OPTIMIZATION OF FREE COOLING BASED ON WEATHER PREDICTION

Klemen Dovrtel, Sašo Medved

Faculty of Mechanical Engineering, University of Ljubljana, Slovenia

1. Introduction

In current practice of building service systems control, the system operation depends solely on the instantaneous parameters of the outdoor and indoor conditions. Such operation decreases the efficiency of the systems and for example in case of free cooling system, decreases the utilization factor of natural sources. In the paper developed controlling strategy of free cooling systems is presented. The strategy is based on weather prediction of energy potential of air temperature of following days.

A schematic diagram of a simple free cooling system with heat storage is shown in Figure 2. The system consists of heat storage, air ducts and a fan with variable flow control. The system takes the outdoor air from the environment and supplies it through the ducts. The inlet air flow is divided in two streams: one flows into the building, other into the heat storage. During the summer time the system operates in two regimes: night-time and daytime. In the night-time regime (Figure 1 – solid line) the system intakes the outdoor air and divides it into two streams: one for free cooling of the building and the other for cooling the heat storage. After cooling the heat storage, this stream is drained back to the environment. In the daytime regime total amount of fresh air flows through heat storage and as pre-cooled air enters into the building (Figure 1 – dashed line).

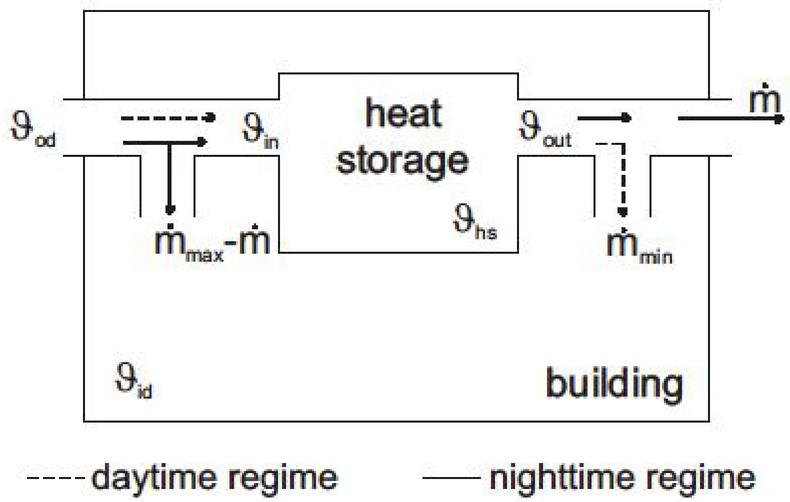

Fig. 1: Scheme of free cooling system

The whether forecast parameters are a key element of weather predicted system operation. Weather data influence on free cooling system operation and thermal response of building – therefore, the available energy potential for system utilization and also the needed energy for proper system operation are variables affected by external weather parameters. In case of free cooling system, the available energy for system utilization and needed energy for system operation is function of outdoor air temperatures and indoor air temperatures.

2. Outdoor and indoor air temperature

The outdoor and indoor air temperatures can be predicted using weather forecast. Weather forecast data can be obtained from weather information centers, in our case from the Environmental Agency of the Republic of Slovenia. The weather forecast is based on ALADIN/SI model. The model simulates the weather in a small bounded area using boundary and initial conditions from the IFS/APRAGE global weather model. The model calculates hourly values of temperature, rainfall, solar radiation, relative humidity, wind speed, wind direction and cloudiness.

The outdoor and indoor air temperature profile was approximated analytically. The weather forecast discreet hurly values of outdoor air temperature were approximated using discreet Fourier transform and composition of temperature (signal) using Fourier series. The outdoor air temperature can be written as:

$$\vartheta_{od} = A_{0,\vartheta od} + 2\sum_{i=1}^{n_{od}}\left[A_{i,\vartheta od}\cos(i\omega t) + B_{i,\vartheta od}\sin(i\omega t)\right] \qquad \text{(eq. 1)}$$

The same way, using discreet Fourier transform, the indoor air temperature profile was approximated. The prediction of indoor air temperatures can be calculated using Trnsys simulation of uncontrolled building thermal response to predicted weather forecast data. The indoor air temperature can be therefore approximated the same way, by composition of temperature (signal) using Fourier series. The indoor air temperature can thus be written as:

$$\vartheta_{id} = A_{0,\vartheta id} + 2\sum_{i=1}^{n_{id}}\left[A_{i,\vartheta id}\cos(i\omega t) + B_{i,\vartheta id}\sin(i\omega t)\right] \qquad \text{(eq. 2)}$$

Usually the higher frequencies of composed temperature (signal) can be neglected (depend on weather conditions). The composition of outdoor temperature is usually sufficiently accurate if we use the fundamental frequency and the second harmonic frequency only (n_{od}=2). Because of temperature amplitude damping in building construction, the composition of indoor temperatures can be done accurate enough using only the fundamental frequency (n_{id}=1).

3. The free cooling system model

The free cooling system consists of three main elements, the ventilator, the inlet flap and the heat storage. A ventilator with variable flow rate can be controlled in the range of 0 to m_{max}. The inlet duct flap enables the variable flow control into the building (in range from 0 to m_{max}) or into the heat storage (in range from m_{max} to 0). The most important element of free cooling ventilation system is a thermal storage. Sensible heat storage with thermal capacity C was selected. Applying the first law of thermodynamics yields:

$$C\frac{d\vartheta_{hs}}{dt} + UA_{hs,out}\left(\vartheta_{hs} - \vartheta_{amb}\right) = \dot{m}c_p\left(\vartheta_{in} - \vartheta_{out}\right) \qquad \text{(eq. 3)}$$

In case of low temperature gradient inside heat storage, one node heat storage model can be assumed. The heat storage outlet temperature can thus be simply calculated:

$$\vartheta_{out} = \vartheta_{hs}\left(1 - e^{-NTU}\right) + \vartheta_{in}e^{-NTU} \qquad \text{(eq. 4)}$$

the eq. 3 can thus be simplified to:

$$C\frac{d\vartheta_{hs}}{dt} + UA_{hs,out}\left(\vartheta_{hs} - \vartheta_{amb}\right) = \dot{m}c_p\left(\vartheta_{in} - \vartheta_{hs}\right)\left(1 - e^{-NTU}\right) \qquad \text{(eq. 5)}$$

$$\text{where: } NTU = hpL/\dot{m}c_p \text{ and } h = \frac{Nu\,k_f}{d} = \frac{(0{,}023\,Re^{4/5}\,Pr^{1/3})k_f}{d}$$

Assuming that heat storage ambient temperature ϑ_{amb} in the eq. 5 equals to building indoor air temperature ϑ_{id} (eq. 2), that the heat storage inlet air temperature ϑ_{in} equals to outdoor air temperature ϑ_{od} (eq. 1), and rearranging yields:

$$\frac{d\vartheta_{hs}}{dt} + \left(k_1 + k_2\right)\vartheta_{hs} = k_1\vartheta_{od} + k_2\vartheta_{id} \qquad \text{(eq. 6)}$$

where: $k_1 = \frac{\dot{m} c_p}{C}\left(1-e^{-NTU}\right)$ and $k_2 = \frac{UA_{hs,out}}{C}$

Combining eq. 6, eq. 1 and eq. 2 thus yields:

$$\frac{d\vartheta_{hs}}{dt} + \left(k_1 + k_2\right)\vartheta_{hs} = k_1\left(A_{0,\vartheta od} + 2\sum_{i=1}^{n_{od}}\left[A_{i,\vartheta od}\cos\left(i\omega t\right) + B_{i,\vartheta od}\sin\left(i\omega t\right)\right]\right) + k_2\left(A_{0,\vartheta id} + 2\sum_{i=1}^{n_{id}}\left[A_{i,\vartheta id}\cos\left(i\omega t\right) + B_{i,\vartheta id}\sin\left(i\omega t\right)\right]\right) \qquad \text{(eq. 7)}$$

Solving the above differential equation for the initial condition yields the heat storage temperature:

$$\begin{aligned}\vartheta_{hs} = {} & e^{-(k_1+k_2)t}\,Const + \\ & \left[\frac{k_1}{k_1+k_2}A_{0,\vartheta od} + 2\sum_{i=1}^{n_{od}}\left[\frac{k_1}{\left(k_1+k_2\right)^2 + \left(i\omega\right)^2}\begin{pmatrix}\left(\left(k_1+k_2\right)A_{i,\vartheta od} - i\omega B_{i,\vartheta od}\right)\cos\left(i\omega t\right) + \\ \left(\left(k_1+k_2\right)B_{i,\vartheta od} + i\omega A_{i,\vartheta od}\right)\sin\left(i\omega t\right)\end{pmatrix}\right]\right] + \\ & \left[\frac{k_2}{k_1+k_2}A_{0,\vartheta id} + 2\sum_{i=1}^{n_{od}}\left[\frac{k_2}{\left(k_1+k_2\right)^2 + \left(i\omega\right)^2}\begin{pmatrix}\left(\left(k_1+k_2\right)A_{i,\vartheta id} - i\omega B_{i,\vartheta id}\right)\cos\left(i\omega t\right) + \\ \left(\left(k_1+k_2\right)B_{i,\vartheta id} + i\omega A_{i,\vartheta id}\right)\sin\left(i\omega t\right)\end{pmatrix}\right]\right]\end{aligned} \qquad \text{(eq. 8)}$$

where: $Const = funct\left(\vartheta_{hs}\left(t_0\right)\right)$

Combining the above equation with eq. 4 yield the heat storage outlet temperature:

$$\vartheta_{out} = \left[\begin{matrix} e^{-(k_1+k_2)t}\,Const + \\ \left[\frac{k_1}{k_1+k_2}A_{0,\vartheta od} + 2\sum_{i=1}^{n_{od}}\left[\frac{k_1}{\left(k_1+k_2\right)^2 + \left(i\omega\right)^2}\begin{pmatrix}\left(\left(k_1+k_2\right)A_{i,\vartheta od} - i\omega B_{i,\vartheta od}\right)\cos\left(i\omega t\right) + \\ \left(\left(k_1+k_2\right)B_{i,\vartheta od} + i\omega A_{i,\vartheta od}\right)\sin\left(i\omega t\right)\end{pmatrix}\right]\right] + \\ \left[\frac{k_2}{k_1+k_2}A_{0,\vartheta id} + 2\sum_{i=1}^{n_{od}}\left[\frac{k_2}{\left(k_1+k_2\right)^2 + \left(i\omega\right)^2}\begin{pmatrix}\left(\left(k_1+k_2\right)A_{i,\vartheta id} - i\omega B_{i,\vartheta id}\right)\cos\left(i\omega t\right) + \\ \left(\left(k_1+k_2\right)B_{i,\vartheta id} + i\omega A_{i,\vartheta id}\right)\sin\left(i\omega t\right)\end{pmatrix}\right]\right]\end{matrix}\right]\left(1-e^{-NTU}\right) + \vartheta_{in}e^{-NTU}$$

(eq. 9)

4. Criteria and regime of operation

There are more criteria that can be used to determine the optimal operation of free cooling system; for instance, the cost of system operation, minimum energy use for auxiliary cooling system, maximal decrease of auxiliary system cooling power or combinations of multiple criteria are possible. In our case, the comparison of system operation based on two different criteria was done: 1st, minimal energy use for additional mechanical cooling and 2nd, minimum ventilator energy use.

The regime of operation a free cooling system is also bounded to the required building thermal comfort level. Both indoor thermal comfort and indoor air quality requirement criteria have to be met. Therefore the system should provide the minimum amount of ventilation fresh air to sustain suitable indoor air quality standard, and additionally, the inlet fresh air temperature should not exceed certain boundary temperature, as required by the comfort criterion.

The system operation time is divided into two time sections: the night-time operation time, when the heat storage is charging with cold, and the day-time operation time, when heat from thermal storage is released to pre cool the fresh air.

Day-time operation

Day-time system operation is based on the boundary condition – the minimum required amount of fresh air of temperatures not exceeding certain inlet temperatures. The day-time regime of operation should therefore be such, that the fresh air is pre cooled in heat (cold) storage when outdoor temperature exceeds the selected boundary temperature. When free cooling system operates optimally, the maximum outlet temperature reaches the selected boundary temperature. Those can be written mathematically:

$$\vartheta_{out}\Big|_{(t,\dot{m})=(t_{max},\dot{m}_{min})} = \vartheta_{bond} \qquad \text{(eq. 10)}$$

$$\frac{d\vartheta_{out}}{dt}\Bigg|_{(t,\dot{m})=(t_{max},\dot{m}_{min})} = 0 \qquad \text{(eq. 11)}$$

Based on the conditions written in eq. 10 and eq. 11, the needed heat storage temperature for day-time operation can be calculated.

Night-time operation

In the night-time regime of operation, the heat storage is cooled down from some initial temperature of heat storage, left from previous day, to the needed heat storage temperature for day-time operation. The needed heat storage temperature for day-time operation is thus the condition of nighttime regime of operation. The way the heat storage is cooled down depends on criteria of night-time operation:

1st: minimal energy use for additional mechanical

The night ventilation system supplies into the building as much cold as possible. The amount of cooling energy carried by the air can be maximized. This can be written as:

$$Q = \int_{t_{ns}}^{t_{ne}} (\dot{m}_{max} - \dot{m})\, c_p \left(\vartheta_{id} - \vartheta_{od}\right) dt \qquad \text{(eq. 12)}$$

2^{nd}: minimum ventilator energy use

The ventilator for free cooling system consumes electricity, which is proportional to the amount of air supplied. To minimize the electricity consumption and the expenses, the amount of air to cool down the heat storage should me minimized. This can be written as:

$$m = \int_{t_{ns}}^{t_{ne}} \dot{m}\,dt \qquad \text{(eq. 13)}$$

Since the eq. 8 and eq. 9 are valid only when $\dot{m}$ is constant, the solution to the eq. 12 and 13 can be found in such a way, that the night-time operation duration interval is divided to n equal partial time intervals $\Delta t=(t_{ns}-t_{ne})/n$ of constant flow. The equation 12 can thus be rewritten:

$$Q = \sum_{i=1}^{n}\left[\int_{t_{ns}+(i-1)\Delta t}^{t_{ns}+i\Delta t} \left[(\dot{m}_{max} - \dot{m}_i)\, c_p \left(\vartheta_{id} - \vartheta_{od}\right)\right] dt\right] \qquad \text{(eq. 14)}$$

and in a same way equation 13:

$$m = \sum_{i=1}^{n}\left[\dot{m}_i\,\Delta t\right] \qquad \text{(eq. 15)}$$

The night-time operation condition is the same for both criterion functions. Using the same principle of dividing night-time operation interval to smaller peaces, the night-time operation condition can be written for n intervals and expressed in recursive form:

$$\begin{aligned} &\text{def}: F(\vartheta, i) = \vartheta_{hs}\left(F(\vartheta, i-1), \Delta t, \dot{m}_i\right); F(\vartheta, 0) = \vartheta \\ &\vartheta_{hs,ne} = F(\vartheta_{hs,ns}, n) \end{aligned} \quad \text{(eq. 16)}$$

The objective function (solution of eq. 13) is linear function of n variables with one nonlinear equality constraint (equation 13). The extreme of the above equation with regards to written constrains was found with Mathematica software using the Interior Point method.

5. Example of system operation

The advantage of weather-predicted operation of a free cooling system was demonstrated on small building test cell of volume 100 m^3. To achieve the required indoor air quality it is assumed that 0.5 air exchange per hour (60 kg/h) should be supplied. The maximum fresh air flow rate supplied by system is limited to $\dot{m}_{max} = 5\dot{m}_{min}$.

The heat storage thermal capacity was selected based on the free cooling potential and available air flow rate through the heat storage. In our case the reference heat storage capacity of 160 kJ/K was assumed. The $UA_{hs,out}$ value of insulation of heat storage was 1,6 W/K. The night-time operation was divided into five equal time sections (n=5) of equal duration. The results of optimization are shown on Figure 2 and Figure 3.

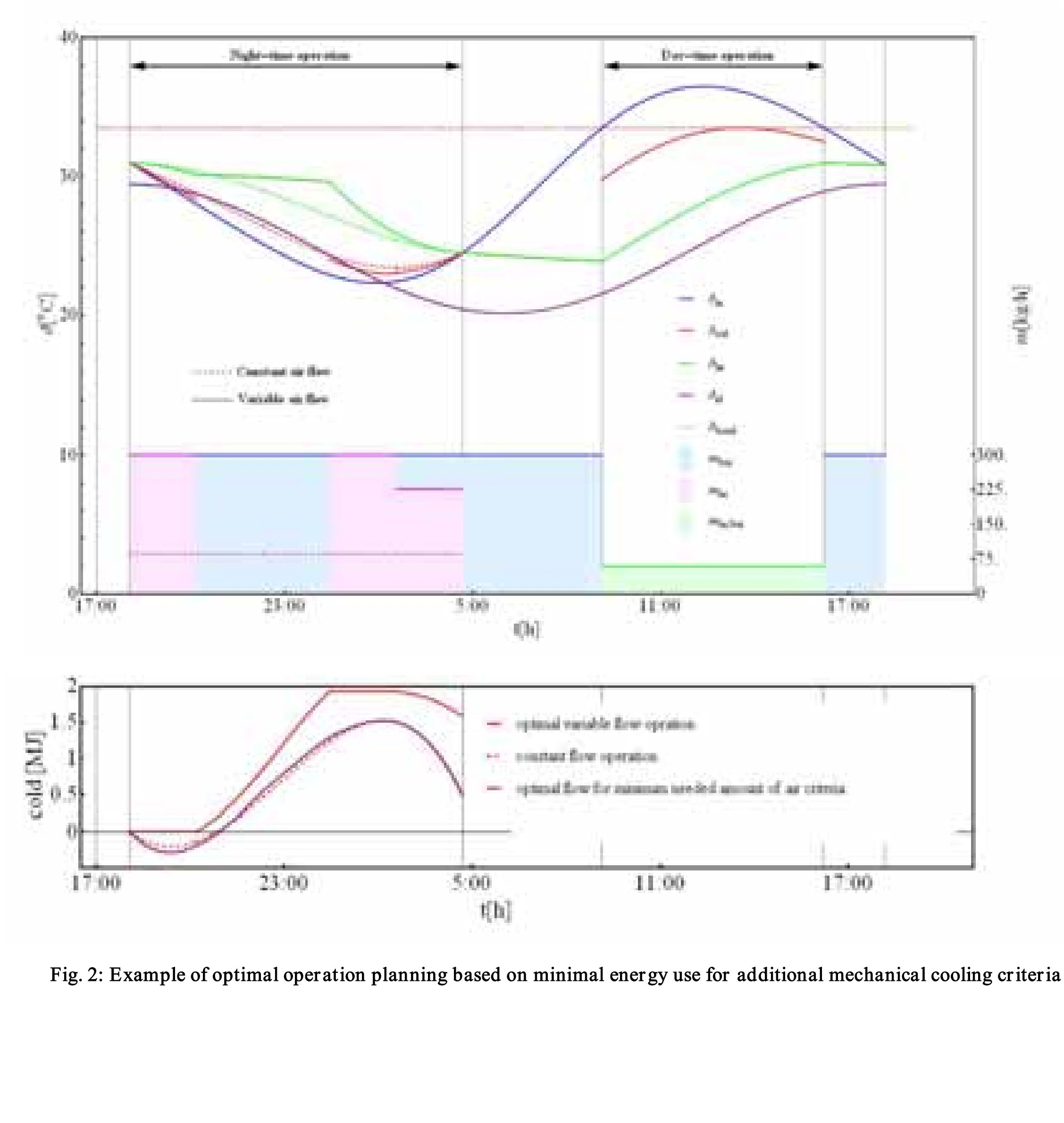

Fig. 2: Example of optimal operation planning based on minimal energy use for additional mechanical cooling criteria

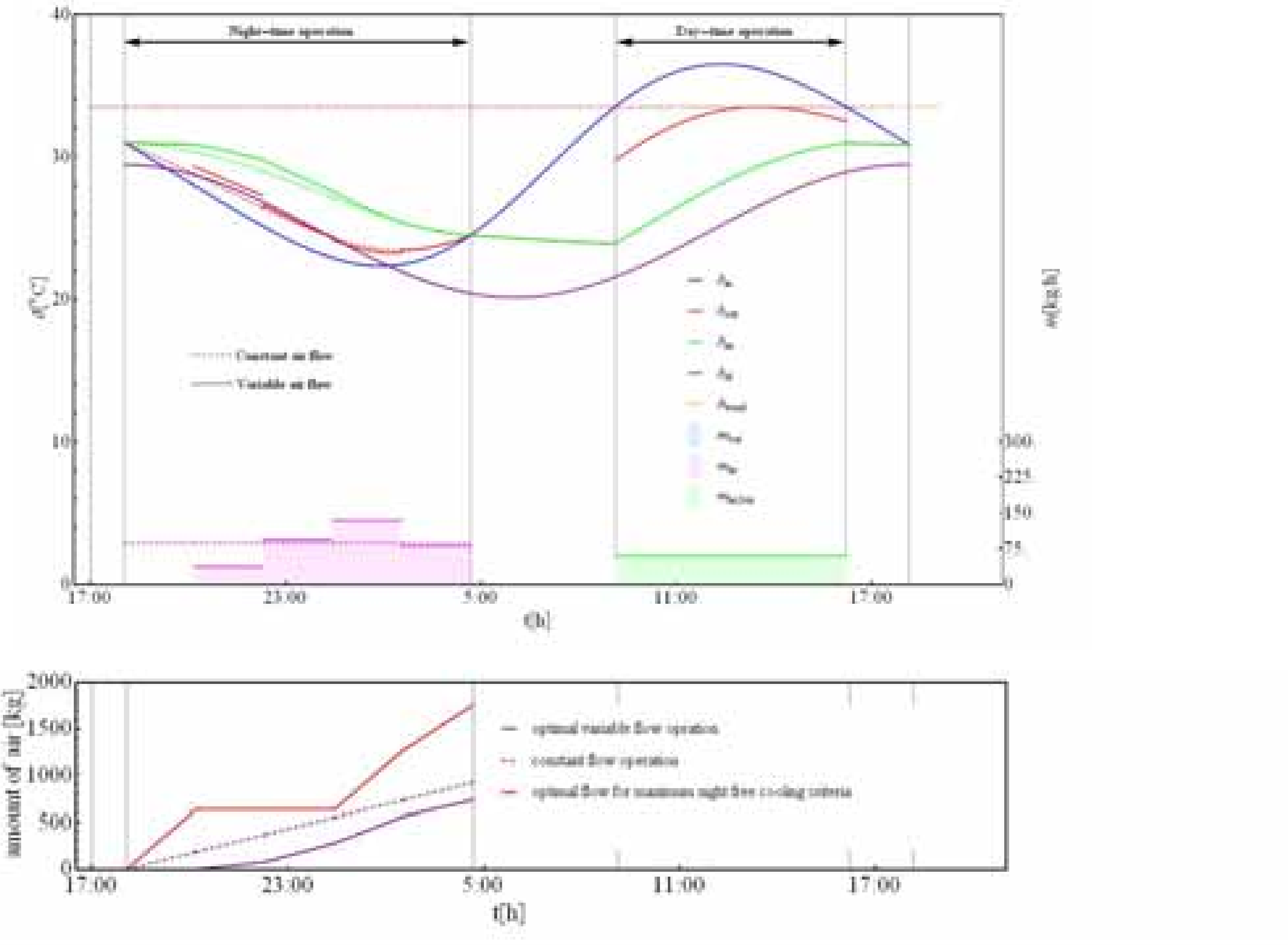

Fig. 3: Example of optimal operation planning based on minimum ventilator energy use criteria

It can bee seen that in case of minimal energy use for additional mechanical criteria, the difference between optimal system operation and constant system operation is 3 to 1 and in case of minimum ventilator energy use 1,25 to 1. It can be seen that the differences based on the operation regime are quite big. The weather predicted operation therefore present great potential for system improvements.

6. Nomenclature

A	surface area, m2
A_i, B_i	Fourier series coefficients
C	heat capacity, J/K
t	time, s
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
cp	specific heat, J/kgK
NTU	number of transfer units, /
Nu	Nusselt number, /

Pr Prandtl number, /

p perimeter, m

L length, m

d diameter, m

U heat transfer coefficient, W/m2K

k thermal conductivity, W/mK

Greek symbols

ϑ temperature, °C

ω angular frequency, rd

Subscripts

id indoor

od outdoor

hs heat storage

ne night end

ns night start

de day end

ds day start

max maximum

f fluid

7. References

Dovrtel K, Medved S, 2011. Weather-predicted control of building free cooling system, Applied Energy, p. 3088-3096.

Vidrih B, Medved S, 2008. The effects of changes in the climate on the energy demands of buildings, International Journal of Energy Research, p. 1016-1029.

Medved S, Arkar C, 2008. Correlation between the local climate and the free-cooling potential of latent heat storage. Energy and Buildings, p. 429-437.

Yang L, Li Y, 2008. Cooling load reduction by using thermal mass and night ventilation. Energy and Buildings, p. 2052-2058.

Pfafferott J, Herkel S, Jäschke M, 2003. Design of passive cooling by night ventilation: evaluation of a parametric model and building simulation with measurements. Energy and Buildings, p. 1129-1143.

Shaviv E, Yezioro A, Capeluto IG, 2001. Thermal mass and night ventilation as passive cooling design strategy. Renewable Energy, p. 445-452.

Brown MJ, 1990. Optimization of Thermal Mass in Commercial Buildings Applications. Journal of Solar Energy Engineering, p. 273-279.

Santamouris M, Sfakianaki A, Pavlou K, 2010. On the efficiency of night ventilation techniques applied to residential buildings. Energy and Buildings, p. 1309-1313.

Henze GP, Felsmann C, Knabe G, 2004. Evaluation of optimal control for active and passive building thermal storage. International Journal of Thermal Sciences, p. 173-183.

Kummert M, André P, Nicolas J, 2001. Optimal heating control in a passive solar commercial building. Solar Energy, p. 103-116.

Gouda MM, Danaher S, Underwood CP, 2006. Quasi-adaptive fuzzy heating control of solar buildings. Building and Environment, p. 1881-1891.

Gwerder M, Lehmann B, Tödtli J, Dorer V, Renggli F, 2008. Control of thermally-activated building systems (TABS); Applied Energy, p. 565-581.

Dincer I, Rosen MA, 2002. Thermal energy storage systems and applications. John Wiley & Sons.