INNO-ST-05

Modelling of a solar thermo-chemical system for energy storage in buildings

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1. Introduction

Design of a heat storage apparatus can be viewed through the different angles of scientific and engineering procedures in a way to fulfil a set of criteria at the every stage of design [1]. Because the global objectives may differ from one project to another [2], the adaptation of existent experience to the concrete goals seems to become tedious if an appropriate methodology is not established. There are several commonly defined objectives leading or not to the commercialisation of thermal storage device [3]: demonstration design, laboratory scale prototyping, full scale design. Since the scaling factor of thermal storage apparatus varies significantly [4], the conceptualizing procedure and basic hypothesis may contribute a substantial part in the total risk of project failure. So far that, if the simplifications in basic hypotheses seem to be enough sufficient to achieve the desired results during the demonstration phase, they are indeed the main source of inadequate behaviour of the full scale device, resulting in a higher risk of project failure. On the other hand, the complication of limiting factors and excessively broad specification of operating conditions may lead to highly contradictory restrictions of design, leaving no possibility for the further optimization.

On the initial stage of the design, depending not of the type of the project, it appears useful to establish the determinative shapes of the future heat storage device, basing on the grounds of the integral analysis of energetic performances [5]. This stage can be regarded as a fast forward designing procedure that points to the plausible set of solutions. Lately, on the second stage, the more detailed modelling [6, 7], involving the geometrical aspects is necessary to undertake to fill up the unprecedented gaps between initial and final study. Surely, with the more inaccurate initial conditions and apparatus working constraints, the gap between simplified and fully developed approaches is considerable, resulting in the increased number of intermediate iterations during the designing procedure before its convergence.

Therefore, the overall process of design is regarded as an iterative convergence of contradictions to the acceptable range of theoretical and technical solutions [8]. Thus, the methodological backgrounds include an iterative scheme, where each iteration is assigned to the optimisation of the operating conditions, energy performances and constructive factors.

The goal of this paper is the demonstration of the methodological design principles within theoretical modelling of thermal heat storage apparatus and simulation of inter-seasonal heat storage system. The designing procedure starts from the modelling of thermal plant behaviour, based on the simplifications in the basic hypothesis. Afterwards, a more detailed modelling, involving dynamic aspects and additional features of plant components, is presented. The accomplishment of the designing procedure follows by the optimisation of key parameters of the thermal storage plant.

2. Material and methods

2.1. Case of system configuration

Suppose the main objective is the development of the inter-seasonal thermal storage system to satisfy the heat demand of a reference building situated in a particular climate zone u. The thermal heat storage system includes a plant and auxiliary equipment, such as solar collectors and heating sector equipment. Plant is a thermal storage apparatus which consists of a chemical reactor filled with a working medium $A \cdot mB$ and connected to a tank to store a component B in a liquid phase. It should be noted, that the modelled plant assumes to work on the grounds of closed thermal chemical apparatus. Medium is a working material suggested to the reversible chemical reaction of decomposition or synthesis $A \cdot mB \leftrightarrow A \cdot (m-n)B + nB$, where A is a solid (adsorbent) and B is an adsorptive. The functional principles and the studied configuration of the system are showed on figure 1.

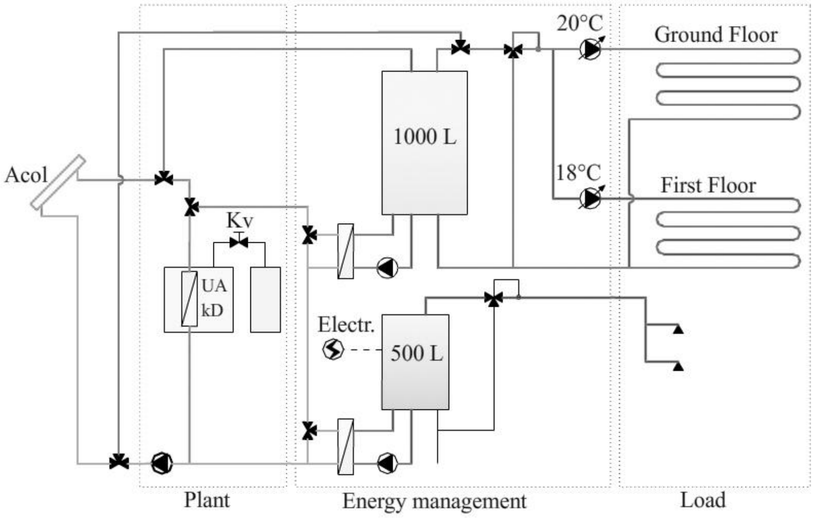

Figure 1. Thermal heat storage system

The system operates in two modes: heat accumulation mode (a) and heat restitution mode (b). In the heat accumulation mode (a) the solar energy captured by solar collectors is used to supply heat to the plant. Moreover, the part of solar energy runs through the hot water storage tank to come across with prior domestic hot water needs. A water storage tank of 500 L stands to meet the daily need of a single-family household. The operating time in mode (a) lasts 3696 hours from May till September. In mode (b), the accumulated energy is transferred from the plant to the building heating sector, which consists of a buffer water tank of about 1000 l and a heating floor circuit. Since the studied building is composed of two zones, the regulation policy relies onto two reference temperatures with distinct heating circuits, namely 20 °C for the ground floor and 18 °C for the first floor. The running cycle for mode (b) counts 3624 hours, starting in November and ending in March.

2.2. Simplified Modelling Approach (Model 1)

The so-called simplified model is based on the knowledge of thermodynamic properties of the storage medium [5], global energy requirements of dwelling and some global data about the plant, such as a volume of thermal storage apparatus. The sizing of storage volume has been done based on the overall heat requirements for the reference building. The model takes into account only a stationary regime. Therefore, the underlaying hypothesis is defined as follows:

Hypothesis 1*. The plant stays in the equilibrium state. The transient phenomenon does not affect the energy output of the thermal storage device.*

The former assumption is suitable to perform simulations of the system in a quick-start way in order to evaluate the correctness of operating conditions and the relevance of the chosen working material in the heat storage application.

However, in real systems the former statement may fail with eventually large perturbations and daily variation of operating conditions. Therefore, the model can be describes by the following energy and mass balance equations:

$$Q_r(\theta,s,u,t) = -Q_{ads}(\theta,s,u,t) + Q_{acc}(\theta,s,u,t) \tag{1}$$
$$m_a(\theta,s,u,t) = m_v(\theta,s,u,t) + m_{\tan k}(\theta,s,u,t) \tag{2}$$

Where $(\theta_{\min} \le \theta \le \theta_{\max}, u_{\min} \le u \le u_{\max}, s_{\min} \le s \le s_{\max}) \in \Omega$ are set of parameters θ, admissible operating conditions u and real plant states s bounded inside of a restricted region Ω.The equations (1, 2) represent both system operation modes (a) and (b). In mode (a) the term $Q_r(\theta,s,u,t)$ is the energy supply from solar collectors to the plant at any time t:

$$Q_r(\theta,s,u,t) = \int_0^t \varepsilon_{col} G_{col}(t) A_{col} d\tau \equiv \int_0^t \dot{m}_{w,col} G_w \left(T_{w,col}(\tau) - T_{w,out}(\tau)\right) d\tau \tag{3}$$

While for mode (b)

$$Q_r(\theta,s,u,t) = \int_0^t G_{\sec t}(t) d\tau \equiv \int_0^t \varepsilon_{\sec t} \dot{m}_{w,\sec t} C_w \left(T_{w,\sec t}(\tau) - T_{w,out}(\tau)\right) d\tau \tag{4}$$

represents the heat demand of the building sector, that takes into account the hot water distribution losses.

The term $Q_{ads}(\theta,s,ut,t)$ is related to the endoergic or exoergic nature of reaction, according to the operating mode and is described by the following relation:

$$Q_{ads}(\theta,s,u,t) = (m_v(t) - m_v(0)) \Delta H_{m,ads} \tag{5}$$

The second term $Q_{acc}(\theta,s,u,t)$ is the energy accumulated by the plant at any time t:

$$Q_{acc}(\theta,s,u,t) = (m_j C_j) \cdot (T_r(t) - T_r(0)) \tag{6}$$

Where $(m_j C_j) = m_{ads} C_{ads} + m_v(t) C_v + m_a(t) C_a + m_{tb} C_{tb}$ with C_j are thermal capacities of every medium of reactor for $j = \{ads, v, a, tb\}$ related to the adsorbent A, gaseous state and adsorbed state of adsorptive B and all metallic components. Equation (2) is the mass balance for the entire amount of adsorptive $m_a(\theta,s,u,t)$, that is equivalent to the sum of mass of adsorptive $m_v(\theta,s,u,t)$ in gaseous state in the control volume of reactor and of $m_{\tan k}(\theta,s,u,t)$ in the storing tank in liquid phase. Both terms are modelled based on the ideal gas equation:

$$m_v(\theta,s,u,t) = \left(\frac{P_v(t)}{T_r(t)} - \frac{P_v(0)}{T_r(0)}\right) \frac{V_v}{R_\mu}, \tag{7}$$

$$m_{\tan k}(\theta,s,u,t) = \left(\frac{P_v(t) - P_{\tan k}(t)}{T_r(t)}\right) \frac{V_v}{R_\mu} \tag{8}$$

Operating conditions u include the following information: $G_i(t)$ are solar gains on the inclined surface of solar collector or heat demand of dwelling with $i = \{col, \sec t\}$; $T_{\tan k}(t)$ is the temperature of condensation or evaporation of adsorptive in the storing tank that depends on the heat source; $P_{\tan k}(t)$ is the vapour pressure inside of the storing tank, related to the temperature $T_{\tan k}(t)$ by Clapeyron's law. The set of constraints Ω takes into account the operating mode and consists of $\{P_v(t) \ge P_{\tan k}(t), 0 \le m_v(t) \le m_{v,\max}, T_r(t) \ge T_{r,\min}\}$ in case of heat accumulation mode (a) and $\{P_v(t) \le P_{\tan k}(t), 0 \le m_v(t) \le m_{v,\max}\}$ in case of heat restitution mode (b). The first constraint provides herein the control over mass transfer between reactor and tank. Thus, the second constraint is related to the first one in order to saturate the chemical reaction in the control volume V_v. The last constraint is used to define the minimal temperature when the chemical reaction starts in order to supply the necessary amount of energy from solar collectors.

According to the hypothesis 1, the plant states $s = \{P_v(t), T_r(t), m_v(t)\}$ are calculated from the equilibrium law in case of pressure:

$$\ln(P_v(t)) = \frac{\Delta S_{\mu,ads}}{R} - \frac{\Delta H_{\mu,ads}}{RT_r(t)} \tag{9}$$

and from steady-state equation in case of reactor temperature, using the mean algebraic temperature drop $\Delta T = const$ across the reactor heat exchanger with output temperature $T_{w,out}(t)$.

$$T_r(t) = T_{w,out}(t) - \Delta T \tag{10}$$

2.3. Detailed Modelling Approach (Model 2)

Similarly to the former approach, the detailed modelling of the plant relies onto the energy and the mass balance, however considering the differential form of the onset composition. Moreover, the heat flux and the mass flow equations bring out separately the states of the working medium, of the heat exchanger and of the control volume. Thus, the enlarged set of parameters is required in order to give a proper description of transient phenomenon of the plant.

It should be noted that the set of operating conditions u remains the same as in the former modelling approach, since no other auxiliary equipment is involved in the further considerations. The main hypothesis is formulated with regards to the perfect mixing behaviour of the plant:

Hypothesis 2. *The working medium is homogeneous and uniform. The process is adiabatic and no heat losses occur for the material transfer.*

With regards to the hypothesis 2, the following enlarged model is defined to describe the energy and the mass balance:

$$\frac{d}{dt}(Q_{ads}(\theta,s,u,t)) = q_{ads}(\theta,s,u,t) + q_r(\theta,s,u,t) \tag{11}$$

$$\frac{d}{dt}(Q_{hex}(\theta,s,u,t)) = q_{r,hex}(\theta,s,u,t) + q_r(\theta,s,u,t) \tag{12}$$

$$\frac{d}{dt}(m_v(\theta,s,u,t)) = F_a(\theta,s,u,t) - F_{\mathrm{tank}}(\theta,s,u,t) \tag{13}$$

Relation (11) states that the rate of heat accumulation in the working medium consists of the sum of heat flux $q_{ads}(\theta,s,u,t)$ caused by the chemical reaction of decomposition/synthesis and the heat flux $q_r(\theta,s,u,t)$ supplied to the plant. Relation (12) describes the energy balance of the heat exchanger that consists of the heat flux $q_{r,hex}(\theta,s,u,t)$ between the heat exchanger and working medium and of the heat flux $q_r(\theta,s,u,t)$ provided to the whole plant.

Relation (13) is interpreted in terms of the incoming and outcoming mass fluxes of adsorptive.

Thus, the key parameters that affect the transient behaviour of the plant are the heat and mass transfer rates. The definition of the rate of heat transfer requires some basic considerations of the preliminary geometry of the heat exchanging parts of the reactor. It should be mentioned withal that the mass transfer consists of two major processes: the decomposition/synthesis of working material $F_a(\theta,s,u,t)$ and the transport of adsorptive, being in a gaseous state, between the control volume of reactor and the storing tank $F_{\mathrm{tank}}(\theta,s,u,t)$.

Unlike the previous approach, the plant states s are defined for the present model as follows: $T_r(t)$ is a temperature of the working medium; $T_{w,hex}(t)$ is a temperature of the fluid inside of the heat exchanger; $\rho_v(t)$ is a vapour density in the control volume V_v of reactor; $x_a(t)$ is the amount of captured adsorptive on the surface of adsorbent (state-of-charge); $P_v(t)$ is a vapour pressure in the control volume V_v of reactor.

The modified set of constraints is implemented to provide the control over the mass fluxes and is $\Omega = \{P_v \geq P_{\tank k}(t), 0 \leq F_{\tank k}(t) \leq F_{a,\max}\}$ for the heat accumulation mode (a) or $\Omega = \{P_v \leq P_{\tank k}(t), 0 \leq F_{\tank k}(t) \leq F_{a,\max}\}$ in case of heat restitution mode (b), where $F_{a,\max}$ is related to the maximal vapour flow defined from equilibrium condition.

Similarly to the equation (6), the rate of heat accumulation in the medium using the differential form is defined as follows:

$$\frac{d}{dt}\left(Q_{acc}(\theta,s,u,t)\right) = \left(m_j C_j\right)\frac{d}{dt}T_r(t) \tag{14}$$

With a difference in the term $m_j C_j = m_{ads}\left(C_{ads} + C_a x_a(t)\right) + V_v \rho_v(t) C_v + m_{tb} C_{tb}$. The heat flux $q_{ads}(\theta,s,u,t)$ is proportional to the rate of change of the state-of-charge:

$$q_{ads}(\theta,s,u,t) = m_{ads}\Delta H_{m,ads}\frac{d}{dt}x_a(t) \tag{15}$$

The heat flux $q_r(\theta,s,u,t)$ supplied to the plant is:

$$q_r(\theta,s,u,t) = \dot{m}_{w,i} C_w \left(T_{w,i}(t) - T_{w,hex}(t)\right) \tag{16}$$

Where $i = \{col, \sec t\}$, depending on the operating mode. Therefore, the state variable $T_{w,hex}(t)$ is derived considering the perfect mixing model of the heat exchanger, so that:

$$\frac{d}{dt}\left(Q_{hex}(\theta,s,u,t)\right) := m_w C_w \frac{d}{dt}T_{w,hex}(t) = UA\left(T_r(t) - T_{w,hex}(t)\right) + q_r(\theta,s,u,t) \tag{17}$$

Where UA is an overall heat transfer coefficient of plant heat exchanger. The rate of change of the free adsorptive in the control volume V_v of reactor is given as:

$$\frac{d}{dt}\left(m_v(\theta,s,u,t)\right) = V_v \frac{d}{dt}\rho_v(t) \tag{18}$$

For the mass balance relation in (13), the mass flow rate of adsorptive between the control volume and the storing tank is proportional to the pressure difference:

$$F_{\tank k}(\theta,s,u,t) = \begin{cases} K_v \cdot \rho_v \sqrt{\dfrac{P_v(t) - P_{\tank k}(t)}{g_a}}, & \text{if mode (a)} \\ K_v \cdot \dfrac{P_{\tank k}(t)}{R_\mu T_{\tank k}(t)} \sqrt{\dfrac{P_{\tank k}(t) - P_v(t)}{g_a}}, & \text{if mode (b)} \end{cases} \tag{19}$$

Where K_v is the flow coefficient of the adsorptive, being in gaseous state, between the control volume of the reactor and the storing tank; g_a is the specific gravity of adsorptive. In the mass flux calculation $F_a(\theta,s,u,t) = m_{ads}\frac{d}{dt}x_a(t)$, the variation of the state-of-charge is supposed to be linear:

$$\frac{d}{dt}x_a(t) = k_D\left(x_a^* - x_a\right) \tag{20}$$

Where $x_a^* = x_{a,\max}$ is the maximum state-of-charge of the chemical compound for the mode (b) and $x_a^* = x_{a,\min}$ is the minimum state-of-charge for the mode (a); k_D is the rate of decomposition/synthesis of the working material. Additionally, the pressure of the control volume is obtained by deriving the ideal gas equation:

$$\frac{d}{dt}P_v(t) = R_\mu T_r \frac{d}{dt}\rho_v(t) + R_\mu \rho_v(t)\frac{d}{dt}T_r(t) \tag{21}$$

3. Optimisation of key parameters of heat storage apparatus

3.1. Formulation of the optimisation problem

The optimisation procedure stands as the intermediate step of the whole designing routine in order to search such technical parameters, so that the convergence of energy performances of the plant and the heat demands of the reference dwelling can be reached. It involves two main stages:

- The choice of response analysers w
- The tracing of suboptimal target response $\text{opt}\, w$

For the goals of empirical study, the whole optimisation procedure can be referred to the Response Surface Methods [9]. This class of methods stands simultaneously as a suitable tool to analyse the effects of interactions between variables, parameters or operating condition and to resolve the complex optimisation problem.

In order to limit a set of possible response analysers, a second-order empirical model is applied here:

$$w(\theta) = a_0 + \sum_i a_i\theta_i + \sum_i a_{i,i}\theta_i^2 + \sum_i\sum_j a_{i,j}\theta_i\theta_j \tag{22}$$

Where a_0 is the main effect for the response, $\{a_i\}$ are linear effects, $\{a_{i,i}\}$ are the quadratic effects of studies parameters and $\{a_{i,j}\}$ represent the interactions between parameters. Solutions $\{a_0, a_i, a_{i,i}, a_{i,j}\}$ are obtained by the help of multiple regression approach. It should be noted that the chosen response analyser (22) allows optimising a set of parameters θ. However, the additional information about operating conditions, i.e. source and sink temperature, can be included if necessary.

A set of parameters θ is evaluated by minimisation technique of polynomials (22), leading to the suboptimal target response $\text{opt}\, w$ within a restricted area Ω. The desirability function method has been utilized to trace the suboptimal target responses.

3.2. Experimental design

The screening experimental domain has been generated using the central composite design scheme. The key parameters to optimise are overall mass of adsorbent m_{ads}, total surface of solar collectors A_{col}, flow rate during the heat accumulation mode $\dot{m}_{w,col}$, fractional volume of storing medium $x_v = 1\Big/\left(1 + V_v \cdot \dfrac{\rho_{ads}}{m_{ads}}\right)$. Additionally, depending on the type of the concerned model, other important parameters have been included: heat distribution efficiency $\varepsilon_{\text{sect}}$ for the first modelling approach (1, 2) and adsorptive flow coefficient K_v in case of the detailed modelling approach (11 - 13).

The following responses have been selected to analyse the overall behaviour of the system during the N years of operation:

- The number of heating degree-hours bellow 20°C $UNDH_{rez} = \sum_{i=1}^{N} UNDH_{rez,i}$ for the ground floor of studied dwelling
- The number of heating degree-hours bellow 18°C $UNDH_{floor} = \sum_{i=1}^{N} UNDH_{floor,i}$ for the first floor
- The difference of accumulated and resituated heat $dQ_{a-b,i} = \Delta m_{\text{tank,i}}(t)\Delta H_{m,ads}$ between modes (a) and (b). Alternatively, from the flow rate of adsorptive $dQ_{a-b,i} = \int_{t_{0,i}}^{t_i} \Delta F_{\text{tank},i}(\tau)\Delta H_{m,ads}\, d\tau$

Here $i = \overline{1, N}$ denotes the current year of operation from the total number of cycles N. Additionally, the response of the variation $\mathrm{var}(\{dQ_{a-b,i}\})$ is also analysed in order to keep the energy performances of the plant equalised.

4. Results and discussions

For the demonstration reasons, the optimisation of plant parameters has been performed using the $CaCl_2/H_2O$ a working pair. The temperature $T_{\mathrm{tank}}(t)$ has been simulated considering the ground as the heat source for evaporation and the heat sink for condensation.

The design conditions u are based on the annual heat consumption for a typical Belgian dwelling, including the space heating demand and domestic hot water production. The reference value of space heat demand is about 42 kWh/m² per year for the reference dwelling and for domestic hot water is 2700 kWh/year. The building is situated in a climate zone with the following conditions:

- Mean annual air temperature $T_{a,m} = 9{,}73°C$ of D.B.
- Minimal and maximal seasonal air temperatures $T_{a,\min} = -8°C$, $T_{a,\max} = 29{,}8°C$ of D.B.
- Global solar irradiance on the horizontal surface $G_{s,gl} = 946{,}4\, kWh/m^2$ per year
- The lowest value of solar irradiance is $G_{s,gl,\min} = 19{,}83\, kWh/m^2$ recorded for January and the highest value is $G_{s,gl,\max} = 147{,}89\, kWh/m^2$ recorded for June on the horizontal surface
- Mean ground temperature $T_{g,m} = 10{,}8°C$
- Minimal and maximal seasonal ground temperatures $T_{g,\min} = 8{,}3°C$, $T_{g,\max} = 13{,}3°C$

The optimisation of parameters for both models has been performed by means of computer-aided software Minitab. The software allows calculating also the significance of each parameter, either the interaction between them. The initial values, constraints and the optimised values of the parameters are presented in the table 1.

Table 1. Optimisation of the parameters of the plant

Name of the parameter	Restrictions	Model 1	Model 2
Mass of anhydrite working medium, tons	$15 \le m_{ads} \le 30$	22470,5	23023,0
Surface of solar collectors, m²	$12 \le A_{col} \le 25{,}52$	14,41	15,42
Fractional control volume	$0{,}1 \le x_v \le 0{,}9$	0,84	0,65
Flowrate in mode (a), kg/h.m²	$15 \le \dot{m}_{w,col} \le 25$	16,82	25,0
Efficiency of heat distribution	$0{,}3 \le \varepsilon_{\mathrm{sect}} \le 0{,}9$	0,87	
Flow coefficient, m³/h.Pa	$4 \le K_v \le 18$		16,6

It can be shown from table 1 that the optimised set of parameters is situated inside of restricted area Ω. Moreover, the values of parameters are located very closed, showing the convergence tendency of solutions for both the modelling approaches.

For the calculated set of parameters the verification simulations over three consecutive years have been carried out. In order to compare the energy performances of both models and at the same time to check the correctness of calculated parameters, the energy density of the working medium has been used, instead of a simple difference between stored and restituted energy. The results of the simulations are listed in the table 2. Therefore, the energy densities of the working medium are similar. For those conditions, the necessary volume of material is equal to 12,35 m³ for Model 1 and to 12,51 m³ for Model 2 respectively.

Table 2. Verification of models and energy densities of the working medium

Number of Cycle	Stored Energy Density, kWh/m³		Restituted Energy Density, kWh/m³		Difference, kWh/m³		Cycling difference of accumulation, kWh/m³		Cycling difference of restitution, kWh/m³	
	M.#1	M.#2	M.#1	M.#2	M.#1	M.#2	M.#1	M.#2	M.#1	M.#2
First year	212,17	213,75	212,19	214,47	-0,021	-0,726				
Second year	205,46	211,33	205,45	212,80	0,011	-1,475	-6,71	-2,42	-6,74	-1,67
Third year	199,90	207,97	199,85	208,93	0,007	-0,968	-5,56	-3,36	-5,60	-3,87

However, due to the transient behaviour, the difference between stored and restituted energy densities is much more significant for the Model 2. A more detailed look at the monthly performances gives an illustrative explanation. The monthly energetic outputs of the thermal storage system over three years of work are showed on the figure 2. It can be seen from the figure 2 that the output differences between each year of overall work occur mainly at transitions from the mode (a) to the mode (b) and vice versa. For the mode (a), the amount of energy stored from June till September is uniform over three cycles for the both models. This period corresponds to the normal heat accumulation operation. The transitory period starts in May and the cycling differences are more considerable for the Model 1. However, the amount of energy, accumulated for the same month in case of Model 2, tends to the equalisation.

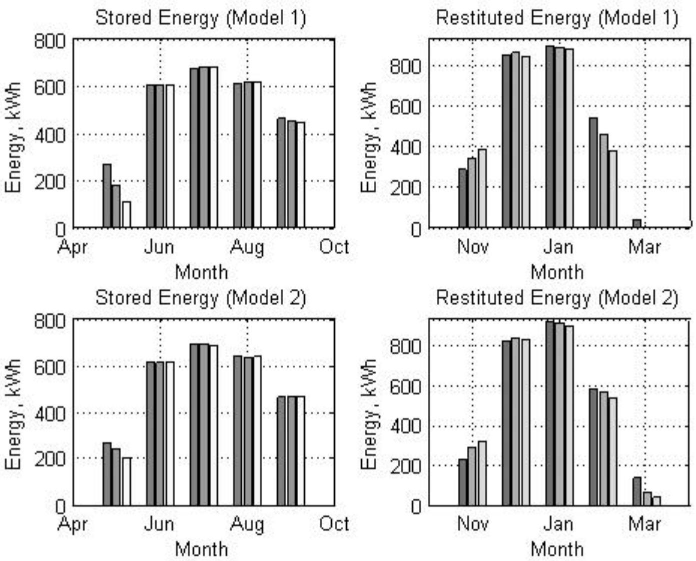

Figure 2. Comparison of monthly energetic output of thermal storage system for three consecutive years

In case of energy restitution mode (b), the transitory period occurs in November and the output is higher for the Model 1 in comparison with Model 2. The normal operating mode, when the energy output remains equalised, occurs only in December and January for the Model 1. It can be extended withal to February in case of the Model 2, since the output differences are less expressed. The end of mode (b) occurs in March and is characterised by low energy output.

A better look at the Model 1 reveals that the high energy output in November may result in zero energy restitution in March, where the building still requires some heat. Somehow different type of situation is observed for the Model 2. Since the modelling encounters the transient character of the plant, the inertia of its dynamic behaviour influences the energetic output. The restitution of overall energy is shifted from November to March providing still the heat. Therefore, the inertia effect plays an important role in energy cycling and should not be discarded when developing the energy management policy.

The number of heating degree-hours below the reference temperatures remains reasonable for both modelling approaches (Table 3). Additionally some values for the utilisation of direct solar heat for the space heating and hot water production are listed in the table 3.

Table 3. Comparison of number of heating degree-hours under the reference temperatures

Model	heating degree-hours below 18°C $UNDH_{rez}$	heating degree-hours bellow 20°C $UNDH_{floor}$	Mean direct solar heating per year, kWh	Mean auxiliary heat input for hot water production per year, kWh	Mean direct solar input for hot water production per year, kWh
#1	799,80	4,44	1692,22	615,70	2620,95
#2	791,20	5,32	1658,67	579,59	2708,03

In this context it appears useful to analyse the utilisation of direct solar heating, which is shown on the figure 3. The transitory periods of the heat restitution mode (b), which occur in November, can fully meet the space heat demand of the building simply by providing the accumulated energy. Alternatively, the direct solar heating may provide the auxiliary energy for this period. In the normal operating mode between January and February the utilisation of direct solar heating is not sufficient and the major part of the heat space demand is covered by the accumulated energy.

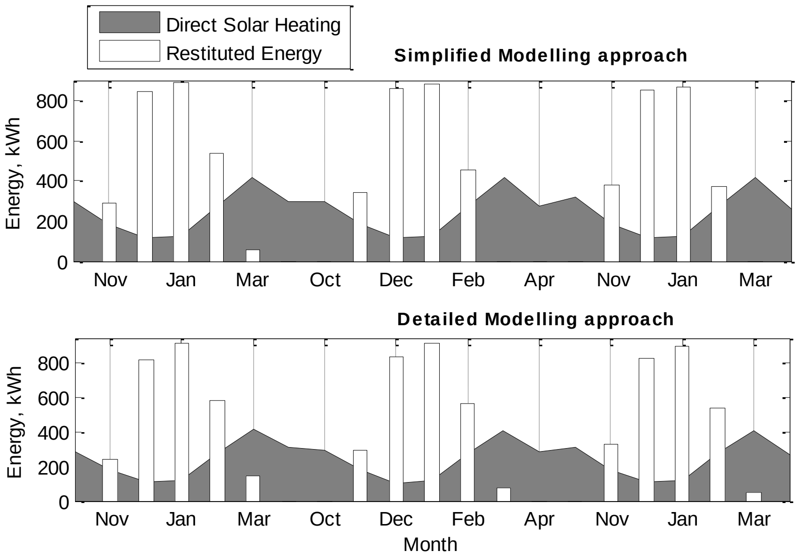

Figure 3. Potential of direct solar heating utilisation in heat restitution mode

However, in March when the energy output of the plant is low, the utilisation of direct solar heating may come across of heat demands. The potential of utilisation of direct solar heating reaches about 400 kWh for the both simulated models. Thus, at the end of heat restitution mode, when the delivered output is at low energetic level, the utilisation of direct solar heating becomes advantageous.

5. Conclusions

Therefore modelling of a thermal heat storage apparatus stands as a part of overall designing procedure, but it is not a main objective of the process. Moreover, modelling has not to be regarded as unique representation of future apparatus and different types of models have to be developed and investigated in order to provide the information about constraints that can be encountered within real implementation.

Modelling of a heat storage apparatus from distinct approaches exhibits a range of advantages, i.e. it allows at the same time to draw the domain of physical constraints and to embrace the interval where the true parameters are situated with respect to the real operating conditions. The proposed models exhibit similar behaviour of the studied system configuration, meaning that the basic assumptions are meaningful. Surely, the calculated parameters are valid only for the studied configuration, which uses the predefined energy management policy for the space heating and domestic hot water production. Notwithstanding, the calculus have been performed on the grounds of overall heat consumption of the studied dwelling, thus the calculated convergence intervals may point to the range of true parameters, depending not on the technical solution or energy management policy.

Additionally, the utilisation of direct solar heating has been analysed. Some advantageous conclusions, related to the calculated parameters can be drawn out. Firstly, the direct solar heating may bring supplementary energy input when the storage capacities are not enough to come across the space heating demands. Secondly, it allows decreasing the volume of working material.

The presented modelling and optimisation techniques do not take into account technological and economical aspects and the future work will dedicated to the integration of those aspects into the entire design procedure of the heat storage apparatus.

6. References

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7. Acknowledgements

This work was realized within the Solautark Project (led by: ESE) financed by the Government of Wallonia (Belgium) within its politic of Competitiveness Poles.