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ANALYSIS OF DYNAMIC CHARACTERISTICS OF HEAT PUMP WITH THERMAL ACCUMULATOR

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Abstract: For the purpose of reducing their production costs, most of domestic scale heat pumps are thermostatic controlled with fixed speed compressors. Often, different temperatures in different rooms of the same building are achieved by using thermostatic valves which, when opened and closed, change the thermal load of the heat pump. To prevent frequent switching On/Off of the pump, as this reduce energy efficiency and pump life cycle, heat accumulators are installed with the task of absorbing the excess heat delivered by the pump. The use of heat accumulators affects both the dynamics of the pump itself and the accuracy of the regulated temperature. The paper address dynamic characteristics of heat pump with accumulator using relay feedback technique.

Key words: heat pump, thermal accumulator, relay feedback, relay with hysteresis, integrator-plus-dead-time systems

1 INTRODUCTION

Heat pump control is standardly realized by two control loops. One loop (high level control) is used to control the amount of heat energy that the pump delivers to the consumer. The second loop is used for the evaporator superheat (ES) control. ES is a quantity that affects the energy efficiency of a system. In the classical approach, for ES control, a local loop with an electronic expansion valve based on information about the actual temperature and pressure at the evaporator outlet is used [1]. Our idea is to use additional information on the dynamics of the entire heating system, in addition to this local control data. In this way, the control variables are coupled via operating and control parameters.

We consider the installation (Fig.1) which uses a thermal accumulator (TA) for heating/cooling in addition to the heat pump. TA is used to prevent frequent switching on/off of the pump. This increases the energy efficiency of the pump and prolongs the working life of the compressor [2].

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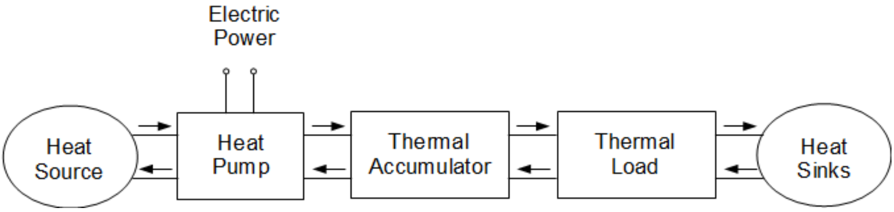


Figure 1. *Domestic scale heat pump with thermal accumulator*

Although HP with variable speed compressor are increasingly being installed today, most of the domestic scale heat pumps are thermostatic controlled (On/Off) with single, fixed speed compressors. Relay nonlinearity with hysteresis shown in Fig. 2 is used to control the compressor.

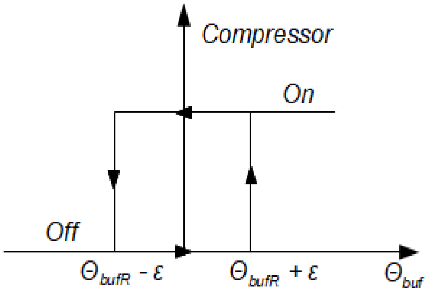


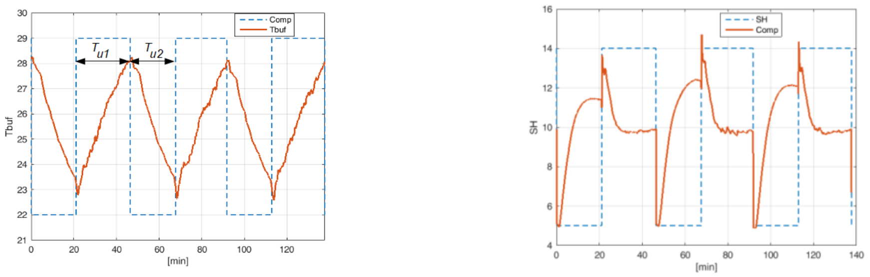
Figure 2. *Heat pump compressor control with relay nonlinearity*

where is:

$$e = \theta_{bufR} - \theta_{buf} \tag{1}$$

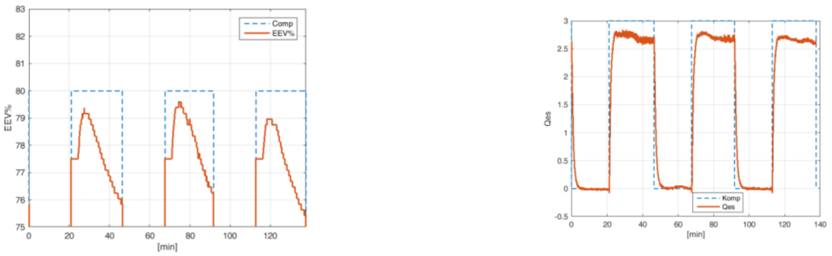
θ_{bufR} - reference temperature of the accumulator, θ_{buf} - actual temperature, ε – hysteresis width

The experimental data for the system in Fig. 1 are shown in Fig. 3. At a time interval of 140 min, the ambient temperature has a slight increase 12 °C-14 °C. Due to the cyclic switching of the compressor, the temperature in the accumulator changes periodically (Fig. 3a). The operating modes of the compressor are shown by dashed line. The changes of the SH value and the corresponding changes in the openness of the EXV are shown in Fig.3b and Fig.3c, respectively. In the mode when the compressor is On, in order for the size SH to remain approximately constant, it is necessary that EXV continuously reduce the flow of working fluid through the evaporator. The reason is that, due to the increase in the water temperature in the accumulator, the heat load increases and the SH starts to decrease. As a result, the EXV must close. Practically, after the end of the transition process, the accumulator temperature has a dominant influence on SH or on the energy efficiency of HP. Therefore, we need to know the change in water temperature in the acmulator in order to, based on that, improve the control algorithm of EXV.



a) *Actual temperature*

b) *Evaporator superheat*



c) EXV *Openness*

d) Heat flux at the input

Figure 3. *HP – Experimental data*

2. TEMPERATURE MODEL OF ACCUMULATOR IN CYCLIC MODE

For modeling the system from Fig. 1 we use the relay feedback system shown in Fig. 4.

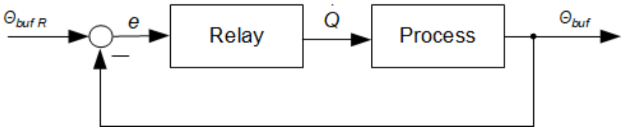


Figure 4. *Model of HP System*

Switching mechanism is modeled as biased relay with hysteresis (Fig. 5).

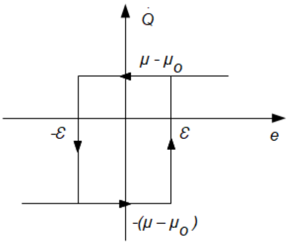


Figure 5. *Switching mechanism of HP system*

Switching On/Off of the compressor results in a change in heat flux ($\dot{Q}$) to and from the system. When the compressor is switched Off, the heat consumption, due to the large inertia of the thermal load and the relatively small change in the outside temperature, has an approximately constant value:

$$\dot{Q}_{out} = -(\mu + \mu_0) \tag{2}$$

When the water temperature in the accmulator θ_{buf} falls below the limit value $(\theta_{\mathrm{bufR}} - \varepsilon)$, it turns on a compressor whose termal power is approximately $\dot{Q}_{\mathrm{in}} = 2\mu$. The change in heat flux at the input to the system is shown in Fig. 3d. The total heat flux is then:

$$\dot{Q} = \dot{Q}_{in} - \dot{Q}_{out} = \mu - \mu_0 \tag{3}$$

The quantity μ_0 represents the asymmetry in the heat flux in the mode when the compressor is switched On/Off. If $\mu_0 = 0$ then the heat flux in both compressor modes is the same per module but of the opposite sign:

$$\dot{Q} = \left|\dot{Q}_{out}\right| = \frac{\dot{Q}_{in}}{2} = \mu \tag{4}$$

When $\mu_0 > 0$ then the heat flux in the *Off* mode of the compressor is greater than the heat flux in the *On* mode so then $T_{u1} > T_{u2}$. When $\mu_0 < 0$ the situation is reversed ($T_{u1} < T_{u2}$) . This means that based on the times T_{u1} i T_{u2} we can conclude what is the ratio of heat flows. In order for the compressor could be switched on/off periodically, the following must be applied:

$$-\mu < \mu_0 < \mu \tag{5}$$

This means that the asymmetry in the heat flow must be less than half the heat flow of the pump.

The basic assumption we use in this paper is that the dynamics of the process, in relation to the accumulator temperature as an output quantity, can be described by a first-order transfer function:

$$G_P = \frac{K}{Ts+1} \tag{6}$$

In the literature, often to describe the process, from control point of view, the first-order plus dead-time (FOPDT) model is used [3]. In our case, the delay is neglected with respect to the time constant *T*. If (6) is valid then it can be written [4]:

$$A_u = A_d = \varepsilon \tag{7a}$$

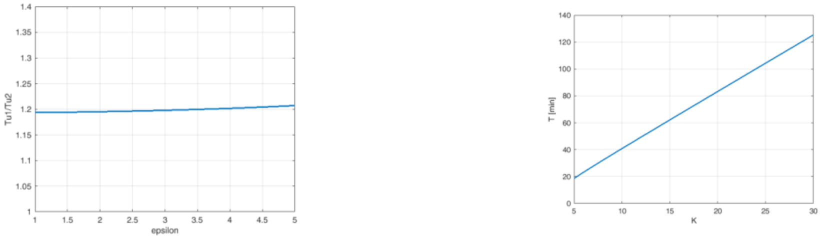
$$T_{u1} = T\, ln \frac{K(\mu-\mu_0)+\varepsilon}{K(\mu-\mu_0)-\varepsilon} \tag{7b}$$

$$T_{u2} = T\, ln \frac{K(\mu+\mu_0)+\varepsilon}{K(\mu+\mu_0)-\varepsilon} \tag{7c}$$

Based on expression (7) for the system from Fig.4, the following dependences can be obtained:

a)) T_{u1}, T_{u2} vs. μ_0 dependence

b)) T_{u1}, T_{u2} vs. ε dependence



c)) T_{u1}/T_{u2} vs. ε dependence

d) *T* vs *K* dependence

Figure 6. *Analytical results*

In Fig.6a, the dependence of time T_{u1}(compressor *On*) and time T_{u2}(compressor *Off*) from asymmetries of heat flows (μ_0) is shown. For example, for experimental data from Figure Fig.3a ($T_{u1} = 25\,min,\ T_{u2} = 21\,min$) it can be found that the asymmetry is about 8.8% and 4.4% of the heat flow of the pump (2μ), respectively.

For the given system, by increasing the width of the hysteresis (ε) time periods T_{u1} i T_{u2} are increased approximately linearly but their ratio is approximately constant (Fig.6b, Fig.6c).

Based on (7), the dependence between the time constant T and the gain K for $T_{u1} = T_{u2}$ ($\mu_0 = 0$) is shown in Fig.6d. This dependence is approximately linear, which means that if K is known, we can easily find T and vice versa.

By numerical integration of experimental results using expression (8) [5]:

$$K = G_P(0) = \frac{\int_0^{Tu1+Tu2} (\theta_{bufR} - \theta_{buf})dt}{\int_0^{Tu1+Tu2} \dot{Q}dt} \tag{8}$$

it can be obtained $K = 16.60$, i.e. $T = 4134\,[sec]$. Based on these data, a comparative presentation of the experimental results and the simulation results is given in Fig. 7.

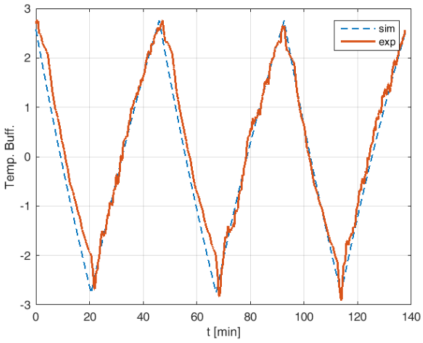


Figure 7. *Experimental vs. simulation results for temperature of thermal accumulator*

3. CONCLUSION

Although the dynamics of thermal processes are complex, due to high inertia, it can be often described by simplified models. A comparative presentation of simulation and experimental results shows that the temperature model of the heat pump accumulator can be described with sufficient accuracy by a linear first-order model. Such model can serve to determine the basic operating parameters as well as to change the control strategy of the electronic expansion valve.

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