Mapping Savings in Energy Demand by Heat Recovery for European Countries Under Consideration of Humidity Control

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Abstract

Sensible (SHR) and total heat recovery (THR) can play a significant role in energy savings in mechanical ventilation usage. Apart from the technical characteristics of the heat exchangers, the savings on the ventilation load depend on the air conditions for the two airstreams, namely the conditions maintained for the indoor air and the actual outside conditions, and on the proper control strategies deployed to minimize the impact on the air processes required after the heat recovery device. In this respect, humidity control can conflict with heat recovery whenever excessive humidity requires dehumidification of the ventilation air. In particular, SHR in heating mode should be preferably by-passed if the outside humidity exceeds the supply conditions required to compensate internal latent loads. For THR, moreover, the control strategy has also to account for the device's latent effectiveness, which may require an even earlier limitation of heat recovery or a by-pass if the system effectiveness cannot be controlled. Depending on the specific climate, the actual heat recovery can be much lower than the expected one and needs to be evaluated in order to avoid overestimating its energy and economic performance. The humidity supply limit required in the analysis of the actual recovery can be defined considering the target indoor humidity ratio (corresponding to the relative humidity setpoints of 50 %) reduced by the amount needed to compensate any indoor humidity source. This reduction can be expressed in terms of a specific latent load, SLL, calculated as the ratio between the mass rate of water vapor produced indoor, and the mass rate of the ventilation air, no matter whether some recirculation exists or not. Since both vapor production and ventilation air mass rate depend on sources and occupants' density inside the conditioned space (0.8, 1.2, 1.6, 2.0 or 2.4 g_v/kg_{da}), SLL is largely independent of the remaining building characteristics. In this research, we studied the savings from ventilation heat recovery in different European climatic zones and countries by applying different control strategies to avoid excess humidity. Only the ventilation system had to be modelled through a simplified effectiveness model, considering different SLL as the only relevant building characteristic. Savings were expressed in terms of energy demand per flow rate, averaged over climatic Köppen-Geiger classes.

1. Introduction

High performance buildings, with enhanced airtight envelopes, often need a mechanical ventilation system for fresh air supply. Furthermore, the maintenance of an adequate indoor air quality for the occupants requires also humidity control. Even though results from field studies (Kosonen and Tan, 2004) lead to a weak sensation of relative humidity by occupants, some works (Tsutsumi et al., 2007) report a negative impact of high relative humidity on the performance of occupants. Moreover, health problems can be caused by excessive humidity, and it may lead to building material damage (Sterling et al., 1985). However, since the share of final energy uses by mechanical ventilation may be relevant for highly insulated buildings, an effective strategy to reduce the overall energy consumption can be based on heat recovery but its impact has to be carefully assessed, considering all air treatment stages. When dehumidification is required, for example, heat recovery is counterproductive and can even enhance the necessity of dehumidification because of excessive humidification, especially in case of latent heat recovery as indicated by the study of Smith and Svendsen (2016).

Developing further the analysis of a previous work by the authors (Tafelmeier et al., 2017), the current

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research investigates the influence of humidity control on the potential energy and economic savings in heating mode, achieved by sensible and total heat recovery (respectively, SHR and THR). The study is performed for the 9 main Köppen-Geiger climate classes present in Europe, including 66 reference cities.

2. Simulation

2.1 Mechanical Ventilation System

The air treatment in the mechanical ventilation configuration is considered as in Fig. 1: outside air (OA) passes through the heat recovery device (HR), after which the exiting air (R) is mixed with the recirculated air (CA) to form the mixed air (MA). The MA is adjusted to meet the supply air (SA) condition by an air handling unit (AHU) equipped with devices for preheating, cooling and dehumidification, humidification and reheating. The return air (RA), is split up into the CA and the exhaust air (EA). The latter one passes through the HR device, whose sensible (ε_s), latent (ε_l) and total effectiveness (ε_t) is described as:

$$\varepsilon_s = (m_{OA}/m_{min})\cdot(T_R-T_{OA})/(T_{RA}-T_{OA}) \quad (1)$$

$$\varepsilon_l = (m_{OA}/m_{min})\cdot(x_R-x_{OA})/(x_{RA}-x_{OA}) \quad (2)$$

$$\varepsilon_t = (m_{OA}/m_{min})\cdot(h_R-h_{OA})/(h_{RA}-h_{OA}) \quad (3)$$

where T is temperature, x is humidity ratio and h is enthalpy.

A unitary mass flow rate ratio is assumed, hence the OA mass flow rate, m_{OA}, and minimum mass flow rate, m_{min}, are considered to be the same. For the THR devices, ε_s, ε_l and, so, ε_t are taken as equal while for SHR devices ε_l is clearly null.

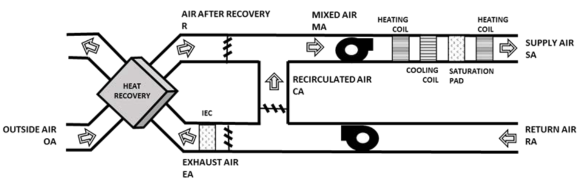

Fig. 1 – Mechanical ventilation configuration with HR device

2.2 Relative Humidity in Winter Mode

The SA humidity ratio has to be high enough to reach the comfort setpoint and low enough to balance the internal latent load m_L. Equation (4) follows from the mass balance and gives the connection between the RA and the m_L.

$$x_{SA} = x_{RA} - m_L/m_{MA} \quad (4)$$

Even though in winter mode humidification is often necessary, when x_{MA} is greater than or equal to x_{SA} dehumidification is required. In common AHUs, it is performed by cooling the air below its dew point, with a consequent pre-heating before air is supplied to the environment.

Hence, the energy consumption in winter mode is the lowest if:

$$x_{MA} \leq x_{SA} = x_{RA} - m_L/m_{MA} \quad (5)$$

and for HR use:

$$x_R \leq x_{RA} - m_L/m_{OA} \quad (6)$$

Equation (6) describes the limitation of x_R as a function of x_{RA} and the latent load per OA flow rate. If there is no HR or SHR (6) turns to:

$$x_{OA} \leq x_{RA} - m_L/m_{OA} \quad (7)$$

with

$$\Delta x = m_L/m_{OA} \quad (8)$$

being the specific latent load, SLL. Its value is derived by considering the suggested minimum fresh air rate per person and the latent load per person, which is classified in different activity levels. In that way, a SLL value is given independently of the building characteristics just knowing the activity of the occupants (Lazzarin et al., 2000). In this work, five SLL values were considered: 0.8, 1.2, 1.6, 2.0, and 2.4 g_v/kg_{da}.

2.3 Control Strategies and Saving Considerations

Two control strategies, A and B, have been defined for the SHR and THR. For determining the limits of control strategy A, it is assumed that HR is beneficial for heating as soon as the OA temperature for SHR or enthalpy for THR is below the EA temperature or enthalpy, respectively (dashed red lines in Fig.s 2 and 3). Control strategy B, instead, has been defined in order to avoid counter-productive dehumidification, which occurs if the humidity limitation of Equation (6) is exceeded. For SHR devices, control strategy B can be implemented through bypass as soon as Equation (7) does not apply (white area in Fig. 2). In the case of THR, Equation (6) can be rewritten by considering that x_R is given by the humidity ratio of OA, EA and ε_l

(Equation (2)). x_R is depending on ε_l, consequently it can be controlled by regulating ε_l by a partial bypass or rotational speed modification:

$$\varepsilon_{l,op} \leq \min[\varepsilon_l; 1-\Delta x/(x_{RA}-x_{OA})] \quad (9)$$

The maximum $\varepsilon_{l,op}$, which in principle could be one, yet is limited to the nominal effectiveness, accounts for x_{OA} smaller than $x_{RA}-\Delta x/(1-\varepsilon_l)$ (blue area in Fig. 3). The partialized $\varepsilon_{l,op}$ is from the maximum value at $x_{RA}-\Delta x/(1-\varepsilon_l)$ to zero at $x_{RA}-\Delta x$ (yellow area in Fig. 3).

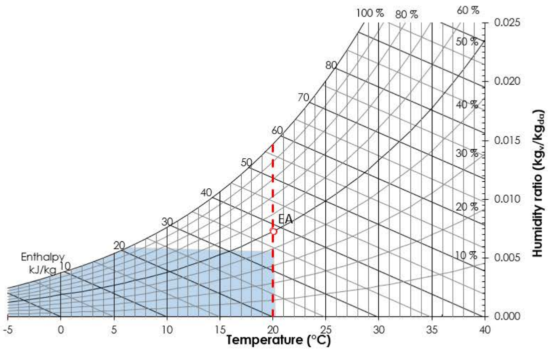

Fig. 2 – Psychrometric chart with highlighted limits in case of SHR for control A (red line) and control B (blue area)

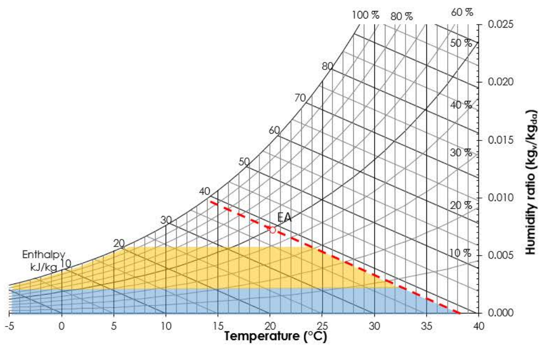

Fig. 3 – Psychrometric chart with highlighted limits in case of THR for control A (red line) and control B (blue and yellow areas)

The simulation is performed by calculating the energy savings on an hourly basis considering a steady state performance during this duration. Energy savings result from enthalpy difference between the OA and R. In order to calculate R, the EA condition equals the setpoint of the conditioned space, i.e. temperature of 20 °C and relative humidity of 50 %. The OA conditions are given by hourly weather data of a representative year provided by the EnergyPlus database (2016) for 66 reference cities in Europe. The energy savings are averaged for each Köppen-Geiger climate class (Table 1), for which a representative city is identified by means of a Kolmogorov-Smirnov test, performed on the cumulative distributions of OA hourly enthalpy values. The cost savings are computed by consulting the natural gas prices and averaged at a national level. Energy and cost savings are given as savings per unit of flow rate and, consequently, they can be generalized independently of the building size.

Table 1 – details about climate classification and considered cities (Peel et al., 2006)

				Description	Number of cities and representative	
Cold	D	s	b	Dry + warm summer	1	Ankara
		f		Without dry season		
			a	+ hot summer	3	Bucharest
			b	+ warm summer	16	Ostrava
			c	+ cold summer	5	Ostersund
Temperate	C	s		Dry summer		
			a	+ hot summer	15	Bari
			b	+ warm summer	3	La Coruna
		f		Without dry season		
			a	+ hot summer	4	Bologna
			b	+ warm summer	16	Amsterdam
arid	B	S	k	Steppe + cold	3	Madrid

Table 1 – Price for natural gas [EUR/kWh] for the main countries

State	natural gas cost	State	natural gas cost	State	natural gas cost
A	0.071	FIN	0.040*	PL	0.05
BG	0.039	GR	0.075	ROM	0.034
CZ	0.058	I	0.091	S	0.12
D	0.068	N	0.07**	SRB	0.04
DK	0.077	NLD	0.077	TK	0.035
E	0.093	P	0.098	UK	0.071
F	0.073				

* (StatisticsFinland, 2016); ** (Gasnor AS , 2016)

The following assumptions have been made:

- The nominal effectiveness for sensible, latent and total heat recovery is chosen to be 70 %.
- The gas boiler for the hot water supply for the heating coils installed in the AHU has an efficiency of 80 %.

- The natural gas prices are provided by the European Union Eurostat (2016) for the considered countries. Exceptions are Finland and Norway, for which a different source is considered, and Belarus, Cyprus, Iceland, Russia and Ukraine, which are excluded due to the lack of price information (Table 2).
- Impacts on the energy and cost savings independent of the HR device choice, such as pressure losses, were not included as well as an eventual reduction in downscaling of the AHU devices in case of HR.

3. Discussion and Result Analysis

Durational-plots (Fig.s 4 and 5) are used to show how the control strategies affect the energy savings in case of SHR and THR. They illustrate the hourly energy savings per flow rate sorted in decreasing order. Bologna, the representative city of the climate class Cfa, is chosen as an example.

As described above, control strategy A for SHR is not correlated to SLL, while control strategy B defines whether the device is bypassed or not and, consequently, if the energy saving becomes zero or remains unchanged. From Fig. 4, it can be seen that the highest energy savings achieved for SHR do not fall into the bypass region, even for the highest SLL value.

Differently from SHR, THR savings depend on the SLL value for both control strategies. In case of control strategy B for THR, the highest energy saving is reduced by the partialization of the effectiveness for SLL values of 2.0 and 2.4 g_v/kg_{da} (i.e. with those SLL, the air condition is included in the yellow region in Fig. 3). Savings in case of control strategy A for THR are generally higher than those for the control B, as the SLL raises the EA enthalpy, and allows a higher recovered energy. Nevertheless, this might reduce the indoor air quality or increases the dehumidification need.

The areas in Fig.s 4 and 5 represent the annual energy savings in heating for a SLL of 1.6 g_v/kg_{da}. Specifically, savings achieved by control strategy A are illustrated by the plane red areas and those by control strategy B by striped red areas. An increasing SLL value in control strategy B for SHR and in both controls for THR leads to a shrinking area and so, to the reduction in savings. In this example, the total saving for heating by SHR considering a SLL of 1.6 g_v/kg_{da} in Bologna are 76.77 kWh/(l/s) and 53.00 kWh/(l/s) for SHR, and 151.50 kWh/(l/s) and 54.78 kWh/(l/s), respectively with control strategy A and B.

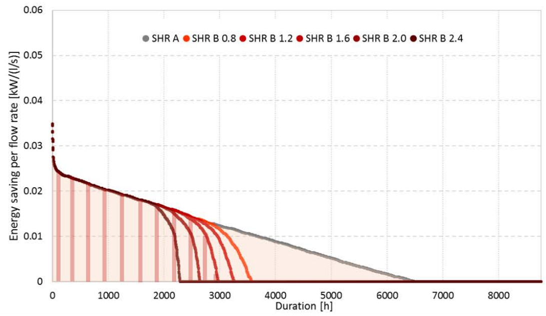

Fig.4 – Durational-plot of energy savings by SHR for Bologna

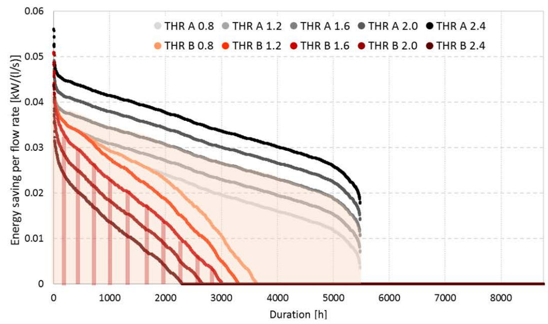

Fig. 5 – Durational-plot of energy savings by THR for Bologna

The hourly savings over the year are represented by means of carpet-style plots in order to visualize when, in terms of daytime and season, the savings due to HR is effected by the control.

Fig. 6 shows the hourly energy savings by SHR throughout the year for each climate class representative city. The order of the climate classes from the top to the bottom equals the increasing ranking of the averaged energy savings for SHR with control strategy A. As in this work only the savings regarding the heating were considered, the lowest saving potential occurs for the months from May to September, the highest from December to February. The cold climates (initial letter D) benefit from the SHR also in the intermediate seasons, especially compared to the temperature climates with dry summer (Csa and Csb). The highest saving potential occurs in the hours between 22:00 to 8:00. Applying the control strategy B leads to an increase of zero-saving occasions, particularly for the temperate climates during the intermediate seasons in the afternoon, and during the summer in the morning and nights.

In case of the THR, again, the order of the climate classes aligns with the computed increasing average savings in energy (Fig. 7). Lowest savings occur between June and September, highest similar as for SHR between December and February. The duration of zero-savings due to bypassing in summer, extents strongly for all climates into the intermediate seasons once the control strategy B is applied. The effect of the partialization of the effectiveness is the most visible in the winter months. The consequences of the above discussed impact of the humidity control on the potential in achieving energy savings by HR are given in the following paragraphs.

The ventilation load – calculated as enthalpy difference between OA and SA air conditions - and the energy savings, expressed in terms of averages and standard deviations for each climate class, are given in Table 3 for SHR with a SLL of 1.6 g_v/kg_{da}. The order is again in line with the increasing savings potential for control strategy A. The savings are the least for the temperate climates with dry summer and the highest for all cold climates in the order of the sub classes: hot, warm and cold summer. The orders do not change for control strategy B, except for BSk and Cfa, since the latter one is slightly more affected by control strategy B. The reduction itself ranges from 49.7 % for the Csa to 9.3 % for Dfc if an SLL of 1.6 g_v/kg_{da} is considered. As a whole, the energy savings contribute to a reduction of the ventilation load, which is in average 50 % for control A and 37 % for control B.

Table 3 – Average energy savings and standard deviation for heating by SHR with a SLL of 1.6 g_v/kg_{da} and the ventilation load [kWh/(l/s)]

	Ventilation load	SHR control A	SHR control B
Csa	86.25±29.40	47.35±11.36	23.82±13.63
Csb	121.19±68.66	67.83±19.86	35.38±35.43
BSk	156.48±34.70	71.46±12.67	57.50±15.96
Cfa	140.08±22.72	73.97±9.03	51.08±11.91
Cfb	178.52±36.50	93.28±15.38	64.01±18.43
Dfa	187.32±14.79	94.14±5.39	75.69±5.31
Dsb	228.75	103.67	91.26
Dfb	242.48±28.84	118.76±3.18	98.91±13.95
Dfc	345.33±47.06	163.21±21.29	148.08±24.75

Table 4 represents the results for THR equivalently to Table 3 for SHR. The ranking of the climate classes in terms of saving potential has changed but, likewise for control strategy A and a SLL of 1.6 g_v/kg_{da} for SHR, the least savings occur for the temperate climate class with hot and dry summers, the highest can be achieved in the cold climate class without dry seasons and cold summers.

Table 4 – Average energy savings and standard deviation for heating by THR with a SLL of 1.6 g_v/kg_{da} and the ventilation load [kWh/(l/s)]

	Ventilation load	THR control A	THR control B
Csa	86.25±29.40	111.90±22.86	23.68±14.65
Cfa	121.19±68.66	148.13±13.47	55.05±18.15
Csb	156.48±34.70	153.77±43.01	34.00±43.29
BSk	140.08±22.72	165.09±24.03	62.30±23.84
Dfa	178.52±36.50	176.59±9.21	90.15±10.32
Cfb	187.32±14.79	193.32±26.43	65.55±22.46
Dsb	228.75	208.36	121.25
Dfb	242.48±28.84	223.11±19.09	123.42±21.60
Dfc	345.33±47.06	284.65±25.29	196.81±41.66

The magnitude of savings and the order of the climate classes are affected more by the control strategy than for SHR. Indeed, with control strategy B, the cold climate Dfa shows higher savings than the temperate climate Cfb without dry seasons and warm summers. However, the savings in Cfb are reduced more by adopting the control strategy B than for Dfa. The climate classes representing the lowest savings (Csa) and the highest three ones (Dsb, Dfb and Dfc) are, in terms of ranking, not affected by the HR or control choice. The maximum reduction is found for Csa, equal to 78.8 %, and the minimum for Dfc, equal to 30.9 %, with a SLL of 1.6 g_v/kg_{da}. Also in that case, the contribution of the energy savings by the THR on reducing the ventilation load has been considered. The average for a SLL of 1.6 g_v/kg_{da} and control strategy A is 104 % while for control strategy B it is 42 %. The reason for the contribution higher than 100 % is due, on the one hand, to no limitation in the effectiveness and, on the other hand, to EA humidity and enthalpy higher than those for SA, which also indicate excessive humidification.

Table 5 – Average national cost savings by SHR [EUR/(l/s)]

Country	SHR control A	SHR control B
SRB	3.53	2.83
BG	3.76	3.04
GR	3.94±0.63	2.25±0.96
TK	4.18±0.82	3.23±0.08
P	4.56±1.10	1.26±0.47
PL	5.55±0.10	4.37±0.16
F	5.63±0.95	3.57±1.02
FIN	5.72±0.22	5.03±0.24
I	5.79±1.63	3.65±1.75
ROM	5.81	4.57
E	6.26±1.40	4.24±2.18
CZ	6.63±0.18	5.42±0.14
UK	6.64±0.44	4.48±0.62
A	6.98	5.82
D	7.12±0.55	5.66±0.60
NLD	7.32	4.58
DK	8.38	6.58
N	8.61±0.21	7.05±0.89
S	17.78±3.86	15.80±4.63

The cost savings achieved by SHR are given in Table 5 for control strategies A and B, considering a SLL of 1.6 g/kg. The highest savings are for both cases for Sweden, due to the high saving potential plus the high natural gas costs. On the contrary, the lowest savings for control strategy A account for Serbia, mainly due to the low natural gas costs. This changes as soon as the control strategy B is applied, and the weakest saving potential is in Portugal, followed by Greece. This control leads to a reduction of more than 70 % for Portugal and more than 35 % for Greece, the Netherlands, Italy, and France. Equally is the trend of the cost savings for THR with both strategies (Table 6). The maximum cost reduction accounts for Portugal, with more than 90 %, and Greece, Spain, and the Netherlands for more than 70 %. However, it should be noted that a large spread of results in the cost calculation occurs for Spain and Italy, because of the many climate classes present in those countries.

Table 6 – Average national cost savings by THR [EUR/(l/s)]

Country	THR control A	THR control B
SRB	6.70	3.20
BG	7.33	3.69
GR	8.36±1.05	1.95±1.18
TK	8.77±2.17	3.75±0.65
P	10.19±2.16	0.81±0.49
FIN	10.24±0.25	6.54±0.44
PL	10.53±0.16	5.27±0.34
ROM	10.74	5.46
I	11.64±2.60	3.55±2.19
F	11.78±1.60	3.68±1.17
CZ	12.92±0.66	6.29±0.25
A	13.85	7.02
E	13.91±3.13	3.91±2.74
D	14.00±0.72	6.19±0.95
UK	14.04±0.76	4.31±0.52
NLD	14.68	4.15
N	16.67±0.14	8.45±2.01
DK	16.79	7.28
S	31.88±4.70	20.68±7.47

4. Conclusion

This work investigates heat recovery in mechanical ventilation systems in heating mode in Europe, focusing on two aspects: 1) the impact of the humidity control on the energy and cost savings, as well as 2) the applicability of Köppen-Geiger climate class based mapping for large scale assessments.

Concerning 1), for a sensible heat recovery with humidity control strategy B, more hours with zero-savings occur, proportionally with the specific latent load, especially in the intermediate seasons. For a total heat recovery, the behavior of the potential saving is similar but, due to an additional partialization to avoid excessive humidification, hourly and annual potential savings are often more affected than for sensible heat recovery, especially in the intermediate seasons, in the morning and night hours. The climate rankings of energy saving potentials do not vary significantly regarding the control strategies, but the humidity control strategy B causes a strong reduction, particularly for temperate and arid climate classes. In terms of cost savings, it follows from the results that heat recovery combined with humidity control reduces the annual savings for heating, especially in Portugal and, with a lower magnitude, in Greece, the Netherlands, Italy, France, and Spain. Conversely, this means that, in those countries, a saving calculation without consideration of the humidity control might overestimate the economic benefits of the heat recovery device.

Regarding 2), as soon as the control is applied the deviation increases. This raises the question of how appropriate the Köppen-Geiger classification is for humidity-correlated investigations. Since the classes are distinguished in terms of temperature and precipitation, it stands to reason that a more sophisticated classification is necessary for a convenient visualization of the saving. Future works shall overcome this limitation by investigating different solutions such as combining the Köppen-Geiger classification with the air humidity and enthalpy, consulting other climate classifications if suited or define a classification based on a larger number of examples on a tighter grid, particularly tailored for heat recovery control.

Acknowledgement

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Nomenclature

Symbols

ε	Effectiveness
h	Specific enthalpy(kJ/kg)
m	Mass flow rate (kg/s)
T	Temperature (°C)
x	Humidity ratio (kg_v/kg_{da})

Subscripts/Superscripts

l	Latent
L	Load
MA	Mixed air
min	Minimum
OA	Outside air
R	Air after recovery
RA	Return air
S	Sensible
SA	Supply air
t	Total

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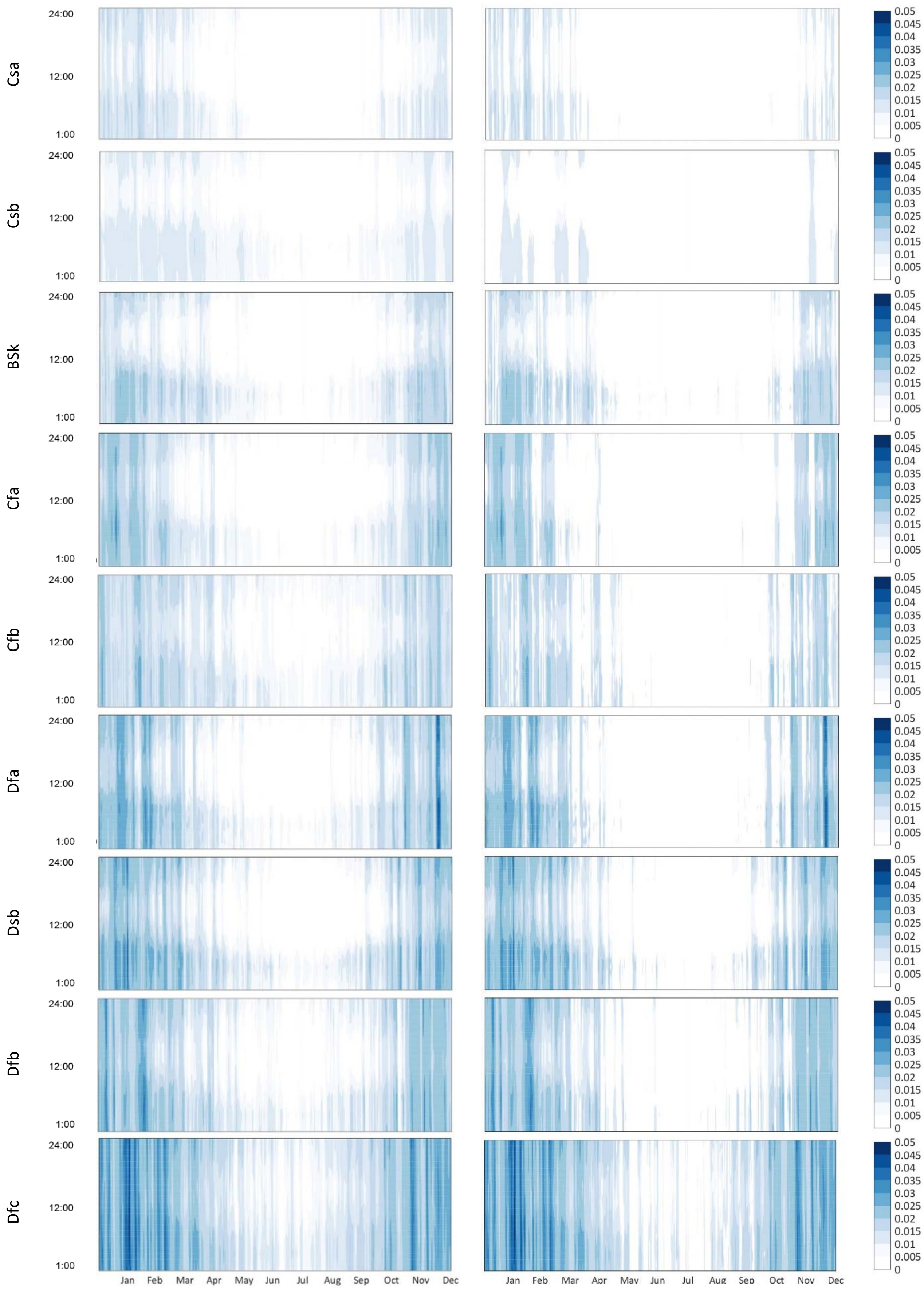

Fig. 6 – Carpet-plot of hourly energy savings for the representative cities for SHR control A (left) and control B and a SLL of 1.6g_v/kg_{da} (right)

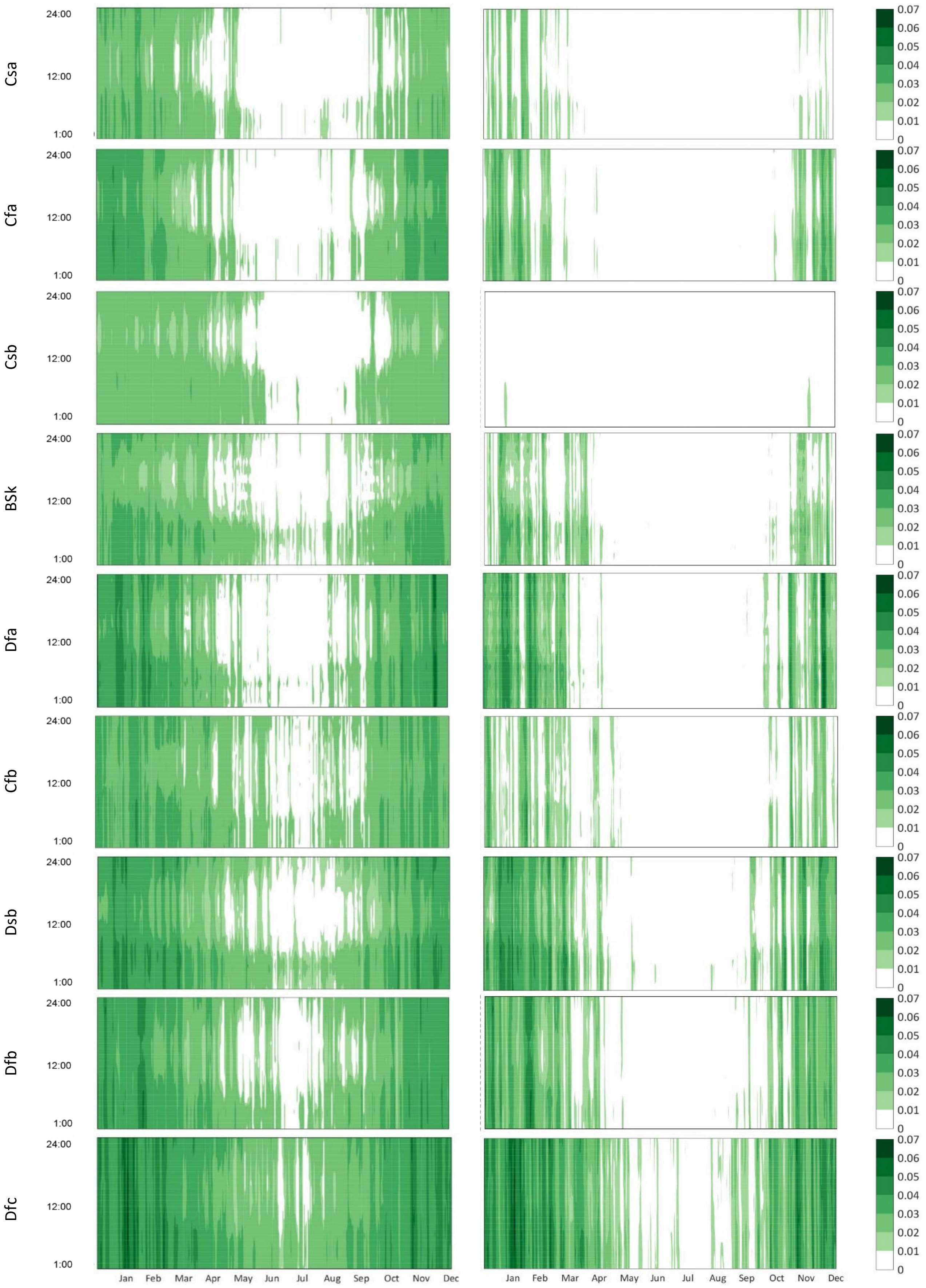

Fig. 7 – Carpet-plot of hourly energy savings for the representative cities for THR control A (left) and control B and a SLL of 1.6g_v/kg_{da} (right)