MODEL OF HEAT EHCHANGE OF PREMISES AND CONTROL OF HEATING SYSTEM OPERATION BY CHANGING HEAT CARRIER FLOW

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Abstract. The model of heat exchange of premises and the heating control system operation is developed using mathematical and imitation modelling methods. The modelling of premises heat transfer processes is carried out, the curves of heat system parameters are obtained by imitation of different changes of outside air temperature. The obtained regularities are useful to develop heating system control devices. By using worked out model it is possible to investigate heat exchange and changes of indoor air temperature depending on external factors.

Key words: heating, premises, modelling, heat transfer, control.

Introduction

Every building needs a heating system to provide the necessary level of comfort conditions inside premises and it have to comply with economical and ecological terms. One way to fulfil these terms is to ensure optimal operation of heating system by using automatic control systems, what provides operation with parameters close to required conditions without reference to variable external conditions. The predictive control circuit is used to overtake the impact of external perturbances to the object thereby providing constant indoor air temperature in comply with comfort conditions when outdoor temperature is changing. To develop such model, there is necessary to carry out heat balance for given object and to obtain mathematical correlation what characterizes the process of heat exchange.

Object of investigations and mathematical analyses of processes

The purpose of the heat system is to keep optimal indoor temperature in premises independently on the outside factors. If the heating system doesn't operate, indoor air temperature changes following outdoor temperature with time delay, what is determined by heat transfer inertia of building constructions. Inertia depends on material, construction mass, heat transfer characteristics and temperature transfer characteristics. Heat flow throw building constructions is determined by the following parameters: thickness of building construction layers and coefficients of its thermal conductivity as well as area of the surface and the coefficient of heat transfer. In the following calculations the coefficients of heat transfer are: for walls inside surfaces α_t = 8,7 W·m^{-2}·^{o}C^{-1}, outside surfaces $\alpha_{\bar{a}}$ = 23 W·m^{-2}·^{o}C^{-1} [1].

It is possible to simulate the impact of outside factors, the change of input and output parameters and operation of the system by using the computer program *Matlab* subprogram *Simulink*. This subprogram allows to construct blocks and connections between them that imitates the system operation on the computer and gives a possibility to investigate the operation of the system overall and the operation of separate blocks during stationary and dynamic processes [2].

A dynamic model is developed for premises (Fig. 1) with area of floor S_{gr} = 16 m^2, area of the outer wall S_z = 34,2 m^2, height h = 3 m. Heat losses occur through outer wall and window. Heat losses occur through outer wall and window. The wall consists of sawdust concrete (thickness 25 cm), a ceramic brick layer (12 cm), an internal plaster layer (1,5 cm) and foam plastic plates (3 cm). The area of the window surface 1,8 m^2, the coefficient of heat transfer k_l = 1,1 W·m^{-2}·^{o}C^{-1}.

The heat transfer coefficient of the wall is k_f = 0,69 W·m^{-2}·^{o}C^{-1}. There is a window in the wall with the heat transfer coefficient k_l = 1,1 W·m^{-2}·^{o}C^{-1}. Generalizing model it is necessary to take into account the parameters of given premises in certain case.

The temperature changes occurs with time delays and the corresponding units in the model are inertial units. It is possible to describe these units with the gain of the unit and time constant [3]. Initial indoor temperature is equivalent to outdoor temperature. The heat transfer balance equation for the heating process is:

$$\Delta Q_g = P_f - Q_z, \tag{1}$$

where $P_f = \alpha_s S_s(\Delta T_s - \Delta T_g)$ – actual heat power (flow) delivered by the heater, W;

$\Delta Q_g = c_g \cdot m_g \cdot \frac{d\Delta T_g}{dt}$ – the heat flow accumulated in air, W;

$\Delta T_g = T_g - T_{ag}$ – the difference between average indoor air temperature and outdoor temperature, °C;

c_g – specific heat capacity of air, J·kg^{-1}·°C^{-1};

m_g – mass of air, kg;

$Q_z = (k_f \cdot S_z + k_l \cdot S_l) \cdot \Delta T_g$;

$\Delta T_s = T_s - T_{ag}$ – difference between the heater surface and outdoor air temperatures, °C.

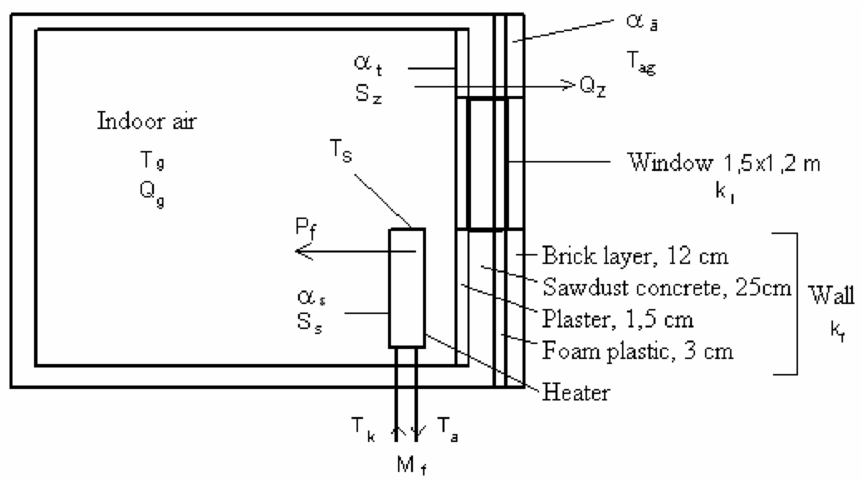

Fig. 1. **Scheme of heat flow in premises**:

α_s – coefficient of heat transfer from the heater surface, W·m^{-2}·°C^{-1};
T_{ag} – outdoor air temperature, °C;
T_g – indoor air temperature, °C; T_k,
T_a – supplied and return heat carrier temperature from the heater, °C;
S_z, S_l – the area of the inner surface of the wall and the area of the window surface, m^2;
k_f – heat transfer coefficient of the wall, W·m^{-2}·°C^{-1};
P_f – actual delivered heat power of the heater, W;
Q_z – flow of heat losses through the wall and the window, W;
Q_g – accumulated heat flow in indoor air, W; S_s – area of the heater surface, m^2.

Putting the equations of the heat flow into formula (1) it is obtained:

$$\frac{c_g \cdot m_g \cdot d\Delta T_g}{dt} = \alpha_s \cdot S_s \cdot (\Delta T_s - \Delta T_g) - (k_f \cdot S_z + k_l \cdot S_l) \cdot \Delta T_g . \tag{2}$$

Applying Laplace transformation to the equation 2 it is obtained:

$$\int_0^\infty (\tau_g \cdot \frac{d\Delta T_s}{dt} + \Delta T_g) \cdot e^{-st} dt = \int_0^\infty k_g \cdot \Delta T_s \cdot e^{-st} dt , \tag{3}$$

where τ_g – time constant of indoor air, s;

$$\tau_g = \frac{c_g \cdot m_g}{\alpha_s \cdot S_s + k_f \cdot S_z + k_l \cdot S_l} = \frac{1003{,}6 \cdot 46{,}5}{14{,}5 \cdot 1{,}4 + 0{,}69 \cdot 34{,}2 + 1{,}1 \cdot 1{,}8} = 1030\,\text{s (17 min)},$$

k_g – gain, that describes the correlation between increasing of indoor air temperature and increasing of heater surface temperature,

$$k_g = \frac{\alpha_s \cdot S_s}{\alpha_s \cdot S_s + k_f \cdot S_z + k_l \cdot S_l} = \frac{14{,}5 \cdot 1{,}4}{14{,}5 \cdot 1{,}4 + 0{,}69 \cdot 34{,}2 + 1{,}1 \cdot 1{,}8} = 0{,}43 .$$

Using Laplace integral (with initial conditions $\Delta T_g = 0$, $\Delta T_s = 0$) it is obtained:

$$(\tau_g \cdot s + 1) \cdot \Delta T_g(s) = k_g \cdot \Delta T_s(s)\,. \tag{4}$$

where s – Laplace variable.

From equation 4 transfer function is obtained:

$$W_g(s) = \frac{\Delta T_g(s)}{\Delta T_s(s)} = \frac{k_g}{\tau_g \cdot s + 1}, \tag{5}$$

where $\Delta T_g(s)$ – the image of indoor air excess temperature;
$\Delta T_s(s)$ – the image of heater surface excess temperature.

In the premises (Fig. 1) the heater is steel battery with the average heat transfer coefficient $\alpha_s = 14{,}5$ W·m^{-2}·^{o}C^{-1}, heating surface $S_s = 1{,}4$ m^2 and mass $m_s = 30$ kg.

The gain k_s describes the changes of average temperature on the heater surface in dependence on the supplied heat flow to the heater by the heat carrier Q_s. The supplied and delivered heat flows are different. During a nominal regime ($T_k/T_a/T_g$ = 75/65/20) the average surface temperature is $T_s = 70$ ^{o}C and the nominal delivered heat power is $P_f = P_n = 1000$ W. That heat power compensates heat losses from premises if outdoor air temperature $T_{ag} = -20$ ^{o}C, and the heat carrier flow, what corresponds to that, is $M_f = 0{,}02374$ kg·s^{-1}. It is possible to express correlation between ΔT_s and P_f by the gain k_r [4]:

$$k_r = \frac{\Delta T_s}{P_f} = \frac{90}{1000} = 0{,}09^{\circ}\mathrm{C} \cdot \mathrm{W}^{-1}\,. \tag{6}$$

The coefficient of heat transfer change $k_{tg1} = \Delta P_f/\Delta T_g$ describes the change of heat delivery with constant heat carrier flow in dependence on indoor air temperature. If $M_f = M_n$ and $T_k = 75$ ^{o}C, then $k_{tg1} = 22{,}54$ W·^{o}C^{-1} [4]. When M_f changes it changes too.

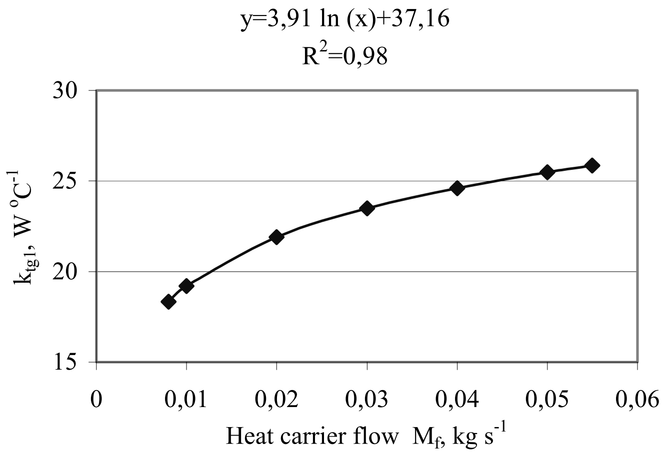

Fig. 2. **Correlation between the coefficient of heat transfer change k_{tg1} and the actual flow of heat carrier**

The developed system model with changing of heat carrier flow rate is shown in Figure 3. Changes k_{tg} of are evaluated in the model by blocks "k_{tg} funkcija" and "reizinatajs 2". The influence of heat losses to T_g is modelled by a circuit consisting of blocks "Siena", that evaluates heat transfer inertia of the wall and blocks modelling changes of outdoor temperature "T_{ag} 1…4". Inertia of the wall is described by the wall time constant τ_z:

$$\tau_z = \frac{\sum_{i-1}^{n}(c_{si} \cdot m_{si})}{\alpha_t \cdot S_z + k_f \cdot S_z} = \frac{c_{sb} \cdot m_{sb} + c_k \cdot m_k + c_p \cdot m_p + c_a \cdot m_a}{\alpha_t \cdot S_z + k_f \cdot S_z} = \tag{7}$$

$$= \frac{840 \cdot 11900 + 880 \cdot 6940 + 126 \cdot 60 + 840 \cdot 612}{295} = 59738\ \mathrm{s} \approx 1000\ \mathrm{min}$$

where c_{si} – specific heat capacity of separate wall layers, $J \cdot kg^{-1} \cdot {}^{o}C^{-1}$;
m_{si} – mass of separate wall layers, kg.

It is possible to control the flow rate of the heat carrier by changing the productivity of circulation pump, what is possible to do using a frequency converter. Nominal frequency of pump rotation is 2800 min^{-1}. The gain is:

$$k_{cs} = \frac{M_n}{f} = \frac{0{,}02374}{50} = 4{,}73 \cdot 10^{-4}\,\mathrm{kg \cdot s^{-1} \cdot Hz^{-1}}. \quad (8)$$

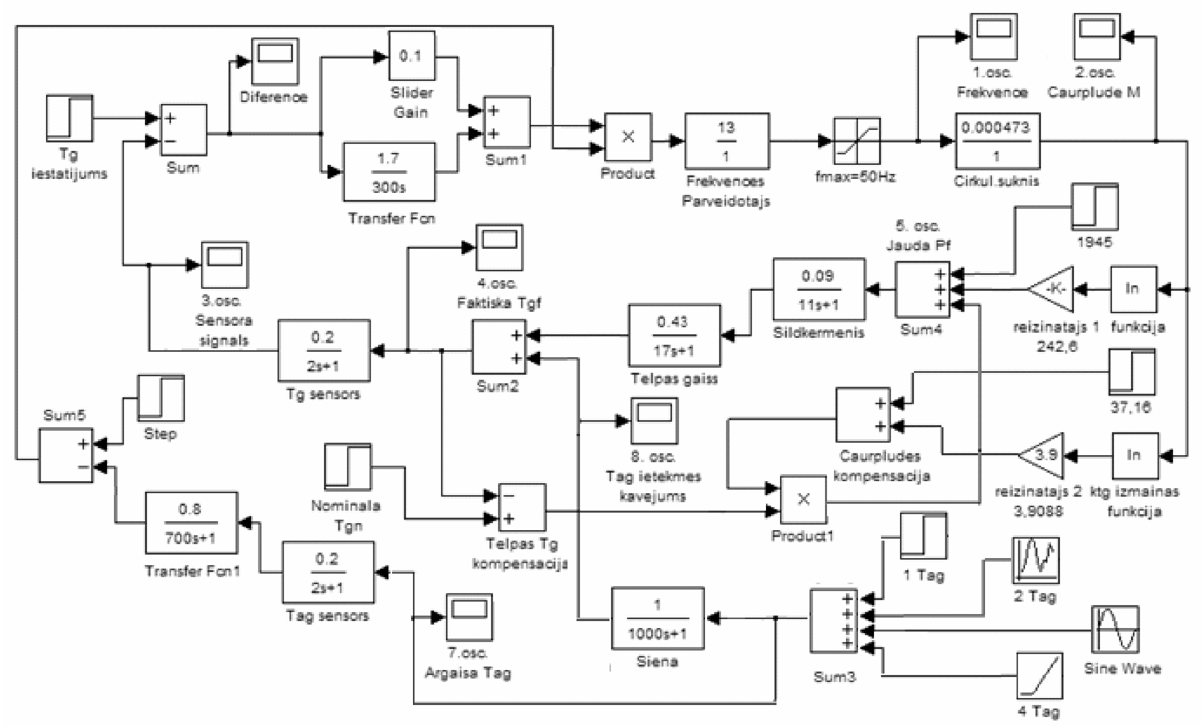

Fig. 3. **Simulation block diagram of the temperature control system for the variable heat carrier flow rate**

The frequency of pump feeding voltage is controlled by a frequency converter (block "Frekvences parveidotajs"). If the heat carrier temperature in the input is $t_k = 75\ {}^{o}C$ the correlation occurs $P_f = 242{,}5\ \ln M_f + 1945$ [4]. Circuit, that evaluates correlation between M_f and P_f, consists of blocks "funkcija", "1945" and "reizinatajs1". Blocks "Telpas T_g kompensacija", "Nominala T_{gn}" evaluate the impact of indoor air temperature to P_f, but the impact of flow rate blocks "37,16", "reizinatajs2", "k_{tg} izmainas funkcija". "Telpas gaiss" simulates the changes of indoor air temperature. The impact of flow rate to temperature compensation is evaluated by the block "Product1" and correction of overall delivered heat power is done by the adder "Sum4". The constant value of T_g, is provided by preemptive impact that compensates changes of outdoor temperature before it affects the indoor air temperature. For this purpose predictive control circuit is used. It consists of blocks: "T_{ag} sensors", "Transfer Fcn1" – block, evaluating delay of affect, "Step" – correction block, that gives the initial value of correction coefficient $k_{ag0} = 1$, if T_{ag} is constant and block"Sum5". This circuit determines the value of k_{ag}, which further is multiplied (block "Product") with the output signal from the controller accordingly to change of T_{ag}. The signal from the sensor "T_g sensors" is supplied to the adder "Sum" and is compared with the necessary T_g value set in the block "T_g iestatijums". Difference signal is fed to the PI controller what consist of blocks "Slider gain", "Transfer Fnc" and "Sum1". Controller fed signal to frequency converter.

To investigate operation of system during dynamic processes and quality of control process, the four different sources of outside impact were chosen: block "1 T_{ag}" gives step signal, block "2 T_{ag}" gives stochastic changes, block "4 T_{ag}" linear change of temperature with different rate of changing, block "Sine Wave" modelling changes of outside temperature like real changes.

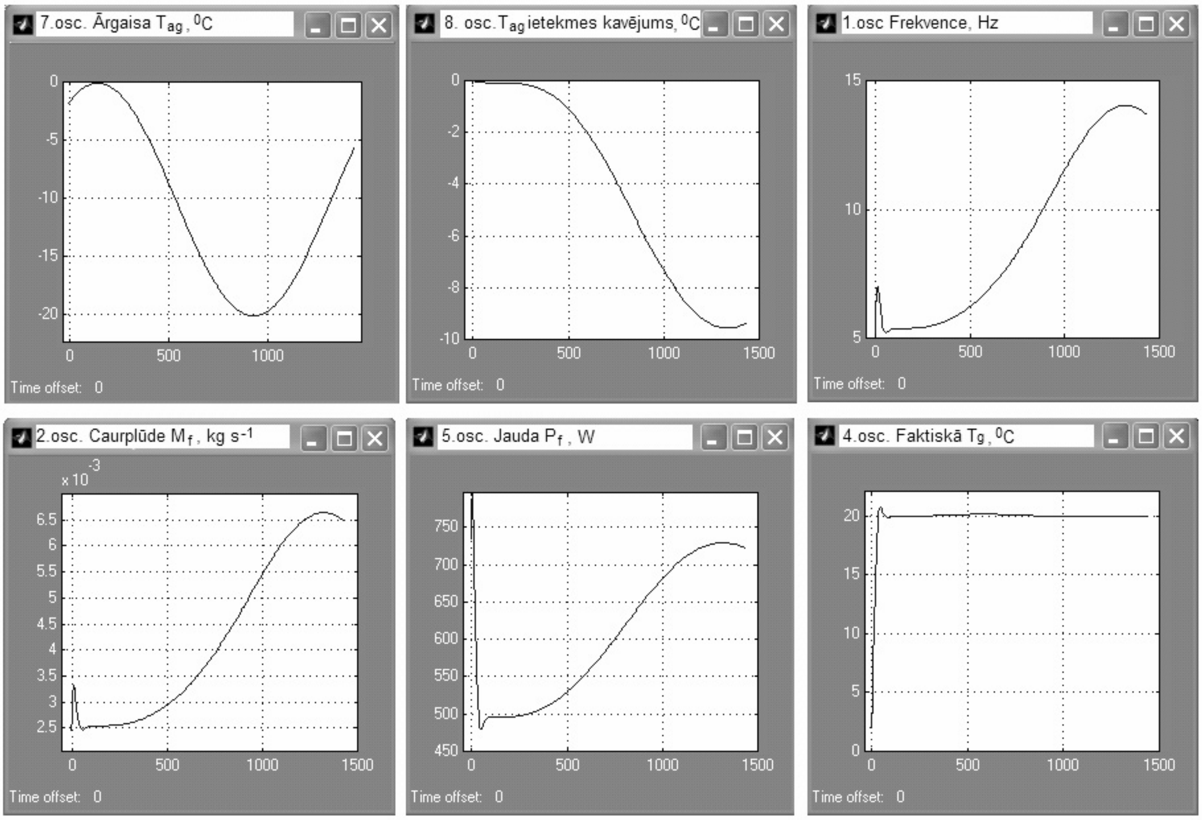

Fig. 4. **Simulation curves for the variable outdoor temperatures (**T_{ag} = 0...-20 ^{o}C):

1.osc. – voltage frequency supplied to pump *f*, Hz;
2.osc. – heat carrier flow M_f, kg·s^{-1};
3.osc. – signal of indoor air temperature sensor U_s, V;
4.osc. – actual temperature of indoor air T_g, ^{o}C;
5.osc. – delivered actual heat flow of heater P_f, W;
6.osc. – difference between nominal and actual temperature of indoor air, ^{o}C;
7.osc. – outside air temperature T_{ag}, ^{o}C;
8.osc. – delay of outdoor air temperature impact, caused by wall thermal inertia ^{o}C.

Results of modelling

Figure 3 shows simulation block diagram of the temperature control system for the variable heat carrier flow rate. The time period for modelling was chosen 1440 min (24 hours). The curves in Figure 4 describe the system operation, when the changes of possible outdoor temperature are imitated. At the beginning of imitation indoor air heats up from the initial value T_{g0} set in the model to the necessary temperature set by the controller. Before stabilization the temperature exceeds the setting value for approximately 0,4 ^{o}C, it is fully acceptable. By further changes of the outside impact the controller realizes qualitative control and keeps constant set indoor air temperature.

Conclusions

1. It is necessary to develop control devices of heating systems using cascade control systems covering all factors that impact the system operation: outdoor air temperature, indoor air temperature, the rate of their change and thermodynamic parameters of building constructions.
2. Enclosing constructions of buildings have high heat transfer inertia, as a result they have high time constants and the impact of outer perturbations occurs with high time delays. It creates a necessity for the outgoing impact from control system to occur with sufficient time delay. Equally, if there is a big difference in the controlled parameter – indoor air temperature from setting it is advisable to use the predictive control impact.

3. If the heat power of the heater is selected according to the calculated heat losses and the necessary temperature and flow rate of the heat carrier is provided, the developed control system provides maintaining of the set indoor air temperature with accuracy ±0,5 ^{o}C independent on outdoor air temperature perturbations.
4. It is possible to apply the developed model of heating system control for modelling the operation and control of heating system and also for modelling of heat transfer processes of premises.

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