Article
Compact Decentral Façade-Integrated Air-to-Air Heat Pumps for Serial Renovation of Multi-Apartment Buildings

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Abstract: To address the huge market of renovation of multi-apartment buildings, minimal-invasive decentral serial-renovation solutions are required. One major challenge in the design of decentral heat pumps is to find the optimal balance between, on one hand, compactness and pleasant design, and on the other hand, efficiency and minimal sound emissions. A comprehensive holistic design and optimization process for the development of decentral heat pumps, from the component level, to the system level, and up to the building level, is developed. A novel façade-integrated speed-controlled exhaust air to supply air heat pump combined with a mechanical ventilation system with heat recovery and recirculation air was developed and simulated in a reference flat. Compared to a traditional supply air heat pump without recirculation, it shows only slight performance improvement, but allows significantly better thermal comfort and control, independently from the hygienic air flow rate and from the heating and cooling loads. Detailed measurement and simulation results are presented for several functional models with heating power of around 1 kW up to 2.5 kW. The design was optimized by means of CFD simulations to allow for low pressure drop, homogeneous flow, and low sound emissions. Moreover, mock-ups of innovative façade-integrated heat pump outdoor units are presented.

Keywords: serial/industrialized renovation; enthalpy recovery; mechanical ventilation; building façades; speed-controlled supply air heat pump

1. Introduction
1.1. Minimal Invasive Serial Renovation

The European directive, concerning the energy performance of buildings [1], stated the goal of the European Union to reduce equivalent carbon dioxide emissions attributed to the building sector by at least 40% by 2030 compared to 1990. Recently, the European Union (EU) has set a milestone of a society with net-zero emissions of overall greenhouse gasses (GHG) by 2050 compared to the values recorded in 1990 [2,3]. Following a continuous improvement in the quality of the buildings, heat pumps (HP) are nowadays regarded as a preferable choice towards the decarbonisation of the building sector [4,5] in both new and existing buildings. Groundwater or brine-source HPs are generally able to achieve a higher coefficient of performance (COP) compared to air-to-water or air-to-air HPs [6] but they often present additional challenges due to the required construction work for the installation of the ground heat exchanger and are challenging, in particular, in densely populated areas and in renovations of the building stock. Additional limitations may occur in renovation cases due to the scarce availability of space concerning the exploitation of a heat source and the recurrent techno-economical unfeasibility of renovating the existing centralised heating system [7]. In such cases, where for technical, economical, structural, social, or other reasons central solutions are not feasible, decentralised partially or entirely...
façade-integrated systems can satisfy the space heating (SH) and/or domestic hot water (DHW) demands of a single apartment with minimal invasive construction works.

1.2. Review of Compact Air-Sourced Heat Pumps for Renovation

 Appropriately designed ventilation systems (air-to-air) have proved to reduce space heating (SH) and space cooling (SC) energy demands and contemporarily improve thermal comfort and indoor air quality (IAQ) [8]. Among them, exhaust air HPs exploit the difference between the enthalpy of exhaust air from the mechanical ventilation with heat recovery (MVHR) and that of ambient air, the latter being lower than the former in winter conditions. Examples of such exhaust air HPs are so-called Passive House (PH) heat pump compact units for mechanical ventilation, SH, and DHW preparation.

In contrast, an extract air HP (frequently also called exhaust air HP in the literature) uses indoor room air as its source (or sometimes the air of the technical room, the cellar, or the garage, etc.). It can, thus, also be used for heat recovery. A detailed classification of heat recovery and MVHR systems is presented in Ref. [9].

Compared to an extract air HP, the temperature level of the evaporator in an exhaust-air HP is lower, but the air is almost saturated. Hence, the enthalpy is still higher than that in ambient air [10,11]. The authors in Ref. [12] reported the development of a physical model (based on the NTU method) for an energy recovery ventilation (ERV) system to calculate the effectiveness of a membrane enthalpy heat exchanger and compare it to a conventional heat recovery ventilation (HRV). Simulations based on the model, which was validated through measurements, demonstrate that enthalpy exchangers exhibit a higher total transferred power compared to traditional heat exchangers. In terms of efficiency (COP), the MVHR outperforms the extract air HP [13,14]. For high-energy-efficient buildings, such air-based systems can supply the design heat load using only supply air. Furthermore, ERV allows for dehumidification in summer in humid climates, as shown also in Ref. [12]. In Ref. [15] a façade-integrated exhaust HP equipped with MVHR was developed, simulated, and monitored. As also shown in Ref. [16], the installation of so-called micro HPs is also suitable in prefabricated timber façades, thus improving the compactness and consistently reducing the construction and installation time. Ref. [17] confirms the need to develop systemic renovation packages for both residential and non-residential buildings with the aim of reducing the primary energy consumption while ensuring at the same time optimal indoor comfort for the occupants. In deep renovations, it was demonstrated that such systems could achieve theoretically the highest overall performance [18]. In an in-situ monitoring study [19], this hypothesis was proven for a renovation case.

In Ref. [20], an economic and environmental analysis of different energy renovation packages in terms of life cycle cost (LCC) and life cycle assessment (LCA) was presented for various European climates, achieving the lowest total costs and environmental impact for buildings with the lowest heating demand. The analysis was additionally extended in Ref. [21], where a database was developed with the purpose of generating energetic, environmental, and economic performance indicators for different combinations of envelope and HVAC systems retrofitting strategies in different climates. An extensive review of multiple building retrofitting strategies is presented in Refs. [22,23].

The possibility of equipping different HVAC configurations with onsite thermal and/or electrical storage was also analysed in Ref. [24]. In Ref. [25] the authors performed a techno-economic analysis of rule-based control algorithms for an exhaust air HP installed in a single-family house and connected to a photovoltaic system with the purpose of minimizing the final energy consumption and maximizing the self-consumption.

However, the maximum deliverable heating power of these supply air systems is limited by the hygienic supply air flow rate [26]. In fact, an air volume flow exceeding the hygienic air requirements can lead to drier room air during winter. This can happen even if the ventilation system is equipped with an enthalpy exchanger (EE), as shown in Refs. [11,12]. Thus, the maximum specific heating load that can be provided by the system corresponds to approximately 10 W/m² for typical residential buildings with typical
occupation density, making this type of solution not adequate in case of renovations in which specific heating loads are higher than 25 W/m$^2$ [27]. One possibility of increasing the heating power is to recirculate secondary air. This recirculation air can be mixed with the supply air or distributed separately. The second option requires additional pipe work but allows a better distribution of the heating load, e.g., by controlling the volume flow of the recirculation air. Moreover, in addition to the supply air rooms, the corridor can be heated, which leads to higher degree of freedom in the control of room temperature and, thus, to an increase in thermal comfort [28]. Furthermore, the increased volume flow allows to operate such systems also for SC. However, to increase the total heating power of the HP, additional ambient air (independent from the fresh air of the ventilation unit) has to be drawn into the evaporator, in addition to the exhaust air. The concept of decentralised heating systems is further discussed in Refs. [28,29], comparing these systems with respect to indoor air quality, thermal comfort, and energy performance.

The authors in Ref. [30] studied through simulations the effect of exhaust air HPs on district heat energy use for renovated buildings in cold climates and concluded that exhaust air HPs can achieve the highest seasonal coefficient of performance (SCOP = 3.6). Besides, in Ref. [31] it was concluded, among others, that the impact of varying boundary conditions on the HP’s performance has to be considered. A market research performed in Ref. [32] highlighted that currently available split HPs present a sound pressure level ranging from 43 dB(A) to 65 dB(A) for the internal unit and 59 dB(A) to 70 dB(A) for the outdoor unit depending on the size, therefore indicating the need to undertake sound reduction measures for both outdoor and indoor units to avoid acoustic discomfort and unacceptance. In Ref. [33], the operation and energy costs and the primary energy consumption of gas boilers and HPs for the production of DHW were compared.

An integrated split air-to-water HP providing SH, SC, and DHW was investigated [34] for newly constructed buildings. Although a similar concept could be developed also for decentralised renovation, the compactness of the system would be heavily limited and less adaptable for apartment-wise refurbishments. To address the challenges of decentralised DHW preparation, a mini split air-to-water façade-integrated HP with a size of 1.5 kW was also developed, as an alternative to decentral electric or gas boilers [35]. Particular focus was given to the optimization of the airflow pattern throughout the façade-integrated element, by means of CFD simulations that are coupled with refrigerant cycle simulations, targeting the reduction of the sound emissions and the minimization of the visual impact. CFD-based component optimization processes were also adopted in Ref. [36] for the analysis of the airflow pattern in a modular fan unit and in Ref. [37] for a novel packaged outdoor air humidifier. A review of CFD-based applications for the design and optimization of heat exchangers instead is also presented in Ref. [38]. The authors in Ref. [39] also implemented a CFD-based procedure to optimize the air-side heat transfer coefficient of a 30 kW R290 air-to-water HP and to improve its icing and defrosting behaviour.

Detailed CFD analyses are also used to solve a wide range of issues that are commonly encountered when designing and testing compact heat exchangers, such as flow maldistribution, fouling, or dead-spots [40]. Extensive studies about the design and the optimization of compact heat exchangers are presented in Refs. [41,42]. However, when the task is to purely optimize the whole functionality of the system, detailed coupled fluid-dynamics and heat transfer analyses can be neglected. CFD simulations of this type might help, for example, in the choice of a proper inflow geometry and in the selection of the type and number of fans.

The modelling of a HP system is often crucial for control optimization and the identification of performance gaps. In Ref. [43], the effect of partial-load and on-off cycles on the system performance of air-to-water HPs is investigated. Air-to-water HPs were analysed in Ref. [44] in order to assess the seasonal performance factor for residential buildings in Chile. The HP was modelled using three individual sub-models for the compressor, condenser, and evaporator, assuming that the compressor is a fixed-speed one. Inverter-based compressors were instead modelled in Ref. [45] under different compressor speeds and different
operating conditions for an air-to-water HP in Poland supplying DHW. Finally, in Ref. [46] a simulation model was developed to optimize the seasonal coefficient of performance and seasonal energy efficiency ratio of air-to-air HP.

To summarize, several options for decentral application of compact air-to-air HPs were investigated. Combining the HP for SH and optionally SC with the MVHR seems to be one of the most promising configurations for decentral flat-wise application. Figure 1 gives a schematic overview of the different systems concepts and the different HP sources and sinks.

![Figure 1. Schematic classification of air-to-air heat pumps (HP) with or without heat recovery and combined with or independent of MVHR; remark: refrigerant lines and fans not displayed for better readability; EXH: exhaust air, AMB: ambient air, EXT: extract air, SUP: supply air, REC: recirculation air, MVHR: mechanical ventilation with heat recovery, INF: infiltration air.](image)

### 1.3. Identification of Research Gaps

HPs have been investigated for many years and there is no doubt that they will play a major role in the future for SH and SC of the building stock. However, based on the literature review, it could be identified that previous studies often miss addressing the very relevant but also challenging market section of existing multi-apartment buildings. Compact HPs for decentral (i.e., flat-wise) applications in deep renovated buildings can be a promoting solution if central (i.e., building-wise) HP solutions cannot be implemented for technical or non-technical reasons. However, those compact air-to-air HPs for decentral (i.e., flat-wise) applications have not been systematically investigated and optimised in previous studies.

Because of the various individual boundary conditions in the huge market of renovations, modular and scalable solutions are required. The solution of decentral compact air-to-air HPs is promising in combination with prefabrication and industrial (so-called serial) renovation. Existing HP design methods lack a holistic and systematic approach allowing to evaluate and rank the competing goals of attractive design, compactness, comfort, low noise level, and high efficiency.

Furthermore, simulation-based optimization on a component level, refrigerant cycle, and building level is required to find optimal solutions considering the constraints of specific applications depending on size, floor plan, location, building energy level, etc. Previous research concentrated either on the component level (e.g., compressor design or air flow optimization), on refrigerant cycle optimization, or on building level design and optimization (e.g., building and HVAC simulation for control optimization). However, in particular for decentral air-to-air HPs, which require for the target of façade-integration a compact design and a visually and architecturally attractive solutions, a systematic and holistic approach including all levels (from component to building level) is still missing.

### 1.4. Contribution and Novelty of this Work

To enable serial and minimal invasive renovations including a switch to HP-based heating systems under the constraint of flat-wise renovation and limited space availability, façade integration is seen as a promising solution. Innovative solutions for minimally
invasive decentral renovation with multi-functional façades that integrate compact, architecturally attractive, and silent as well as efficient HPs are presented. It is shown by experimental and by simulation results of functional models, by building and HVAC simulations as well as by mock-ups, that these systems can represent an enabler for the deep renovation of multi-apartment buildings.

The novelty of this contribution lies in the holistic and complete optimization approach from the component through the system to the building level, involving combined flow guidance (i.e., fan power) and sound emission optimization, refrigerant cycle, and air-flow (i.e., CFD) simulations, as well as coupled building and HVAC simulations. In particular for compact and small-capacity HP systems (in the range of 1.0 kW to 2.5 kW) low pressure drop and low thermal losses are of major importance to obtain high performance and low sound emissions.

Laboratory experiments and simulation results allow to compare fan and evaporator configurations in terms of flow homogeneity, fan power, and sound emissions. Different refrigerant cycle options are compared on the refrigerant cycle level as well as by building and HVAC simulations on building level. The holistic design approach is completed by the development and testing of real-scale mock-ups.

2. Concept and Methods

Figure 2 summarizes the conceptual structure of this research work. Firstly, after presenting the different decentralised HP-based concepts for SH (and optionally SC), MVHR, and DHW, the component-level design optimization is performed focusing on the façade-integrated HP-outdoor unit. Secondly, on a system level, the performance of a supply-air HP with recirculation is evaluated through both refrigerant cycle simulations and laboratory measurements. Then, on a building level, these results are integrated in a coupled dynamic building and HVAC simulations of a typical multi-apartment building in Austria. Several key performance indicators (KPIs) are evaluated both on the building and on the flat level for the system’s overall performance, the thermal comfort, and the indoor air quality. Finally, the concept of serial renovation with prefabricated multi-functional façades is introduced and real-scale mock-ups prove the feasibility of the proposed solutions.

![Figure 2](image)  
**Figure 2.** A holistic approach to support the development of façade integrated decentralised HPs—from component to system and to building level optimization.

2.1. Concepts of Façade-Integrated Air-to-Air Heat Pumps for Renovation

Two concepts of façade-integrated split exhaust air-to-supply air HPs and one concept of split air-to-air HP are introduced.
2.1.1. Exhaust Air-to-Supply Air Heat Pump System

A simplified scheme of an exhaust-air to supply-air HP is shown in Figure 3. The ambient air, pre-heated in a pre-heater and in the MVHR (with either HRV or ERV), is heated in the condenser of the HP and, optionally, in a post-heater if the power of HP is insufficient, before being supplied and distributed to the flat as supply air. An additional electric radiator is typically mounted in the bathroom for comfort reasons. The heating system is controlled depending on the operative temperature of a defined room (e.g., the corridor) or as a weighted temperature of several rooms.

![Figure 3. Scheme of exhaust-air to supply-air heat pump with MVHR and with post-heater (PostH) and pre-heater (PrH).](image)

The unit can be simply mounted in a compact fashion on the ceiling (e.g., in the kitchen) or, alternatively, integrated in the façade (see section below) while the air distribution system is typically located in the corridor, supplying fresh air to the sleeping room, child room, and living room. The main advantages of this solution are the cost-effective use of the ducts, which are already in place for the ventilation system, to supply heat, and its compact design. A limitation of such a system is represented by the hygienic airflow rate, which in turn constrains the maximum heating power. With such an exhaust air-to-supply air HP, the heating power is limited by the hygienic flow rate and is usually 10 W/m² (i.e., one of the criterions for achieving PH standard). Furthermore, as the heating power is connected to the air distribution, the control options are limited (see also Section 2.2.3) and do not allow for a room-wise temperature control. Room-wise control is possible only if, e.g., a room-wise electric radiator is additionally installed.

It is important to mention that the COP of the exhaust air-to-supply air HP and its overall performance (i.e., including pre-heating, de-icing, etc.) directly depends on the effectiveness of the HRV or ERV, which in turn is strongly affected by the use of the building and its operation (see also Ref. [14]).

2.1.2. Air-to-Air Split-Type Heat Pump with MVHR

This system is introduced here rather as a reference for the energetic evaluation than as a practical solution. It consists of two separated parts, as depicted in Figure 4. SH is performed by a split-type air-to-air heat pump that exploits the ambient air as a source and heats up the recirculated air in the corridor of the flat. A separated MVHR system equipped with pre-heater (for frost-protection operation) and an optional post-heater guarantees fresh air supply and adequate ventilation. In this concept, an electric radiator is used in each room to post-heat and guarantee comfort conditions. This system represents on the one hand a reference of a heating system where the control of the heating is decoupled from the hygienic ventilation rate (in contrast to the supply air heating system) and on the other hand can be understood as a step between the supply air HP presented in the previous section and the supply air heat pump with recirculation air presented in the following section. Three different qualities of the performance of the split HP are assumed: poor, medium, and good, i.e., COP = 2, 3, or 4. Thus, it represents an instructive rather than a practical solution.
2.1.3. Exhaust Air-to-Supply Air Heat Pump System with Secondary Air Recirculation

In a renovation, typically, the PH standard with 15 kWh/(m² a) and ca. 10 W/m² cannot be easily achieved, instead the so-called EnerPHit standard for renovation allows 25 kWh/(m² a) and also in some cases higher levels of the HD corresponding to ca. 20 W/m² to 30 W/m², see e.g., Ref. [47]). In such a case, pure air heating is not possible and either additional room-wise post-heaters have to be used or the volume flow has to be increased. Both options are clearly limited: electric post-heaters have a poor performance, and their application must therefore be chosen with care. In case of increased volume flow, additional ventilation losses need to be considered and it would be at least in colder climates necessary to use an ERV, which enables the increase of volume flow without resulting in excessively dry air. With higher air flow rates, higher electrical consumption of the ventilators has to be considered too.

Alternatively, recirculated air can be used to increase the heating power and enable for more individual room control, as shown in Figure 5.

In the primary mode of operation, both condensers are used. In this case, the superheated vapour flows into the desuperheater of the recirculation air and subsequently through the condenser of the supply air. In the second mode of operation, the desuperheater is bypassed using two magnetic valves (not shown in the figure) and the superheated vapour directly flows into the condenser of the supply air. In both modes, subcooling is realized in the condenser of the supply air, and thus, a refrigerant receiver is not necessary. Defrosting is performed by hot-gas bypass (not shown in the figure). An enthalpy membrane exchanger is used in the ventilation unit. The volume flow of the supply air can be controlled between 60 m³/h and 120 m³/h. Recirculation air can be controlled between 0 and 135 m³/h.

Figure 6a shows the temperature-enthalpy diagram of the refrigerant cycle (solid line) of the exhaust air HP with recirculation of secondary air. The corresponding psychrometric chart is shown in Figure 6b. The two condensers are connected in series with respect to the refrigerant cycle. Dashed lines indicate the air temperatures. Compared to HPs using
ambient air as a source with typically several hundreds of m$^3$/h, for a low capacity exhaust air HP of a few kW, the temperature differences of the air between the inlet and outlet are relatively high because of the limitation to the hygienic flow rate with some 120 m$^3$/h to 150 m$^3$/h. Thus, the pressure drop in the evaporator (see dotted line) has no significant effect, and a relatively high subcooling can be reached. The effective subcooling depends on the presented configuration, but also on the compressor speed, i.e., for low speeds, a higher subcooling can be achieved. This behaviour can be observed because no refrigerant receiver is foreseen. Moreover, the heat transfer coefficient on the refrigerant side in the two-phase region is significantly higher than that on the air side [48]. In this example the recirculation air is heated to 50 °C, which is close to the limit of the maximum air heating temperature.

In cold climates, a more satisfactory solution can be achieved with the use of ERV, instead of HRV, to avoid dry air during winter [12]. Moreover, with ERV, lower hygienic airflow rates can be realized, leading to a substantial reduction in the energy consumption. It is remarkable that MVHR systems that are equipped with ERV can be mounted horizontally (e.g., ceiling mounted), since the amount of condensate can be reduced to zero compared to systems with HRV where condensate removal is critical.

2.2. Holistic and Systematic Optimization Approach

2.2.1. Component-Level Optimization

CFD Simulation-Based Optimization of HP Outdoor Unit

For compact decentral façade-integrated systems with an attractive appearance (or as much as possible hidden), the space constraints need to be carefully considered in the early design phase. Sufficient air inflow through the façade-integrated element needs to be provided while guaranteeing both sufficiently high energy performance and acceptably low sound emissions. Hence, this requires minimal pressure drop and air guidance that enables homogeneous flow through the evaporator of the HP (and where applicable of the heat exchanger of the MVHR). The aspects of accessibility for maintenance and condensate removal must also be tackled.

The following criteria are investigated and, by means of CFD simulation, a system optimization is performed:

- Flow homogeneity on the evaporator surface;
- Air-side pressure drop minimization;
- Effect on the overall performance of the refrigerant cycle by means of coupled refrigerant cycle simulations.

Obtaining a possibly homogeneous flow guarantees that the evaporation pressure of the refrigerant stays almost constant for each refrigerant circuit, whereas highly inhomogeneous flows can have a negative effect on the efficiency of the HP.
The complexity of the interactions between heat transfer, fluid dynamics, and fan efficiency as well as sound propagation requires to evaluate a number of flow configurations for the evaporator. Different configurations and geometries of the façade-integrated outdoor unit were designed and analysed by means of CFD simulations in the software ANSYS 19.2 with the use of the solver CFX concerning their flow patterns, and pressure drop, depending on:

- Fan type (axial or radial);
- Evaporator geometry;
- Evaporator configuration compared to the position of the fans.

For the nominal conditions of the supply-air HP of heating capacity of 1.5 kW at ambient air temperature of $-7 \, ^\circ \text{C}$, the volume flow on the evaporator is 350 m$^3$/h, which is composed of 150 m$^3$/h of exhaust air plus 200 m$^3$/h of ambient air.

Table 1 summarizes the adopted boundary conditions for the models involving radial fans and axial fans, respectively. In Table 2, the different variants that were analysed in the CFD software are listed. Each variant is characterised by a particular choice regarding the type of fan and the position of the evaporator. Then, the geometries of the simulated models are shown in Figure 7 (Variants 1 to 4).

### Table 1. Boundary conditions in CFD simulations for the inlet and outlet of the computational models; rotational speed in case of axial fans is varied between 900 rpm and 1700 rpm.

| Type of BC | Inlet | Total pressure = 0 Pa | Mass flow rate = 0.1152 kg/s |
|------------|-------|-----------------------|-----------------------------|
| Radial fans| Outlet|                       |                             |
| Axial fans | Inlet  | Static pressure = 0 Pa |                             |
|            | Outlet | Average static pressure = 0 Pa |                     |

### Table 2. Overview of simulated variants with indications about the type of fans adopted and their position related to the evaporator.

| Variant number | Geometry   | Type of Fan | Position of the Fans |
|----------------|------------|-------------|----------------------|
| 1              | Square     | 4 axial     | Suction side         |
| 2.1            | Rectangular| 4 axial     | Suction side         |
| 2.2            | Rectangular| 4 axial     | Pressure side        |
| 3              | Rectangular| 1 radial    | Suction side         |
| 4              | Rectangular| 1 radial    | Suction side         |

Variant 1 consists of four axial fans with an impeller diameter of 225 mm together with a square evaporator with dimensions 350/350/60 mm (L/H/W).

The impeller of the single fan in variants 2.1 and 2.2 has the same dimensions as variant 1. The position of the fans is on the suction side with respect to the evaporator in variant no. 2.1 and on the pressure side in variant 2.2, which can have advantages in terms of sound emissions and condensate removal. In variants 3 and 4, a single radial fan with a 220 mm impeller is implemented. To properly simulate the inflow of ambient air, a control volume of air was added to the physical inlet of the outdoor element.

During a pre-analysis phase, the standard deviation of the air velocity on the incident surface of the evaporator was used to assess the homogeneity of the flow pattern. The evaporator was for the sake of simplicity treated as an anisotropic porous media with a loss coefficient in the flow direction corresponding to the pressure drop of commercial finned evaporators (of about 5 Pa at 350 m$^3$/h). Radial fans were modelled as a simple circular surface with defined perpendicular mass flow. Axial fans were instead modelled in a more detailed way. A generic geometry of an axial fan was assumed, and its volume was subtracted from the fluid domain, thus, creating a void simulating the obstruction caused by the fan. This procedure is repeated for each single fan. A local reference system was
introduced for the domain corresponding to each fan and a rotating motion was finally assigned to it with given angular velocity. For the modelling of the interface between the fixed and rotating fluid domains, the option “Frozen rotor” was selected.

Figure 7. Simulated CFD models for simulation-based optimization of the outdoor unit. Variant no. 1 with four axial fans and flat square evaporator on the low-pressure side of the fans. Variant no. 2.1 with four axial fans and thin rectangular evaporator on the under-pressure side of the fans. Variant no. 2.2 with four axial fans and a thin rectangular evaporator on the overpressure side of the fans. Variant no. 3 with a single radial fan and compact evaporator on the low-pressure side of the fan. Radial outflow takes place at an angle of 20° with the building façade. Variant no. 4 with a single radial fan and a more compact evaporator (compared to var. 1 and 2) on the low-pressure side of the fan. Radial outflow takes place at an angle of 50° with the building façade. The inlet region is indicated in green, the outlet region in blue, and the evaporator in red.

The first simulation runs were performed with a laminar turbulence model to check the overall functionality of the computational model. Then, independently from the type of the fan, the k-ε turbulence model was selected for the simulations as it:
- Represents a good compromise between accuracy, robustness and convergence time;
- Requires fine meshing in the near-wall areas and in nearby rotating domains;
- Is generally suggested for flows over complex and/or rotating geometries.

The walls of the fluid domain are smooth and the velocity on the boundaries equal to zero (no-slip wall condition).

Fan Performance and Sound Emissions Testing

A measurement campaign was carried out to determine the sound pressure level of axial and radial fans, corresponding to the geometrical variants 2.1, 2.2, and 4 indicated in Figure 7, with varying pressure boundaries and volume flows. A test rig (sketched in Figure 8a,b and shown in Figure 9) was developed with the purpose of selecting the fan type and category for the outdoor unit, with the lowest sound emissions in the operating
range. The pressure drop upstream of the fans can be imposed via a ventilation flap coupled to a PI controller, while the pressure difference with the environment downstream of the fans is equalled to zero to correctly replicate outflow into the ambient. For this purpose, a PID-controlled support fan is used. Volume flow is measured with the help of a calibrated hot-wire anemometer (with an accuracy of 1.2% of the measured value).

Figure 8. Schematic representation of the test-rig with flap, evaporator, and support fan used for the assessment of the electrical power consumption and sound power level of axial (a) and radial (b) fans. (1) Measurement point of pressure difference on the under-pressure side of the fans, controlled by means of ventilation flap, (2) measurement point of pressure difference on the overpressure side of the fans, controlled by means of a support fan, (3) measurement point of air volume flow through a hot wire anemometer.

Figure 9. (a) Four parallel axial fans mounted in the channel section for acoustical and fan performance measurements and (b) the set-up of the test rig at UIBK.

2.2.2. System-Level Optimization—Refrigerant Cycle

The performances of the supply air HP without (Section 2.1.1) and with secondary air recirculation (Section 2.1.3) were measured in the lab (in the so-called HP compact unit test rig of UIBK [11]). The air ducts were connected to two climatic chambers, one for the emulation of ambient air conditions (e.g., 0 °C, 80% r.H.) and one for the emulation
of room air conditions (e.g., 20 °C, 50% r.H.). External pressure conditions can be kept constant throughout the measurement thanks to the use of ventilation valves and support fans. The electrical power consumptions of the fans as well as of the compressor (including the inverter) are also logged. The pipe surface temperatures within the refrigerant cycle are measured by means of calibrated thermocouples. Pressure transducers were also adopted to measure evaporation and condensation pressures. All air ducts that connect the functional model to the test-rig are insulated with at least 25 mm polystyrene. Based on the measurement results, the performance maps of the heating power and COP of the HPs were simulated depending on the power (speed/rpm) of the compressor and ambient air temperature (see Refs. [11,49] for the details of the refrigerant cycle model and also Refs. [11,12] for a detailed description of the test-rig specification including error propagation analysis). The refrigerant cycle was filled with R134a. Two different compressors with a nominal total heating power of approximately 1.4 kW and 2.5 kW were used and the system was equipped with either a HRV or ERV unit.

2.2.3. Building-Level Optimization

In order to test the innovative HP-based heating system and to evaluate and compare the performance against alternative solutions under realistic dynamic conditions, a virtual case study was developed. With that virtual case study, the system performance of the different HP-based SH concepts and the impact of the different variants on the TC and IAQ in particular with respect to air heating were investigated. The reference for the TC and IAQ is the flat heated with ideally controlled room-wise radiators with an HRV. In the reference building model, six electric radiators (one for each room) were implemented to maintain an operative room temperature of 21 °C.

The dynamic building and system simulations were performed in MATLAB/Simulink with an extended version of the toolbox CARNOT [50,51]. The reference building is a multi-apartment building first mentioned in Ref. [28] and is shown in Figure 10a. The building is refurbished to the EnerPHit standard [47], corresponding to a HD of 25 kWh/(m² a). Each flat consists of six thermal zones, one for each room, as shown in Figure 10b. The staircase and the basement are assumed to be non-heated. A monthly average of the ground temperature is further assumed (based on Ref. [52] with an annual average temperature of 14.9 °C). A balanced MVHR, equipped with either HRV or ERV, supplies fresh air to the sleeping room (SL), child room (CH), and living room (LI) and extracts air from the kitchen (KI) and from the bathroom (BA). A constant air change rate of 0.07 h⁻¹ due to infiltration (exfiltration) through the façade is assumed. Air exchange between two adjacent zones through the opening of the doors was modelled as a function of the temperature and humidity difference based on an empirical model developed by Ref. [53]. The flat is occupied by three persons and the daily occupancy schedule is also applied during the weekend. Active SC is not considered, and overheating can be prevented by means of shading and night ventilation. A hygrothermal wall model [54] was implemented in the transient multi-zone model to include the aspect of a moisture buffer, which is required to accurately consider the impact on the humidity. Further assumptions and information about the building model are reported in Refs. [28,54].

As an (low investment cost) alternative to the supply air heating systems, a variant can be obtained when a split HP is used to heat the corridor (with recirculation air). In this case, the electric radiators installed in each room work as post-heater. Three different qualities of this split HP concept were assumed in this simulation study corresponding to a coefficient of performance (COP) for the split HP of 2, 3, or 4.
2.2.4. Mock-up Testing

The façade-integrated mock-ups were installed on the façade of the PASSYS test cells of the University of Innsbruck. Each test cell has a test room with a volume of 38 m$^3$ and a floor area of 14 m$^2$. The test room of each cell is equipped with a very good quality thermal envelope with a thermal transmittance of less than 0.1 W/(m$^2$ K) and is nearly airtight ($n_{50}$ lower than 0.5 h$^{-1}$). The test facades are installed in a thermally insulated interchangeable frame and screwed to the test cell on the south wall. An air-based heating and cooling system inside the test cells allows for an accurate control of the temperature inside the test room. A mobile climate chamber can also be screwed on the test façade to emulate different outdoor air temperature conditions (see also Ref. [15]).

2.3. Key Performance Indicators

2.3.1. Component and System Level Heat Pump Performance and Flow Optimization

The HP performance depends on its operation conditions (i.e., source and sink temperature), the effectiveness of the HRV/ERV and of particular interest, here, on the air distribution (indicated as standard deviation) and on the fan consumption. In addition, the sound emissions were determined for different evaporator-fan configurations and air distribution (indicated as standard deviation) and on the fan consumption. In addition, the sound emissions were determined for different evaporator-fan configurations and air volume flows.

The coefficient of performance of the HP is determined as the ratio of the condenser power (in case of secondary air recirculation, as the sum of desuperheater and condenser) and the total electricity consumption, i.e., the compressor and fan consumption.

$$\text{COP}_{\text{HP}} = \frac{Q_{\text{cond}}}{P_{\text{el,HP}} + P_{\text{el,fan}}}. \quad (1)$$

The temperature effectiveness for the HRV is defined by:

$$\eta_T = \frac{\vartheta_{\text{sup}} - \vartheta_{\text{amb}}}{\vartheta_{\text{ext}} - \vartheta_{\text{amb}}}. \quad (2)$$

and the enthalpy effectiveness for the ERV is defined by:

$$\eta_H = \frac{H_{\text{sup}} - H_{\text{amb}}}{H_{\text{ext}} - H_{\text{amb}}}. \quad (3)$$
The standard deviation of the flow velocity is used to evaluate different evaporator-fan configurations, with \( v_{x,y} \) being the velocity distribution and \( v_{vol} \) the volume-averaged velocity.

\[
\sigma_{\text{flow,vol}} = \sqrt{\left( v_{vol} - v_{x,y} \right)^2}. \quad (4)
\]

In addition, the relative standard deviation is assessed, allowing for an immediate comparison of the flow configurations:

\[
\sigma_{\text{flow},\%} = \frac{\sqrt{\left( v_{vol} - v_{x,y} \right)^2}}{v_{vol}} \cdot 100. \quad (5)
\]

Furthermore, sound emissions are measured for different evaporator and fan configurations and given as A-corrected sound power level \( L_{W,A} \) by means of a compact commercial sound level meter (see Ref. [55] for more details) in 1/3 octave bands from 1.6 Hz to 16 kHz according to the norm (see also Refs. [56,57]).

\[
L_{W,A} = 10 \log_{10} \frac{W}{W_0}. \quad (6)
\]

2.3.2. Building Level Thermal Comfort and Indoor Air Quality

Thermal comfort (TC) and indoor air quality (IAQ) as well as energy performance of the proposed concepts of MVHR and HP systems for serial renovation are compared by means of simulation using the reference building. As reference for TC and IAQ, a flat equipped with room-wise controlled electric radiators, i.e., an ideally controlled system, is used.

For the TC (operative Temperature \( \theta_{op} \), relative humidity \( \varphi \)) and the IAQ (indicated by means of CO\(_2\) concentration), dynamic building performance simulations using a 2* multi-zone model are performed (see also Ref. [58]) on hourly basis.

2.3.3. Building Level Energy Performance

The system performance of the HP is characterised by the thermal power \( Q_{\text{Condenser}} \) and the COP depending on the boundary conditions (i.e., source and sink temperature) and the compressor speed (measured in rpm). For the building level evaluation, the seasonal performance factor (SPF) is used as an indicator and is evaluated based on hourly data for different system boundaries:

\[
\text{SPF}_{\text{HP}} = \frac{Q_{\text{Condenser}} + \dot{Q}_{\text{Desup}}}{(W_{el,\text{Compressor}} + W_{el,\text{Fan,sec}} + W_{el,\text{Fan,amb}})}, \quad (7)
\]

\[
\text{SPF}_{\text{sys}} = \frac{(Q_{\text{HP}} + \dot{Q}_{\text{Postheater}} + \dot{Q}_{\text{Radiator}})}{(W_{el,\text{HP}} + W_{el,\text{Postheater}} + W_{el,\text{Radiator}})}, \quad (8)
\]

\[
\text{SPF}_{\text{HRV}} = \frac{(Q_{\text{Preheater}} + \dot{Q}_{\text{HX}})}{(W_{el,\text{Preheater}} + W_{el,\text{Fan,HRV}})}, \quad (9)
\]

\[
\text{SPF}_{\text{tot}} = \frac{(\dot{Q}_{\text{HRV}} + \dot{Q}_{\text{sys}})}{(W_{el,\text{HRV}} + W_{el,\text{sys}})}. \quad (10)
\]

In the definition of SPF\(_{\text{HP}}\), the electricity consumption of the two additional fans for the secondary air and the additional ambient air was considered (SFP = 0.093 Wh/m\(^3\) and SFP = 0.057 Wh/m\(^3\), respectively). Instead, the electricity consumption of the supply and extracted airflow fans (SFP = 0.400 Wh/m\(^3\)) are included in the definitions of SPF\(_{\text{HRV}}\),
An additional electric power of 10 W was assumed for the cooling of the compressor of the HP through a fan.

3. Results and Discussion
3.1. Component Level—Evaporator
3.1.1. CFD-Based Design Optimization

Figure 11 shows the simulated models of the variants 1 to 4 (see Figure 7 for the basic features) and their results in terms of incident velocity and streamlines. In Figure 11 (Variant 1), the air inlet occurs laterally, adjacent to the façade, while the discharge takes place orthogonally. The combination of the air inflow with the position of the fans leads to a heavily inhomogeneous flow pattern, with a standard deviation of 56.3%.

Figure 11, variant 2.1 and variant 2.2, show a layout with four parallel axial fans and a rectangular and slightly thinner evaporator compared to variant no. 1. The configuration in variant 2.1 allows on one hand for a better sound insulation on the suction side, as the evaporator acts as a partial sound absorber, and on the other hand a reduced turbulence of the airflow on the inlet surface of the evaporator and thus improved flow homogeneity. This leads with 7.1% to a significantly reduced standard deviation, compared to variant no. 1. A similar configuration has been tested in variant 2.2 with the fans being on the opposite side of the evaporator (i.e., blowing) compared to the one presented in Figure 11, variant 2.1. The inverted position of the fans leads to a worse overall homogeneity of the flow, as also testified by a standard deviation of 23.9%.

Variants with radial fans were also evaluated in the pre-design phase. Figure 11, variants 3 and 4, show the designs and CFD results of two design variants, which include a single radial fan with a 220 mm impeller. The configurations with radial fan delivered a standard deviation of 16.7% for variant no. 3 and 13.7% for variant no. 4, respectively, which
outperform all variants with axial fans, except the one with four axial fans on the suction side of the evaporator. As a conclusion, variants 2.1, 3 and 4 were further considered for design evaluations and combined acoustical and electricity consumption laboratory measurements. Table 3 shows a summary of the investigated configurations with the resulting standard deviations, minimum and maximum velocities on the evaporator surface.

Table 3. Summary of the fan configurations and resulting standard deviations, and minimum and maximum velocities.

| Variant Number | Standard Deviation [%] | Maximum Velocity at the Evaporator [m/s] | Minimum Velocity at the Evaporator [m/s] |
|----------------|------------------------|------------------------------------------|-----------------------------------------|
| 1              | 56.3                   | 1.1                                      | −0.7                                    |
| 2.1            | 7.1                    | 1.1                                      | −0.04                                   |
| 2.2            | 23.9                   | 1.9                                      | 0.06                                    |
| 3              | 16.7                   | 2.0                                      | 0.07                                    |
| 4              | 13.6                   | 2.0                                      | −0.3                                    |

3.1.2. Fan Electric Power Consumption and Sound Emissions

Figure 12a shows the results of the acoustical measurements of two fan configurations. At first, a configuration with four parallel axial fans (as in variant 2.1) with a diameter of 140 mm was tested and compared against a layout with a single radial fan (as in variant 4) with a 220 mm diameter impeller. The measurements shown were performed for a design air volume flow of 350 m³/h and the pressure drop upstream of the fans was controlled by means of a ventilation flap.

Although the single radial fan is able to withstand the pressure drop without almost any repercussions on the sound emissions, the results show that four parallel axial fans are a more favourable choice than a radial fan concerning sound emissions. This applies when the pressure drop upstream of the fans can be kept below 24 Pa, when only the discharge side is considered and below 21 Pa when also the sound pressure level of the suction side is taken into consideration. In addition, the axial fans outperform the radial fan in terms of power consumption, as can be seen in Figure 12b.

3.2. System Level—Refrigerant Cycle
3.2.1. Refrigerant Cycle Characterisation and Optimization

The different developed functional models of the supply air heat pump and the supply air heat pump with recirculation air were tested in the laboratory and performance maps were derived for different source temperatures, compressor speeds, and operation modes. Figure 13 shows the exemplarily experimental results of four characteristic system states for
the case of 2.5 kW nominal heating power for the supply air heat pump with recirculation (see also 2.1.3) for the following cases:

(a) No recirculation and no additional ambient air;
(b) Recirculation air, no additional ambient air;
(c) Recirculation and additional ambient air;
(d) Recirculation and only ambient air.

Figure 13. Measured heating power and system COP for the 2.5 kW heat pump under different compressor speeds as a function of the ambient air temperature. (a) 100 m³/h supply air with no additional ambient air. (b) 100 m³/h supply and secondary air with no additional ambient air. (c) Similar to (b) with 200 m³/h additional ambient air. (d) 100 m³/h supply and secondary air with no exhaust air but only 300 m³/h ambient air [11].

The heating power and COP$_{\text{HP,sys}}$ are plotted for ambient air temperatures between $-7 \, ^\circ C$ to $+10 \, ^\circ C$ with different compressor speeds as a parameter (1500 to 6500 rpm). Figure 13a shows the operation mode for 100 m³/h supply-air flow only. The recirculation volume flow is set to zero. The maximum heating power is limited to approximately 1.2 kW, and a COP$_{\text{HP,sys}}$ of 1.6 can be expected. The power and performance are quasi-independent from the ambient air temperature. However, even at the lowest compressor speed, COP$_{\text{HP,sys}}$ is never higher than 3.0. Figure 13b shows a second mode of operation with an additional recirculation air flow of 100 m³/h. The heating power can be increased to approximately 2.0 kW with a COP$_{\text{HP,sys}}$ of approximately 2.0. Between 5500 and 6500 rpm, the heating load does not further increase because of the limitation of the available source (only exhaust air). Figure 13c shows the same mode with additional 200 m³/h of ambient air. In this mode, at lower compressor speeds, the power and COP decrease. However, for speeds above 5000 rpm, the maximum heating power can be significantly increased to above 2.5 kW with a COP$_{\text{HP,sys}}$ of 1.9. Figure 13d shows the HP in the ambient air only operation mode to allow for a comparison with ambient air HPs. Again, a heating power of 2.5 kW can be reached, but the maximum heating power for low ambient temperatures decreases. However, the COP$_{\text{HP,sys}}$ is only in the range of 1.5. See also Ref. [11] for further
details. It is noteworthy that, because of the façade integration of the outdoor unit of the HP, the maximum possible volume flows are comparably small.

3.2.2. Performance Maps for Building and System Simulation

Two different sets of performance maps (PMs), derived from laboratory tests, were implemented in the (Matlab/Simulink) building and HVAC system simulation model. Figure 14 shows a comparison of the heating power and performance (COP) of a supply air HP with (Figure 14a,b) and without recirculation air (Figure 14c,d).

![Performance maps](image)

**Figure 14.** Performance map of the total thermal power of the heat pump and the COP for supply air heat pump system (top, (a,b)) and supply air heat pump with secondary air recirculation (bottom, (c,d)) with an additional ambient air of 200 m³/h as the source. The total heating power was split between the condenser (50%) for the supply air and the desuperheater (50%) for secondary air.

Both HPs are combined with an ERV. The performance maps are reported between 1000 rpm and 6000 rpm and the ambient temperatures between −10 °C and +10 °C, while the operation of the HP is limited between about 1800 rpm and 6000 rpm and for the supply air heating system to around 1000 W (typically max. 10 W/m²) in order not to exceed the temperature limit of air heating of approximately 55 °C.

In case of supply air HP with a secondary air recirculation system, the thermal power is split between the condenser (50%) and the desuperheater (50%), and an additional ambient air of 200 m³/h was used as the source, in addition to the exhaust air.

An airflow rate of 100 m³/h is supplied to the SL/CH/LI (40/20/40 m³/h, respectively) and is extracted from the KI/BA (60/40 m³/h). An ERV model was implemented using the eta-NTU analytical approach. Sensible (\(\eta_T\)) and latent (\(\eta_w\)) effectiveness of 0.77 and 0.60 are assumed, respectively. The doors of the kitchen and living room were considered to always be open, while the other doors of the flat (i.e., bathroom, child room, and sleeping room) were closed.
Because of the different capacities of both HPs, the COP of the HP is comparable for both systems. If, for better comparability, the exhaust air HP had a larger compressor than the supply air HP with recirculation, so that both delivered approximately 2500 W thermal power at full speed (6000 rpm) and ambient temperature of 10 °C, the COP of the supply air HP with recirculation at full speed would be with 3.5—significantly higher than that of the supply air HP (1.5). The influence of the ambient air temperature is small in case of the supply air HP due to the pre-heating for ambient temperatures below freezing conditions and because of the damping of the MVHR.

Based on the component level (i.e., evaporator) optimisation, as shown in the previous sections, where the influence of the combinations of fan type, evaporator geometry and inflow characteristics was identified, the influence of the air flow guidance on the performance of the HP is discussed. A homogenous airflow allows for a higher evaporation pressure due to the lower pinch point temperature difference \( \Delta T_{pp} \) and consequently better performance. An improved design with better air flow guidance and more homogeneous flow in particular allows for higher compactness or lower fan consumption and consequently reduced fan power. In this work, the evaporator pinch point temperature difference can be improved from 6 K to 4 K, leading to an improvement of the COP from 2.4 to 2.5.

Figure 15 shows this influence of 2 K pinch point temperature difference on the evaporation temperature and, thus, on the performance of the HP in case of an inlet temperature of \(-3\,^\circ\text{C}\) (Figure 15a) with a volume flow in the evaporator of 350 m\(^3\)/h (150 m\(^3\)/h exhaust air with 2 °C and 200 m\(^3\)/h ambient air of \(-7\,^\circ\text{C}\)) and of an inlet temperature of 2 °C (Figure 15b) with 150 m\(^3\)/h exhaust air of 2 °C and 200 m\(^3\)/h ambient air of 2 °C. The main influence of the compact design of the outdoor unit with improved air flow guidance is the reduced fan power and sound emissions, while the influence on the COP is only of second order relevance here.

![Figure 15](image_url)

**Figure 15.** Influence of inflow characteristics and evaporator geometry on the refrigerant cycle for inlet air temperature of (a) \(-3\,^\circ\text{C}\) and (b) 2 °C, with a homogeneous (green dash line) and non-homogeneous (yellow dash line) combination of airflow and evaporator geometry with 350 m\(^3\)/h.

### 3.3. Building-Level—Dynamic Building and System Simulations

#### 3.3.1. Thermal Comfort and Indoor Air Quality

Figure 16 shows the operative temperature of each room of the reference flat (a) and the heating load versus the ambient temperature (b) for the reference case with room-wise electric radiators. The flat requires, as shown above, a maximum hourly heating load of 1220 W corresponding to (17.2 W/m\(^2\)), while during the summer night, cooling and opening of windows can avoid excessive overheating of the flat with no need for active SC systems. Figure 16c,d depict the box plot of relative humidity and CO\(_2\) concentration of each room (thermal zone), respectively. Good thermal comfort and indoor air quality (IAQ) are always ensured as the relative humidity does not fall below the recommended values of 30% and the concentration of CO\(_2\) remains well below 1200 ppm, respectively.
Figure 16. Analysis of reference system (room-wise electric radiator): (a) daily average operative temperatures for each room of the flat; (b) hourly heating load versus ambient temperature of the flat (b) as well as box plot of (c) relative humidity and (d) CO$_2$ concentration (CO$_2$ ambient concentration is 400 ppm) for each room of the flat (KI: Kitchen, SL: Sleeping Room, CO: Corridor; BA: Batch room; CH: child room, LI: Living room).

A very good temperature distribution without underheating and little overheating of all rooms can be seen for the reference case in Figure 16a. There is no need to prove that all three investigated heating concepts enable sufficient IAQ and that there are no significant deviations between these concepts with respect to the IAQ. However, a clear difference among the systems can be recognized with respect to the TC. In contrast to the systems with room-wise electric radiators (see Figure 17a,b) where the temperature control of each room of the flat works independently of the hygienic air flow, in the case of the analysed supply air heating systems ((c) supply air without and (d) supply air with recirculation), the influence of the interdependency between heating power and hygienic air flow can be clearly seen in Figure 17c shows the temperature distribution with the traditional supply air heating system. Relatively high over- and underheating cannot be avoided when the SH is coupled to the hygienic air exchange. The supply air rooms (especially sleeping room, child room, and living room) show a significant number of overheating hours. The disadvantage of coupling the space heating power to the hygienic air flow rate (i.e., fresh air supply and...
heat supply) results in the overheating of supply air rooms in case of the supply air heating system where some underheating occurs in all rooms, except for the bathroom with the additional electric post-heater. The main disadvantage is the overheating of the sleeping and the child room. Extract air rooms tend to be underheated, but the underheating remains within acceptable limits. This is typically not accepted and would lead to excessive window ventilation with correspondingly increasing thermal losses.

![Box plot of operative temperatures (hourly average) during the heating season for (a) the reference (with room-wise electric radiators) and for the three presented heating concepts, (b) split HP with room-wise electric radiators (c) supply air heat pump, and (d) supply air heat pump with recirculation.](image)

Figure 17. Box plot of operative temperatures (hourly average) during the heating season for (a) the reference (with room-wise electric radiators) and for the three presented heating concepts, (b) split HP with room-wise electric radiators (c) supply air heat pump, and (d) supply air heat pump with recirculation.

In case of the system with a supply air HP with secondary air recirculation (see Figure 17d), due to the additional heating of the corridor with the secondary air recirculation, overheating in SL, CH, and LI can be avoided and an improved room-wise control is possible. Still, some underheating cannot be avoided but further improvement of the control is possible to avoid underheating. Here, secondary air heating is always activated if the HP is running. A significant improvement, i.e., a reduction of the overheating in the sleeping room, can be observed. However, some deviation of the room temperatures from the SP during the winter could not be completely avoided. Around 25% of the time the temperature is between 19.0 °C and 20.0 °C. Another 25% of the time the temperatures were higher than 20.8 °C and in only around 50% of the time they were close to the set point of 20.5 °C.

### 3.3.2. Comparison of System Performance

Table 4 shows the heating demand (HD), the heating load of the flat (HL) and the electricity demand (ED) of the three heating systems discussed in the previous section, based also on the set of PMs presented in 0 for an exhaust-to-supply air HP with and without recirculation. In case of the system with a split HP combined with the room-wise electric radiators (Figure 4), three different electricity demands are shown depending on the assumed performance of the split HP (i.e., COP = 2, 3 or 4). The HD is 29.7 kWh/(m² a) for the ideally controlled system with room-wise radiators.

**Table 4.** Annual space heating demand (HD), maximum hourly heating load (HL), and annual electricity demand (ED) of the three heating systems (split HP combined with room-wise electric radiators, supply air HP, and supply air HP with secondary air recirculation.

|                     | Split HP with Electric Radiator (Room-Wise) | Supply Air HP ** | Supply Air HP with Secondary Air Recirculation ** |
|---------------------|---------------------------------------------|------------------|--------------------------------------------------|
| HD/[kWh/(m² a)]    | 29.7                                        | 30.5             | 23.8                                             |
| HL/[W/m²]          | 17.2                                        | 17.3             | 14.8                                             |
| ED/[kWh/(m² a)]    | 27.1/26.3/25.9 *                           | 14.8             | 12.3                                             |

* in case of used COP of 2, 3, or 4, respectively. ** The electric power of fan for compressor cooling) and for secondary air are considered in the calculation of electricity demand.
The slight increase in the heating demand (+3\%) in case of the system with the supply air HP compared to the reference case is due to the overheating of the supply air rooms (see SL, CH, and LI in Figure 10b). Similarly, the system with a supply-air HP with secondary air recirculation features a lower SH demand (−20\%) because of the under-temperature of all the supply air rooms. Correspondingly, the HL is also the lowest with 14.8 W/m² in this case. Because of the lower HD, the electricity demand (ED) is also the lowest in this case and is approximately 50\% of the best variant of the split HP (COP = 4) with the room-wise electric radiators. The relative high electricity demand of the split HP as well as the relative small influence of its COP on the electricity demand prove that in this concept the room-wise electric post-heaters have a significant contribution to the SH and that such a concept cannot be recommended from the performance point of view (but instead this concept is introduced, as mentioned above, to explain the influence of coupling the SH to the hygienic air flow and to see the influence of overheating the corridor with recirculation air).

The annual SPFs (determined during the heating season according to Equations (7)–(10)) for the three different air HP-based heating concepts are shown in Figure 18. The SPF_{HP} is slightly higher for the case of a supply HP with recirculation air (2.43) compared to the case without (2.03) because of the lower supply temperature of the HP. The system with a simple supply air HP presents the highest SPF_{sys} (2.07) and this value is higher compared to the one with additional recirculation air (1.93) because of the lower electricity consumption for the heating of the bathroom. SPF_{HRV} is slightly lower for the case with room-wise electric radiators and for the case with a split HP (5.60) compared to heating concepts based on supply air HPs, without or with recirculation air (6.05 and 6.00, respectively) because of the higher W_{el,FanHRV} for the split-HP systems due to the slightly longer heating period. SPF_{tot} is 3.2 for the supply air HP with additional secondary air recirculation, slightly higher compared to the case without (3.15) and significantly higher than the system based on a split HP (2.03), even in case a COP of 4.00 is considered for the split HP. Both supply-air HPs represent a significant improvement compared to the split-HP system with room-wise radiators.

| COP   | SPF_{HP} | SPF_{sys} | SPF_{HRV} | SPF_{tot} |
|-------|---------|-----------|-----------|-----------|
| 2     | 2.00    | 1.95      | 1.13      | 2.01      |
| 3     | 3.00    | 2.01      | 1.15      | 2.03      |
| 4     | 4.00    | 2.03      | 1.15      | 2.03      |

Figure 18. Annual (heating season) SPFs of the heat pump (“HP”), heating system (“sys”), HRV and total HVAC (“tot”), according to Equations (7)–(10) for the three investigated systems.

The results of the dynamic building and system simulations determined the thermal comfort and the overall system efficiency as well as the potential to improve the control of the system, supporting the potential of such developed concepts. The system performance SPF_{sys} is around 2.0 (including a comfort E-heater in the bathroom). A further optimization of the control strategy has the potential to allow a room-wise temperature control without threatening the performance. However, algorithms for such an application need to be
further developed and tested in real conditions. Furthermore, improved de-icing control is of particular importance for exhaust air HPs.

3.4. Façade-Integration and Mock-Ups

3.4.1. Mock-Up of Supply-Air Heat Pump with Recirculation of Secondary Air

Functional models were developed based on simulation results and tested in the laboratory. Figure 19 shows a sketch (a) and a photo (c) of the functional model of the indoor unit. The outdoor unit in Figure 19b,d includes the evaporator, compressor, and the inverter. The exhaust air flows over the upper part of the evaporator. A minimum volume flow of 35 m$^3$/h of ambient air is maintained during the heat-pump operation to cool the compressor and the inverter. If necessary, additional ambient air can be used to increase the heating power. An electrical expansion valve (EEV) is used to control the superheating. The evaporator is a plate-fin-tube heat exchanger (PFTHE).

Figure 19. Scheme of the indoor unit with MVHR, condenser, and desuperheater as well as sound absorbers (a) and outdoor unit with evaporator, compressor, and inverter (b) of the developed functional model of an exhaust-air to supply-air heat pump with recirculation air, see also Ref. [11]. (c,d) photos of the indoor and outdoor unit, respectively. (1) HRV/ERV; (2) condenser (supply air); (3) condenser/desuperheater; (secondary air); (4) fresh air; (5) exhaust air; (6) evaporator; (7) radial fans; (8) inverter; (9) compressor.

3.4.2. Mock-Up of a Façade Integrated HP Outdoor Unit

In refurbishment projects, the available space is limited, and typically no additional space is available for HVAC systems (see also Ref. [19]). Besides, integration into a prefabricated façade element allows for short construction times and thus enables a minimal disruptive renovation. Considering the outcomes of the fluid-dynamics analysis and the measurement
results for the sound power level and electrical power consumption, two mock-ups of façade-integrated outdoor unit were conceived, as shown in Figures 20a–d and 21a–d.

Figure 20. Façade integrated outdoor unit of an exhaust to supply air heat pump with external compressor (a), (b) airflow streamlines (for velocity) inside the mock-up from CFD simulations with two radial fans, picture of the functional model of the outdoor unit ((c) with open casing and (d) with closed casing) resulting from the CFD analysis realised within the Austrian (FFG) research project Saluh! (see also Ref. [11]).

Figures 20a and 21a show simplified schemes of the outdoor unit integrated in the façade. The main differences between the two developed designs are in the number and type of fans and in the position of the compressor, which is housed inside the outdoor unit for the design of Figure 20a while for the design of Figure 21a it is directly integrated in the indoor unit inside the flat, containing also the condenser of the HP.

For the design shown in Figure 20, it was not possible to achieve sufficiently good flow homogeneity, also due to the space constraints of the developed unit (see also Figure 20b). To guarantee a safe de-icing operation, two radial fans were adopted despite a generally higher electric power consumption. In the case of Figure 21, a more homogeneous airflow was realised with a different concept and with the adoption of four axial fans. A standard deviation of 13% was obtained in the corresponding simulated CFD model (see Figure 21b).

The mock-ups were then built and installed in PASSYS test cell for testing, see Figure 20c,d, Figure 21c,d and Figure 22. With this experiment, the handling during the installation was tested and the functionality was proven. A thorough investigation of the details including a moisture-proof design is required. A key aspect of façade-integration is accessibility and maintenance. The appropriate solution depends on the type of building and in particular on the floor plan, and it has to be proven individually.
Figure 20. Façade integrated outdoor unit of an exhaust to supply air heat pump with external compressor (a), (b) airflow streamlines (for velocity) inside the mock-up from CFD simulations with two radial fans, picture of the functional model of the outdoor unit ((c) with open casing and (d) with closed casing) resulting from the CFD analysis realised within the Austrian (FFG) research project Salüh! (see also Ref. [11]).

Figure 21. Façade integrated outdoor unit with four axial fans (a), (b) airflow streamlines (for velocity) inside the mock-up from CFD simulations and picture of the functional model ((c) with open casing and (d) with closed casing) of the outdoor unit resulting from the CFD analysis realised within the Austrian (FFG) research project FiTNeS (see also Ref. [35]).

Figure 22. PASSYS test cell (a) and functional model of a prefabricated wooden façade element with integrated outdoor unit of the heating-ventilation heat pump in the PASSYS test cell of the UIBK. (b) Installation of the façade element. (c) Execution (Holzbau Kulmer, Pischelsdorf am Kulm, Styria, Austria).

4. Conclusions

Accomplishing the goals of an energy efficient building stock as one of the major challenges in the transition of the energy system to a sustainable one requires cost-effective industrialized solutions for the deep thermal renovation of buildings and for the change from fossil heating systems to HP-based ones. Often a central renovation of the HVAC
system is not possible, due to several reasons such as flat-wise renovation (e.g., in case only few owners or tenants agree to renovate) or due to space restrictions. Prefabricated façades with integrated HPs for minimal invasive deep serial renovation of residential multi-apartment buildings were developed and optimized following a comprehensive and holistic design and optimization procedure. Starting the optimization from the component-level, i.e., HP outdoor unit (evaporator), via system level, i.e., refrigerant cycle, up to the building level (coupled multi-zone building and HVAC simulations), the goal of a compact efficient and silent HP-based heating solution with high degree of prefabrication and façade integration could be achieved.

The presented serial renovation concept is based on prefabricated façades in combination with an exhaust air to supply air HP combined with a mechanical ventilation system. Several functional models and mock-ups were developed and tested in order to prove the feasibility of the concept and to calibrate the simulation models (evaporator, refrigerant cycle) for further optimization.

Various optimization options and levels are presented addressing different (and partly competing) goals such as compactness and attractiveness, minimal sound emissions, and high efficiency. The different optimization levels are connected, but there is not one unique concept and design, since depending on the application and specific site conditions (i.e., quality, size and floor plan of the building, etc.), different combinations and options should be chosen.

An innovative exhaust-air to supply-air HP in combination with a mechanical ventilation system and with additional secondary air on the sink side and ambient air on the source side proved to be a promising solution and also feasible when PH renovation standards (i.e., 10 W/m$^2$) cannot be reached and thus traditional supply air heating systems cannot be applied. One of the advantages of the developed solution with secondary air recirculation (compared to a traditional supply air heating system), is the possibility to control the heat distribution and to avoid overheating in the supply air rooms. The overall thermal comfort can be increased, while at the same time a higher heating load can be provided. A heating power of up to 2.5 kW with a SPF$_{HP}$ of more than 2.7 and a system SPF of 2 could be obtained (for the climate of Innsbruck). Instead, with a traditional supply-air heating system, the heating power is limited to approximately 1.0 kW because of the limit of hygienic air flow rates.

An important finding is that in such very compact HPs with relatively low overall heating power (max. 2.5 kW), the heat losses in all components have to be minimised in order to obtain a sufficiently high energy performance. To achieve that, a holistic simulation-assisted design is recommended, which should include optimisation on component, system, and building levels (i.e., control). Compactness is required to enable façade integration and allow for attractive design. Minimal sound emissions are required to obtain a high degree of acceptance. Speed-controlled compressors can be very beneficial in such an application (no SH buffer storage, air heating).

The proposed system can represent a breakthrough for serial minimal invasive renovation, in particular in cases where central solutions are not possible and when a hydronic heating system does not exist.

However, further development is still necessary. In particular, the accessibility for maintenance requires new concepts and further improvement. Ongoing work within the Austrian FFG projects FiTNeS and PhaseOut concentrates on the development of an effective and attractive design of façade-integrated heat pump outdoor units, and their demonstration in real buildings.

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Nomenclature

| Symbol | Description |
|--------|-------------|
| E      | Energy [kJ] |
| h      | Specific enthalpy [kJ/kg] |
| L_{w,p} | Sound pressure level [dB(A)] |
| P      | Pressure [Pa] |
| Q      | Heat [J] |
| W      | Work [J] |
| P_{el} | Electrical power [W] |
| Q      | Heat flux [W] |
| T      | Temperature [K] |
| v      | Velocity [m/s] |
| \eta   | Effectiveness [-] |
| \theta | Temperature [°C] |
| \varphi | Relative humidity [%] |
| \sigma | Standard deviation |
| \omega | Humidity ratio [kg/kg] |

Abbreviations

| Abbreviation | Description |
|--------------|-------------|
| BA           | Bathroom    |
| CFD          | Computational Fluid Dynamics |
| CH           | Child room  |
| COP          | Coefficient of performance |
| DB           | Dry bulb    |
| DESUP        | Desuperheater |
| DHW          | Domestic hot water |
| ED           | Energy demand |
| EE           | Enthalpy exchanger |
| EER          | Energy efficiency ratio |
|ERV           | Energy recovery ventilation |
| EEV          | Electronic expansion valve |
| EXH          | Exhaust air |
| EXT          | Extract air |
| H            | Enthalpy   |
| HD           | Heating demand |
| HE           | Heat exchanger |
| HL           | Heating load |
| HG           | Hot gas    |
| HP           | Heat pump  |
| HRV          | Heat recovery ventilation |
| HVAC         | Heating ventilation and air conditioning |
IAQ  Indoor air quality  
INF  Infiltration air  
KI  Kitchen  
KPI  Key Performance Indicator  
LCA  Life cycle analysis  
LCC  Life cycle cost  
LI  Living room  
MVHR  Mechanical ventilation with heat recovery  
NTU  Number of transfer units  
op  Operative  
PFTHE  Plate-fin-tube heat exchanger  
PH  Passive House  
PostH  Post-heater  
PP  Pinch point  
PrH  Pre-heater  
REC  Recirculation air  
SC  Space Cooling  
SCOP  Seasonal coefficient of performance  
SEC  Secondary air  
SEER  Seasonal energy efficiency ratio  
SH  Space heating  
SL  Sleeping room  
SFP  Specific fan power  
SPF  Seasonal performance factor  
SUP  Supply air  
SYS  System  
T  Temperature  
TC  Thermal comfort  
tot  Total  
TP  Two phase

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