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Evaluation of the Performance Criteria of Combined Thermo-Chemical Energy Storage Systems for Building Applications

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Abstract

The collaborative research activities of the industrial and academic communities for the development of the thermo-chemical energy storage (TES) technology are growing dynamically and new concepts of systems are appearing. The building sector presents the particular interests field for the implementation of such technology in accordance with the new EU energy policy. The well-known limitations of the existent TES technology may be overcome by the development of novel materials as well as by the design of new type of reactors. In the present work the enhanced energy efficient TES configurations, destined for space heating applications, have been established and studied basing on simple models using performance data of the reactive medium. Two major heating techniques have been assessed: low temperature air-to-water integrated circuits and combined warm air heating. The analysis of the performance criteria of the proposed configurations allowed concluding about the usability of the TES in the predefined range of operating conditions.

Key-words: Thermo-chemical energy storage, performance criteria, low temperature heating, combined warm air heating

1. Introduction

The necessity of the development and promotion of the TES technology arises from the context of energy saving programs for the building sector, whereas the TES system is liable to reduce the emissions of the "greenhouse gases", to fill the gap between the periods of thermal energy consumption and generation, to boost the heat supply for the peak energy demand.

According to the Annual Report of the Market Observatory for energy analyses, the significant change for the decline in greenhouse gases and CO_2 emissions between 2008 and 2009 from fossil combustion is due to the damped growth of the renewable energy utilisation. In contrast to the early EU energy regulation policy in 2001, where the promotion of the renewable energy has been related directly to the internal electricity market, the recent directive of the European Parliament (EC 28/2009) has stimulated the support for the technological demonstration and innovations of every decentralised renewable energy technology. The dedicated budget for the projects, among which storages design and demonstration, has been estimated of about of 4 billion Euros (EC 663/2009). The national overall targets for the share of energy from renewable sources in gross final consumption of energy are fixed at 13 % for Belgium and Czech Republic, 23 % for France and 34 % for Austria to be achieved by the end of 2020 (EC 663/2009).

Around one quarter of final energy consumption is accounted for the residential sector and the heating demand encounters around 70 % of the overall household energy consumption, which is estimated to represent 14 % of greenhouse gases and CO_2 emissions. The predomination of the heat energy end use in the residential sector is the known issue. The losses for the heat delivery and energy transformation include more

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than half of the total energy supply and therefore this huge share can be retrieved be recycling existing heat losses or by using the thermal storages.

As one may see, the growth of interest to the innovative heat energy systems integrated in the residential sector from the part of experts of the TES community is due to the recently adopted policy by the European Commission (from 2009 to 2011) in promoting the scientific research projects on the renewable energy sources, which allowed within the several past years the formation of the solid basis for the so-called "thermo-chemical energy storage technology". This technology is established on the parallel fundamental research on the suitable materials with high energy density properties and on the applied engineering approaches to handle those materials in the most efficient way.

Typically the high energy densities in between of 0.9 and 2.2 GJ m^{-3} are reported for the chemical adsorbents (the pure salt hydrates), *e.g.* $LiSO_4 \cdot H_2O$, $MgSO_4 \cdot 7H_2O$ or $SrBr_2 \cdot 6H_2O$ (Ferchaud, 2012; N'Tsoukpoe, 2014). The TES applications involving commonly studied physical adsorbents such as silica gel and zeolites are characterized by much lower energy densities in the range between 0.1 and 0.5 GJ m^{-3} (Finck, 2014; Hongois, 2011). The novel composite materials are being synthesised to reach the energy density over 1.4 GJ m^{-3} and further experimental framework is going on.

Numerous results issued from the laboratory built apparata including the different design concepts of the TES plants show the feasibility of the domestic integrated systems (Zondag, 2013; Mette, 2013; Marias 2014). The design affords are mostly dedicated to the seasonal storage concepts, wherein the prototyping framework addresses the demonstration of the reactor design and experimentation within controlled conditions. The extensive simulations based on the mathematical modelling of the real-scale systems forecast the onset performances for the employed domestic integrated systems realising various configurations (Skrylnyk, 2012; Hennaut, 2012; Hennaut, 2014).

2. Materials and methods

In the assessment of existing configurations, the prevailing part of TES prototypes are designed on the grounds of fixed bed reactors (Zondag, 2013; Finck, 2013), although a few alternative storages embodying the moving bed reactor concept has been proposed (Mette, 2013; Zondag, 2009). The former apparata comprise the coated finned heat exchangers, atmospheric-packed beds, uncoated adsorbers, whilst in the later installations the motion of material is organised by the mean of mechanical operation.

Considerable shortcoming of closed TES with integrated heat exchanger is the poor heat and mass transfer between the material and the working fluid. The adsorption column type reactor stands as a better idea, but may result in the increased pressure drop across the bed and the appearance of preferential paths for the gas (Marias, 2014). Therefore, the fine layer moving bed reactor is considered to be a good alternative to overcome the mentioned shortcomings.

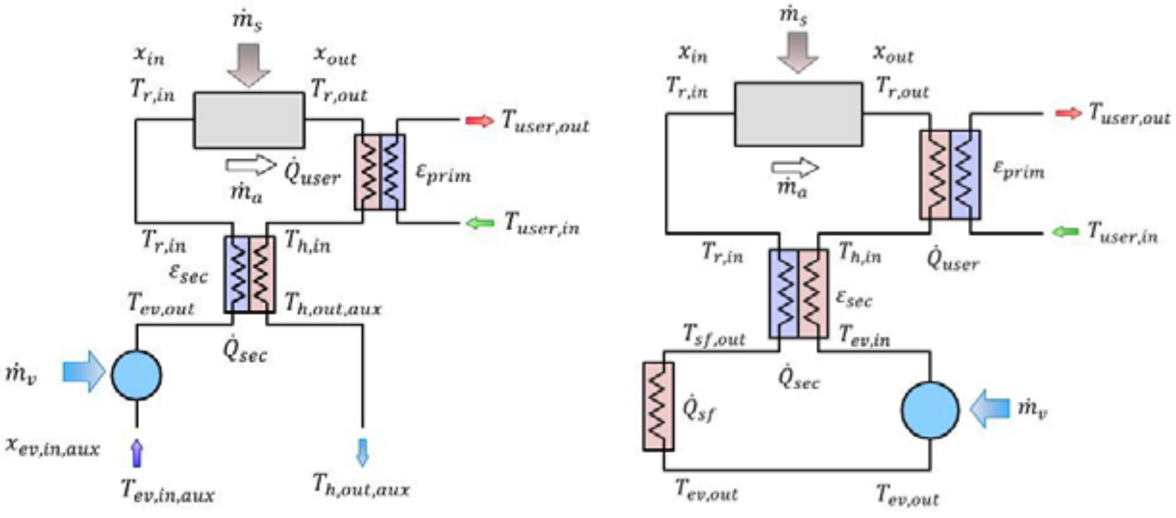
The moving bed reactor can be regarded as the counter- or the cross-flow solid-to-gas heat exchanger being at the steady-state. The particular reason of such assumption is the evaluation of performance criteria purely from the thermodynamic point of view. It should be also noted that the idealised conditions for the transportation of the material from the separate storage to the reactor have been considered at the first time.

The list of potentially interesting configurations has been established with regards to the commonly used heating techniques in the residential sector (Hennaut, 2012; Tanguy, 2012), namely the low temperature air-to-water integrated circuits (a) and combined warm air heating system (b). In the case (a) the working fluid at the outlet of the reactor (hot and dry air) is used to warm the heating fluid (water). In the case (b) the reactor and auxiliary heat exchangers are integrated directly to the ventilation system.

2.1. Air-to-water low temperature heating

In this type of heating technique the output of the reactor is connected to the air-to-water heat exchanger that delivers the produced thermal power to the user's low temperature heating circuit. The low temperature heating technique is particularly adapted for the TES applications to meet the optimal efficiency of heat generation process. The conceptual schemes of the selected TES systems with moving bed reactor are illustrated on the Figs.1 and 2.

The operation process of the illustrated configurations is described as follows. The granular dehydrated material is constantly supplied with the feed rate $\dot{m}_s$ to the reactor, where it reacts with water vapour carried by the air with the mass flow rate $\dot{m}_a$. The air, being isenthalpically moistened at constant water humidification rate $\dot{m}_v$, is blown by the fan unit through the reactor circuit and the related heat exchangers. The air charged with humidity x_{in} enters to the reactor input at the temperature $T_{r,in}$. Due to the chemical reaction, the air is then heated to the temperature $T_{r,out}$. At the output, the TES system delivers the useful heat $\dot{Q}_{user}$ through the primary heat exchanger to the user's heating circuit at the temperature $T_{user,out}$. The secondary heat exchanger placed in parallel to the reactor, is used to retrieve the heat losses $\dot{Q}_{sec}$.



(a) *(b)*

Fig.1: Open-loop circuit with heat recovery (a) and closed-loop circuit with heat recovery (b)

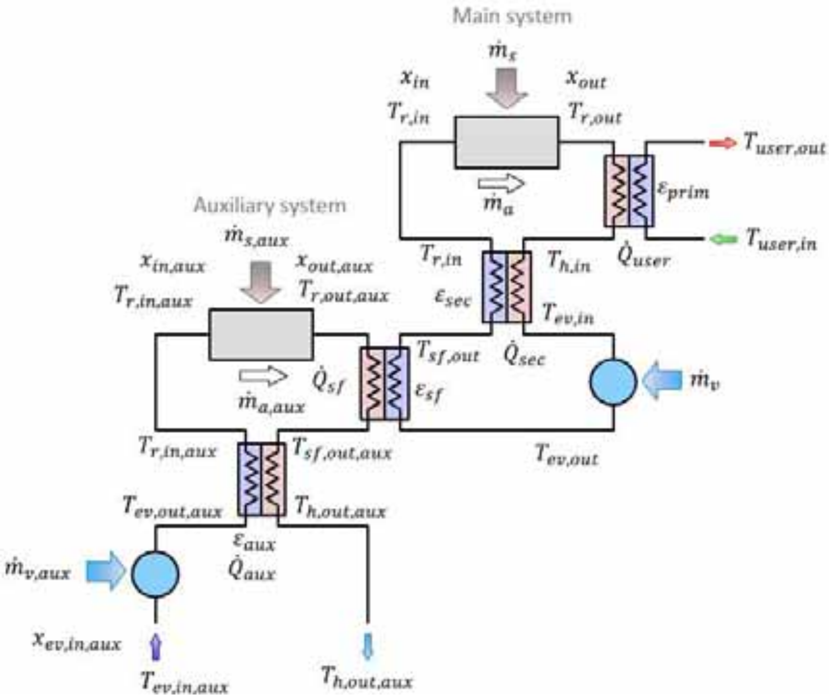


Fig.2: Two-cascaded TES system

The open-loop configuration (see Fig.1 (a)) is designed to use the outside cold air charged with initial humidity $x_{ev,in,aux}$. The processed air is rejected outdoor. The utilisation of the humidifier for the open-loop configuration is optional if the outside humidity content $x_{ev,in,aux}$ is fully sufficient for the generation of the useful heat $\dot{Q}_{user}$.

In the closed-loop configuration (see Fig.1 (b)) the utilisation of the humidifier is permanent and the additional heat exchanger, the so-called *cold source* injecting the amount of heat $\dot{Q}_{sf}$, is preferably added in order to compensate the temperature drop $T_{ev,out} - T_{ev.in}$ due to the air humidification process.

The combination of the two former circuits results in the two-cascaded TES configuration shown on the Fig.2. The *main system* is the closed-loop circuit and is designed to provide the useful heat $\dot{Q}_{user}$ at the system output. The *auxiliary system* is the open-loop circuit which stands as a cold source, thus injecting the necessary amount of heat $\dot{Q}_{sf}$.

2.2. Combined warm air heating

The configurations of the combined warm air heating (see Fig.3 (a) and (b)) represent the particular interest, since those systems can be proposed to the final user as plug-ins to the existing heating circuits. For example, the traditional mechanical ventilation heat recovery unit contains already air-to-air heat exchanger that would facilitate the integration of the TES to the domestic infrastructure.

The low temperature heating circuits as well as the ventilation ducts may be used for the combined warm air heating. Nevertheless, the integration of the TES circuits to the combined warm air heating is restricted by the national norms of ventilation.

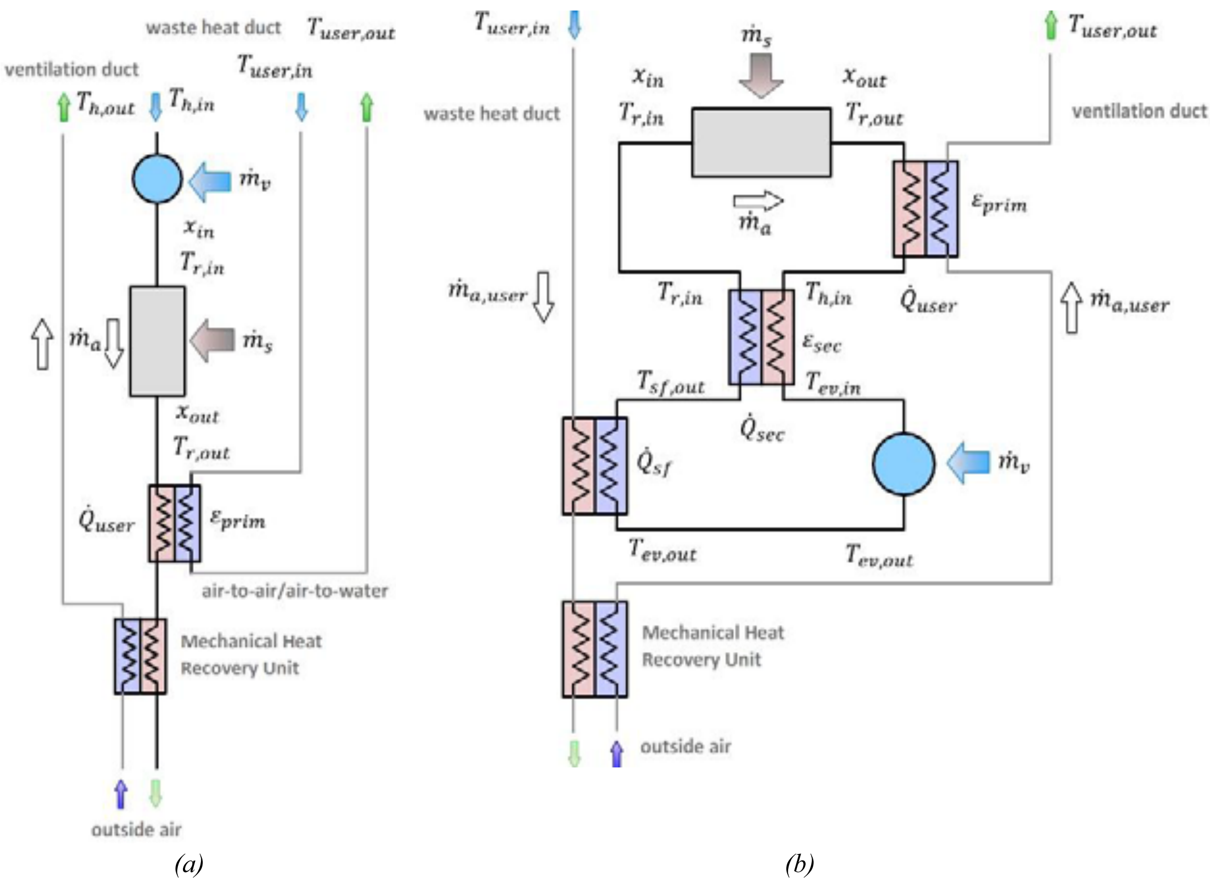


Fig.3: Combined warm air heating systems: (a) open-loop plug-in scheme and (b) closed-loop plug-in scheme

The advantage of the open-loop plug-in scheme (see Fig.3 (a)) consists in the simplicity of the integration to the existent ventilation and heating circuits. The ventilation duct serves to supply the fresh and warm air to the room. The reactor is plugged to the waste air ventilation duct. The amount of useful heat $\dot{Q}_{user}$ is then delivered to the user's heating circuit by the primary heat exchanger.

In case of closed-loop plug-in scheme (see Fig.3 (b)), the reactor and auxiliary components are placed in loop between waste and fresh air ventilation ducts. This scheme enables injection of the warm and fresh air to the house. The waste heat coming out from the building performs the role of the cold source to compensate the temperature drop at the humidifier. The design of this configuration depends on the national ventilation norms and thus the amount of heat retrieved from the waste air is limited by the domestic air renewal system.

2.3. Modelling background

The modelling principles of the depicted configurations have been based on the hypotheses of the steady-state homogenous process of heat and mass transfer over time-invariant conditions for input/output temperatures T and humidity contents x_{in} and x_{out}. Furthermore, it has been considered that the chemical reaction takes places at the equilibrium conditions. The energy balance for the reactor is written as follows:

$$\dot{m}_a C_{P,a}\left(T_{r,in} - T_{r,out}\right) + \dot{m}_a(x_{in} - x_{out})\Delta h_s = 0 \qquad \text{(eq.1)}$$

The process of air humidification is considered to be isenthalpic for which the energy balance in terms of input and output enthalpy flows is written below:

$$C_{p,a}\left(T_{ev,in} - T_{ev,out}\right) + x_{out}\left(C_{p,v}T_{ev,in} + \Delta h_{w,ev}\right) - x_{in}\left(C_{p,v}T_{ev,out} + \Delta h_{w,ev}\right) = 0 \qquad \text{(eq.2)}$$

Here $\Delta h_{w,ev}$ is the heat of water vaporisation and Δh_s is the isosteric heat of adsorption. For every heat exchanger, the energy balance includes the set of equations to define the heat flow $\dot{Q}$ from the hot side to the cold one:

$$\dot{m}_h C_{p,h}\left(T_{h,out} - T_{h,in}\right) - \dot{Q} = 0 \qquad \text{(eq.3)}$$

$$\dot{m}_c C_{p,c}\left(T_{c,out} - T_{c,in}\right) + \dot{Q} = 0 \qquad \text{(eq.4)}$$

$$\varepsilon_{hex} \cdot \min\left\{\dot{m}_h C_{p,h}, \dot{m}_c C_{p,c}\right\} \cdot \left(T_{h,in} - T_{c,in}\right) - \dot{Q} = 0 \qquad \text{(eq.5)}$$

Here ε_{hex} is the effectiveness of the heat exchanger. For the general description purposes, it is supposed that the inlet and outlet temperatures on the hot side are $T_{h,in}$ and $T_{h,out}$, and similarly on the cold side are $T_{c,in}$ and $T_{c,out}$. Moreover, the product of the heat capacity of the fluid on the hot side and its mass flow rate is $\dot{m}_h C_{p,h}$, while at the cold side it is $\dot{m}_c C_{p,c}$.

a. Performance criteria

The performance criteria used for the evaluation and comparison of different configurations are:

- ***Coefficient of performance*** (COP), which is defined as the ratio of the *useful heat* and the *overall energy consumption* by the auxiliary equipment:

$$COP = \frac{\dot{Q}_{user}}{\sum_i W_{f,i}} \qquad \text{(eq.6)}$$

The consumption of the electric power by the fan unit can be found from the next relation:

$$W_f = \dot{V}_g \Delta P_f \cdot {\eta_f}^{-1} \qquad \text{(eq.7)}$$

Where $\dot{V}_g$ is the volume flow rate of the gas, ΔP_f is the total pressure rise from the fan inlet to outlet and η_f is the overall fan efficiency which includes the drive efficiency, the hydrodynamic fan efficiency and the fluid hydromechanical efficiency.

- ***System productivity rate*** (SPR), which is the ratio of the *useful heat* and the *total hydration power*:

$$SPR = \frac{\dot{Q}_{user}}{\sum_i \dot{Q}_r} \qquad \text{(eq.8)}$$

The hydration power is defined by the Eq. (9):

$$\dot{Q}_r = \dot{m}_a(x_{in} - x_{out})\Delta h_s \qquad \text{(eq.9)}$$

- ***Total solid feed rate***, that indicates on the mass of the material to be processed for the heat generation. At the same time this criterion determines the size of the material storage:

$$\dot{m}_s = \dot{m}_a(x_{in} - x_{out}) \cdot (\Delta x_s)^{-1} \qquad \text{(eq.10)}$$

Here Δx_s is the difference of the water uptake in the material between hydrated and dehydrated states.

- ***Total air humidification rate*** that points out the water consumption by the TES system:

$$\dot{m}_v = \dot{m}_a\left(x_{ev,out} - x_{ev,in}\right) \quad \text{(eq.11)}$$

Where $x_{ev,out}$ corresponds to the humidity content at the outlet of the humidifier and is strictly x_{in} for all systems; $x_{ev,in}$ is the humidity content at the inlet of the humidifier and is strictly x_{out} for all closed-loop circuits or respectively is $x_{ev,in,aux}$ for all open-loop circuits.

- ***Cold source necessity*** $\dot{Q}_{sf}$ that means if the additional heat has to be supplied to the TES. This quantity is determined from the energy balance equations (Eq. 1 – 5) at the appropriate heat exchanger.

3. Results and discussion

The numerical values of the known process variables and parameters are presented in the Tab.1.

Tab.1: Numerical assumptions and working conditions of the TES configurations

Name	Parameter	Value
Isosteric heat of adsorption (kJ kg^{-1})	Δh_s	3000
Water uptake difference (kg kg^{-1})	Δx_s	0.2
Useful heat delivered to the user (kW): • Air-to-water low temperature heating (a) • Combined warm air heating (b)	$\dot{Q}_{user}$	 3 -
Reference temperature to the user heating circuit (K): • Air-to-water low temperature heating (a) • Combined warm air heating (b)	$T_{user.out}$	 303.15 299.35
Return temperature from the user heating circuit (K): • Air-to-water low temperature heating (a) • Combined warm air heating (b)	$T_{user.in}$	 298.15 293.15
Total pressure at the reactor input (kPa)	P	101.32
Pressure drop per heat exchanger (Pa)	ΔP_{hex}	150
Heat exchangers effectiveness	$\varepsilon_{prim}, \varepsilon_{sec}, \varepsilon_{sf}, \varepsilon_{aux}$	0.8
Nominal fan unit efficiency	η_f	0.48
Nominal outdoor relative humidity	φ	0.6

The isosteric heat of adsorption and the water uptake difference are issued from the characterisation conditions of the composite material (porous matrix – inorganic salt) for which the estimated theoretical energy density reaches 1.4 GJ m^{-3}. The useful heat $\dot{Q}_{user}$ and the reference temperature $T_{user.out}$ are the boundary conditions for the configurations with low temperature heating (a). The value of 3 kW is chosen from the condition of the power that the average heating installation must provide to come across with heating demand. For the case (b) this value is not fixed, since the thermal power that must be carried with warm air depends on the outdoor temperature and only the reference temperature $T_{user.out}$ is taken for the boundary conditions.

The return temperature represents the boundary conditions at the outlet of user's heating circuit for case (a) and is the comfort temperature for the case (b). The total pressure P is the pressure at which the installation is supposed to run. This pressure corresponds to the standard atmosphere pressure. The pressure drop per heat exchanger is set to be constant for the reasons of simplicity and its value supposed to fit the elevated air flow rates. The effectiveness of heat exchangers is some arbitrary parameter that is put constant in the present study for the reasons of simplicity and the absence of full information about characteristics of real heat exchangers. The value of the nominal fan unit efficiency represents the average characteristics of the axial flow fan type; however it is a subject of the variation according to the type of fan which would be used in the TES configuration.

3.1. Evaluation of working cycle

In order to find the unknown variables, the calculations have been performed in the range of outdoor temperatures T_a between 271.15 K and 283.15 K. However, it should be noted that the solutions obtained for the configuration from the Fig.1 (a) are not meaningful in this range of outdoor temperatures and for chosen boundary conditions, since this system requires very high air flow rate to deliver 3 kW of the useful heat. Therefore, the temperature range T_a for this configuration has been shifted to 284.15 – 288.15 K.

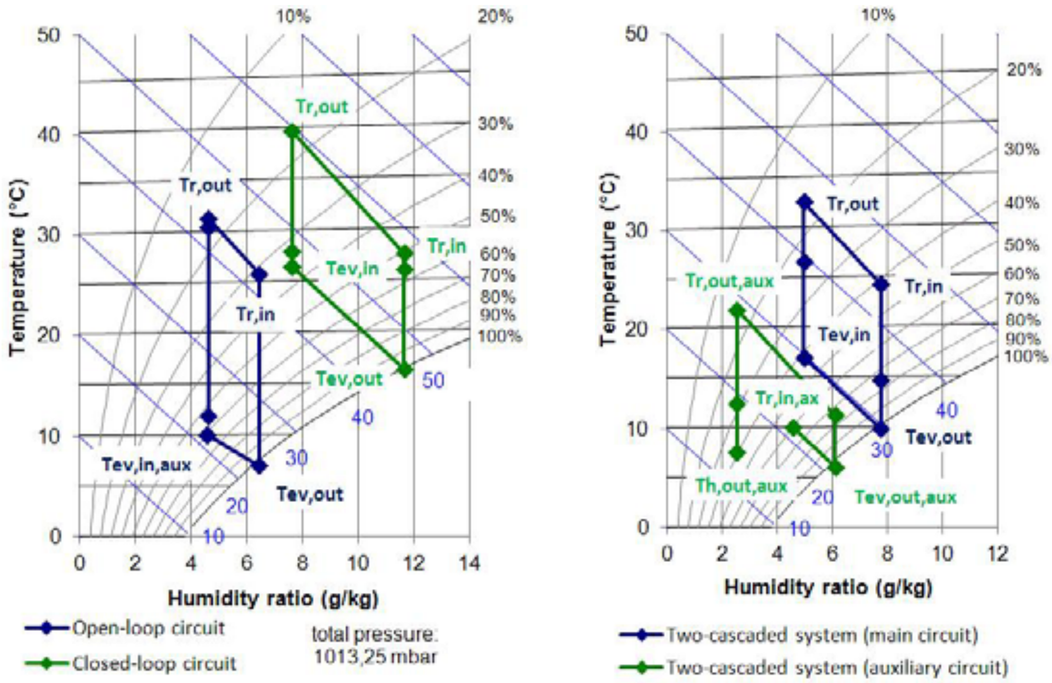


Fig.4: Working cycle diagrams of the configurations used for the low temperature heating

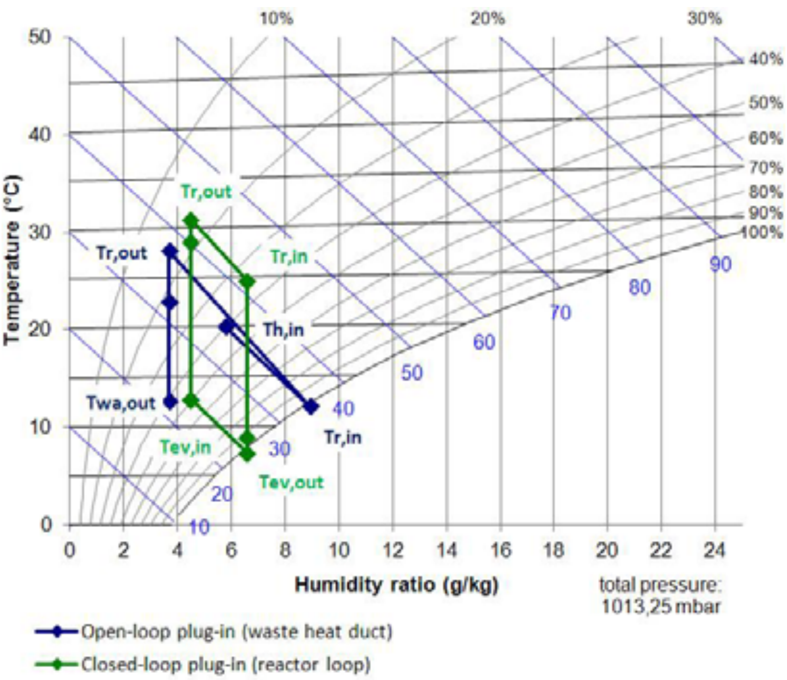


Fig.5: Working cycle diagrams of the configurations used for the combined warm air heating

The most representative results for all listed configuration are published on the Figs. 4 and 5. The temperatures and humidity levels for the open-loop circuit are calculated at $T_a = 284.15$ K, while for the rest of configurations the outdoor temperature is equal to $T_a = 283.15$ K.

It has been concluded that the temperatures above 283.15 K approach to the *building base temperature*, where the heating is not needed due to the thermal inertia of building and to the solar gains. Therefore, the operating range of the stand-alone open-loop TES configuration is limited for the adopted useful heat of 3 kW. The utilization of this TES is unreasonable for the high power applications.

It was found that the configurations destined for the low temperature heating require quite large air flow rates, because of the imposed high power at the system output. Thus the designer's temperature drop across the primary heat exchanger of 3 – 5 K must be maintained either with temperature boost from the cold source, or by increasing the flow rate. The closed-loop circuit configuration from the Fig. 1 (b) does not depend on the outdoor temperature conditions, but only on the flow rate of the heating medium $\dot{m}_{a,user}$ on the cold side of the primary heat exchanger and the amount of heat injected from the cold source $\dot{Q}_{sf}$; whilst the rest of configurations depend on the outdoor temperature and the air flow rate $\dot{m}_a$ (with the effectiveness ε_{prim}), because the ambient air plays the role of the cold source for those configurations.

3.2. Evaluation of the performance criteria

The main results of the performance criteria evaluation of the selected configurations are summarized in the Tab.2. It has to be reminded that the evaluation of the configurations taken from the case (b) is based on the constant reference temperature equal to 299.35 K, while for the case (a) the constant output heat flow equal to 3 kW is used. Therefore, the performances criteria are also dependant on the useful heat.

Tab. 2: Performance characteristics of the studied configurations

Configuration name	Outdoor temperature (K)	air flow rate (kg s^{-1})	COP	SPR	Solid feed rate (kg s^{-1}) $\times 10^{-3}$	air humidification rate (kg s^{-1}) $\times 10^{-3}$	Cold source necessity (kW)
Air-to-water low temperature heating (a)							
Open-loop circuit	284.15	3.75	3.25	15.7 %	31.78	5.23	-
	285.15	0.39	31.25	75.0 %	7.00	0.69	-
	288.15	0.53	23.06	65.8 %	7.59	0.85	-
Closed-loop circuit	-	0.28	1.84	53.7 %	5.16	1.03	2.48
	-	0.35	1.74	51.6 %	5.63	1.13	2.43
	-	0.49	1.57	47.6 %	6.59	1.32	2.33
Two-cascaded TES[1]	273.15	0.54 (1.44)	2.48	26.5 %	18.81	2.79	2.29
	275.15	0.54 (0.62)	2.99	37.2 %	13.48	2.05	2.29
	283.15	0.54 (0.24)	3.31	44.2 %	11.31	1.75	2.29
Two-cascaded TES[2]	271.15	0.97 (0.80)	1.16	29.3 %	17.04	2.75	1.96
	275.15	0.97 (0.41)	1.72	33.5 %	14.94	2.49	1.96
	283.15	0.97 (0.19)	1.75	36.1 %	13.84	2.34	1.96
Combined warm air heating (b)							
Open-loop plug-in	271.15	0.08 (0.041)	4.42	14.4 %	2.07	0.42	-
	275.15	0.08 (0.044)	5.83	20.7 %	2.07	0.42	-
	283.15	0.08 (0.073)	8.00	39.8 %	2.07	0.42	-
Closed-loop plug-in	271.15	0.71 (0.080)	1.28	36.4 %	6.83	1.37	0.800
	275.15	0.64 (0.080)	1.27	38.5 %	6.19	1.24	0.804
	283.15	0.45 (0.080)	1.54	44.3 %	4.91	0.98	0.809

The overall useful heat for the open-loop plug-in has been calculated as the sum of heat amount transmitted to the primary heat exchanger and the heat injected by the ventilation. Therefore, the overall useful heat varies proportionally in between of 0.18 and 0.421 kW in accordance to the outdoor temperature range for the indoor humidity evaluated as 40 %. Concerning the closed-loop plug-in configuration, the useful heat on the primary heat exchanger has been calculated between 1.491 and 1.305 kW.

It can be seen that the performances of the open-loop circuit are high only for the given range of outdoor temperature, however as it was pointed out before, this configuration is not relevant for the stand-alone domestic heating application. The advantage of the simple closed-loop circuit is the independence on the outdoor temperature, but the significant shortcoming of this configuration is a large demand of the cold source reaching up to 80 % in the share of the useful heat. The price of diminishing of the cold source share on 6 % by increasing the air flow rate results in the drop of the COP on 15 % because of the electric power consumption by the fan unit. Moreover, the cold source technique, *e.g.* the geothermal energy, can be an expensive option.

The two-cascaded system has more degrees of liberty for the design and merges together the advantages of the open-loop and closed-loop circuits. The most representative results in the range of outdoor temperature taken between 271.15 K and 283.15 K are shown for the lower ([1]) and upper bounds ([2]) of the air flow rate in the main circuit (see Tab.2). The air flow rate for the auxiliary circuit is cited in the parentheses. It was found that the higher flow rates in the main circuit allow extending the operating range of the outdoor temperatures and reducing the cold source demand on 14 %. However, the COP in this case dramatically drops down onto 42 ... 47 %. The quantity of the material and water for the humidification process also increase for the higher flow rates in the main circuit. Therefore, the special regulation technique has to be implemented to keep the performances of this configuration optimal.

The open-loop plug-in configuration exhibits indeed quite interesting performance characteristics and does not require the presence of the cold source, since it is as a matter of fact included in the indoor comfort conditions. But its SPR factor is insufficient in comparison with other configurations. Besides, the hydration power of this configuration has to be controlled in accordance to the internal humidity comfort level. The closed-loop plug-in allows overtaking the lack of SPR resulting in the amplification of the useful heat. The utilization of the warm waste air allows retrieving up to 46 % of the heat to assure the cold source demand. No internal humidity control is required for this configuration. The air flow rates in the parentheses for the configurations (b) are those denoted as $\dot{m}_{a,user}$.

4. Conclusions

The closed-loop systems are easy to integrate into existent domestic systems and they have more degrees of freedom for the air flow rate adjustment and useful heat generation, but they require the presence of the cold source and the humidifier. Open-loop systems may strongly depend on the existent air ventilation, sanitary norms and ambient conditions; however they do not require any additional heat input. Additionally, the open-loop plug-in scheme may disturb the internal comfort acting as a dehumidifier. The two-cascaded system is a combination of the closed-loop and open-loop circuits and is a self-alimentation process that covers the needs of the cold source, but the increased autonomy degree results in a double quantity of the material in comparison to the simple loop TES.

5. Acknowledgments

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