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Variable Speed Exhaust Air Heat Pump with Enhanced Deicing Control

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Abstract
A micro-exhaust air heat pump in combination with a mechanical ventilation system with heat recovery (MVHR) offers great potential for a cost effective heat supply for highly energy-efficient buildings. Particularly, for the renovation of multi-family houses, where central solutions are often not possible for various reasons, a decentralized MVHR with heat pump represents an appropriate solution for ventilation, heating and cooling. A major advantage of such a compact system is the ability of façade integration: a space-saving and attractive solution can be achieved while the renovation can be accomplished with minimum disruption, which is one of the main objectives of the EU-project iNSPiRe. A challenge in the development of such a compact solution is maximizing the efficiency within the space limits and in the same time reducing noise emission to minimum. In addition, accessibility for maintenance and repair has to be guaranteed. The concept of the micro-exhaust air heat pump is described, opportunities and challenges of optimization are discussed. Results of component, system and building simulations as well as measurement results are presented. Measurements include indoor laboratory measurements of the icing behavior, outdoor laboratory measurement results of the system efficiency in so-called PASSYS test cells as well as monitoring results of a demo building in Ludwigsburg (Germany).

Keywords - MVHR; micro-heat pump; deep renovation; cost-effective; efficiency

1. Introduction

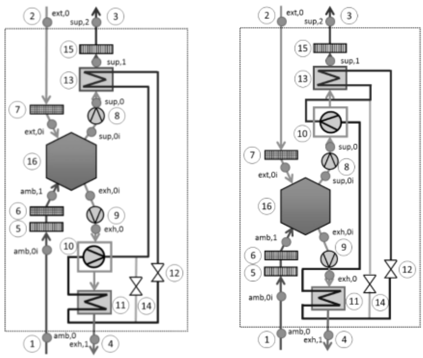
The majority of existing building stock in Europe and worldwide is poor energy performance buildings and renovation plays a major role in achieving climate protection and energy independence. Deep renovation solutions in combination with integrated HVAC systems are developed within the framework of the European project iNSPiRe [1, 2]. In this work the development, testing and modelling of a façade integrated micro-heat pump (µ-HP) in combination with mechanical ventilation with heat recovery (MVHR) is presented. Different functional models are developed in the framework of the EU-project iNSPiRe and are measured in PASSYS test cells (Passive Solar Systems and Component Testing) at the laboratory of Innsbruck University (UIBK) and will be later monitored in a demo building in Ludwigsburg, Germany. It is an example of social housing built in the 1970s, which contains 4 flats on 4 stories. During the renovation process a prefabricated timber frame façade was fitted onto the building.

2. Motivation and Concept

In the framework of the project iNSPiRe a new variable-speed micro-heat pump (micro-HP) with a heating capacity of approx. 1 kW is developed for use in highly efficient buildings. The development of the micro-heat pump is led by UIBK together with the Austrian company SIKO Energiesysteme (in collaboration with the companies Pichler Luft and WGT Elektronik). The compactness of the unit allows to integrate it into the façade. Within iNSPiRe the exhaust-air heat pump in combination with a mechanical ventilation system with heat recovery is integrated in a prefabricated wood frame façade element from the German company Gumpp & Maier. Aspects of façade integration of the micro-heat pump can be found in [2, 3].

The exhaust air of mechanical ventilation with heat recovery (MVHR) is the source of the micro-heat pump. Pre-heating (defrost) and post-heating (backup) for peak load coverage are required. Additionally, a bathroom radiator is recommended for comfort reasons. Domestic hot water preparation is done separately in this concept, e.g. with an additional air-to- water HP.

In the framework of the project iNSPiRe two functional models have been built for measurements in the PASSYS test cells of the UIBK and for thermodynamic analysis. A further functional model was created for the measurement of a façade integrated MVHR. With this functional model the hygrothermal behavior has been investigated and the façade integration has been optimized (e.g. avoiding thermal bridges and condensation, accessibility, etc.). Furthermore, the calibrated effective heat transfer capability (UA-value) for the air-to-air heat exchanger were derived from these measurements. A fourth functional model with micro-heat pump was built for measurements in the acoustic test cell of UIBK. Sound pressure levels were measured and the system was optimized for reduced sound emissions. Fig. 1 shows the hydraulic scheme and the measuring points for the first (left) and second (right) functional model.



1 Ambient Air
2 Extract Air
3 Supply Air
4 Exhaust Air
5 Defroster MVHR
6 Filter Ambient Air
7 Filter Extract Air
8 Ventilator Supply Air
9 Ventilator Exhaust Air
10 Compressor (variable speed)
11 Evaporator
12 Throttle (Expansion valve)
13 Condenser
14 Hot gas defrost bypass
15 Post heater
16 Air-to-Air Heat Exchanger

Fig. 1 Hydraulic Scheme and measurement points for both functional models

3. Simulation Model

a. Steady-State Simulation

In the framework of this research, a simplified steady-state simulation model of the micro-heat pump and the MVHR was developed in MATLAB using the CoolProp library. The model uses a simplified approach for the thermodynamic refrigeration cycle. The condenser is divided into three sections (i.e. desuper-heating, condenser, and sub-cooling) with different heat transfer coefficients for determining the energy balance between refrigerant and moist air. The evaporator is also divided into three areas (evaporation with high liquid content up to a steam quality of 0.85, evaporation and superheating). Sub-cooling and superheating temperature difference are inputs to the model and determine the share of area of the condenser and evaporator, respectively. The each two remaining areas of condenser and evaporator are determined with a numerical minimization methods using Matlab built-in functions.

The efficiency of the compressor is described with a characteristic curve of the isentropic efficiency, depending on the speed of the compressor and the evaporator or condenser temperature, or pressure ratio, respectively. The mass flow of the refrigerant is a function of the so-called "displacement" as a function of the volumetric efficiency, which is also described by a characteristic curve depending on the compressor speed and again the evaporator or condenser temperature. The corresponding characteristics can be computed from the manufacturer's data sheet. The expansion valve is modeled as an isenthalpic throttle. Pressure losses of the components can be considered but are negligible in this study.

The mechanical ventilation system with heat recovery is simulated also stationary. The supply and exhaust air temperature are calculated with a calibrated heat transfer capability (effective UA-value) and the logarithmic temperature difference for a given ambient and exhaust air temperatures.

The model is validated against measurements (see next section) and is used for the simulation based evaluation of measurements and for optimization of the micro-heat pump. A detailed description of the model including the physical and mathematical correlations, as well as a parametric study can be found in [4, 5].

b. Transient Simulation

For the transient simulations, a performance map model of the MVHR and the micro-heat pump was derived from the calibrated physical model. Time constants for the simulation model with PT1 transfer function are derived from the measured data and take the thermal losses during the start-up or cooling of the heat pump and its components into account (e.g. during deicing). The influence of the start-up of the heat pump on the electrical power input can be neglected. The de-icing of the heat pump is realized by means of a hot gas bypass, which is modeled in the simulations by a constant electrical power consumption of the compressor at maximum speed (regardless of the current evaporator temperature). The optimization potential of the speed control of the compressor with respect to the overall system efficiency are presented in [8].

c. Simulation and Experiment in HiL Environment

The experimental setup (PASSYS Test Cell, see e.g. [3]) has been recently enhanced by means of implementing hardware in the loop (HiL) environment. Currently, dynamic tests are conducted to improve the controller of the micro-heat pump for the behavior in the demo building in Ludwigsburg using a calibrated building simulation model. Results will soon be available, see [11].

4. Experimental Investigations and Simulation based Analysis

The two functional models of the micro-heat pump were measured in the PASSYS test cells at UIBK which also allow to consider the additional thermal losses (influence of the façade integration). The measuring system is based on components from National Instruments. Pt100 sensors are used for temperature measurements, which have an accuracy of approx. $(0.05 + 0.0005 \cdot \vartheta)$ °C after a calibration over the entire measurement chain. The humidity measurement is carried out using digital SHT75 sensors, which have an accuracy of some ± 3% in relative humidity. The volume flow measurements are realized with measuring rings and with differential pressure sensors. The volume flow measurements were compared to measurements with a Wöhler Compensation Flow meter. The accuracy that can be achieved is in the range of ±5%. The simulation model described above was calibrated against the measured data. The position of the sensors is indicated in Fig. 1. For a more detailed description of the measurement setup see [6, 7].

5. Increasing the system performance factor

a. MVHR

The mechanical ventilation with heat recovery is the key component for the overall system efficiency. The following optimization approaches could be developed:

i. Effectiveness of the MVHR

The effectiveness of the air-to-air heat exchanger plays a crucial role. With a higher heat recovery efficiency the heating load of the building can be decreased and thus the use electrical energy for the heat pump and the backup (post-heater) can be reduced. A higher heat recovery efficiency leads to lower exhaust air temperatures and thus to lower source temperatures for the exhaust air heat pump. Fig. 2 (left-hand side) shows the system efficiency (COP system, eq. 2) and heat pump efficiency (COP heat pump, eq. 1) depending on the relative heat transfer capability ($UA_{ratio} = UA_{eff}/UA_{eff,norm}$), which is equivalent to a scaling the heat exchanger area (the overall heat transfer coefficient can be considered constant because of the laminar flow regime).

$$COP_{HP} = \dot{Q}_{cond}/P_{el,comp} \qquad (1)$$

$$COP_{sys} = (\dot{Q}_{cond} + \dot{Q}_{MVHR})/(P_{el,comp} + P_{el,fan}) \qquad (2)$$

The ambient air temperature is -6.5 °C, which corresponds to the temperature limit just before the switch-on point of the electric pre-heating for frost protection.

Fig. 2 right hand side shows the power of the system and of the heat pump, respectively, depending on the relative UA-value. While the COP of the system is increasing with increase of the UA-ratio the COP of the heat pump decreases. The reduction of exhaust air temperature with increasing heat transfer capability leads only to a relatively small reduction of the COP of the heat pump. From this, it can be derived that increasing the effectiveness of the MVHR has a positive impact on the overall efficiency of the system in any case.

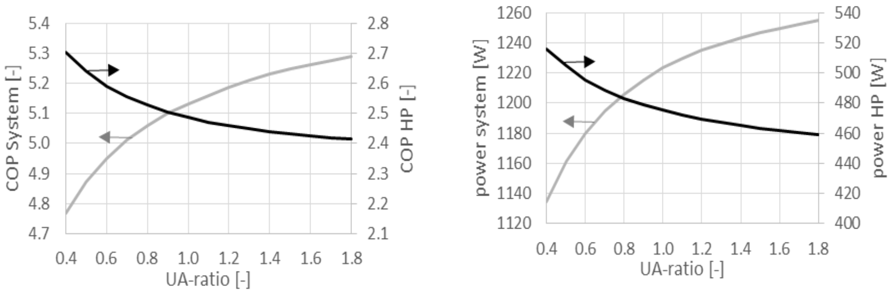


Fig. 2 Dependence of the HP and system COP (left) and of the HP and system power (right) from the heat recovery efficiency expressed in relative heat transfer capability ($UA_{eff}/UA_{eff,norm}$)

The effectiveness of heat recovery depends on the balance of the two mass flows (supply and extract air). This balancing should be controlled automatically by the MVHR. In particular the reduction of the flow rate of the exhaust air due to the freezing of the evaporator must be prevented. This was not yet specifically considered in the framework of this project but represents an important aspect in future work.

ii. Pre-heating

A frost-free operation of the air-to-air heat exchanger is necessary. Below an ambient air temperature of about -6 °C, frost protection is realized by means of a heating coil (pre-heater), which is activated if exhaust air temperatures are measured below approx. 1 to 2 °C. The pre-heating register has a heat output of approx. 560 W, which is almost independent of the outside air or exhaust air temperature despite the use of a PTC element. This leads to on/off operation of the heater and significant overheating of the exhaust air. A control (for example, PI controller) using the exhaust air temperature as variable would significantly reduce the demand for electricity for the electric pre-heating. This influence is investigated by means of dynamic simulation and is part of future work.

With the application of enthalpy exchangers the pre-heating for frost protection can be avoided or at least reduced. However, currently the available products are not competitive with respect to the costs. Future developments might enable a more compact construction and operation without frost protection.

b. Micro-Heat Pump

The optimization potential of the micro-HP is limited due to the limit of the hygienic flow rate at the evaporator and condenser. The performance of a variable speed compressor, the choice of the refrigerant and the control strategy have the highest influence on the system performance.

i. Compressor

Two different compressors from the company EMBRACO have been tested in the two functional models, which significantly differ in their cooling capacity. In case of the first functional model, refrigerant R290 is used. The compressor has an electrical power consumption (at full speed) of approx. 550 W (at 2 °C exhaust air temperature and a supply air temperature of about 47 °C). For the second functional model refrigerant R134a was used. The compressor has an electrical power consumption (at full speed) of approx. 200 W (at 2 °C exhaust air temperature and a supply air temperature of approx. 38 °C).

Fig. 3 shows the COP (as ratio of thermal performance on the condenser divided by the power consumption of the compressor) of both micro-heat pumps depending on the ambient air temperature and the compressor speed. In Fig. 3 left hand side, the first functional model with R290 has significantly lower COP compared to the second functional model with R134a, as in Fig. 3 right hand side. With R290 the compressor has a thermal power of 830 W with a COP of 1.65 at full speed and 420 W at a COP of 2.18 at the lowest speed at an ambient air temperature of 2 °C. The second functional model with R134a achieves a maximum output of 530 W at a COP of 2.60 and a minimum capacity of 320 W at a COP of 4.30. The significant improvement of the COP can be explained on the one hand with the fixed air flow rate of 90 m³/h (hygienic flow rate) for which the first compressor was over dimensioned. Furthermore, the compressor was not optimized for R290.

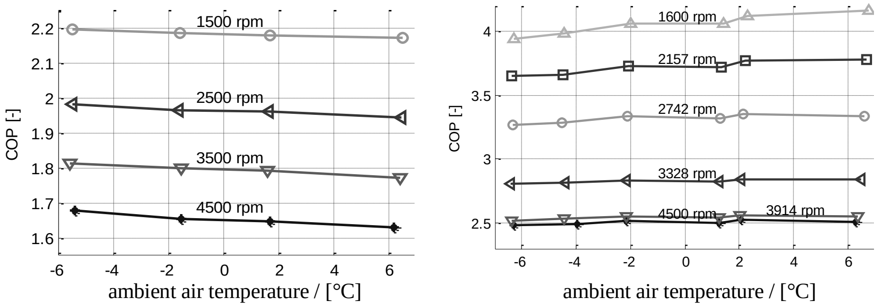


Fig. 3 COP of the heat pump for refrigerant R290 (left) and R 134a (right)

Only a small dependency of the COP on the ambient air temperature can be recognized. This can be explained by the MVHR, the source is the exhaust air which varies less than the ambient air temperature. The thermal power decreases with decreasing ambient air temperature. The control range of the heat pump has optimization potential. Between -6 °C and 7 °C ambient air temperature the control range between maximum speed for low ambient air temperature (-6 °C) and minimum speed for higher ambient air temperature (7 °C) is for the first functional model 40%, but for the second functional model only 27%.

ii. Sizing of Evaporator and Condenser

The choice of the surface of the evaporator and condenser directly depend on the required thermal capacity. The challenge of maximizing the efficiency of such a compact solution is the space limits. Furthermore, the power and performance of evaporator and condenser are limited to the hygienic flow rate, i.e. 80 m³/h to 150 m³/h for a standard flat with three or four persons.

In the evaporator, there is a tradeoff between the distance of the fins and the evaporator surface for a given volume. Due to the icing during operation, the distance between the fins is reducing leading to an increase of the pressure drop and consequently to a higher electric consumption of the exhaust air fan (and or disbalance in the MVHR). An excessive distance of the fins leads to a larger volume of the evaporator for a given surface and to potentially also to a poorer heat transfer coefficient. Figure 4 shows power and COP vs. evaporator (left) and condenser (right) surface area.

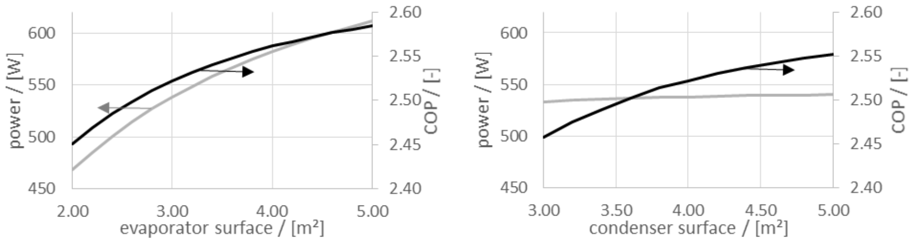


Fig. 4 Power and COP vs. evaporator (left) and condenser (right) surface area

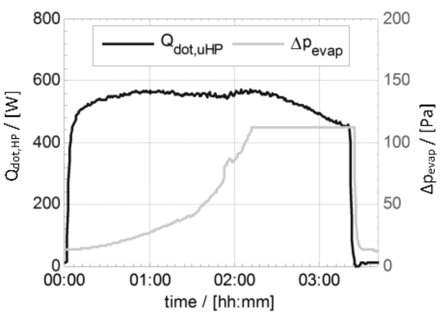
iii. Deizing Control

The freezing of the evaporator has a significant impact on the annual performance factor of the heat pump. This transient ice formation process results on the one hand in a decrease of the performance over the operation period due to increase of the overall heat transfer coefficient, and on the other hand in an increase of pressure drop. The pressure loss can be reduced by a larger distance of the fins at the expense of the heat transfer coefficient. The decrease in performance concerns in particular the first 45 min. run time. Defrosting is realized here via a hot gas bypass, i.e. during defrost period, the electrical output of the compressor is converted directly into heat to defrost the evaporator. As a consequence, there is no heating provided during this period by the micro-HP, and heating has to be provided fully by the post-heater. It can easily be concluded that the aim must be to maximize the runtime considering pressure loss and keep the deicing interval as short as possible.

Usually, a simple time based control of the defrost cycle is implemented with e.g. a 60 min. operation of the heat pump followed by a 10 min. defrosting period. Such a control is obviously not depending on the actual degree of icing (depending on the boundary conditions). Hence, the operating time must be reduced to a safe period of time. For fixed speed heat pumps, the degree of icing depends only on the source temperature and humidity which varies only slightly in case of an exhaust air heat pump. In case of variable speed heat pumps the degree of icing depends further on the actual heating capacity.

One way for indirect measurement of the degree of icing is the measurement of the pressure loss through the evaporator. Usually such a measurement is affected by the wind induced over or under pressure. The façade integration and the automatic balance compensation of the air handling unit reduces this influence strongly. Fig. 4 (left) shows the results for a test run at -5 °C ambient temperature and full speed of the heat pump. The performance of the heat pump and the measured pressure difference via the evaporator are shown (the sensor value is limited to approx. 110 Pa). A slight decrease of the heating power can be detected after approximately 60 minutes of operation.

After about 100 minutes, the balance compensation of the MVHR leads to a change of the working point which results in an increase of the flow rate and thus to an increase of the pressure loss. This higher volume flow results in a higher exhaust air temperature. This leads to melting of some ice and a short period of increased power output. The sensor for static pressure difference reaches its limit of some 110 Pa after 120 minutes, the electric power consumption of the MVHR is approx. 50 W (reference value 38 W) at this time. The heating capacity decreases until ca. 130 minutes of operation only slightly and then significantly. This is the latest point when the de-icing cycle has to start. The electrical power consumption of the MVHR is increased to approx. 57 W at this time. Compared to a conventional control, the run time of the heat pump at full load can be more than doubled. In variable-speed operation mode or in case of increased exhaust air temperature a complete defrost cycle might be avoided, see Fig. 5 right hand side as an example.



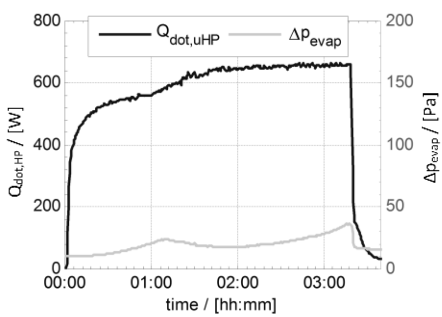


Fig. 5 Pressure difference via the evaporator and thermal power; (left) maximum speed (right) change of exhaust air temperatur due to an increase of higher indoor air humidity [9]

In the full load operation the electric power consumption can be reduced with 130 min. instead of 60 min. runtime and a defrost period of 10 minutes with a direct electrical coverage of the heating load during defrosting by the post-heater from 6.29 kWh/d to 5.43 kWh/d, i.e. by 13.8%. The optimization potential of improved deicing control will be further investigated by means of simulation and experiment in a hardware-in-the-loop test environment (coupling of simulation with hardware measurements in real time in the PASSYS test cells) in ongoing research projects.

A further measurement method with excellent robustness represents an automated optical monitoring of the degree of icing, see Fig. 6 left: evaporator with and without icing. Further results can be found in [9]. However, those detailed measurement methods are relatively expensive and technically complex. A prediction model by means of simulation of the icing based on less sophisticated sensors (such as temperature and humidity) allows also to extend the life of the heat pump (see Fig. 6).

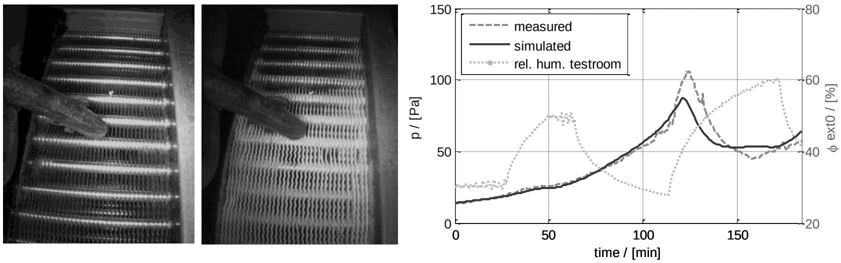


Fig. 6 Pressure difference via the evaporator and thermal power; (left) maximum speed (right) change of exhaust air temperatur due to an increase of higher indoor air humidity

Such prediction models can be based either on physical equations (white box models) or on empirical or so-called black box models. Existing physical models for ambient air for evaporators seem to be not accurate enough for the prediction of the icing behavior of the icing of an exhaust air heat pump with different temperature and humidity range, see [10] for more details. Empirical prediction models must be calibrated and linearized over a wide range. Fig. 5 shows the result of a simple empirical model, which uses only the exhaust temperature of the MVHR as a parameter [9]. The dynamic step of moisture during the operation leads to a change of the exhaust air temperature and absolute humidity, which can be predicted with sufficient accuracy. Such a control would be cost-neutral as only sensors are used that are required anyway (temperature sensor in the exhaust air). Due to the uncertainties of the model caused by the linearization, an absolute time limit of run time is still required.

6. Conclusions and Outlook

The objective of this work is the development and optimization of a façade-integrated ventilation system with heat recovery combined with a micro-heat pump. The most important optimization parameters were examined and evaluated according to their technical feasibility. There is a significant optimization potential in the defrost strategies. The use of an exhaust air heat pump allows new and robust concepts with a considerable cost saving potential. The concept of micro- heat pump will be further developed in ongoing projects with focus on optimizations in moisture recovery, cooling, sound, pressure losses and control are examined.

A new test rig for ventilation systems, compact units (MVHR and heat pump for ventilation heating and DHW preparation), micro-heat pumps and Split Units is soon in operation at UIBK, where such systems can be further developed and optimized.

Acknowledgment

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