A micro-heat pump combined with mechanical ventilation including heat recovery - simulation and in situ monitoring

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Abstract

An innovative heating and ventilation system is developed within the European fp7 project iNSPiRe for the deep energy retrofit of residential buildings. A low capacity heat pump of about 700W is combined with mechanical ventilation including heat recovery. The evaporator of the heat pump is located in the exhaust air flow of the mechanical ventilation system and the condenser heats further the supply air. The whole system is meant to be cost-effective, compact and thus, suitable to be integrated into a prefabricated timber façade. Several functional models were already measured in the laboratory of university of Innsbruck (AT) and finally one is installed in a demonstration building in Ludwigsburg (D). The building is a multi-family house with four stories and one flat per story. Within the refurbishment the heat pump with the mechanical ventilation is installed in one flat and designed to cover the heating demand of it; an additional electric radiator is installed in bathroom for comfort reasons. A detailed monitoring system is installed in the demo building to investigate the performance and the dynamic behavior of the HVAC system as well as the comfort in the flat. A simulation model of the system and flat is developed in Matlab/Simulink for dynamic simulations. The model is calibrated and validated against the monitoring data and is used for simulation based analysis of the performance.

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Keywords: micro-heat pump; façade integration; deep renovation; refurbishment of residential buildings; HVAC

1. Introduction

The majority of existing building stock in Europe and worldwide is poor energy performance buildings. Deep renovation solutions in combination with integrated heating, ventilation and air-conditioning systems (HVAC) are developed within the framework of the European project iNSPiRe [1]. Passive renovation measures are already known and used worldwide, e.g. EnerPHit standard [2]. The very low heating load of such buildings gives the opportunity to develop new compact and economic HVAC systems. The present study focuses on a system composed of a micro-heat pump (micro-HP) combined with a mechanical ventilation unit with heat recovery (MVHR), which is integrated into a timber frame prefabricated façade. The main advantages of the proposed system are the compactness (providing the possibility of integration into the façade) and cost reduction. A functional model including a monitoring system is installed within the refurbishment of a demonstration residential multi-family house in Ludwigsburg, Germany.

Analytical monitoring and simulation results for the building and the flat are presented in [3]. The present study focuses on the micro-HP. The HVAC system, the monitoring concept and the demonstration building are presented in section 2. Simulation models of the micro-HP and the MVHR are developed and further calibrated

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and validated against monitoring data (section 3). System and building annual simulations results are performed to investigate the performance of the HVAC system (section 4).

Nomenclature	
ϑ_{amb}	Ambient air temperature
ϑ_{exh}	Exhaust air temperature
ϑ_{ext}	Extract air temperature
ϑ_{sup1}	Supply air temperature after the heat pump condenser
$\dot{m}$	Air mass flow
c_p	Heat capacity of air
n_{MVHR}	Efficiency of the MVHR unit
P_{fan}	Power consumption of the electric fans
Q_{MVHR}	Heating energy supplied by the MVHR unit
$Q_{condenser}$	Heating energy supplied by the micro-heat pump
$Q_{post\ heater}$	Heating energy supplied by the post heater
$Q_{bath\ radiator}$	Heating energy supplied by electric radiator in bath
W_{el_MVHR}	Electrical energy consumption of the MVHR unit
$W_{el\_compressor}$	Electrical energy consumption of the micro-heat pump
$W_{el\_post\ heater}$	Electrical energy consumption of the post heater
$W_{el\_bath\ radiator}$	Electrical energy consumption of the electric radiator in bath

2. HVAC and monitoring system on the demonstration building

The micro-HP combined with MVHR system addresses very good building standards (e.g. EnerPHit 25 kWh/(m² a) [2] or better), corresponding to a specific heat load in the range of 10 W/m². Hence, the heat power of the heat pump is about 700 W. An additional radiator in the bathroom is suggested for comfort reasons. Basically, the concept would work also for water (radiator, floor heating, radiant ceiling) and air based systems (supply air and principally also recirculated air). As source ambient air and/or exhaust air or brine are possible. The exhaust air-to-supply air has the highest potential to be micro and thus compact. The proposed system is designed for hygienic volume flow rates i.e. between 90 m³/h and 120 m³/h (based on 25-30 m^3/(h·person)).

Figure 1 shows the hydraulic scheme of the unit. The ambient air (1) will be heated with the electric heater (5) (frost protection of the heat exchanger) when the ambient temperature is below -5 °C. The filter for the ambient air (6) is situated in front of the heat exchanger (16). The ventilator for the supply air (8) is situated after the heat exchanger. The supply air will be further heated due to the losses of the frequency controlled compressor (10) and then by the condenser (13) of the micro-heat pump. An additional heater (15) will heat the supply air (3) up to (in maximum) 52 °C in case the heat pump cannot cover the entire heating load. The extract air (2) is filtered (7) before the heat exchanger. The extract/exhaust air ventilator (9) is situated in the air flow of the exhaust air (4) in front of the evaporator (11). The expansion valve (12) reduces (as usual) the pressure between condenser and evaporator. A defrost cycle is implemented using a hot gas defrost bypass valve (14) which is used to bypass the condenser, so the hots gas of the compressor flows directly into the evaporator and the heat melts the ice that formed during operation.

The demonstration building is a multi-family house of the company “Wohnungsbau Ludwigsburg GmbH” (WB-L) built in 1971 in Ludwigsburg (Germany). The building consists of four stories and four flats: one small in cellar (39.5 m^2), two identical in ground and first floor (99 m^2) and one in attic floor (61.4 m^2), see Figure 2 right.

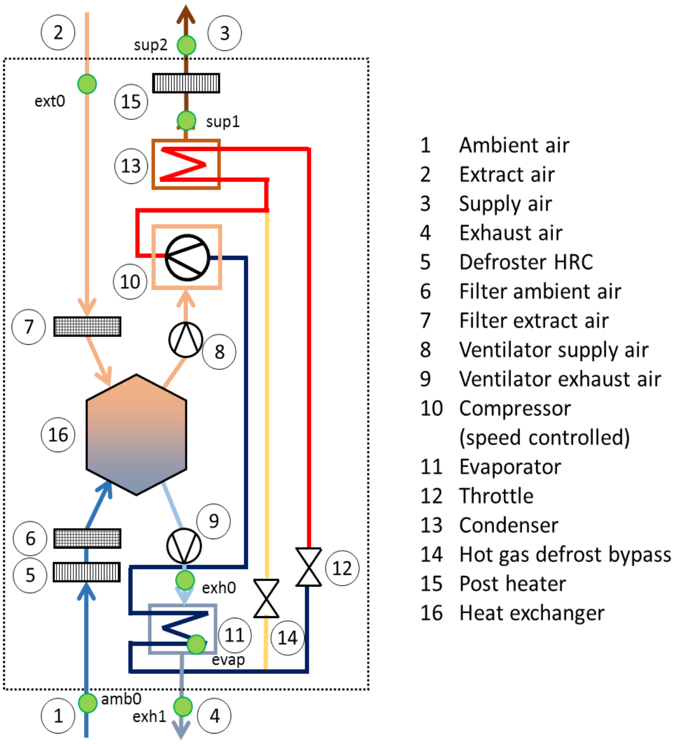

Fig. 1. Hydraulic scheme [3] of micro-HP and MVHR including the temperature sensors (green dots)

The building was renovated with prefabricated timber frame façade elements in passive house quality. The air-tightness was measured in 1.5/h via a blower door test, which does not meet the aimed target of 1/h but is still within the allowed range according to German energy standard (EnEV). All flats have a MVHR with a recovery efficiency in the range of 0.85. The flats in the ground floor and attic floor were occupied during the renovation, but the flats in the first floor and cellar were empty and were also refurbished inside. A central air-to-water heat pump (with a brine circuit) is used to cover space heating in all flats and for domestic hot water preparation. The central heating system is switched off in the flat in ground floor, where the micro-HP is installed (Figure 2 left). The floor plan and distribution ducts are shown in Figure 3. Since all extract air rooms are situated in the north, all extract air ducts are integrated into the prefabricated timber frame façade (the inlets are placed in the reveal of the windows). Thus only the supply air ducts are inside the flat minimizing the disturbance to the tenants during the renovation fostering the concept of a minimal disruptive renovation [4].

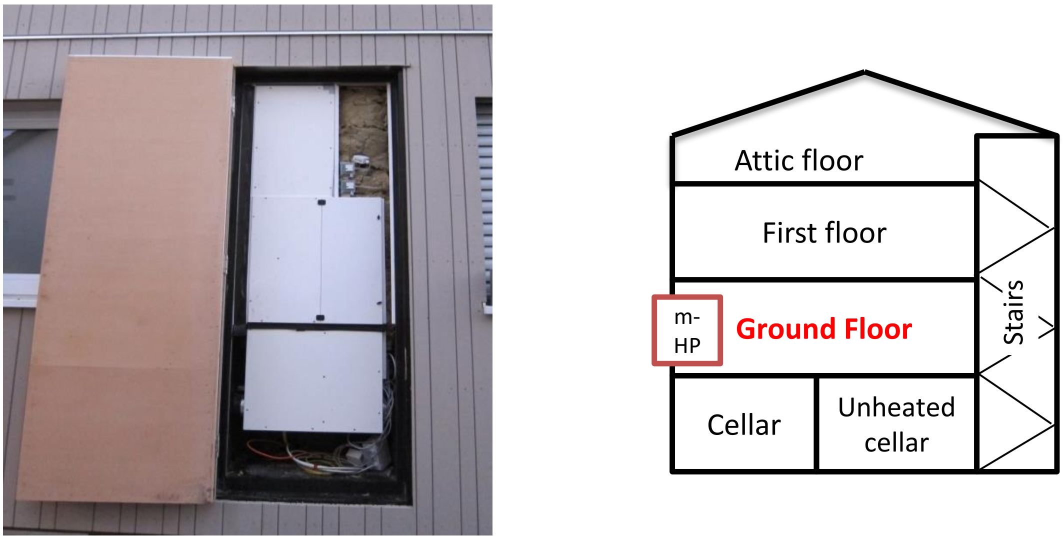

Fig. 2. Left: Outside view of the micro-HP and MVHR (company: Siko Energiesysteme) within the prefabricated timber frame façade (company: gumpp&maier). Right: simplified scheme of the demonstration building

Fig. 3. Floor plan of the renovated flat in ground floor with a simplified scheme of ducting for ventilation and heat distribution [5]

2.1. Monitoring system

A monitoring system was installed with the micro-HP and the MVHR. The position of the temperature sensors are also shown in Figure 1. In total, eight temperature sensors (PT1000 with an accuracy of $\pm 0.15+(0.0002\cdot\vartheta)$), one humidity sensor, two pressure difference sensors for measuring the volume flows, and three electricity meters (compressor, MVHR and post-heater) are used to monitor the dynamic behavior of the system. The measurement accuracy is similar to [6]. An additional temperature sensor is installed in the corridor of the flat (green dot in Figure 3) and is used for the control of the system.

The building is also separately monitored (comfort, heat and electricity consumption) before and after the renovation [3]. The flat in ground floor was renovated in December 2015, and the full monitoring started in

February 2016 until September 2016 (there might be an extension of one year more). Thus, within the project iNSPiRe, which ended in September 2016, data are not available for one whole winter.

3. Calibration and validation of the HVAC system

3.1. Simulation models

The simulation models are developed in Matlab/Simulink [7] with the additional Carnot blockset [8]. The MVHR is modelled using as input the heat recovery efficiency according to [9], see eq. (1), to calculate the heat transfer between the two air flows. Then the temperature and humidity of exhaust and supply air flows are calculated based on enthalpy balance.

For the micro-heat pump a dynamic performance map model was calibrated against measured data from laboratory tests. The power of the micro-HP ($P_{m\text{-}HP}$) and COP are modeled as functions of ambient temperature (or exhaust air temperature of MVHR) and the compressor frequency - see Figure 4. The performance map data are steady state test points measured at the laboratory of University of Innsbruck [10] using a constant air volume flow of 120 m^3/h. For sake of simplicity, the slightly increased flow in the Demo case (around 10 m^3/h) is not considered in the performance map data. The dynamic behavior of the heat pump i.e. the time duration to reach the steady state condition after switching on or off, is modelled using a transfer function with different time constants for start and stop function. The time constants for on/off cycling are similar to those of the defrost switches.

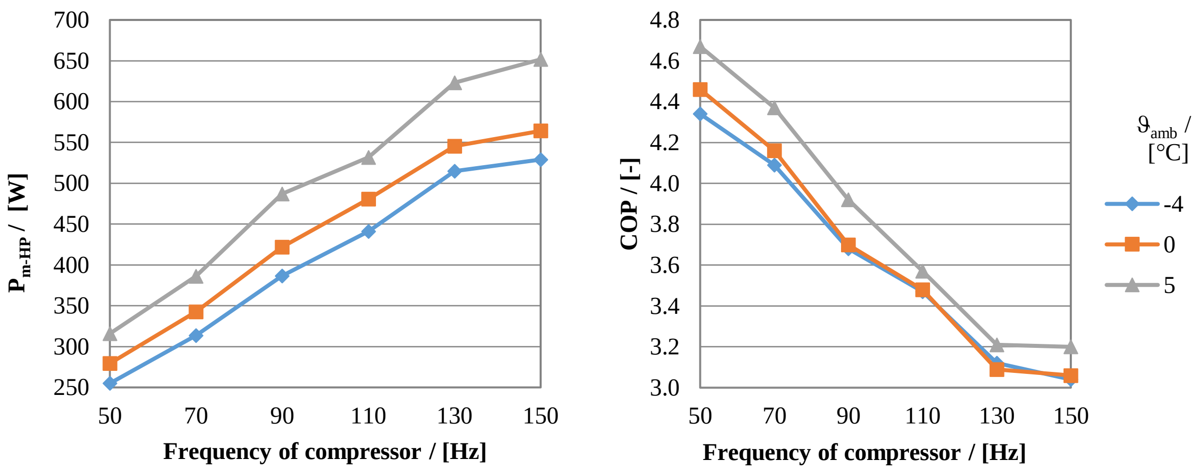

Fig. 4. Performance map of the power (left) and of the COP (right) of the heat pump depending on ambient air temperature and compressor frequency [10]

$$n_{MVHR} = \frac{\vartheta_{ext} - \vartheta_{exh} + \frac{P_{fan}}{\dot{m} \cdot cp}}{\vartheta_{ext} - \vartheta_{amb}} \tag{1}$$

A PI controller is implemented having as process variable the indoor temperature. The output is the required supply air temperature and then correspondingly the compressor speed and the power of the post-heater are calculated via a look-up table [10]. In the functional model, the controller of the defrost cycle is based on operation time and evaporator surface temperature (when the evaporator surface is below the set point for two hours, the defrost cycle starts for 15 minutes). Since the evaporator surface temperature is not calculated in the simulation model, the outlet exhaust air temperature of the evaporator is used. The value of 15 minutes is set as minimum operation time of the compressor.

3.2. Calibration

Measured values of ambient and extract air temperature, relative humidity and mass flow are used as inputs for the simulation model with a time step of 30 seconds. Additionally, the measured signals of the controllers are directly given to the heat pump model i.e. the frequency of the compressor and the signal of the defrost cycle.

The electrical consumption of the fans and the controller has been calibrated to the measurement data and turned out to be significantly higher compared to the specifications of the MVHR data sheet due to higher pressure losses mainly because of the condenser and the evaporator.

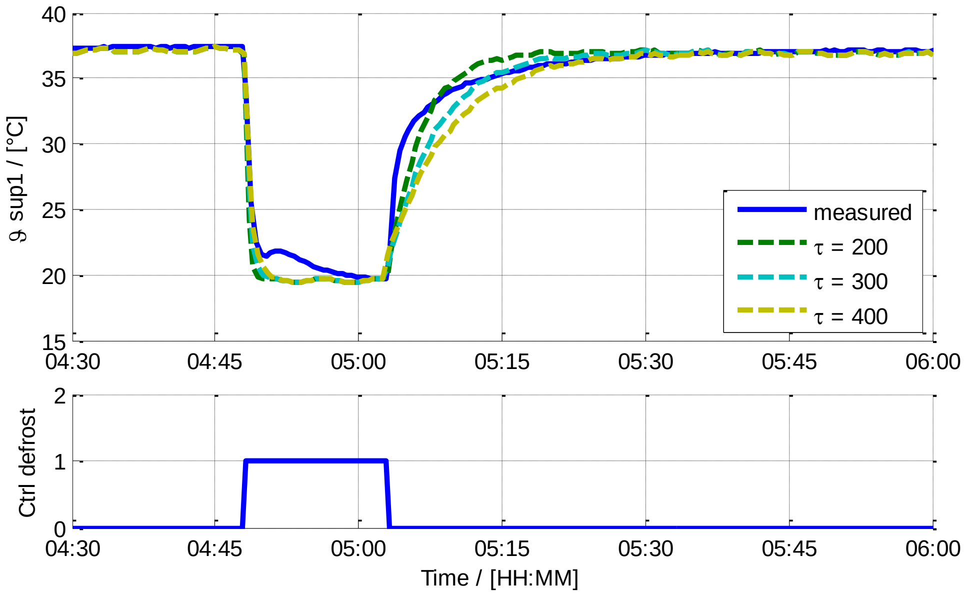

Fig. 5. Parametric study varying the time constants of the condenser (the time constant when condenser is cooled down is set to 10 times lower than the one in case of heating up)

The time constants used for heating and cooling phase of condenser during on/off operation are calibrated against the monitored data via a parametric study (Figure 5). The time constant, when condenser is cooled down, is set to 10 times lower than the one in case of when the condenser is heated up. The value of 300 sec is used further in simulations.

3.3. Validation

Two parameters are important for the validation of the simulation model: the supplied power to the flat and the consumed electricity of the HVAC. Since the supplied power is not measured directly, the supplied air temperature is used for comparison of measurement and simulation. The measured temperature after the condenser (ϑ_{sup1}) is used for the validation of the model.

The measured and simulated supply air temperature (ϑ_{sup1}) in average values of 15 minutes is presented in Figure 6. A good agreement between simulation and measurement results can be observed both in low and high supply air temperature. The measured and simulated electrical consumption of the compressor are shown in Figure 7. The simulated electricity is slightly lower than the measured one in maximum operation (ca. 15 W) and quite similar in low operation.

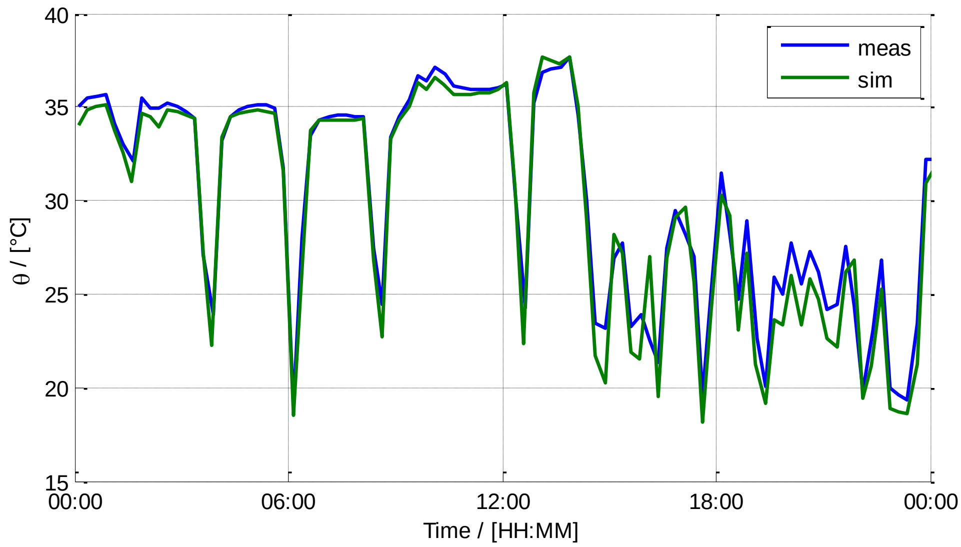

Fig. 6: Comparison of simulated and measured air temperature after the condenser (ϑ_{sup1}) for one day in February. Results in average values of 15 minutes.

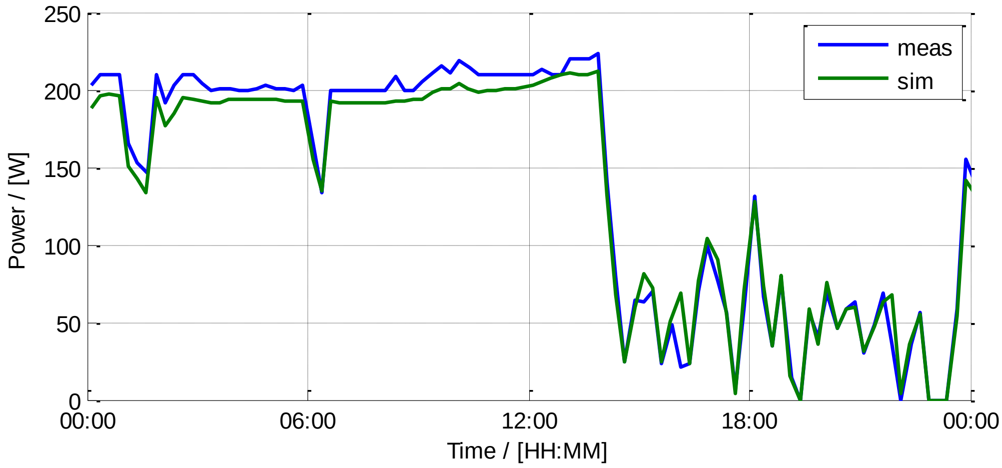

Fig. 7: Comparison of simulated and measured electrical consumption of the compressor for one day in February. Results in average values of 15 minutes

As mentioned in the previous section, the two flats below and above the investigated flat were renovated and unoccupied during the monitoring period. Therefore, the transmission losses from the monitored ground floor flat to the other two flats were relatively high leading to relative high heating demand compared to thermal losses to the ambient only. As a consequence, the micro-HP was most of time in operation and often with maximum compressor speed and also at higher ambient temperature than would be normally expected. As a further consequence, also the post-heater was almost continuously in operation. Therefore, the monitoring data can be used to calibrate the micro-HP model but the results do not represent normal operation.

4. Simulation study

Annual building and system simulations are performed to investigate the performance of the micro-HP in annual operation under several boundary conditions by varying a) the set point temperature of the flat and its neighboring flats and b) the minimum operation time of the micro-HP.

4.1. Boundary conditions

A six-zone two star building model of the refurbished ground floor flat is used [5] for the annual simulations. The six thermal zones are presented in Figure 3. Another 6-zone model of the whole renovated building is used to calculate the temperature of the neighboring flats as well as the unheated zones (cellar and staircase) [3]. These temperatures are used as boundary conditions in the simulation model of the ground floor flat. An occupancy (with three persons) and electric profile based on [11] is used, while the distribution to each thermal zone is presented in [5]. The climatic data of Stuttgart (WP2, iNSPiRe [1]) is used. The post-heater is assumed to be located in corridor instead of the north children room (remark: this corresponds to the real situation after autumn 2016). An additional electric radiator is modelled in bathroom, which is required for comfort reasons (this is also the situation in the real building). Its set point temperature is assumed to be same as the one of the flat here, but it could be set differently.

4.2. Results and discussion

The seasonal performance factor (SPF) is calculated for three different balance boundaries as shown in equations (2), (3) and (4):

$$SPF_{m-HP} = \frac{Q_{condenser}}{W_{el\_compressor}} \tag{2}$$

$$SPF_{sys} = \frac{Q_{condenser}+Q_{post\ heater}+Q_{bath\ radiator}}{W_{el\_compressor}+W_{el\_post\ heater}+W_{el\_bath\ radiator}} \tag{3}$$

$$SPF_{tot} = \frac{Q_{MVHR}+Q_{condenser}+Q_{post\ heater}+Q_{bath\ radiator}}{W_{el\_MVHR}+W_{el\_compressor}+W_{el\_post\ heater}+W_{el\_bath\ radiator}} \tag{4}$$

Remark: the Q_{MVHR} and W_{el_MVHR} are calculated for the heating period assuming that measured temperature is lower than the set point plus 2 K.

4.2.1. Parametric study: varying the set point temperature

The set point temperature of the ground floor flat and of its neighbor flats (in cellar and first floor) is varied from 20 to 22°C. The corresponding set point temperatures and the results are shown in Table 1. The sum of the supplied energy by the micro-HP, the post-heater and the electric radiator in bathroom is considered as heating demand of the flat, while the MVHR is considered a “passive component” in the energy balance (such as in PHPP [12]).

Table 1. Varied parameters: set point temperatures. Results: heating demand and SPF of the micro-HP (SPF_{m-HP}) and of the system with (SPF_{tot}) and without MVHR (SPF_{sys}) for the simulated cases

	Parameters			Results		
Case	ϑ_{set} of ground floor flat / [°C]	ϑ_{set} of neighbour flats / [°C]	HD / [kWh/m2a]	SPF_{m-HP}	SPF_{sys}	SPF_{tot}
A (ref)	20	20	8.5	3.6	2.11	4.0
B	21	21	12.1	3.5	2.14	3.8
C	22	22	16.4	3.4	2.12	3.6
D	21	20	20.7	3.2	2.11	3.3
E	22	20	38.1	3.1	1.77	2.6

In reference case A, the heating demand is 8.5 kWh/(m²·a) and the SPF_{tot} is relative high with a value of 4. In case C ϑ_{set} 22°C the heating demand is 16.4 kWh/(m²·a) quite increased compared to cases A and B but still the micro-HP can cover the heating demand (post-heater almost not operated see Figure 9). In extreme case E the heating demand is quite high (38 kWh/(m²·a)) and the HVAC is at the limit to cover the whole demand. The share of direct electricity increases to 36% from 27% in case A.

Although, the SPF of the micro-HP is decreasing with increasing heating demand - since the micro-HP operates in high frequency (and thus low COP see Figure 4) - , it is always in a relative high values between 3.1 and 3.6. The cumulative distribution function of the frequency of the compressor for all simulated cases is presented in Figure 8. In the first three cases the compressor operates in full speed less than 22%, resulting to relative high SPF. The SPF of the system has a value a bit higher than 2 (except case E), which is acceptable since in combination with the low heating demand the resulting electrical consumption is not high e.g. in case C it is 765 kWh/a. The SPF_{tot} is increased compared to the SPF_{sys}, due to the high efficiency of the MVHR, and follows the same trend as the $SPF_{m\text{-}HP}$. The minimum value of 2.6 is shown in case E.

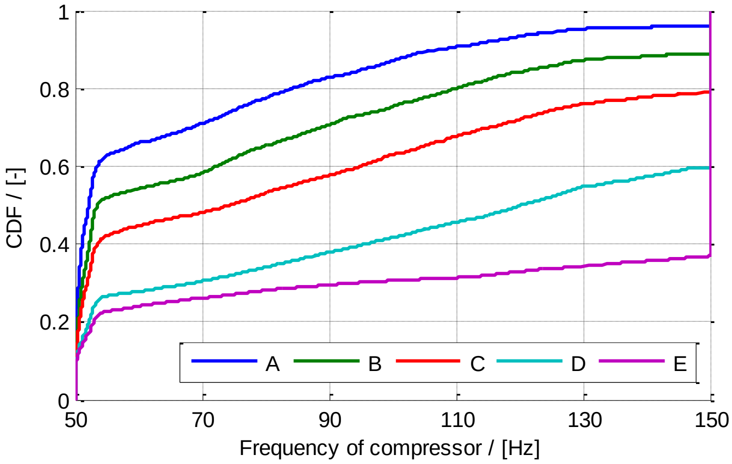

Fig.8. Sorted frequency of the compressor for the simulated cases A to E

The load duration curve of the investigated flat, which is the sum of the supplied power to the flat (by micro-HP, post-heater and electric radiator) for case C is shown in Figure 9. The maximum supplied power to the flat in daily average is almost 900 W. The post-heater operates for a short period (five days), while the electric radiator operates often with a share of 24% to the total supplied heat.

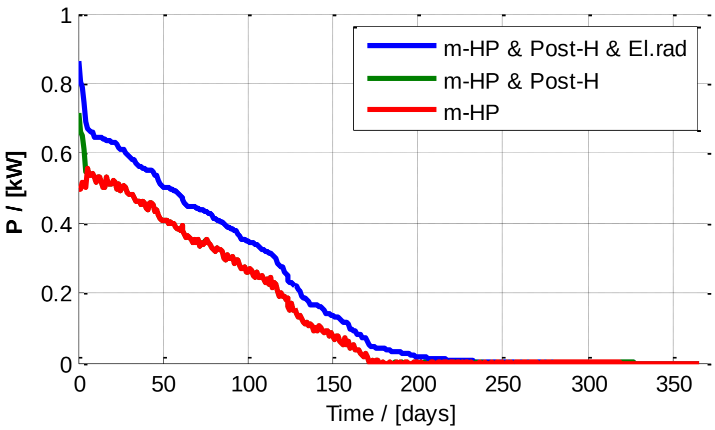

Fig. 9. Sorted daily supplied power to the flat (load duration curve) by micro-HP, post-heater and electric radiator

In general, the micro-HP delivers comfort conditions all over the year with an $SPF_{m\text{-}HP}$ of 3.1 even for the case with the highest heating demand. However, the operation time of the post-heater is significant for the system performance. In case of higher heating demand, another heat pump with higher thermal capacity would be recommended.

4.3. Parametric study: varying the minimum operation time of the micro-HP

The minimum operation time of the compressor is varied from 15 min to 90 minutes in order to show its influence on the performance of the micro-HP. An increased operation in low frequency would lead to improved performance due to higher COP in low frequency. The simulation results for the reference case A are presented in Table 2. Although, the $SPF_{m\text{-}HP}$ is slightly increased with higher operation time (the supplied energy to the flat is also slightly increased), the electric consumption remains almost the same. Thus, there is no significant benefit to increase the minimum operation time.

Table 2. Electrical consumption and SPF of the micro-HP for different minimum operation time of the compressor

Minimum operation time / [min]	Wel_comp / [kWh/a]	$SPF_{m\text{-}HP}$
15	273	3.43
30	272	3.48
60	271	3.57
90	271	3.60

5. Conclusions

The concept of a micro-heat pump (micro-HP) combined with mechanical ventilation with heat recovery (MVHR) unit is presented. A functional model of the system integrated into a prefabricated timber frame façade was developed within the EU-project iNSPiRe [1] and was installed in one flat during the refurbishment of a multi-family house in Ludwigsburg (D).

The micro-HP combined with MVHR system is proposed in combination with deep refurbishment of buildings to very good energy standard such as EnerPHit or Passive House. The integration of the system into a prefabricated façade has many advantages i.e. minimized use of space, reduction of installation time and effort. The low heating capacity of the heat pump, the prefabrication and the combination of m-HP with the MVHR have the potential to be an economically attractive solution.

The system was monitored in detail for several weeks. The monitoring data are used to calibrate and validate the simulation models, which are developed in Matlab/Simulink [7] using the Carnot Blockset [8]. Since there are no available data for a whole heating period, annual dynamic building and system simulations were performed to investigate the performance of the HVAC system. Two parametric simulation studies are performed a) varying the set point temperatures of the investigated flat and its neighboring flats and b) varying the minimum operation time of the micro-HP. The results show good performance with a total SPF of 2.6 in the worst case and 4 in the best case. The SPF of the micro-HP varies between 3.1 and 3.6. The influence of minimum operation time of the compressor is negligible on the performance of the system.

Acknowledgements

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