Controlled thermal mass for cooling of buildings

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Introduction

The future heating and cooling systems of buildings will have to focus on utilization of natural energy sources, for instance cold ambient air or soil. Ambient air is sufficiently cold only during nighttime in climate conditions of middle Europe. Therefore, a suitable thermal storage is needed for daily utilization of colder night air. Pre-cooling of building mass through enhanced night ventilation is a standard way. The substantial disadvantages are limited controllability and slowness of the charge/discharge process as surface heat transfer relies on natural convection.

This disadvantage could be possibly eliminated by controlled use of thermal mass – actively driven air-to-mass heat exchanger/storage (see Figure 1). Its principle is based on the alternation of the forced mass warming (extraction of daily heat gains) and forced mass cooling (equalizing the energy balance of the exchanger). A better control of charge/discharge process is expected due to the forced air flow through exchanger. Moreover, the exchanger can be suitably integrated into building components (e.g. wall, floor, ceiling panel [1]) and it does not have to lead to increase in cost of building.

The dependency of cooling power on the temperature during previous night is the disadvantage. The air temperatures noticed during some recent years (e.g. hot spells in year 2003 and 2006) showed that night temperatures need not to decrease below 20 C considerably, primarily in urban areas. Therefore, the cooling techniques based on night temperature utilization should not be over expected. Moreover, operation relies on expensive electricity, which is recently dominantly converted from non-renewable energy sources with all negative environmental consequences. Therefore, the optimization of relationship power vs. consumption will be a very important issue. The technical development of the device is multi-criteria and nontrivial task.

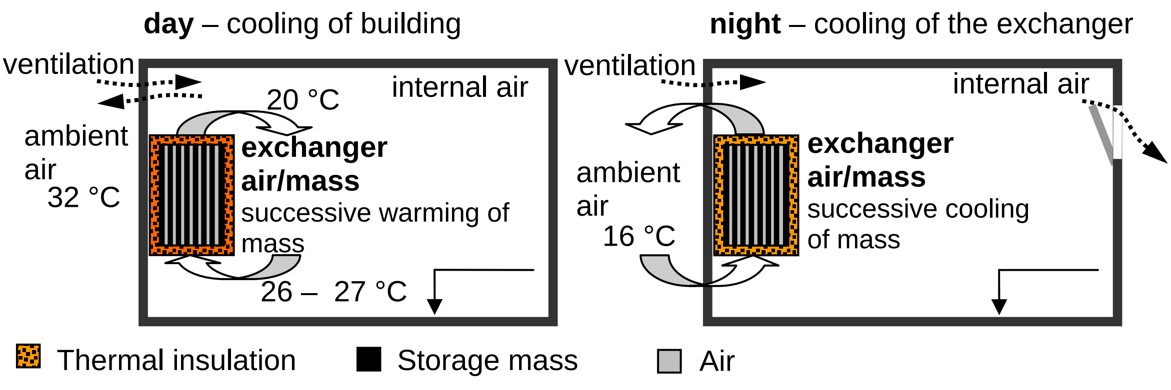

Figure 1: Circulation mode of the air-to-mass exchanger – basic drawing

Model of the air-to-mass heat exchanger

A lot of models for so called packed beds can be found in literature. The goal is to introduce a procedure how to implement a simple ODE based model of air-to-mass heat exchanger in Simulink.

Problem description:

The considered system is air-to-mass heat exchanger with total length L and external adiabatic boundary (Figure 2). The part of the basic block with cross-sectional area A and volume Adx is somehow filled by storage mass with volume V_s and heat exchange surface A_s.

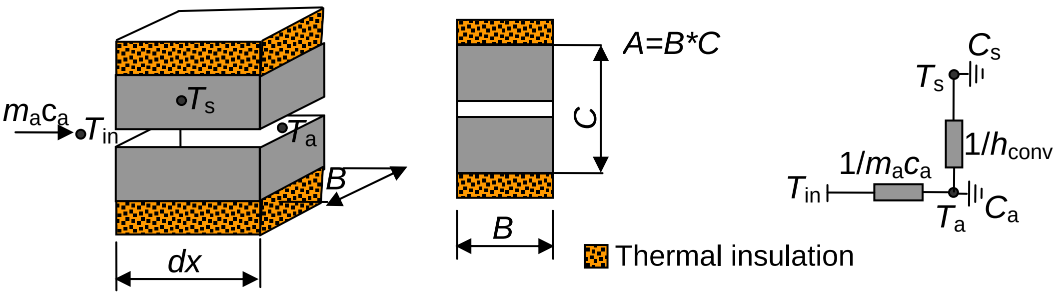

Figure 2: Layout of air-to-mass heat exchanger segment and analogy representing its simplified model

Assumptions:

a) Mass in the segment is substituted by one temperature node T_s with thermal capacity C_s. This simplification will only be possible, if temperature gradient between mass surface and inside is negligible („sufficiently thin mass“)

b) Conductive heat flow parallel with longitudinal axis is negligible.

c) Mass air flow rate through exchanger m_a [kg/s] is equally distributed within cross-sectional surface and leads to internal flow velocity v_a [m/s] and free velocity v_{a0}:

$$v_a = \frac{m_a}{\eta A \rho_a} \quad (1)$$

$$v_{a0} = \frac{m_a}{A \rho_a} \quad (2)$$

where:

η is	ratio of air cross-sectional area to total cross-sectional area [-]
A	total cross-sectional area [m^2]
ρ_a	air density [kg/m^3]

Governing equations:

- thermal balance of node T_a:

$$C_a \frac{dT_a}{dt} = m_a c_a (T_{in} - T_a) - A_s h_{conv} (T_a - T_s) \qquad \text{[W]} \quad (3)$$

- thermal balance of node T_s:

$$C_s \frac{dT_s}{dt} = A_s h_{conv} (T_a - T_s) \qquad \text{[W]} \quad (4)$$

where:

T_a is	air temperature [°C]
C_a	thermal capacity of air node [J/K]
T_s	mass temperature [°C]
C_s	thermal capacity of mass node [J/K]
T_{in}	inlet air temperature [°C]
h_{conv}	convective heat transfer coefficient between air and mass [W/(m^2.K)]
t	time [s]

Capacities C_a and C_s are defined as:

$$C_a = \rho_a c_a \eta A dx \qquad [J/K] \qquad (5)$$

$$C_s = \rho_s c_s (1-\eta) A dx \qquad [J/K] \qquad (6)$$

where:

c_a is	specific thermal capacity of air [J/(kg.K)]
ρ_s	storage material density [kg/m^3]
c_s	specific thermal capacity of storage material [J/(kg.K)]

Moreover, notice the following:

a) General arrangement of mass in the exchanger can be dealt with.

b) Partial (and important) problems as calculation of convective heat transfer coefficient h_{conv}, calculation of heat exchange surface A_s, e.g. for boxes filled by gravel or ball-shaped particles, are not described.

c) Thermal conductivity of heat storage material λ_s is not the input parameter into model. This is the result of model assumption of negligible temperature gradient between mass surface and inside. Validity of the assumption can be verified by fulfillment of the following condition, see [2]:

$$Bi = \frac{L_c}{d} \leq 0.2 \qquad [-] \qquad (7)$$

where:

Bi is	Biot number [-]
L_c	characteristic length [m], which is approximated as:

$$L_c = 2\frac{V_s}{A_s} \qquad [m] \qquad (8)$$

d is ratio:

$$d = \frac{\lambda_s}{h_{conv}} \qquad [m] \qquad (9)$$

Condition (7) will not be always perfectly fulfilled, affecting the accuracy of calculation.

Model representation in Simulink:

Graphical representation of governing equations in Simulink is depicted in Figure 3.

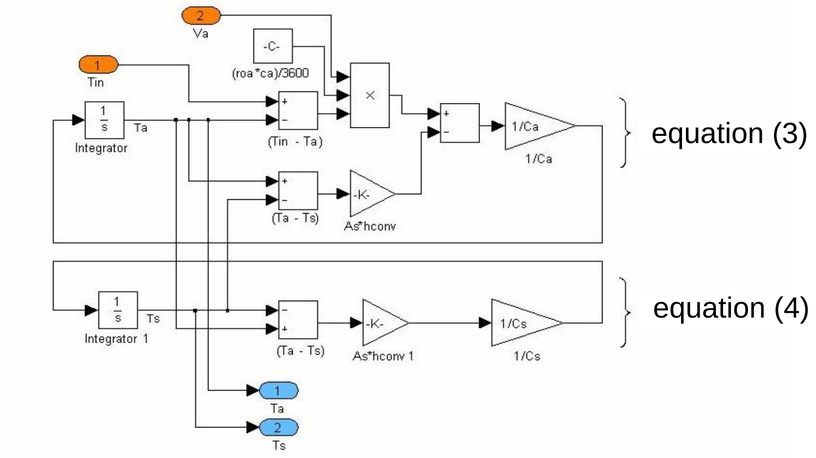

Figure 3: Representation of governing equations in Simulink, subsystem representing one segment of the exchanger according to Figure 2

Analytical validation:

The validation is important, since it enables to remove basic programming errors. The analytical solution published in [3] was used for validation of the developed simplified model. The following case were simulated, see Table 1.

Table 1: Exchanger configurations used for model validation

specification	L [m]	B [m]	C [m]	η [-]	A_s [m^2]	V_a [m^3/h]	v_{a0} [m/s]	v_a [m/s]	h_{conv} [W/m^2.K]
v1	3	0.25	0.25	0.44	5.25	100	0.44	1.00	10
v2	12	0.25	0.25	0.44	21.0	100	0.44	1.00	10
v3	3	0.25	0.25	0.16	15.75	100	0.44	2.78	10
v4	12	0.25	0.25	0.16	63.0	100	0.44	2.78	10
v5	3	0.25	0.25	0.16	15.75	33.3	0.15	0.93	10
thermo physical properties: λ_s = 1,5 W/(m.K), ρ_s = 2500 kg/m^3, c_s = 1000 J/(kg.K); exchanger of total length L was split into 30 longitudinal segments									

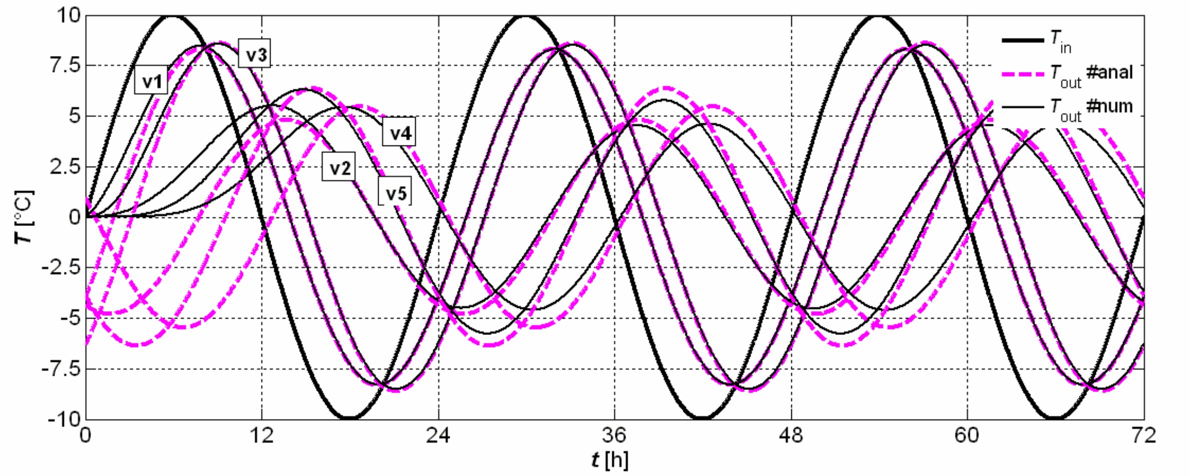

Figure 4: Comparison of numerical solution with analytical solution (#anal – analytical results, #num – numerical results)

A reasonably good agreement of the developed model with the analytical solution has been achieved (Figure 4). However, the performance of real exchangers will be influenced by other effects, which were not bargained for neither in the model formulation nor in the analytical solution (e.g. longitudinal heat conduction, imperfection of adiabatic boundary and particularly non-uniformity of air flow rate within cross section).

Case study – a building with controlled thermal mass for cooling

The goal of this case study is to integrate the thermal model of air circulation through air-to-mass heat exchanger into building thermal model [4] (HAMbase).

Building description:

Building geometry (Figure 5) is identical with test case [5]. However, window surface was assumed different, more corresponding to reasonable values (approximately 40 %). External walls and roof consist of 2 cm plasterboard and 25 cm mineral wool. Floor consists of 2.5 cm wooden board, 25 cm mineral wool, 20 cm concrete slab and 100 cm soil with isothermal boundary condition 10°C on the bottom side. No other internal constructions are placed inside the zone. Thermal transmittance of the window is considered to be 1.0 W/m^2.K. Solar gain factor of the window is assumed to be 0.5 for unshaded window and 0.05 for shaded window.

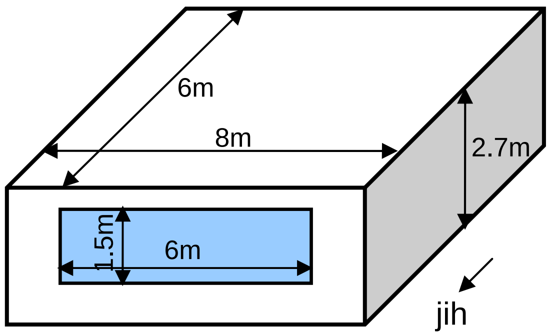

Figure 5: Simulated building

Model description:

External building components are replaced by thin surfaces having dimensions according to Figure 5. Air volume is calculated from external dimensions (8 x 6 x 2.7 m). It is supposed 150 W of internal heat gains, 60 % radiative, 40 % convective. Heat losses due to air circulation through the air-to-mass heat exchanger are considered to be 100 % convective. Absorptivity of the external surfaces is considered to be 0.6, emisivity for long wave radiation is considered 0.9. For internal surfaces mean radiative heat transfer coefficient is considered to be 5 W/(m^2.K) and convective coefficient is considered to be 2.7 W/(m^2.K). Measured data from weather station Prague Karlov in year 2006 was used as climate data into simulation. The term of twenty two days was simulated; the simulated period started on 1st July 2006.

Scenarios:

- „var0“ – basic alternative, without shading, air change rate 0.3 h^{-1} constant during the whole day;
- „var1“ – var0 with shading, which is activated if the solar irradiance on the window exceeds 100 W/m^2 and if the internal temperature exceeds 20 °C;

- „var2" – var1 with night ventilation (2,5 h^{-1}) which is activated between 22:00 and 7:00, this ventilation does not occur if internal air temperature is lower than 20 °C (in this case there remains 0,3 h^{-1});
- „var3" – var2, plasterboard of walls and roof is replaced by 15 cm thick monolithic concrete;
- „varPB" – var2 with circulation cooling. Standard concrete blocks form the air-to-mass heat exchanger (see Figure 6). There is assumed a parallel connection of four sections, which will lead to air change rate 2 h^{-1} (4 x 65 m^3/h). For exchanger properties see Table 2 and Table 3. The surface area of building components was not adapted so that the exchanger is assumed as an additional component, which does not replace the part of walls. A control system activates air circulation between building and the heat exchanger between 07:00 – 22:00. Thus, the night discharge of the exchanger occurs for 8 hours only, but with doubled air flow rate (4 x 130 m^3/h).

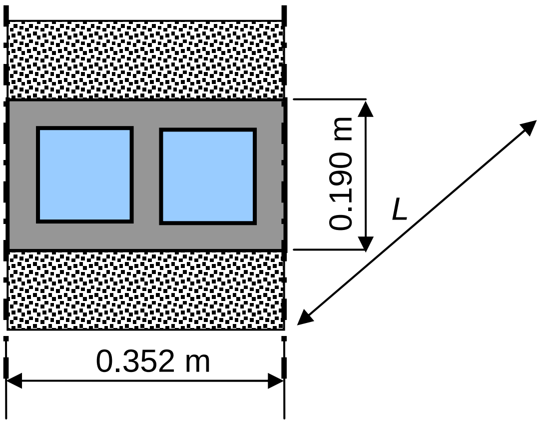

Figure 6: The segment of the air-to-mass heat exchanger

Table 2: Basic information

L [m]	B [m]	C [m]	η [-]	A_s [m^2]	weight [kg]
5.7	0.35	0.19	0.54	6.12	395
thermo physical properties of storage material: λ_s = 1,33 W/(m.K), ρ_s = 2250 kg/m^3, c_s = 1020 J/(kg.K)					

Table 3: Air flow rate and related properties

V_a [m^3/h]	v_{a0} [m/s]	v_a [m/s]	V_a/A [m^3/h/m^2]	$V_a/(\eta A)$ [m^3/h/m^2]	V_a/A_s [m^3/h/m^2]	h_{conv} * [W/m^2.K]
65	0.27	0.50	977	1805	319	~3
130	0.54	1.00	1955	3610	637	~6
*calculated from $Nu = 0.023Re^{0.8}Pr^{0.3}$ [6]						

<u>Model representation in Simulink:</u>

The explanation of the shaded areas in Figure 7: area 1 is building simulation (HAMbase), area 2 is the simulation of the air-to-mass heat exchanger (the developed model), area 3 is a simple control system (see the description of alternative varPB) and area 4 is the calculation of cooling power.

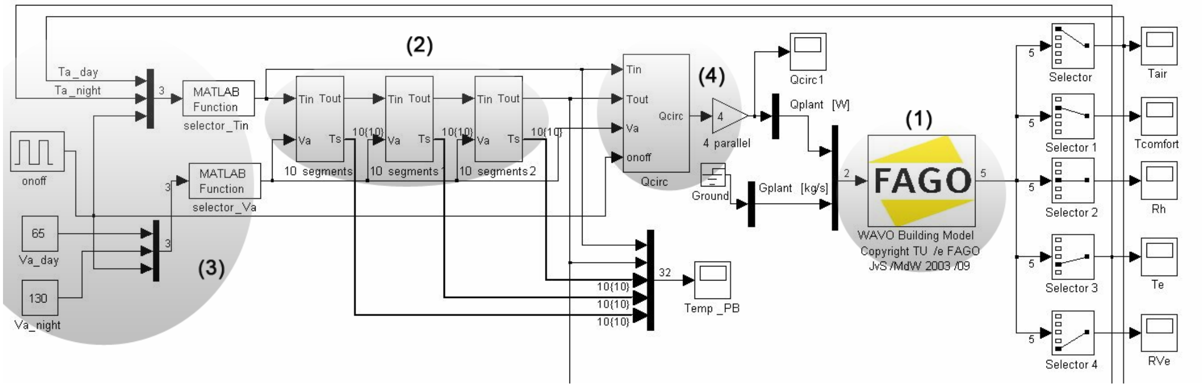

Figure 7: Modeled system in Simulink

<u>Simulation results:</u>

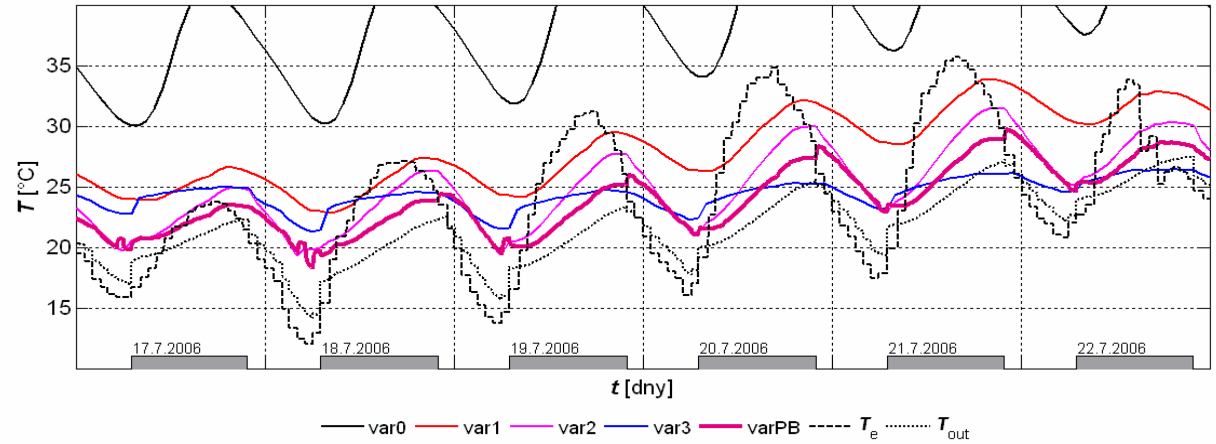

Figure 8: Calculated air temperature in the zone for each scenario (var0 – varPB), T_e is ambient air temperature, T_{out} is outlet air temperature from the air-to-mass heat exchanger

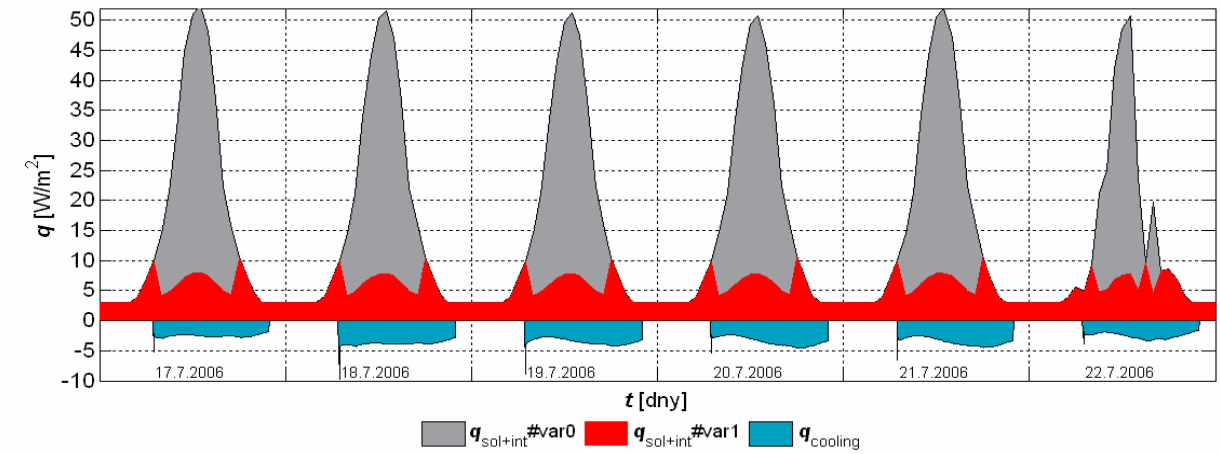

Figure 9: Calculated thermal load from solar and internal heat sources $q_{sol+int}$ for alternatives var0 and var1 and cooling power of the air-to-mass heat exchanger $q_{cooling}$. The values are related to 1m^2 of floor area (8 m x 6 m = 48 m^2).

Figure 9 emphasizes the role of shading for reduction of solar heat gains.

Conclusions

The decrease in the peak heat load and thus the reduction of the cooling system size, especially for lightweight buildings with higher internal heat gains, might be the main benefit from controlled thermal mass. Excessive building heat load should be of course primarily reduced by building concept and reduction of internal heat gains.

A logical follow-up action should be a parametric study of the thermal performance of the air-to-mass heat exchangers (maximization of energy effectiveness). There are many possibilities how to increase heat exchange surface and convective heat transfer coefficient. However, it generally also implies the higher pressure drop. Where is the optimum? The theoretical study could lead to design of some prototypes and verification in a laboratory facility.

Use of phase change materials could substantially decrease the amount of storage mass. For instance, a cylindrical exchanger composed of small spheres filled with PCM was studied in [7]. The developed numerical model was validated with measured laboratory data. The follow up case study of a low-energy building demonstrated the use of this technique. The necessary amount of the PCM material was 6 kg/m^2 of floor area (approximately one sixth of mass used in the case study of building in Figure 5).

References

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