Energy and Economic Performance Analysis of Heat Recovery Devices Under Different Climate Conditions

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ABSTRACT

High performance buildings typically rely on extreme envelope performances in terms of insulation level, solar gains valorization and airtightness. The energy needs are limited and the system sizing is reduced to facilitate the recourse to renewable energy sources. The only exception to this trend is related to the ventilation and the associated energy needs, which is even more crucial because of the tight infiltration control. Mechanical ventilation is a common option to ensure an appropriate ventilation rate and a suitable indoor air quality. Heat recovery devices are widely used to save energy in buildings mechanical ventilation systems. Their contribution in the design or operating phases is typically assessed in terms of effectiveness, which expresses the ratio between the amount of energy – sensible and/or total – actually recovered and the maximum recoverable under ideal configuration. However, the effectiveness of recovery devices is not a constant value and depends on the operating – outdoor and indoor –conditions and control strategy. In order to evaluate the energy and cost savings allowed by heat recovery systems, these aspects have to be considered, and their relative impact on different heat recovery technologies accounted for. In this work, some of those aspects are analyzed in detail and their contribution to the device effectiveness is discussed. The objective is to understand the influence of the air conditions (fresh and exhaust) on the energy savings considering different sensible and total heat recovery devices and operative constraints which could affect the control strategy. The analysis starts from hourly weather data, representative of different climate types, to quantify the actual effectiveness of different heat recovery systems. The impact of indoor conditions on the sensible and latent recovery is analyzed, defining appropriate control strategies to prevent excessive moisture recovery and discomfort issues when the indoor humidity is considered. Moreover, the effect of condensation and frosting on the hourly and annual performance of each kind of device is analyzed. Three different climatic conditions have been considered. Sensible fixed-plate cross flow heat recovery system has been compared with enthalpy wheels. By means of a parameterization of the operating constraints and the expression of the main quantities in specific terms, depending only on the building use, it has been possible to generalize the main results, making them independent of a specific case study, and to compare different technologies and climatic contexts. Seasonal and annual energy and economic performances have been quantified, providing a synthetic overview of the technical and economic aspects of the considered technologies.

1. INTRODUCTION

The movement toward better and better insulated buildings is accompanied by not only a reduction in energy consumptions but also the necessity of better ventilation. The energy consumption of buildings equals 41% of the final energy uses in Europe. More than 60% of it is due to space heating (Bosseboeuf et at.; 2012). Space heating is more efficient if the building is equipped with an airtight envelope, which reduces heat losses by unwanted infiltration. In contrast, the lack of natural ventilation and of fresh air, compromises indoor air quality, and thermal comfort can hardly be reached without additional devices (Rasouli, Simonson, & Basant, 2010). To overcome these problems it is possible to install a mechanical ventilation cycle that supplies fresh air for achiving indoor air quality. The need of a suitable ventilation rate, and the use of those devices, contributes to an additional energy demand and raises the total energy consumption of the building. The most effective solution to reduce energy need due to mechanical ventilation are heat recovery devices.

The effectiveness of an air-to-air heat recovery device can limit the energy demand that is necessary to take the supply air to the indoor comfort conditions, which is known as the building ventilation load. To contain these needs while providing appropriate comfort conditions, a control strategy is required aimed at maintaining suitable indoor air conditions, namely temperature and particularly relative humidity. The influence of these conditions was investigated though field tests and experiments. Generally, a high tolerance in the sensation of relative humidity is often reported by field studies (Hwang, Lin, & Kuo, 2006) (Mishra & Ramgopal, 2014). Nevertheless, some further works discuss an influence of the relative humidity not only on the comfort sensation but also on the occupants' performance. A theoretical approach by Kosonen and Tan (2004) based on reviewing results – using the Predicted Mean Vote (PMV) as a tool to estimate the productivity correlations – led to the conclusion that a rising relative humidity is expected to increase productivity losses. In addition, they showed that occupations requiring a higher activity level, and therefore leading to a higher specific latent load simply due to a rising metabolic rate, could result in a more significant performance loss if the humidity is not controlled. Tsutsumi et al. (2007) investigated experimentally the correlation between a step change in humidity level and occupants' comfort as well as productivity. Even though there were no significant correlations between the sensation or performance and the humidity variation following a step-change, it could be seen that the total number of complaints for relative humidity of 70% were higher than for any lower humidity value. The authors stated that longer exposure time to higher humidity in their climate test chamber possibly leads to a performance loss. In a previous paper similar observation were made, within a similar study with college-aged test persons, which had more complaints about fatigue, but for low humidity (Y. Chen *et al*., 2003). Apart from the thermal comfort sensation and the productivity of occupants, there are also effects on the building materials and occupants' health, which can occur when humidity is not properly controlled, as for instance fungi and mold growth or surface or interstitial condensation (Sterling, Arundel, & Sterling, 1985). Thus, the effect of ventilation device on not only the temperature, but also the humidity, should be considered in the control strategy to avoid the described inconveniences.

Not only the control strategy of heat recovery devices has to adapt to the dynamic control of the indoor conditions, but it has also to take into account the effects of the fresh air conditions on the device's actual effectiveness, especially under the so-called wet operating conditions. These occur in case of condensation within the recovery device, which may happen if the temperature of the separation interface falls below the dew point of the return or supply air. As the return air temperature from the room is quite high, it would only induce condensation when very humid outside conditions corresponding to high dew points. In contrast, saturation on the returning air in case of cold climates is a quite common situation. In theory, condensation of vapor on a plate brings along a release of the latent heat. That additional energy can contribute to the increment of the heat recovery effectiveness. This has been quantified by Nam and Han (2016) who studied a plate type heat exchanger for broiler houses modelled using computational fluid dynamics. The analysis showed an effectiveness increase with increasing return air humidity. In their example it is shown numerically and experimentally a rise up to 15% of effectiveness from dry return air to very humid (rH=90%) condition for outside air temperature between 0°C and 10°C. The numerical simulation of Anisimov et al. (2015) of plate heat exchanger (crossflow) reports an effectiveness increase of 20% for a relative humidity of 90% and an outside air temperature of -20°C. By reaching such low outside temperatures the separation interface approaches the freezing/frosting limit. Broad overview of the findings on frosting in air-to-air heat and energy exchangers is given in the review of Rafati Nasr et al. (2014), showing clearly that frost must be avoided in order to prevent additional pressure drop and especially to avoid mechanical damages to the devices.

The onset of frosting depends largely on the humidity of the return air, and on the outside temperature ((Deshko, Ya, & Sukhodub, 2016) (Fisk et al., 1985)). In the case of total heat recovery, the danger of frosting is reduced by the fact of mass transfer. Nevertheless, studies based on numerical modelling (Simonson & Besant, 1998) showed that for outside temperature of -20°C blocking starts also within an enthalpy wheel for a relative humidity of the return air of 30% and a total effectiveness from ca. 65% to 75% . Fisk et al. (1985) observed for an enthalpy-type cross flow heat exchanger a frosting onset of -8 to -12°C and Liu et al. (2016) a limit of -10°C (return air condition: T≈22°C, rH=30%; outside air rH≈55%) for a crossflow plate enthalpy exchanger. To give some examples regarding the actual applications, suppliers suggests a frost protection starting from -11°C for enthalpy wheel (for return air temperature of 21°C & relative humidity of 50%) and from -5°C for cross-flow plate total heat exchanger THR. A similar threshold temperature is proposed for sensible heat recovery SHR. This value is in line with the experimental and simulated results of Fisk et al. (1985), that are in the range of -7°C to -3°C for a cross-flow plate heat exchanger and ca. -6°C for a countercurrent one. Liu et al. (2016) reports a frosting limit of – 5°C for a return air relative humidity of 30% (return air: T≈22°C; outside air rH≈55%) for a cross flow heat exchanger. Summarizing it can be stated that for cold climate the frosting limit should be considered for both sensible and total heat recovery allowing for a dependency on the relative humidity of the return air. Once that limit is reached the use of the recovery device requires a frost protection strategy.

Frost protection can be realized in multiple ways and contributes to the annual energy consumption. Its impact was investigated by Kassai et al.(2015). Within their study, the pre-heating frost protection appears to be more convenient than a supply side closing strategy both for the climate of Helsinki and Krakow. The work of Zhang and Fung (2015) includes the influence of frost protection (defrost cycles) for heat and energy recovery in Toronto, reporting that the heat recovery annual defrost period is 3,5 time than for total heat recovery. Consequently, for a comparison between different recovery devices, its necessary to consider defrosting needs

The aim of this research is the evaluation of sensible and total heat recovery devices for ventilation purposes. The analysis has been generalized with respect to any specific applications. To do so the specific latent load was introduced (Lazzarin *et al*., 2000), which is calculated by considering the latent load based on a person's activity and the necessary air flow rate per person in an air conditioned space. Showing that the ventilation load reduction can be analyzed taking into account only the specific latent load of the air-conditioned space and the impact of the outside air conditions. The impact of condensation and freezing/frosting within a recovery device have been estimated, both for sensible and total heat recovery, and considered in the analysis. A control strategy for total heat recovery has been proposed and investigated in order to avoid excessive humidity due to excessive humidity recovery when total heat recovery devices are adopted. All considerations have been repeated for 7 different cities (representing 3 climate classes after Köppen) and evaluated by assessing the annual savings. To allow a comparison not only from an energy point but also from a financial point of view, an economic analysis is included based on currents year's price for fuel (natural gas) and electricity and on the investment related to the heat recovery system.

2. METHODS

2.1 Air Handling Unit and Heat Recovery System

For the determination of the energy savings due to sensible and total heat recovery, a simple air handling unit (AHU) has been considered, in which the Return Air (RA) from the conditioned space is partially recirculated (CA) and partially exhausted (EA). An equivalent amount of outside air (OA) and the recirculated air are combined into a Mixed Air (MA) which is then taken to the supply conditions (SA) in a set of heating/cooling and dehumidification, humidification and reheating coils. Heat recovery is typically operated between EA and OA, can be characterized by the values of the sensible ε_s, latent ε_l and enthalpy or total ε_t effectiveness, according to the following equations:

$$\varepsilon_s = \frac{\dot{m}_{OA}\,(T_R - T_{OA})}{\dot{m}_{min}\,(T_{RA} - T_{OA})} \tag{1}$$

$$\varepsilon_l = \frac{\dot{m}_{OA}\,(x_R - x_{OA})}{\dot{m}_{min}\,(x_{RA} - x_{OA})} \tag{2}$$

$$\varepsilon_t = \frac{\dot{m}_{OA}\,(h_R - h_{OA})}{\dot{m}_{min}\,(h_{RA} - h_{OA})} \tag{3}$$

Where $\dot{m}_{OA}$ and $\dot{m}_{min}$ are the OA mass flow rate, and the minimum between the OA and the EA mass flow rates, T is the temperature, x the humidity ratio, h the enthalpy and the remaining subscripts refer to the positions in in Fig. 1. In the following, the ratio of OA and EA mass flow rates is considered as unitary. Moreover, sensible heat exchangers have typically null latent heat effectiveness, while it is quite common to assume an equal value for sensible, latent and total effectiveness of total heat recovery devices.

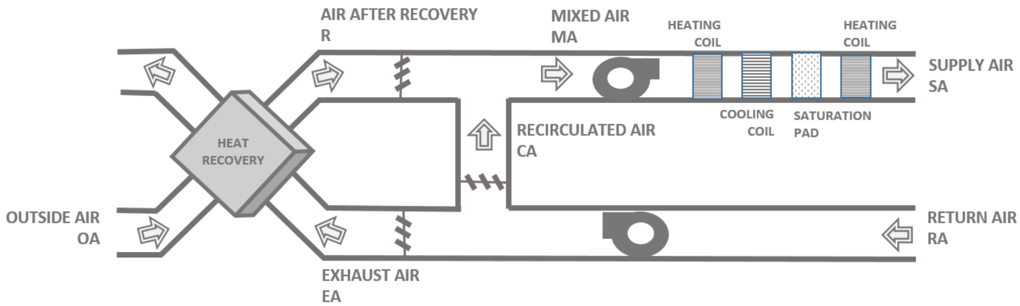

Figure 1: Ventilation and air conditioning cycle with heat recovery.

2.2 Relative Humidity Control and Heat Recovery Limitation

In winter mode operation, humidification of the MA is often necessary to the supply conditions, because of the quite low winter outside humidity ratio. Nevertheless, SA should be provided with a humidity ratio lower than the internal one, in order to compensate for the internal latent load:

$$x_{SA} = x_{RA} - \Delta x_{load} \tag{4}$$

where

$$\Delta x_{load} = \dot{m}_l / \dot{m}_{MA} \tag{5}$$

is the ratio between the mass rate of moisture internal production $\dot{m}_l$ and the dry supply air flow rate MA or SA.
While for $x_{MA} \leq x_{SA}$ a quite energy inexpensive process of humidification by water injection can fill the gap, when the $x_{MA} > x_{SA}$ a very expensive dehumidification by cooling process is required.
Mixing conditions depends on the EA and OA conditions, but also on the heat recovery when total heat exchangers are operated. Excessive moisture recovery, can compromise the benefits of ventilation heat recovery because of the need of dehumidification of the MA.
It can be shown (Lazzarin et al., 2000) that in case of humidity recovery an equivalent condition to $x_{MA} \leq x_{SA}$ can be expressed just as a function of the OA mass flow rate, the moisture production and the outside humidity ratio:

$$x_R \leq x_{RA} - \Delta x_{spec,load} \tag{6}$$

$$\Delta x_{spec,load} = \dot{m}_l / \dot{m}_{OA} \tag{7}$$

In case of no heat recovery or sensible heat recovery only, x_R in equation (6) corresponds to x_{OA}. The potential of the equations (6) and (7) is clear when considering that both terms in the expression of the specific latent load can be referred just to the activity level in a given environment, neglecting any consideration about the actual building characteristics, as reported in table 1.

Table 1: Thermal loads gain depending on the activity and building type

		Thermal load per person [1]					
Activity	**Type of Building**	**Total W**	**Sensible W**	**Latent W**	**Latent g/h**	**Air exchange[2] l/(s person)**	**Specific latent load g_v/kg_{da}**
Seated, relaxed	Theatre, Cinema	100	60	40	57.6	5.5	2.4
Seated, writing	Offices, Hotel, Apartments	120	65	55	79.2	11.0	1.7
Eating	Restaurants	170	75	95	136.7	10.0	3.2
Seated, light activity, typing	Offices, Hotel, Apartments	150	75	75	108.0	11.0	2.3
Slowly walking	Retail store, Bank	185	90	95	136.7	11.5	2.8
Moderate dancing	Dance Hall	375	120	255	367.1	16.5	5.1
Heavy activity	Gymnasium	525	185	340	489.4	16.5	6.9

(1) ISO 7730; (2) UNI 10339

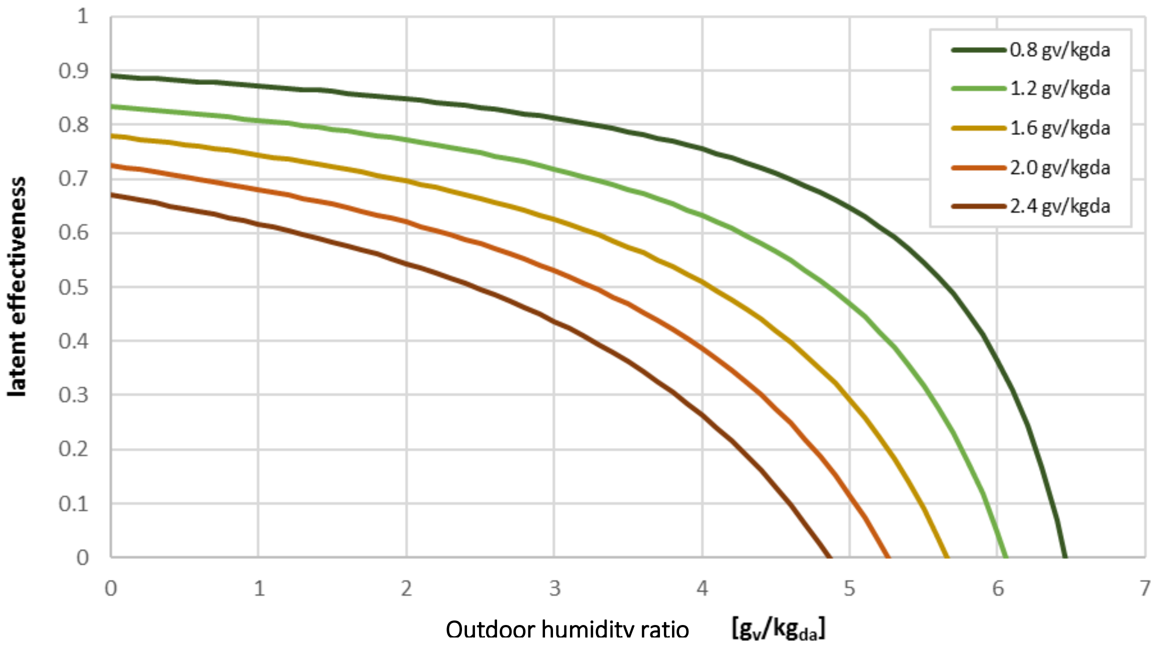

Figure 2: Correlation between the maximum allowed effectiveness and the outside air in case of humidity control & sketch of heat exchanger nomenclature (ASHRAE, 2004)

There, the thermal load and suggested air changes per person are listed, depending on the person's activity, together with the resulting specific latent load. Reference values may vary from country to country because of different standards and regulations, but the specific latent load for a given combination of values stays quite constant along the occupation time, which allows conducting the analysis from a more general perspective.
To prevent excessive humidity, a control strategy can be applied to limit the recovery effectiveness depending on the outside humidity ratio. In practice the control can be realized by changing the rotational speed of an enthalpy wheel or by introducing a bypass for a plate heat exchanger. Following the THR with controlled effectiveness is called THRc. The limitation in (6) can be expressed in terms of a maximum allowed latent effectiveness - and in practice any effectiveness - and the specific latent load:

$$\varepsilon_l \leq 1 - \frac{\Delta x_{spec,load}}{x_{RA} - x_{OA}} \tag{8}$$

Once the level of activity in a building is known, the effectiveness control depends only on the outside humidity. It can be seen (Fig. 2) that for specific latent loads higher than 2, the maximum operative effectiveness should be lower than 0.73, which in other words means that total heat recovery with every device with higher latent effectiveness has to be partialized. When the outside humidity ratio overcomes the limit in (6) the recovery device has to be totally bypassed. In similar cases, heat recovery has to be excluded also for SHR (controlled SHRc).
For values of the supply humidity ratio higher than the return air, such as in summer operation, dehumidification is always required and there is no reason to limit to the effectiveness, provided heat recovery reduces enthalpy.
For the investigation a slightly higher value of 60 % besides the nominal 50 % relative humidity has been taken into consideration for RA, to analyse the sensitivity of the saving potential on the recovery constraints, while maintaining still acceptable humidity levels.

2.3 Condensation and Frosting

The onset of saturation of vapor in the exhaust air outlet has been assumed to occur as soon as the air temperature falls below the dew point. Furthermore, an estimation of the latent heat gain contributing to heat recovery can be based on the amount of condensed vapor, calculated from the difference between the humidity ratio of RA and of its value at saturation. For total heat recovery, the mass transfer tends to reduce the RA humidity within the process and condensation is less likely.

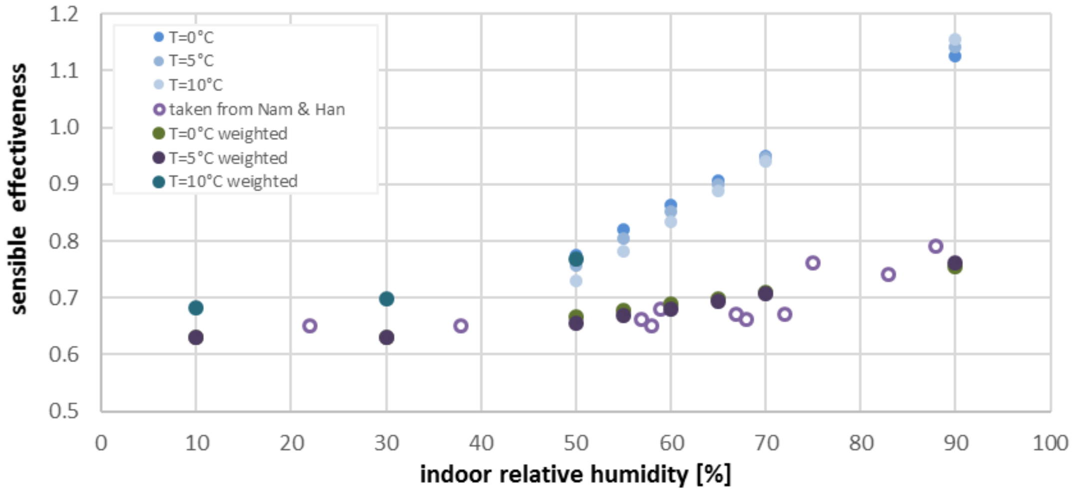

Figure 3: Sensible effectiveness on the indoor relative humidity: calculated vs of Nam and Han (2016)

In real applications, part of the air stream does not reach thermal equilibrium with the cold surfaces of the device to (bypass effect) (Keller, 2009). Consequently, the contribution from the latent heat is reduced. For that reason, a weighting factor has been introduced to consider the actual fraction of latent heat actually recovered. To validate this value, the results were compared with the experiments of (Nam & Han, 2016) for a plate-type heat exchanger, for different relative humidity and temperature (Figure 3) showing an good agreement with a factor of 0.25.
The impact of frosting conditions was considered in terms of amount of time when defrost must be applied and accounts for an additional energy demand. A pre-heating system was assumed for frost avoiding. As freezing onset depends as mentioned on many factors, an outside temperature limit for the start of the pre-heating has been chosen in line with threshold limits suggested by the manufacturers. For sensible heat recovery the onset of pre-heat is at -5°C and for an enthalpy wheel at -11°C for a relative humidity of 50%, and -8°C for a relative humidity of 60%.
The calculations on savings were repeated with and without consideration condensation respectively frosting.

2.4 Energy Savings and Economic Analysis

The savings have been determined considering hourly weather data (outside temperature and relative humidity) for a whole representative year, as provided by the EnergyPlus weather data base (energyplus, 2016). The savings in heating and cooling demand have been calculated in terms of recovered enthalpy, and expressed per unit of outside air flow rate [l/s]. This way the results are independent of the set actual size of the ventilation or air conditioning system and can be generalized for buildings of different dimensions.The following assumptions have been made for the analysis:

- a nominal effectiveness for sensible and total heat recovery of 70%
- a seasonal efficiency of 0.9 for the gas water heater [onset heating for T_{OA}<20°C for SHR, h_{OA}<38.55 kJ/kg (20°C;50%) for THR]
- a Seasonal Energy Efficiency Ratio (SEER) of 2.2 for the chiller [onset cooling for T_{OA}>26°C for SHR, h_{OA}<52.91 kJ/kg (20°C;50%) for THR]
- a pressure drop of 200 Pa for the heat exchanger and a fan mechanical efficiency of 0.3
- costs charged per MWh (Table 2) for natural gas referred to final users with annual gas consumption between 20 and 200 GJ (Eurostat, 2016) and electricity between 2500 and 5000 kWh (StatisticsFinland, 2016).

Investment costs per flow rate are based on the data given by the price list for performance of public works and maintenance of the municipality of Milan (Milano, 2015) and are for SHR 0.7 €/(l/s) and for THR 2.3 €/(l/s).
Savings for smaller sizing of the air conditioning equipment have been neglected.
The performed economic analysis includes the payback time of the recovery devices and its present worth value. The payback helps to understand after what time period the savings overcome the investment. The higher the payback time is the lower the savings. The present worth value represents the value an investment would have in the future considering the discount. For the calculation a time period of 10 years and a discount rate of 5% have been assumed.

Table 2: Natural gas and electricity costs MWh for the considered cities

costs [€/MWh]	Berlin	Helsinki	Lisbon	London	Madrid	Rome	Warsaw
natural gas	68	38	98	64	73	77	50
electricity	295	155	228	213	231	245	144

3. RESULTS AND DISCUSSION

3.1 Specific Latent Load and Recovery Limitation

The percentage of time in heating mode (T_{OA} lower than T_{RA}) with outside air humidity not overcoming the limit given by Equation (6) is reported in fig. 4 (i.e. with conditions lie in the area A). Under these conditions introducing fresh outside air or sensible heat recovery for winter operation would need dehumidification to meet the supply air condition. Obviously, the percentage is decreasing with increasing specific latent load, but it also strongly depends on the climate condition. In London, Lisbon and Madrid the annual outside humidity often reaches an uncomfortable condition if no air dehumidification is available, whereas for Helsinki it is comparably less problematic.
Figure 5 (solid color bars) shows the percentage of time in heating mode (h_{OA} lower than h_{RA}) with outdoor humidity leading to excessive humidity in the MA, when using THR with 70 % effectiveness if not properly controlled (i.e. with conditions in the area A). For all cities, assuming a specific latent load of 1.6 g_v/kg_{da}, which is comparable to the load in an office, only in less than 10% of the heating time the use of such THR does not lead to excessive humidification. In dashed it is shown the additional percentage of time (i.e. with conditions in area B) for heating mode in which controlled THR effectiveness allows preventing excessive humidity. The percentage of operative time without additional dehumidification is almost always more than doubled.

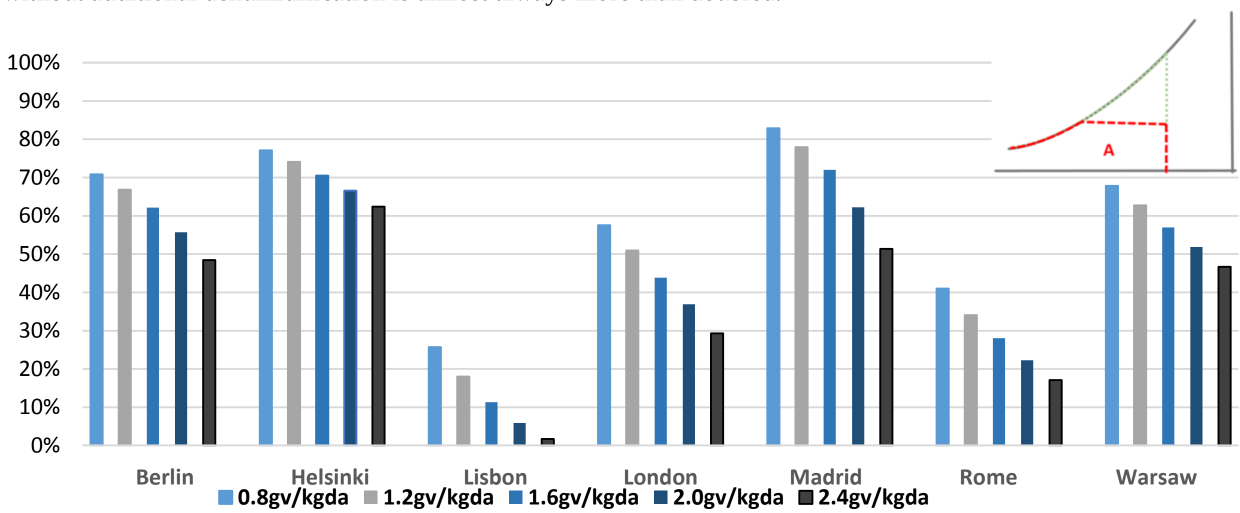

Figure 4: Percentage of time within heating conditions with outside air under excessive humidity limit, RH_{RA}=50%. Top right: corresponding conditions in the psychrometric chart.

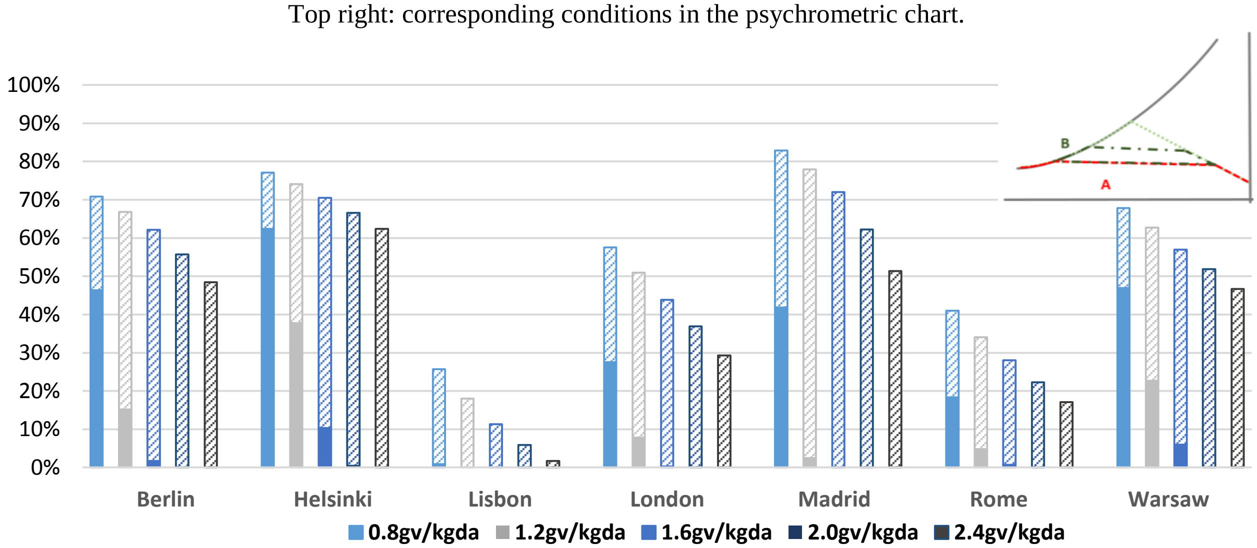

Figure 5: Percentage of time within heating conditions with outside air under excessive humidity limit without (A solid color) and with effectiveness control (B dashed), RH_{RA}=50% for a THR with 70 % effectiveness. Top right: corresponding conditions in the psychrometric chart.

3.2 Condensation and Frosting

The impact of condensation, is significant for SHR while it takes place for THR only for outside temperature far below 0 °C and mainly under the defrost limit excluding gains related to the latent heat. In SHR the additional heat transfers due to phase change results in an average annual effectiveness rise from 1 % in Berlin/Warsaw to 4 % for Helsinki. The maximum rise of annual costs due to frost protection occurs for Helsinki. The presumed frost limits for SHR, lead to an annual additional pre-heating cost per flow rate of 0.20 €/(l/s). The range of frost limits provided by the suppliers depending on the return air humidity leads to a pre-heating cost from 0.06 €/(l/s) to 0.10 €/(l/s) when using THR. In the case of a higher RA humidity, the subsequent rise in the frost limit strongly influences the additional costs: moving from 50 % to 60 %, the pre-heating costs nearly double. It is worth mentioning, that just because the outside temperature doesn't reach the frosting limit so often it does not mean it is not necessary a pre-heating system, as damage should be avoided.

3.3 Energy Savings and Economic Analysis

The energy and cost savings per flow rate of annual demand are listed in Table 3 for both SHRc and THRc. As expected the savings in heating are the lowest for the warm climate (Csa), and due to the highly humid conditions in the cold seasons the THRc for Lisbon and Rome does not bring along a significant benefit for heating conditions. The high costs for natural gas in Germany compared to Finland and Poland lead to cost savings that are higher than for Helsinki or Warsaw, even though the reduction in energy demand is lower. The humidity controlled THRc reduces the savings with respect to uncontrolled recovery, but avoids excessive indoor humidity and dehumidification costs. The actual operative effectiveness could be much lower than the nominal one which could be misleading to designers. Benefits can be not significantly larger than, and in some cases even lower than those provided by SHRc.
If an indoor relative humidity of 60 % is assumed, the benefits in cost savings are higher (Table 4). Depending on the climate, the payback time halves or nearly halves as for Rome and Berlin/Madrid respectively. For Lisbon the impact of the humidity limit is the most significant. This is due to the fact that, in heating season many conditions occur with humidity close to the return air humidity. In general, if it is feasible in terms of thermal comfort the allowance of a higher indoor humidity could be taken into account for reducing expenses.
For the following observations it is distinguished, additional to THRc and THR, between SHRc and SHR. The SHR and THR are not controlled depending on the outside humidity. Hence, this results determine the economic aspects without taking into account the dehumidification needed. The difference in the present worth value is illustrated in Figure 6 (left). The lower the value the less beneficial is the investment. The THRc allows a high present values for cold climates for which heating is responsible for the main part of the annual costs. One exception is Lisbon.

Table 3: Annual reduction in energy for heating and cooling demand [kWh/(l/s)] and cost savings [€/(l/s)] per air flow rate (assumed specific latent load of 1.6 g_v/kg_{da})

	SHRc				**THRc**				
City	heating	costs	cooling	costs	heating	costs	cooling	costs	Climate class
Berlin	74	4.97	0.072	0.02	79	5.32	0.10	0.03	Dfb
Helsinki	110	4.21	0.002	0.00	140	5.39	0.02	0.00	Dfb
Lisbon	7	0.72	0.571	0.13	3	0.29	2.07	0.39	Csa
London	52	3.28	0.019	0.00	49	3.11	0.01	0.00	Cfb
Madrid	49	3.58	0.000	0.00	49	3.62	0.00	0.00	Csa
Rome	21	1.62	0.339	0.08	19	4.56	7.22	1.47	Csa
Warsaw	79	3.98	0.057	0.01	79	5.32	0.23	0.03	Dfb

Table 4: Payback time [a] for different indoor humidity values for THRc (assumed specific latent load of 1.6 g_v/kg_{da})

Indoor relative humidity	Berlin	Helsinki	Lisbon	London	Madrid	Rome	Warsaw
50%	0.61	0.61	11.17	1.05	0.90	2.27	0.72
60%	0.38	0.45	1.57	0.55	0.47	1.00	0.48

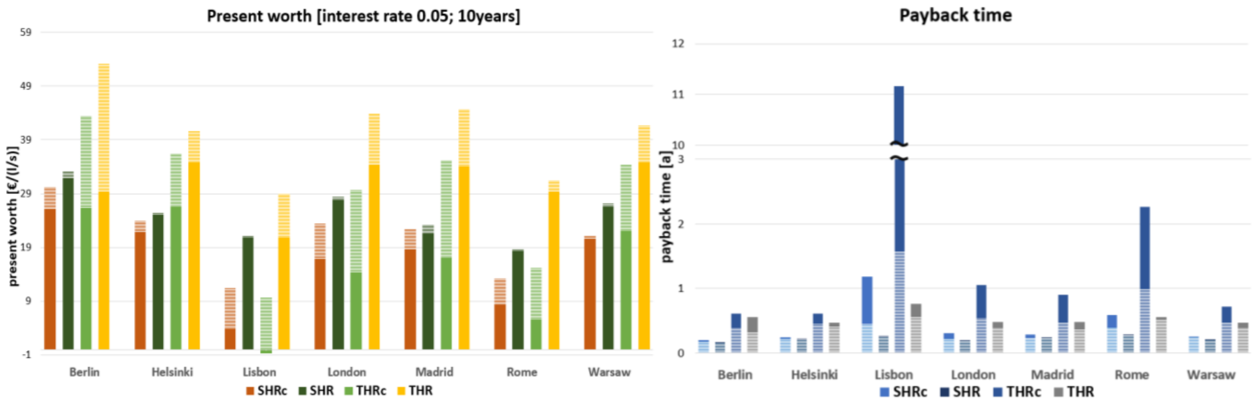

Figure 6: Present worth – interest rate 0,05 10 years – (left) and payback time (right) for RH_{RA}=50% (dark color) and RH_{RA}=60% (light color)

For that climate the humidity control leads to a very high payback time and a negative present worth value, due to that fact that the outside air is in cold condition comparable humid and in order to avoid the excessive humidification the effectiveness is lower, but also the resulting savings. Generally, energy savings are depending on the climate class, while the economic savings are independent of the climate due to different natural gas and electricity costs.

4. CONCLUSIONS

Heat recovery through sensible and total heat exchanger represents an opportunity for the improvement of energy efficiency in existing buildings and an important resource to move towards nearly zero energy buildings. However, the actual saving potential is often overestimated when only the driving gradient is considered, i.e. temperature difference for SHR and enthalpy difference for THR. In fact, in winter mode operation, when the outside air humidity ratio produces excessive mixed air humidity for SHR or even more when latent recovery with THR leads to a similar effect even with lower external humidity, a control strategy (SHRc and THRc) has to be introduced to limitation or exclude any heat recovery and avoid the need for an energy expensive dehumidification by cooling process. This appears particularly challenging with tighter internal humidity control and higher internal latent loads.
Considering the energy savings for SHRc and THRc, the saving potential is higher for heating than for cooling. Only cities in warm climates (Csa) and with relatively humid summer benefit significantly also for cooling. The energy savings of THR is not much higher than that of SHR, when the control strategy is assumed.
As for the economic performance, the payback time for SHRc is much lower than a year in most of the cases. Since the recovery limitation is stronger with lower internal setpoint humidity ratios, the analysis has been repeated with a higher internal relative humidity (60 % instead of 50 %), finding significant improvements of the economic performance (present worth and payback time in particular), but leaving THRc quite far from SHRc.
The contribution of condensation of the exhaust air in the recovery system appears quite limited, with a maximum increase of the annual effectiveness of 4% for cold climates (Helsinki) and SHR. Defrosting has been shown to impact the savings up to 0.2 €/(l/s) in Helsinki for SHR.

NOMENCLATURE

ε	effectiveness	
h	spesific enthalpy	(kJ/kg)
$\dot{m}$	mass flow rate	(kg/s)
RH	relative humidity	(%)
T	temperature	(°C)
x	humidity ratio	(kg_v/kg_{da})

Subscripts

CA	recirculated air
da	dry air
EA	exhaust air
l	latent
MA	mixed air
min	minimum
OA	outside
R	air after recovery
RA	return air
SA	supply air
s	sensible
t	total
W	water
V	vapour

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