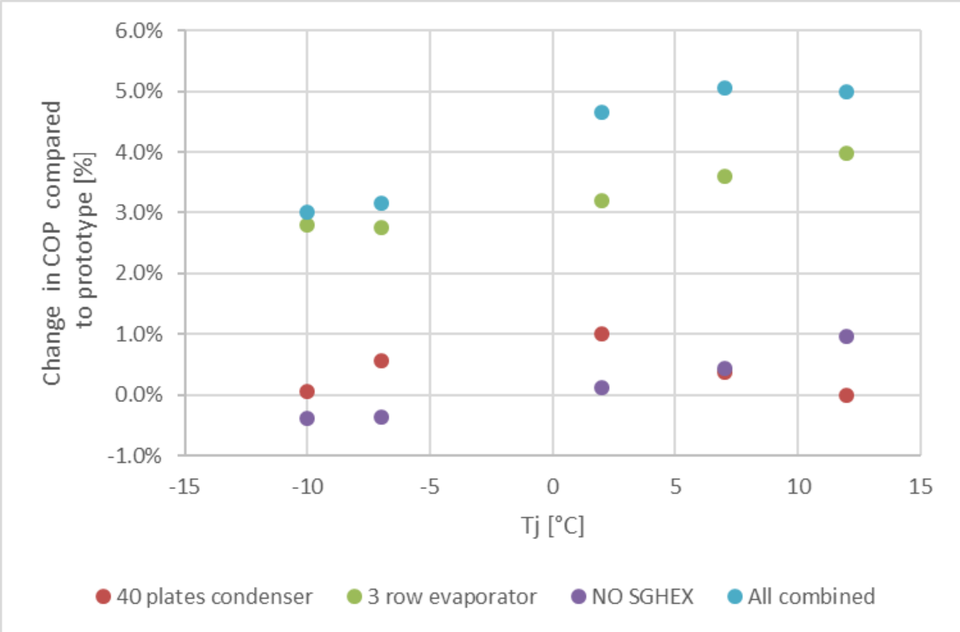

Figure 17:Impact on system COP of alternative configurations using the prototype configuration as the reference.

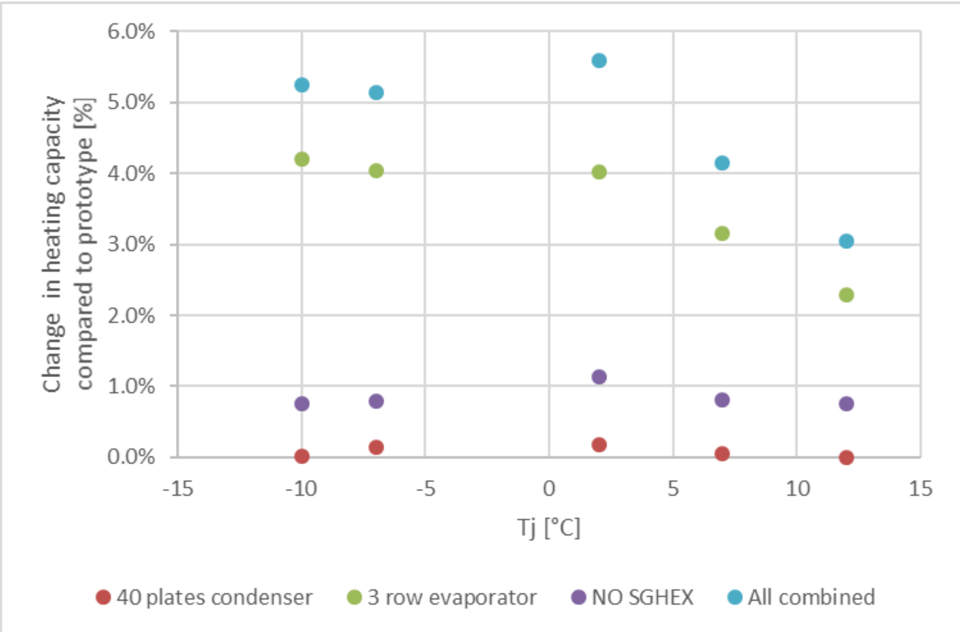

Figure 18:Impact on system heating capacity of alternative configurations using the prototype configuration as the reference.

Configuration	Heating capacity [%]	COP [%]	Refrigerant charge [g]
3-row evaporator	≈ + 3.5	≈ + 3.3	+15 to +20
40 plates condenser	≈ + 0.1	≈ + 0.4	+5 to +10
Removing SGHEX	≈ + 0.9	≈ + 0.15	-2 to -3
All combined	≈ + 4.6	≈ + 4.2	+18 to +27

Table 12: Estimated impact on overall efficiency and refrigerant charge of the alternative configuration using the prototype configuration as the reference.

The calculations show that changing the condenser from 30 to 40 plates, which is a 33% increase in heat transfer area, has only marginal impact on system efficiency. In other words, the selected 30 plates condenser is close to the thermal optimum. The calculations also show that changing the evaporator from 2 to 3-rows, which is a 50% increase in heat transfer area, results in an approximately 3.5% increase in system COP and heating capacity, albeit at the cost of increasing refrigerant charge with 15 to 20 grams.

9 CONCLUSIONS

A compact domestic heat pump with a total refrigerant charge of 150 g Propane and a nominal heating capacity of 3.5 kW and COP of 4.5 at the standard NEN-EN14511 reference condition A7/W35 has been designed, built and evaluated.

The selection of Propane (R-290) as the natural refrigerant is based on good cycle performance in combination with commercial availability of refrigeration components. The main downside of using R-290 is its high flammability. This is covered by designing the heat pump for a maximum refrigerant charge of 150 grams following IEC 60335-4-2:2018.

Within the project, it was decided to develop a "through the roof" solution. This finally resulted in a design that is integrated in the roof and can be serviced from the inside of the house. For this design, the distance between the air side and water side heat exchanger is small and a large volume is available to install the complete heat pump unit. Based on this and the ease of installation the monobloc configuration is selected.

Component selection, based on performance, refrigerant quantity, weight and commercial availability resulted in the selection of an 18.1 cm^3 variable speed twin cylinder rotary compressor from GMCC, an asymmetric condenser from SWEP, a mini channel evaporator (5 mm tubing) from LU-VE, a 350 mm variable speed fan from PAPST, an electronic expansion valve and a reversing valve from Sanhua and a suction line heat exchanger from Danfoss.

Practical evaluation of the prototype showed a system COP of 4.5 and a heating capacity of 3.4 kW at the NEN-EN14511 reference condition A7/W35 for using a refrigerant charge of 150 gram. The Seasonal Primary Energy Ratio (SPER) of the system is calculated for the HP-Launch reference house (Work package 1), while assuming hybrid operation in combination with a natural gas boiler. This resulted in a SPER of 1.53 and a SPER of 1.37 for operation at respectively low and medium temperature application at the reference climate average. Comparing the COP and the SPER values with the measurement results of the reference system, shows that the efficiency of the HP-Launch heat pump is comparable to best in class appliances available on the market.

Refrigerant charge optimisation showed that a 3.6% larger system COP can be obtained for the prototype by increasing the refrigerant charge from 150 to 170 grams. Evaluation of system design showed that the volume at the refrigerant side can be further reduced to meet this optimum COP at a charge of 150 gram or below. This, however, requires some changes of the selected component.

Calculation of system performance showed that changing the condenser from 30 to 40 plates, which is a 33% increase in heat transfer area, has only marginal impact on system efficiency. In other words, the selected 30 plates condenser is close to the thermal optimum. Changing the evaporator from 2 to 3-rows, which is a 50% increase in heat transfer area, results in an approximately 3.5% increase in system COP and heating capacity, albeit at the cost of increasing refrigerant charge with 15 to 20 grams.

Remark: The project idea was that heat pump can be installed indoors for which a charge limitation of 150 g of flammable refrigerant applies. Due to this and the general desire to have a compact low refrigerant charge system, heat pump development focussed on charge minimisation. The final total system design, however, resulted in a "through the roof" unit that is physically separated from the indoor environment. This basically is an outdoor unit, and for such units the charge limitation of 150 g of flammable refrigerant does not apply.

9.1 Recommendations

The protype included a suction gas heat exchanger (SGHEX). The idea behind this was to control the expansion valve opening based on the superheating measured after the SGHEX. Thereby, making it possible to operate at minimum evaporator superheat and therefore at maximum evaporator heat absorption. The testing, however, showed that optimum efficiency is obtained at an evaporator superheat of approximately 5 K. Next to this the practical validations showed that the superheating after the SGHEX does not provide a stable control parameter for a low charge system. Therefore, and due to the additional costs, refrigerant charge and pressure drop, it is recommended not to apply a SGHEX and to control the expansion valve based on evaporator superheat.

The project is concluded with a functional prototype. It is recommended to further study the impact of fan speed, water flow and compressor speed control on heat pump performance and efficiency.

APPENDIX A: DISTRIBUTOR