HPT Annex 50

Heat Pumps in Multi-Family Buildings

Task 3.1: Optimized refrigeration cycle configuration for Air-Water Heat Pumps for renovated, unrenovated and new MFH

Country Report

AUSTRIA

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Contents

1. Summary

Based on the requirements determined in IEA HPT Annex 50 Task 1 (IEA HPT Annex 50 Task 1, 2017) for the systems for use in the various multifamily houses (MFH) categories (new, refurbished, unrefurbished), possible refrigeration cycle variants and suitable refrigerants for the use of Air-Water Heat Pumps were investigated on a numerical basis.

The different refrigeration cycle variants were specified for the modelling under consideration of various refrigeration cycle configurations (with/without hot gas technology, single stage circuit / EVI, with/without subcooler, with/without intermediate heat exchanger, choice of refrigerant). In addition to the refrigerant cycles the refrigerants R290, R410A, R134a and R1234zee were selected for the numerical comparisons.

The refrigerant cycle variants were modelled in the simulation environment Dymola/Modelica[1], a suitable software tool for the investigation of thermohydraulic systems. The parameterisation and validation were done by using results from experimental investigations (GreenHP, 2016).

Based on seasonal coefficient of performance (SCOP) calculations, the most efficient refrigerant cycle variants together with the most suitable refrigerant were selected. It turned out that that for all applications (new, unrenovated existing, renovated existing MFH) the refrigeration cycles with the refrigerant R290 had the highest SCOP. Especially for the higher temperature applications the refrigeration cycle variants with EVI lead to the highest SCOP. Beside the efficiency R290 offers the possibility to reduce the amount of refrigerant mass due to its thermodynamic properties.

Based on the selected refrigeration cycle variants, characteristic curves by means of regression analysis per use case for implementation in TRNSYS [2] were generated.

2. Introduction

Heat pumps (HP) are an environmentally friendly, renewable technology for hot water production, space heating and cooling of residential buildings, and are suitable for those buildings that are not yet connected to a district heating network. Air-Water Heat Pumps are particularly suitable for use in refurbished MFH in urban areas. They require little space, have comparatively low acquisition costs and are easy to install (GreenHP, 2016).

In Task 3 (Modelling and simulation of components and systems), optimized refrigeration cycle configuration for Air-Water Heat Pumps for new, refurbished and unrefurbished MFH buildings are identified and the most efficient configuration for the respective building category is determined by means of dynamic circuit calculations in Dymola/Modelica. In the course of the system classification in Task 1, the requirements for the systems are taken into account. During modelling, particular attention is paid to ensuring that different refrigeration circuit configurations can be calculated with different refrigerants. From the optimal variants for each building category, characteristic models for the respective application area are created for further use in the system simulations in TRNSYS in Task 3.2.

[1] https://www.3ds.com/de/produkte-und-services/catia/produkte/dymola/
[2] http://www.trnsys.com/

3. Methodology

The following steps were taken to select the optimum refrigeration cycle configuration for Air-Water Heat Pumps for the different application:

- Selection of suitable refrigerants and definition of typical applications based on preliminary work and literature research
- Definition of the refrigeration circuit variants to be investigated
- Design of the Air-Water Heat Pump model in the simulation environment Dymola/Modelica
- Validation of the model with experimental data from preliminary work
- Simulation of the defined variants
- Calculation of the SCOP for selecting the most suitable refrigeration circuit configuration for each application
- Generation of the characteristic curves by means of regression analysis per use case for implementation in TRNSYS

4. Definition of use cases

Based on the elaborated building stock characteristics in the HPT Annex 50 Task 1 report (IEA HPT Annex 50 Task 1, 2017) and investigations in the GreenHP project (GreenHP, 2016), three use cases were defined (Table 1). From the preliminary work, a typical Austrian MFH with 8 to 10 housing units and with living spaces per housing unit of approx. 70 to 100 m² could be described. In addition, in connection with the standard EN 14825 (EN 14825, 2018), the design temperatures for the heating system of the different use cases were defined. Table 1 shows the heating load, the specific heating load, the specific heating demand and the design temperature of the heating system for the three applications.

Table 1: Definition of MFH use cases

use cases - MFH	heat load	specific heat load	specific heating demand	system supply temperature
-	kW	W/m²	kWh/m²	°C
new	20	30	45	35
refurbished	30	50	65	55
unrefurbished	50	80	120	65

In the course of the GreenHP project (GreenHP, 2016), the heating capacity for a refurbished MFH was set at 30 kW and the system temperatures for the design at 55 °C. The subsequent experimental investigations served to validate the Air to Water heat pump model for the simulations in this IEA HPT Annex 50.

5. Selection of refrigerants and definition of refrigeration cycles

On the basis of literature research and from preliminary work (GreenHP, 2016 and Store4Grid, 2015) the refrigerants and definition of refrigeration cycles were selected.

The refrigerants selected for the numerical comparison are R290, R410A, R134a and R1234zee. The idea was to use state of the art refrigerants (R410A and R134a) as a benchmark and to have a closer look on using refrigerants with low global warming potential (GWP). In Figure 1 the GWP of the selected refrigerants are shown.

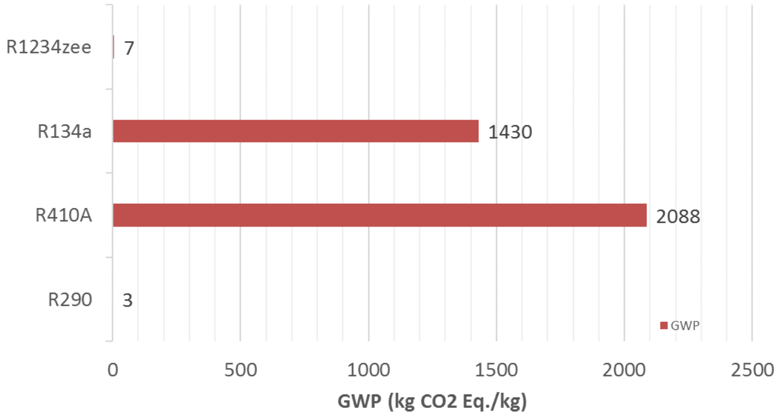


Figure 1: GWP of the selected refrigerants (Bitzer, 2018)

Therefore, R290 has been selected based on the investigations in the EU-funded project GreenHP and R1234zee (Honeywell, 2019) as an alternative to replace R134a for applications with high supply temperatures. Within the project GreenHP (GreenHP, 2016) only refrigerants with low GWP were considered. To determine the best suited refrigerant from environmental and technical aspects, a life cycle climate performance (LCCP) analysis was performed. For the analysis, four refrigerant options were selected: two hydrocarbons (R290, R1270) and two low GWP hydrofluorocarbons (R152a, R1234yf). During the anaylsis in IEA HPT Annex 50 the slightly different R1234zee was used instead R1234yf as R1234zee has close thermodynamic properties to R134a (Honeywell, 2019 and Bitzer, 2018). The LCCP of the GreenHP project showed that indirect emissions dominate the total lifetime emissions, indirect emissions accounted for over 99%. Therefore, choosing the most environmentally friendly refrigerant was also a choice of the most efficient refrigerant. Of the low GWP refrigerants, R1234yf had the highest lifetime emissions, while R152a, R290 and R1270 emitted almost the same CO_2 equivalents over their lifetime, with hydrocarbons showing the best environmental performance. Taking into account the possible limitations in the use of HFC refrigerants as well as other aspects of refrigerant use, preference was given to hydrocarbons, of which the refrigerant R290 (propane) was selected as the refrigerant on which the heat pump design is based (GreenHP, 2016).

Additional to the selection of the refrigerants, three different refrigerant cycle configurations were defined for the comparisons, see Figure 2:

- Single stage
- iHX (internal heat exchanger)
- EVI (enhanced vapour injection)

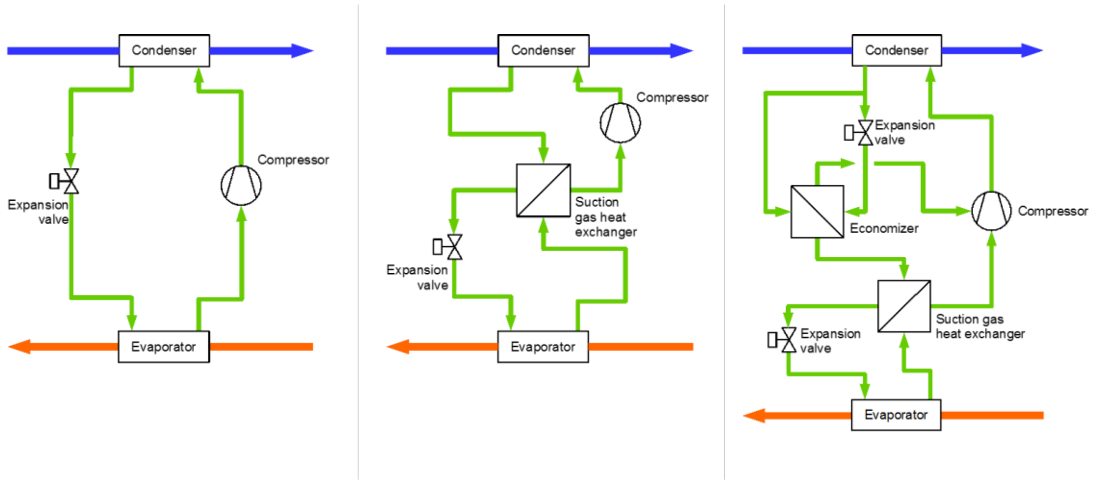


Figure 2: Refrigerant circuit configurations, from left to right: single stage, suction gas HEX, EVI

The single stage refrigeration cycle, Figure 2 (left) consists of the main components: evaporator, condenser, compressor and expansion valve. This refrigeration cycle configuration represents the simplest variant in terms of refrigeration cycle design and control.

A suction gas superheater, Figure 2 (middle) is usually used when some superheating of the refrigerant is required before compression e.g. R290 needs higher superheating for a safe compressor operation. After the refrigerant has condensed on the high-pressure side, the liquid is cooled down by the suction gas heat exchanger to heat the suction gas before entering the compressor. For small capacities a tube-in-tube heat exchanger is usually installed for this purpose.

In the refrigeration cycle configuration EVI (Figure 2 right), compared to the single stage configuration, a compressor with injection port, a second expansion valve (injection valve) and an economizer are provided. The compressor design is divided into two compressor stages by the injection of refrigerant. The pressure level of the injection is fixed by the position of the injection port on the compressor. When controlling the injection valve upstream of the economizer, care must be taken to ensure that the compression cycle remains within prescribed limits. The main focus is on sufficient hot gas overheating at the compressor outlet (avoidance of compression into the wet steam area, oil separation in the hot gas) and to improve the efficiency at operating conditions where high temperature lifts between heat source and heat sink occur e.g. Air-Water Heat Pump operation at low out door temperatures. The main benefit of the EVI configuration in terms of efficiency is due to the fact, that not all the refrigerant is expanded to the evaporator pressure. A certain fraction is expanded to intermediate pressure level, then evaporated in the economizer and injected back to into the compressor. As a result, the compressor work is slightly reduced depending on the present pressure ratio. The higher it is, the higher is the benefit by the EVI configuration. A second benefit arises from the fact that the during injection on intermediate pressure level, the discharge temperature decreases. Depending on the refrigerant, a significant extension of the compressor operating envelope can be achieved.

6. Dymola Simulation

The refrigeration cycle simulations were done in the simulation environment Dymola/Modelica[3] based on the input parameters described in chapter 4 and 5.

6.1. Variants

Based on the determined use cases for different multi-family house categories (new, unrenovated existing, renovated existing), the specified refrigerant cycle variants and the selected refrigerants the simulation variants have been defined (Table 2). In total 24 different variants have been specified for the numerical comparisons.

Table 2: Overview of the investigated variants

For the new building category, a total of six different variants were compared: 1. Single stage refrigerant cycle R290 2. With suction gas heat exchanger R290 3. EVI (Enhanced Vapour Injection) R290 4. Single stage refrigerant cycle R410A 5. With suction gas heat exchanger R410A 6. EVI (Enhanced Vapour Injection) R410A	Nine different variants have been identified for each of the categories refurbished and unrefurbished building: 1. Single stage refrigerant cycle R290 2. With suction gas heat exchanger R290 3. EVI (Enhanced Vapour Injection) R290 4. Single stage refrigerant cycle R134a 5. With suction gas heat exchanger R134a 6. EVI (Enhanced Vapour Injection) R134a 7. Single stage refrigerant cycle R1234zee 8. With suction gas heat exchanger R1234zee 9. EVI (Enhanced Vapour Injection) R1234zee

6.2. Modelling

In order to compare and evaluate different refrigeration circuit configurations, numerical models were created within the afore mentioned simulation environment Dymola/Modelica. For the thermal modelling of the heat pump components the TIL library[4] was used, which is widely used in refrigeration engineering. The modelling of the compressor - the core component in heat pumps - was performed using measured data and an efficiency-based approach. The compressor is described by the isentropic and volumetric efficiency, which change depending on the pressure ratio and the condensing or evaporating temperature, as well as the compressor speed. The isentropic efficiency is based on the ideal isentropic compression and also takes into account suction and discharge pressure losses, internal leakage, re-expansion of the compressed refrigerant and mechanical friction. The volumetric efficiency describes the ratio between the geometrically possible mass flow and the actual mass flow through the compressor.

The numerical model is shown in Figure 3. The generic structure makes it possible to cover both the single stage refrigeration circuit and the more complex variants with EVI by simply activating or deactivating individual components (such as suction gas superheater, EVI injection, economizer) with a single model.

[3] https://www.3ds.com/de/produkte-und-services/catia/produkte/dymola/
[4] https://www.tlk-thermo.com/index.php/de/software/til-suite

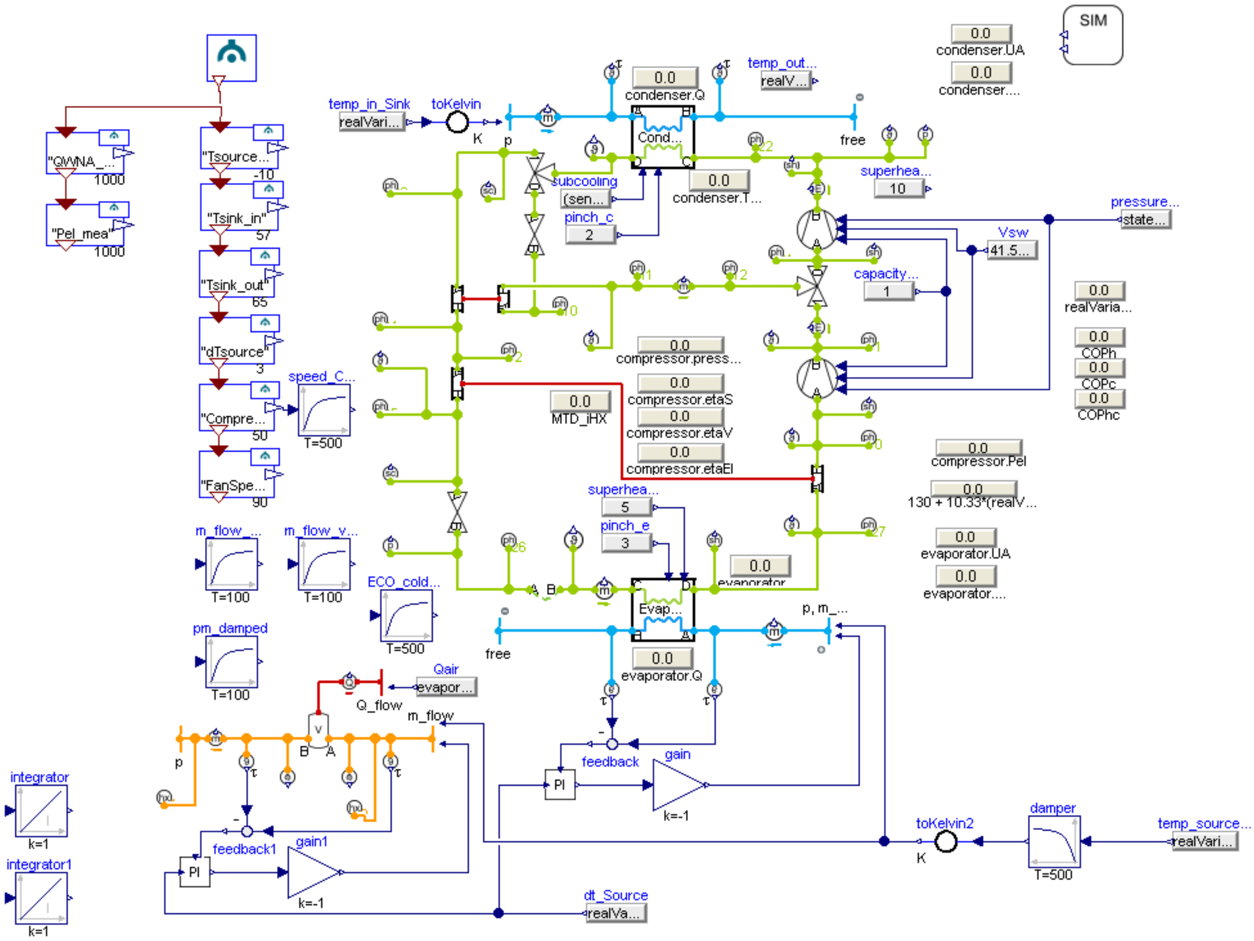


Figure 3: Numerical model in /Dymola/Modelica®

6.3. Model validation

In the first step, the completed model was calibrated with the measurement data of the GreenHP-prototype unit (Figure 4).

Figure 4: Green HP prototype (GreenHP, 2016)

The refrigeration cycle configuration of the GreenHP-prototype unit is equipped with EVI as shown in Figure 2. The refrigerant used is R290 (propane) and the unit is operated with a scroll compressor. The calibration is mainly aimed at the compressor characteristics, which have a significant influence on the overall efficiency and performance of the refrigeration circuit. Table 3 shows the results of GreenHP-measurements based on EN 14825 (EN 14825, 2018), which means part load tests at different outdoor temperatures and supply temperatures. From the entire measurement campaign, seven test points were used for the model calibration.

Table 3: Results of the GreenHP-measurements based on EN14825

	average heating capacity	average power input	Compressor speed	Fan speed	EVI valve operation	**COP**
	[kW]	[kW]	[%]	[%]	[-]	[-]
A-7/W52	25.07	11.42	92	82	auto	**2.19**
A2/W42	15.71	4.51	44	70	auto	**3.48**
A7/W36	10.66	2.61	27	64	auto	**4.09**
A12/W32	11.15	2.32	25	60	auto	**4.80**
A7/W55	19.96	6.67	50	64	auto	**2.99**
A7/W55 [1]	19.54	6.51	48	62	auto	**3.00**
A-10/W55 [2]	29.00	14.36	-	-	-	**2.02**

[1] measured with higher air humidity at the beginning of the icing of the evaporator

[2] calculated based on compressor data

The isentropic and volumetric efficiency of the compressor were approximated with 2nd degree polynomials depending on pressure ratio and speed. Figure 5 shows a comparison of the measured data and the simulation. In this diagram, the measured values are plotted on the x-axis and the simulated comparison values on the y-axis. If the points are congruent on a 45 °C straight line, then one speaks of a good model quality over the entire operating range.

The heating power and the electrical power consumption show a good agreement. The largest deviation is about 5 %. The coefficient of performance (COP) of the heat pump, which relates the heating output and the electrical power consumption, shows a difference of around 15% at the operating point A12/W32, i.e. at high source inlet temperatures and low user outlet temperatures. Except for this one operating point, however, the difference is of the same order of magnitude as for the heating power and the electrical power consumption.

The larger deviation for A12/W32 is an indication that not all influencing variables have been captured in the compressor modelling. As described above, the pressure ratio and the speed were taken into account. These two variables are sufficient for the majority of the operating points. For a more accurate model, more influencing variables, such as evaporating temperature, superheat, etc. would have to be included in the numerical compressor characteristics.

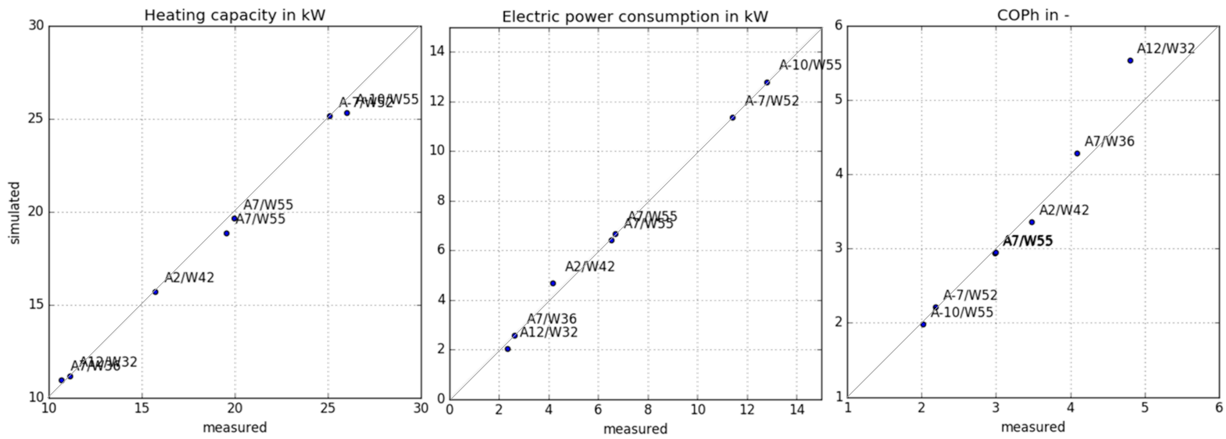


Figure 5: Validation of the numerical model with the GreenHP-measurement data

As a result of the model calibration, so-called compressor maps are available which describe the volumetric and isentropic efficiency as a function of pressure ratio and speed. Figure 6 and Figure 7 show the mentioned compressor maps. On the left, an e axisymmetric view is shown of the polynomial surface for the volumetric efficiency and the isentropic efficiency. On the right, the same information is shown as contour plot with distinctive isolines. The blue dots refer to the “GreenHP” measurement data.

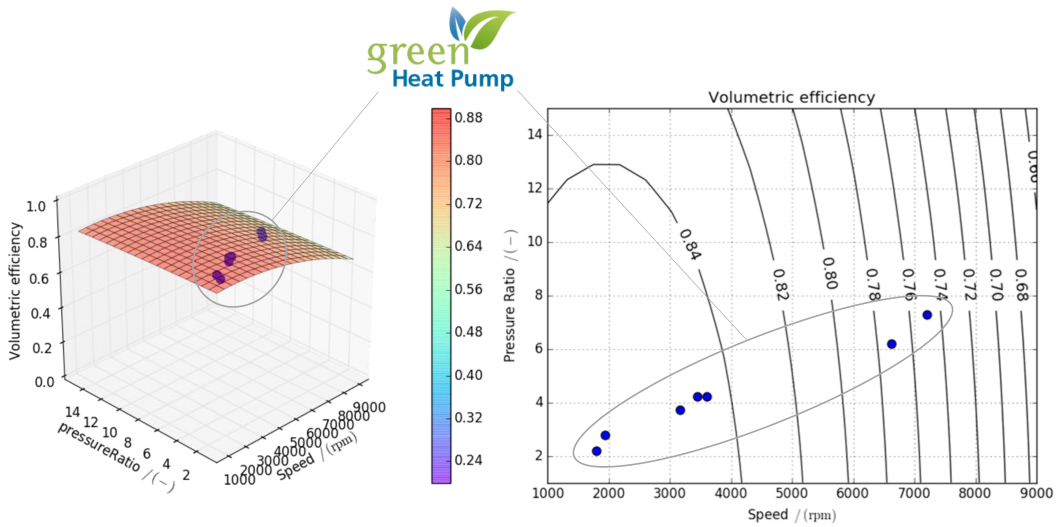


Figure 6: Calibrated volumetric efficiency with GreenHP-measurement data (blue)

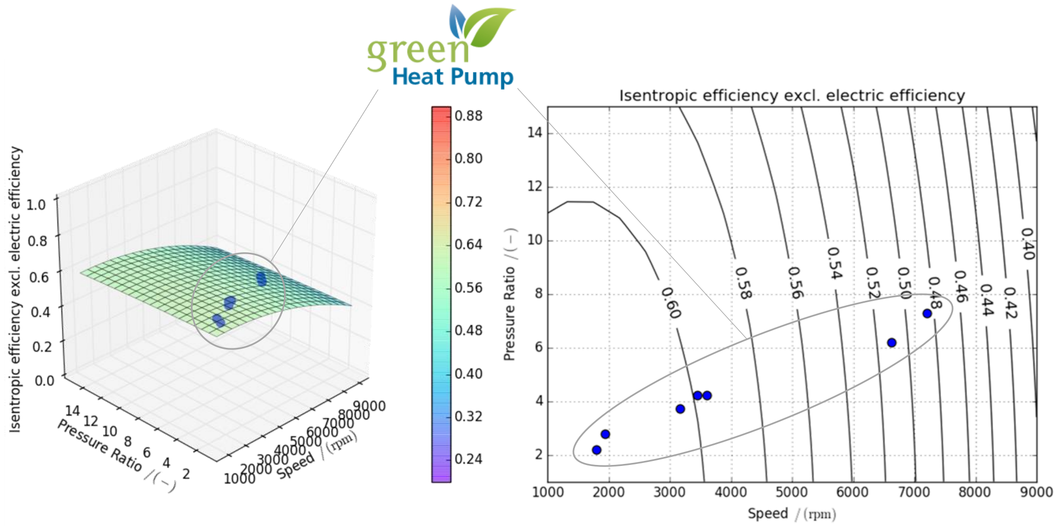


Figure 7: Calibrated isentropic efficiency with the GreenHP-measurement data (blue)

For both maps it is true that the GreenHP-measurement data - which are also the support points for the 2nd degree polynomial - show a distinct distribution with respect to the speed but give little information about the influence of the pressure ratio. Along the GreenHP operating points the modelled compressor characteristics can represent the measured data very well, but beyond that - and especially with deviating pressure ratios, less information is available from the measurements. In order to compensate, we took information from literature on how the compressor characteristics change with varying pressure ratios. In (Granryd et al. 2005) a correlation for the pressure difference and the isentropic and volumetric efficiency is given, see Figure 8 in the middle (literature). With this information it was possible to create a compressor map as seen in Figure 8 (right), that better reflects the influence of the different refrigerants than the current underlying map, see Figure 8 (left). The polynomial surface is now fitted with the constructed support points over the entire range (the red ones) rather than from the limited GreenHP measurement data (the blue ones).

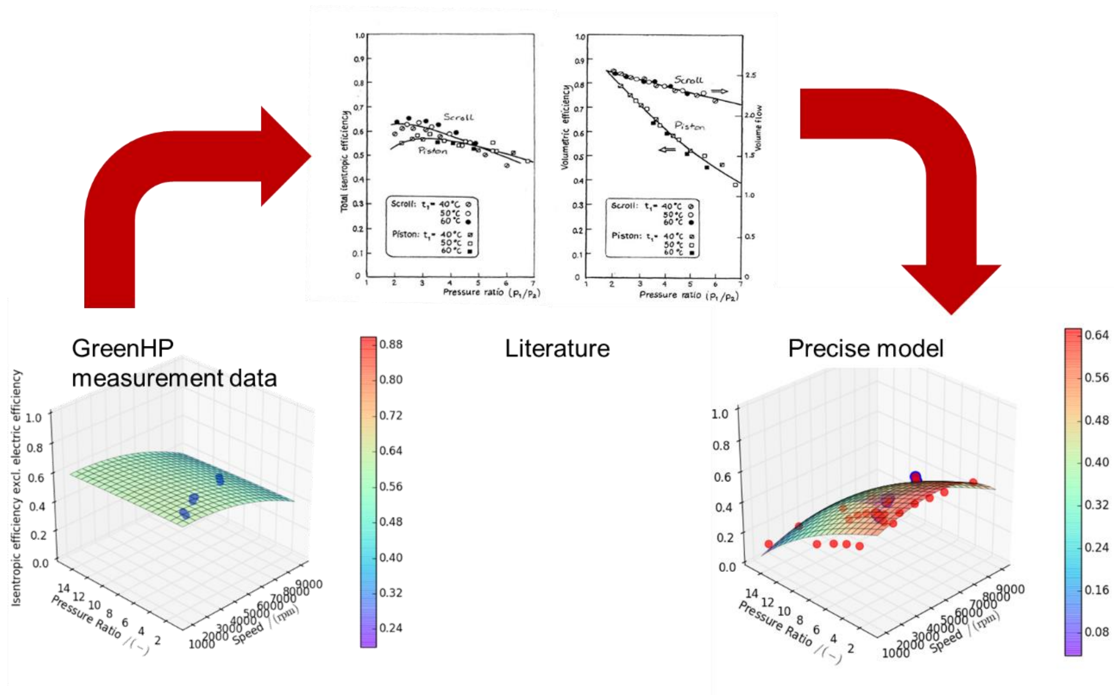


Figure 8: Generation of an improved compressor maps: GreenHP-measurement data combined with literature.

Figure 9 shows the improved compressor maps. While the volumetric efficiency has a rather low dependency on the pressure ratio, the isentropic efficiency shows significant decrease at higher pressure ratio values. The following results are based on the updated map in order to get a better understanding of the influence of the different refrigerants and temperature levels.

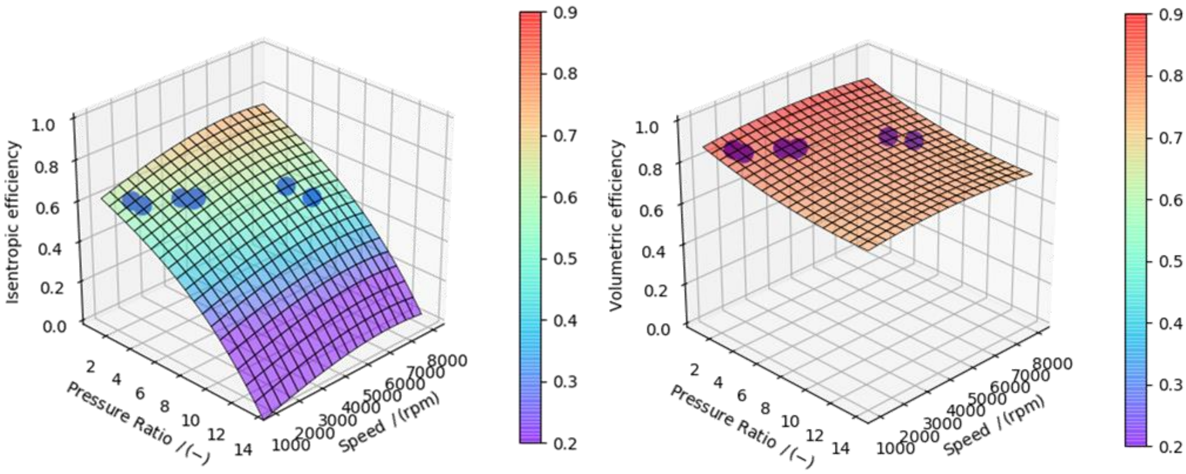


Figure 9: Improved compressor maps. Isentropic efficiency (left) and volumetric efficiency (right). The original GreenHP-measurement data are indicated as blue dots

6.4. Simulation results

The 24 defined variants were simulated in 29 defined operating points in accordance with EN 14825 using the model that was created and validated. Table 4 shows operating points as a function of temperature and compressor speed. For each operating point, the heat output, the electrical power consumption of the fan, the coefficient of performance (COP), the electrical power of the heat pump and the ratio of the compressor displacement compared to the design variant were calculated (area marked red in the table).

Table 4: Calculated operating points

	Case	T source in	T sink out	Compressor speed	FanPower	Heating capacity	Pel	COPh	Size
	-	°C	°C	%	W	kW	kW	-	-
0		-10	55	25.0					
1		-10	55	35.0					
2		-10	55	54.0					
3		-10	55	88.0					
4		-10	55	100.0					
5		-7	52	25.0					
6		-7	52	35.0					
7		-7	52	54.0					
8		-7	52	88.0					
9		-7	52	100.0					
10		2	42	25.0					
11		2	42	35.0					
12		2	42	54.0					
13		2	42	88.0					
14		2	42	100.0					
15		7	36	25.0					
16		7	36	35.0					
17		7	36	54.0					
18		7	36	88.0					
19		7	36	100.0					
20		12	32	25.0					
21		12	32	35.0					
22		12	32	54.0					
23		12	32	88.0					
24		12	32	100.0					
28		-7	52	89.3					
27		2	42	42.0					
26		7	36	25.0					
25		12	32	25.0					
29		-10	55	99.9					
29		-10	55	99.9					

Based on the operating points explained in Table 4 an operating map for each heat pump variant was generated. In Figure 10 the operating map for the refrigeration cycle “EVI with R290” is shown as an example for the variant existing refurbished MFH. The green line represents the heating curve according EN14825 and the red dots show the operating points which are needed for the later SCOP calculations.

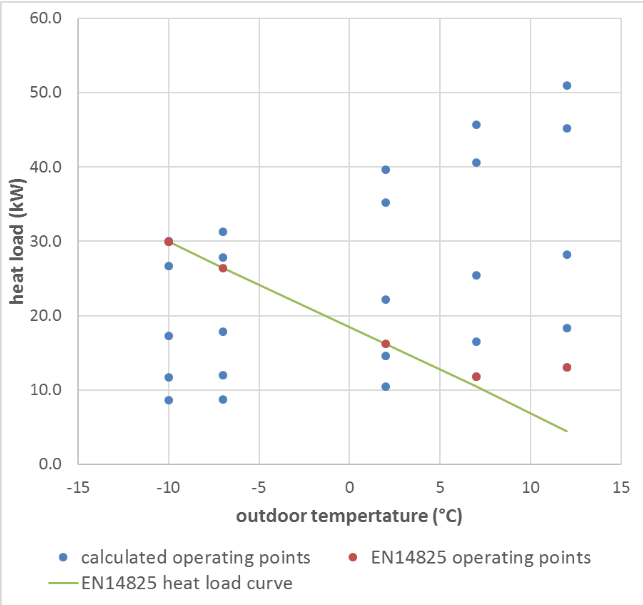


Figure 10: Operating map example for the variant refurbished MFH with EVI and R290

Figure 11, Figure 12 and Figure 13 show the results of the numerical comparison for the new building, the refurbished existing building and the unrefurbished existing building. The bars represent the partial load points as defined in EN14825. The points show the results of compressor speed variations in the range between 25 % and 100 %. Each refrigerant belongs to a certain colour palette. Green for R290, red for R410A, orange for R134a and blue for R1234zee. The first configuration is the EVI variant followed by single stage and suction gas heat exchanger iHX.

The comparison for all building categories shows that the EVI configuration is the best option in terms of COP for all selected refrigerants, except for R290 at 12 °C outside air temperature within the building category new and for R1234zee within the building categories refurbished and unrefurbished. In these cases, the EVI injection has been deactivated in the model because of the low pressure ratio.

The difference between the single stage variant and the variant with suction gas heat exchanger is hardly or not at all noticeable, depending on the refrigerant. This is mainly due to the alignment of the isentropes and the saturated vapour curve in the log pH diagram.

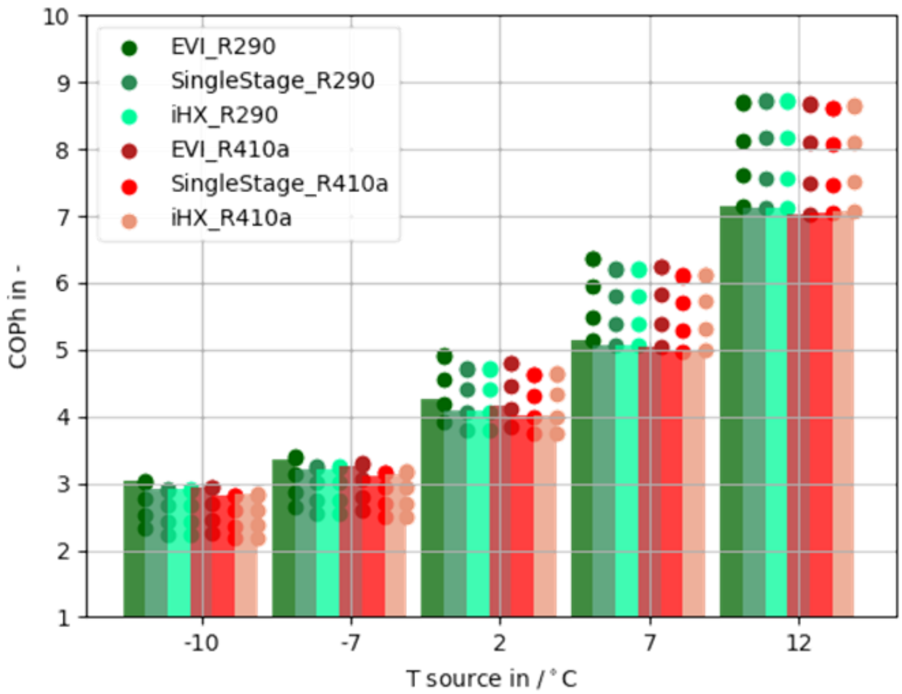


Figure 11: Results for the new building

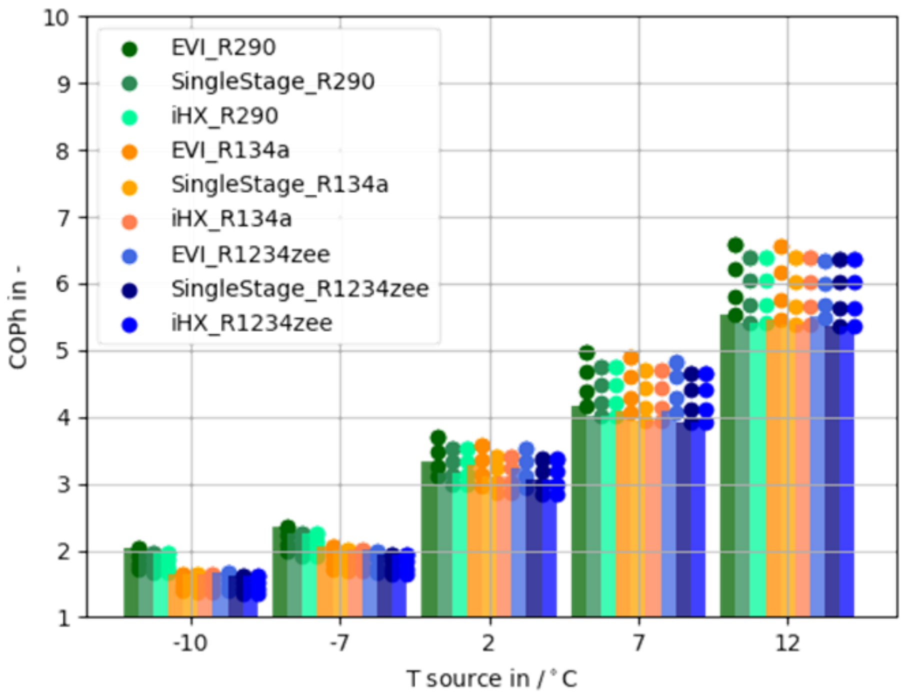


Figure 12: Results for refurbished existing buildings

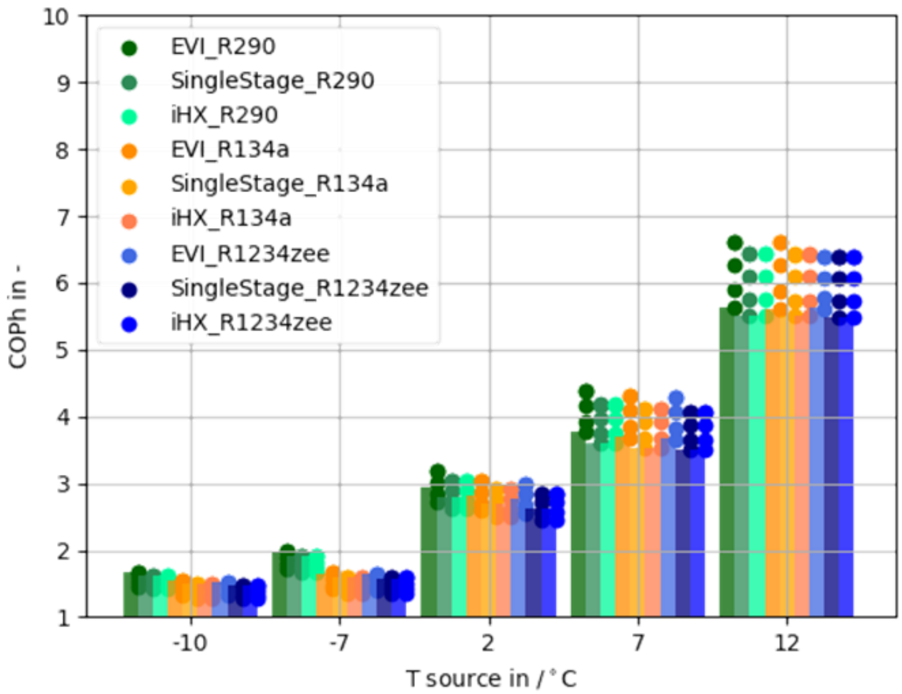


Figure 13: Results for unrefurbished buildings

For the building category “new”, the EVI variant with R290 as refrigerant has the highest COPs. This is most evident for the outside air temperatures of 2 °C and 7 °C. At an outside air temperature of 12 °C the EVI variant with R410A is the most efficient.

For “refurbished” and “unrefurbished” existing buildings, the EVI variants also have the highest COPs - regardless of the refrigerant. At an outside air temperature of 12 °C, the values for the unrefurbished and the refurbished existing building are in the same range, slightly higher for the unrefurbished category. Following the EN14825 operating points, the delivery temperature at 12 °C outside air temperature are almost identical for both building categories. The higher delta T at the supply side for the unrefurbished category results in a more favourable operating point in the unrefurbished stock than in the refurbished one.

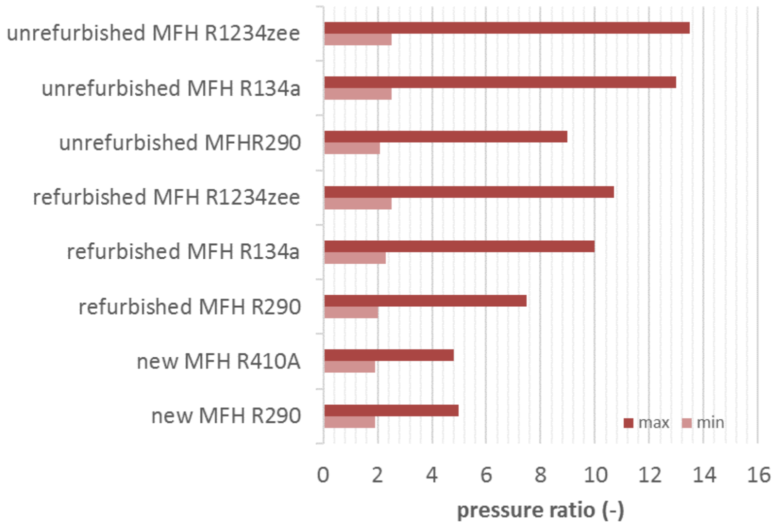


Figure 14: Pressure ratio depending on refrigerant and use case

As described in Figure 9 the isentropic efficiency is dependent on the pressure ratio, therefore the pressure ratio also influences the efficiency. In Figure 14 the pressure ratios depending on the refrigerant and use cases is shown. It can be seen that for the higher temperature applications, R290 has the most favourable pressure ratios, for the new MFH application R410A is slightly better than R290 (Figure 18).

In addition to the performance of the different variants the compressor displacement was analysed. Depending on the use case and the refrigerant, different compressor displacements were needed to reach the design heating capacities. The compressor displacement is also an indicator for the refrigerant mass needed to operate the unit and how compact the components of the unit can be designed.

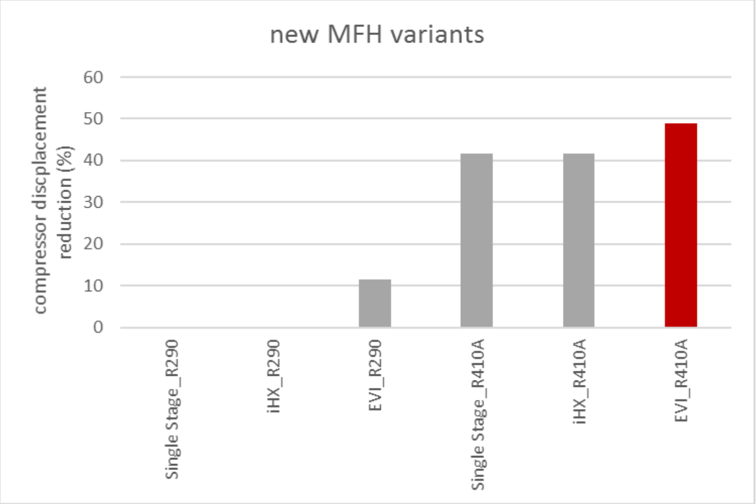


Figure 15: Compressor displacement reduction for new MFH variants

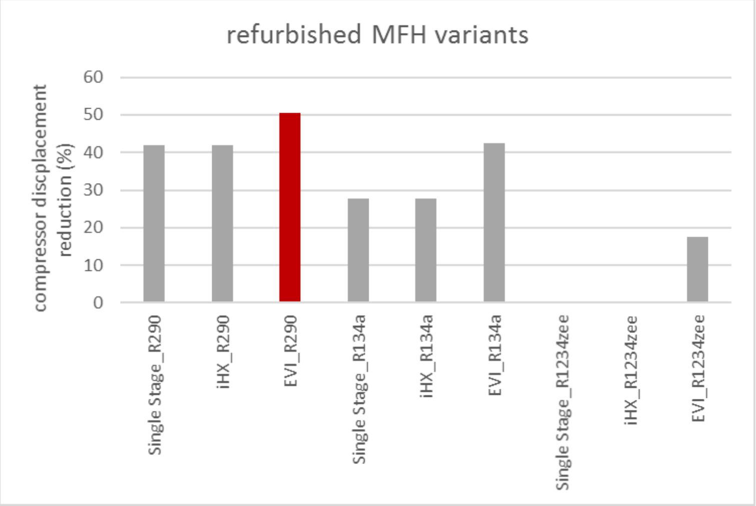


Figure 16: Compressor displacement reduction for refurbished MFH variants

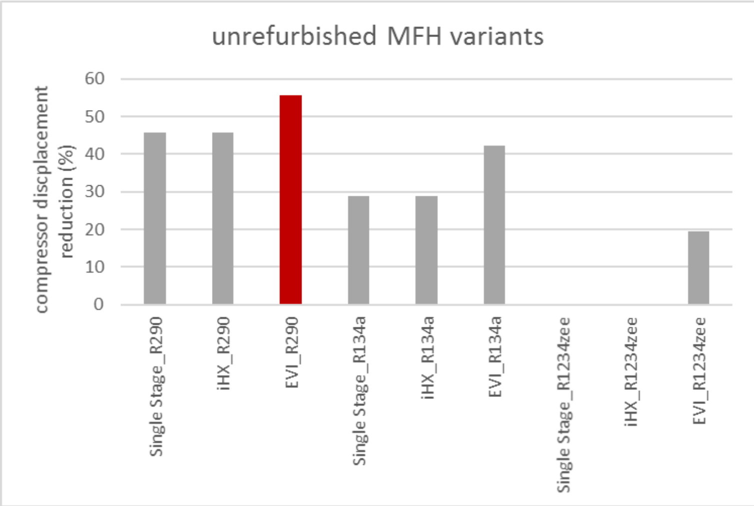


Figure 17: Compressor displacement reduction for unrefurbished MFH variants

In Figure 15, Figure 16 and Figure 17 the compressor displacement size reduction is shown. The basis for the reduction potential is always the worst case (highest displacement) per use case. For the high temperature applications R290 with EVI offers a reduction potential of more than 50 % compared to R1234zee with the single stage cycle. For all refrigerants it can be seen that only the use of EVI can reduce the compressor displacement by more than 10 % in comparison with the single stage cycle. Only for the new MFH use case R410A has a higher size reduction potential than R290.

7. SCOP Calculation

In addition to the criteria discussed in the previous chapters, such as GWP and refrigerant charge quantity, the most efficient of the 24 variants was calculated for the three use cases new, refurbished and unrefurbished MFH based on SCOP calculations. The SCOP represents the seasonal efficiency according EN14825 and can be defined as calculated seasonal performance factor. In contrast to the COP, not only the efficiency of a single operating point is considered here, but the entire operating behaviour of a season.

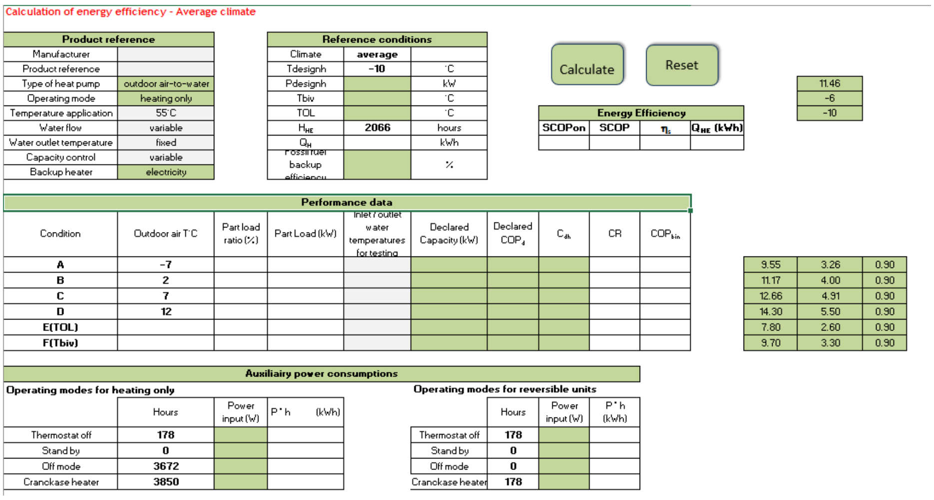


Figure 18: SCOP calculation template (HP-Keymark, 2020)

Figure 18 shows the input template for the SCOP calculation according to EN14825. For calculating the SCOP the results from the numeric calculation described in Figure 10 are needed for the single variants. The input parameters are P_design and the performance data at the specified part load conditions declared in EN14825, the template then calculates the SCOP and eta_s.

In Figure 19, Figure 20 and Figure 21 the results of the SCOP calculations are shown. Generally, the variants with EVI lead to higher SCOPs than the single stage cycle or the cycle with suction gas heat exchanger. For all use cases the highest seasonal efficiency can be reached by using R290 as refrigerant in combination with the EVI refrigeration cycle (marked red). Therefore, this combination of refrigerant and refrigeration cycle (R290 plus EVI) were selected und used for generating the characteristic curves per use case.

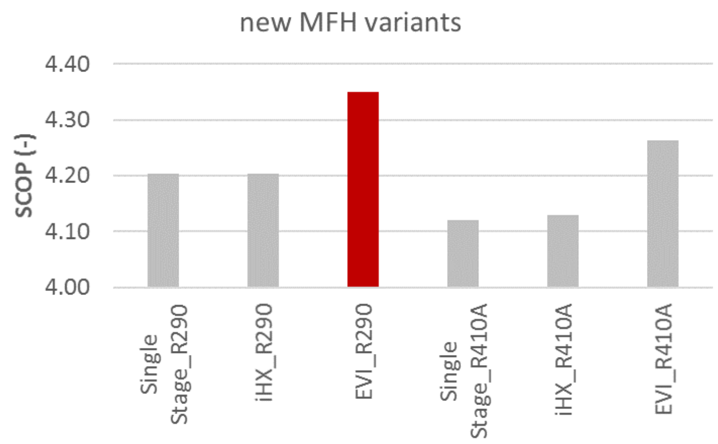


Figure 19: SCOP results for new MFH variants

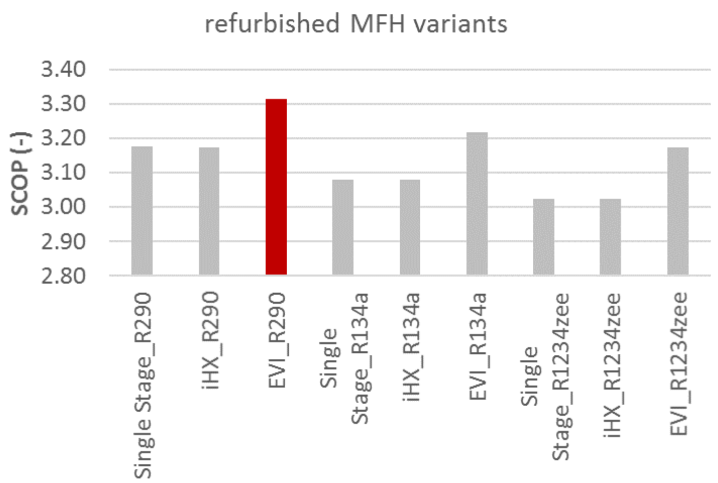


Figure 20: SCOP results for refurbished MFH variants

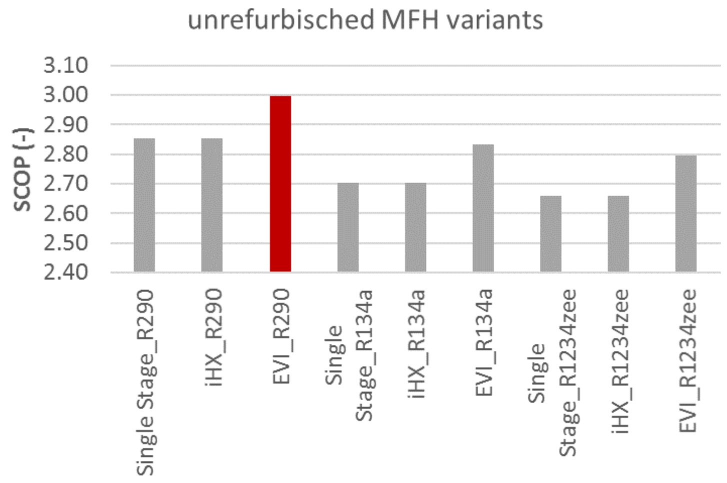


Figure 21: SCOP results for unrefurbished MFH variants

8. Characteristic curves per use case

For the further use of the results of the refrigeration cycle simulations, the EVI configuration with R290 as refrigerant is exported by means of regression analysis in order to use the information in the following TRNSYS system simulations (Task 3.2).

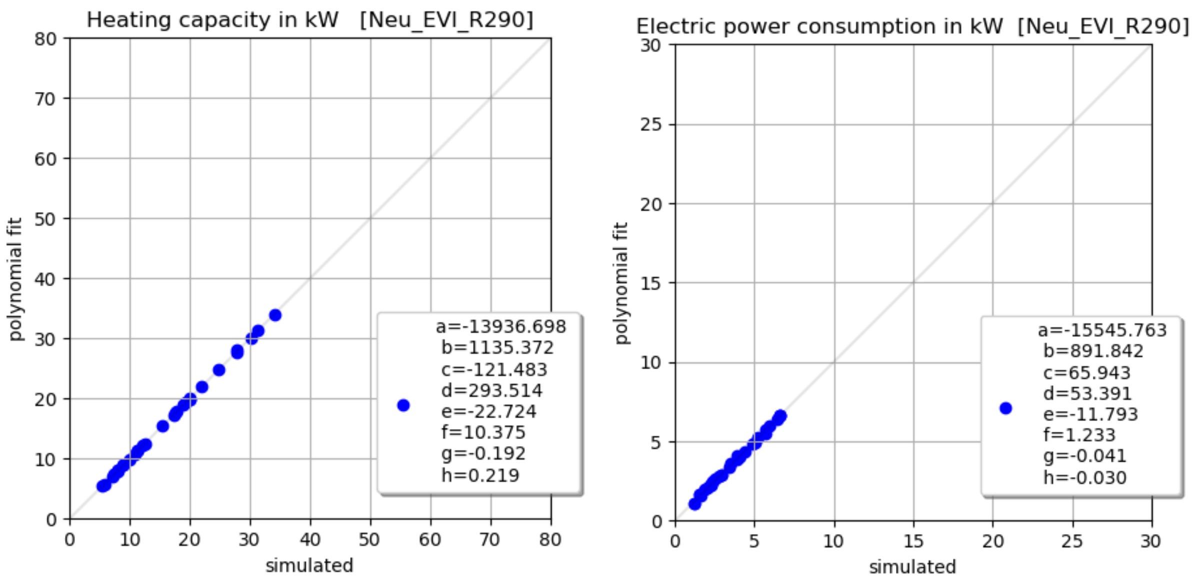
These characteristic curves will be used for the parameterisation of the TRNSYS heat pump model. A regression model is used which gets "trained" by the output data from the Dymola/Modelica model. For the inherently nonlinear characteristics of the heat pump system, a polynomial regression (for example, also used for trendline fitting in Microsoft Excel) is the most promising method. The independent values – also referred as "features" in the literature - are:

- x….T sink out °C → Delivery temperature
- y….T source in °C → Outside air temperature
- z….Compressor speed in % → Compressor speed

The structure of the polynomial is as follows**:**

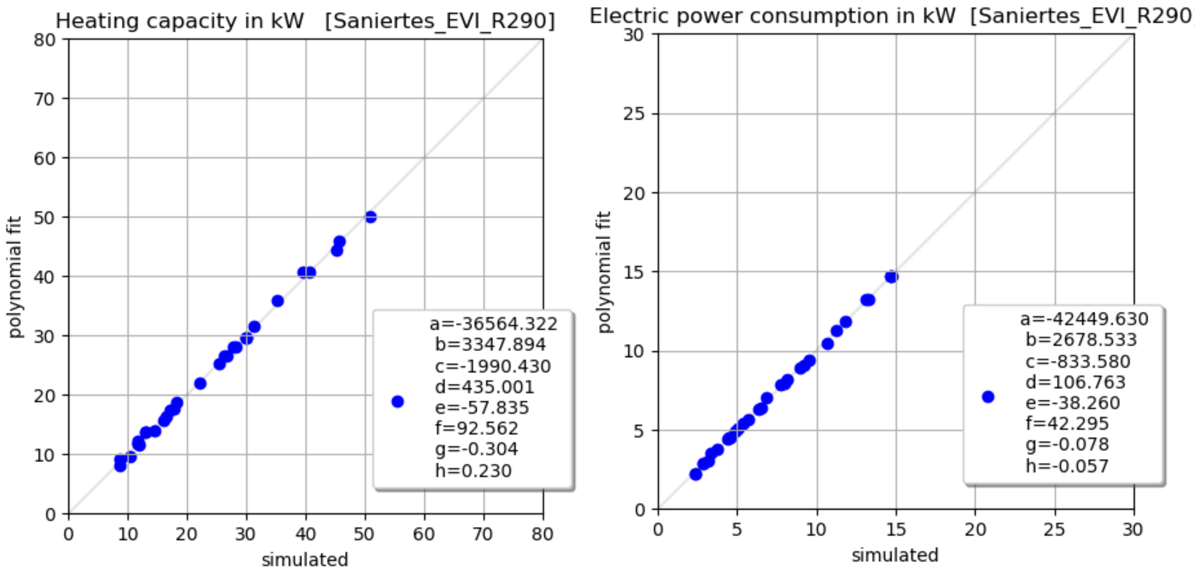
Heating capacity kW/Electric power consumption kW = $a + b \cdot x + c \cdot y + d \cdot z + e \cdot x^2 + f \cdot y^2 + g \cdot z^2 + h \cdot x \cdot y \cdot z$

In the following figures (Figure 22 to Figure 24), a comparison between the regression result and the values of the physical model from Dymola/Modelica simulation environment is given for EVI R290 and each of the three building categories. The "predicted" values are the heating capacity in kW (left) and the electrical power consumption in kW (right). A good regression quality is indicated, when the majority of the points are located on a 45 °C straight line. The values of the corresponding coefficients (a to h) for the polynomial are shown in the figure legend.



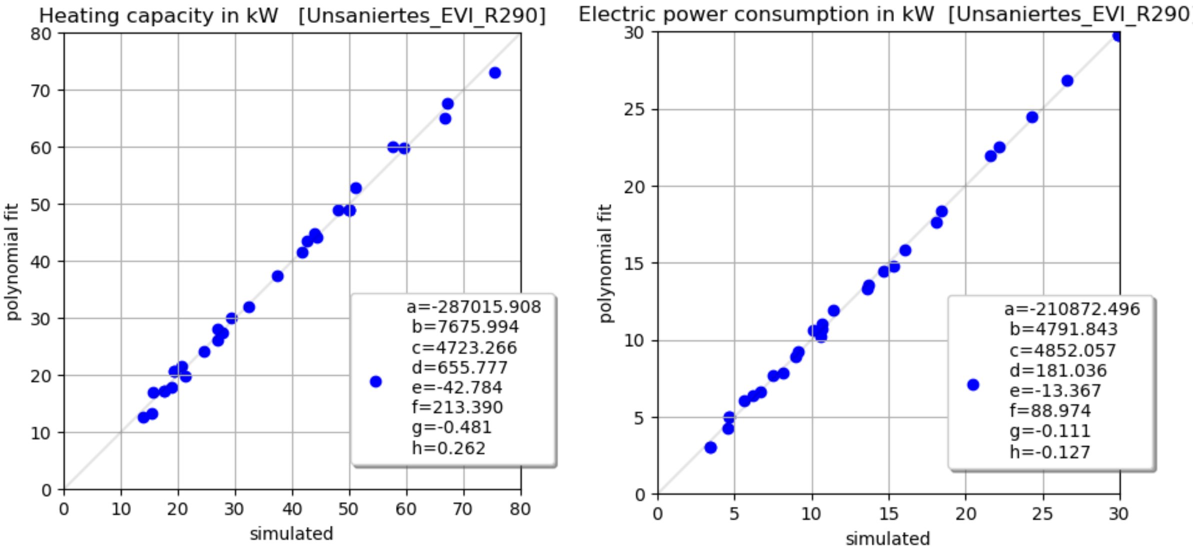
Feature	Lower	Upper
T sink out °C	24	35
T source in °C	-10	12
Compressor speed in %	25	100

Figure 22: Polynomial fit vs. Dymola/Modelica simulation for EVI R290: New building.



Feature	Lower	Upper
T sink out °C	30	55
T source in °C	-10	12
Compressor speed in %	25	100

Figure 23: Polynomial fit vs. Dymola/Modelica simulation for EVI R290: Refurbished building.

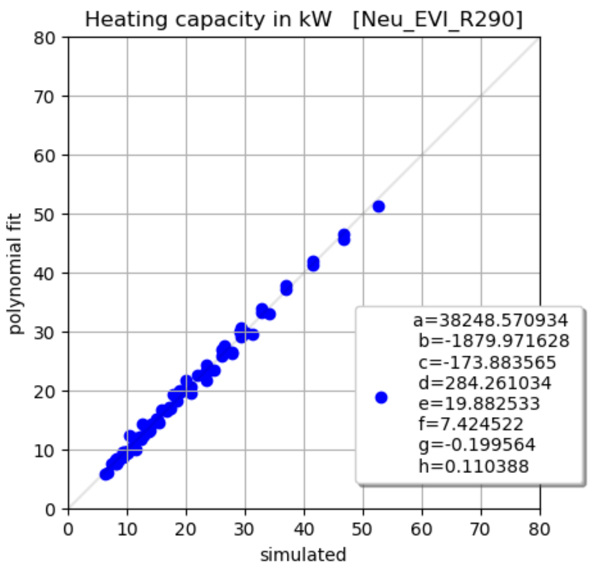


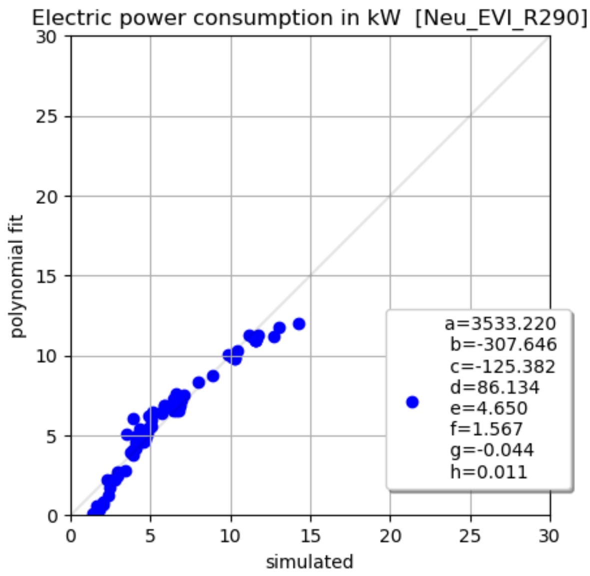
Feature	Lower	Upper
T sink out °C	32	65
T source in °C	-10	12
Compressor speed in %	25	100

Figure 24: Polynomial fit vs. Dymola/Modelica simulation for EVI R290: Nonrefurbished building.

Each of the given polynomials in Figure 22 to Figure 24 has its own range of validity for the independent values. This range originates from the EN14825 which covers the temperature levels for space heating operation. Especially for polynomial functions it is important to keep the validity limits, because extrapolating can lead to completely wrong results due to the higher order terms. The incorporated table in the figures indicates the range of validity for the independent values. The distribution of the operating points (blue dots) reveal, that the characteristic curves by means of regression analysis show a good agreement with the simulated values for all three building categories.

For Task 3.2 the characteristic curve had to be prepared for a new building where additionally to space heating the domestic hot water operation is conducted by the heat pump. Therefore, the characteristic curve also needs to cover the domestic hot water operating points. Here, the regression analysis still produces a feasible result, but due to the wider validity range, the quality of the fit decreases slightly. Especially for the electrical power consumption at low values, the regression model underestimates the prediction. However, as seen from the perspective of a yearly simulation, most of the operating points are in a range where the regression model predicts the values reliably.





Feature	Lower	Upper
T sink out °C	32	65
T source in °C	-10	12
Compressor speed in %	25	100

Figure 25: Polynomial fit vs. Dymola/Modelica simulation for EVI R290: New building with domestic hot water included.

9. Conclusion

In Task 3.1 optimized refrigeration cycle configuration for Air-Water Heat Pumps for new, refurbished and unrefurbished MFH buildings are identified and the most efficient configuration for the respective building category is determined by means of dynamic circuit calculations in Dymola/Modelica and SCOP calculations.

For the thermal modelling of the heat pump components the TIL library[5] was used, which is widely used in refrigeration engineering. The numerical model has a generic structure, that makes it possible to cover both the single stage refrigeration circuit and the more complex variants with EVI by simply activating or deactivating individual components (such as suction gas superheater, EVI injection, economizer) with a single model. The completed model was calibrated and validated with the measurement data of the GreenHP-prototype unit.

With the model it was possible to simulate the three different refrigeration cycles (single stage, iHX, EVI) in combination with the selected refrigerants (R290, R410A, R134a and R1234zee) at different part load conditions, supply- and air temperatures. Thus 24 defined variants were calculated. Based on SCOP calculations, the most efficient refrigerant cycle variants together with the most suitable refrigerant were selected. It turned out that that for all applications (new, refurbished and unrefurbished MFH) the refrigeration cycles with the refrigerant R290 had the highest SCOP. Especially for the higher temperature applications the refrigeration cycle variants with EVI lead to the highest SCOP. For all use cases the refrigeration cycle with EVI in combination with the refrigerant R290 were

[5] https://www.tlk-thermo.com/index.php/de/software/til-suite

selected in order to reach the highest SCOP. In Table 5 the SCOP of the selected variants are summed up.

Table 5: SCOP for selected MFH use cases

use cases - MFH	heat load	system temperature	variant	SCOP
-	kW	°C	-	-
new	20	35	EVI_R290	4.35
refurbished existing	30	55	EVI_R290	3.32
unrefurbished	50	65	EVI_R290	3.00

Beside the efficiency R290 offers the possibility to reduce the amount of refrigerant mass due to its thermodynamic properties. It turned out that for the higher temperature applications, compared to the selected refrigerants, R290 has the lowest and thus most favourable pressure ratios, as the isentropic efficiency is dependent on the pressure ratio and therefore influences the efficiency. In addition, the compressor displacement was analysed, as the compressor displacement is also an indicator for the refrigerant mass needed to operate the unit. For the high temperature applications R290 with EVI offers a reduction potential of more than 50 % compared to R1234zee with the single stage cycle. For all refrigerants it can be seen that only the use of EVI can reduce the compressor displacement by more than 10 % in comparison with the single stage cycle.

Based on the selected refrigeration cycle variants, characteristic curves by means of regression analysis per use case were generated. These characteristic curves are used for the parameterisation of the TRNSYS heat pump model in Task3.2.

References

Bitzer, "Bitzer Kältemittel-Report 20", https://www.xn--bitzerkltemittelreport-64b.com/fileadmin/user_upload/A-500-20.pdf, 20. Aufl., 2018

EN 14825, "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", EN standard, 2018

Granryd, E.; Ekroth, I.; Lundqvist, P.; Melinder, A.; Palm, B.; Rohlin, P. (2005): Refrigerating Engineering. 5. Aufl.

GreenHP, "GreenHP – Next Generation Heat Pump for Retrofitting Buildings", European funded project under FP7 (GA 308816), 2012 – 2016, https://cordis.europa.eu/project/id/308816 (coordinated by AIT), 2016

Honeywell, "Solstice® ze Refrigerant (HFO-1234ze (E)) – the environmental alternative to traditional refrigerants", https://www.honeywell-refrigerants.com/europe/wp-content/uploads/2018/11/4651_SolsticeZe_Brochure_06Nov.pdf, brochure, 2019

HP-Keymark, "SCOP_SEER_water based HP_V1.xlsm ", excel calculation tool, https://drive.google.com/file/d/1ZnwihJ1yBOLTJnjdzWEeuJE6oB07P9Bl/view?usp=drivesdk, 2020

IEA HPT Annex 50 Task 1, "Task 1 - Market Overview", Autrian Country Report, 2017, Austria

Store4Grid, "Optimierte Erdbecken-Wärmespeicher für Wärmenetze (Optimized earth basin heat storage for local heating networks)", Austrian national funded research project (FFG No. 838664), 2013 – 2015, https://www.energieforschung.at/projekte/753/optimierte-erdbecken-waermespeicher-fuer-nahwaermenetze, 2015