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Robust control of refrigeration system based on disturbance observer

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Abstract. This paper presents a robust control algorithm combining the disturbance observer to compensate the serious parameter perturbation and model uncertainty in the refrigeration system. The nonlinear disturbance observer is designed to estimate the composite disturbance value. Based on the 5th-order simplified model of the refrigeration system, the sliding mode control method is designed for the evaporator and the condenser respectively to compensate the disturbance. Comparison result with the integral nonlinearity method demonstrates the superior capability of the proposed method on disturbance rejection, which can effectively guarantee the efficient and stable operation of refrigeration system.

1. Introduction

In vapor compression refrigeration systems, the operating efficiency and stability depend strongly on the superheat at the evaporator outlet. In order to improve the energy efficiency, the superheat value is regulated to a reasonably low setting[1]. To achieve the additional energy saving, Fallahsohi et al. changed the minimum superheat setting value depending on four different operating condition points[2]. However, when the superheat value is low, the evaporator sometimes exhibits an unstable behavior, the so-called hunting problem. A lot of researches show that if the superheat of the evaporator is lower than the corresponding MSS value, the drastic changes in the heat transfer coefficient of the evaporator may cause the dynamic performance and stability of the system worse.

In addition, there are inevitably model uncertainties such as unmodeled dynamics in the refrigeration system model. Although adaptive methods and H_∞ methods can deal with the uncertainty of the system model, they are generally indirect compensation method in the form of feedback control. When the system has strong uncertainties, the performance of the feedback suppression means is conservative and the response is not fast enough to meet the requirements of the refrigeration system. In recent years, the Disturbance Observer Based Control (DOBC) combines the disturbance observer with the existing control method to provide a new idea for solving the disturbance problem. DOBC mainly has the following two advantages: (1) can directly compensate the disturbance amount of the system; (2) does not affect the control performance of the original control method. Therefore, some scholars have begun to apply it to solve the control problems of various practical systems [3-5].

The rest of this paper is organized as follows. In Section 2, the simplified nonlinear model of refrigeration system is presented. In Section 3, Control algorithm based on disturbance observer is described. Section 4 shows the results of simulation and discussion. Finally, in Section 5, the main conclusions of the work are summarized.

2. Simplified nonlinear model of refrigeration system

According to the literature[6], the evaporator model is expressed as:

$$\begin{cases} \dot{T}_e = c_0(T_{chw} - T_e)l_e + c_1\dot{m}_{com} \\ \dot{l}_e = c_2(\dot{m}_e - \dot{m}_{com}) \end{cases} \tag{1}$$

The modeling process of the condenser is similar to that of the evaporator, and the condensing temperature dynamic equation is:

$$\dot{T}_c = c_3(T_{wa} - T_c)l_c + c_4\dot{m}_{com} \tag{2}$$

Considering that the control unit chilled water pump is located outside the chiller unit, in order to more accurately describe its influence on the internal variables of the chiller, the thermal dynamics of the condenser tube wall is considered in the modeling process of the condenser, and the wall temperature H_∞ is used as the state variable. Therefore, the cooling water inlet temperature H_∞ in equation (2) has been replaced with the tube wall temperature H_∞. According to the wall energy conservation equation, the temperature change rate of the condenser tube wall can be expressed as the sum of the heat absorption power of the tube wall and the heat release power to the cooling water:

$$\dot{T}_{wa} = c_5(T_c - T_{wa}) + c_6(T_{wa} - T_{clw})V_{clw} \tag{3}$$

Where H_∞ is heat transfer coefficient on the refrigerant side, H_∞ is the heat transfer coefficient on the cooling water side.

Since the condenser inlet outlet mass flow is opposite to the evaporator, the two-phase zone length dynamic equation of the condenser can be expressed as:

$$\dot{l}_c = c_7(\dot{m}_{com} - \dot{m}_e) \tag{4}$$

In summary, equations (1), (2), (3), and (4) are simplified nonlinear models of the refrigeration system. Finally, based on the experimental data, and using the Matlab System Identification Toolbox to identify the above model parameters, the identification results are shown in Table 1.

Table 1 Simplified nonlinear model parameter table

c_0	c_1	c_2	c_3	c_4	c_5	c_6	c_7
0.12	-188.76	15.5	-0.71	-395.7	0.1	-0.032	-30

3. Control algorithm design

After establishing the system to simplify the nonlinear model, this section is based on the model and combined with the disturbance observer to design the robust control algorithm. The structure of the refrigeration system controller is shown in Figure 1.

The control objective of the refrigeration system control algorithm is to design the control input H_∞ to ensure that the evaporator outlet superheat, evaporation temperature and condensation temperature can still track the set value when affected by model mismatch and parameter perturbation.

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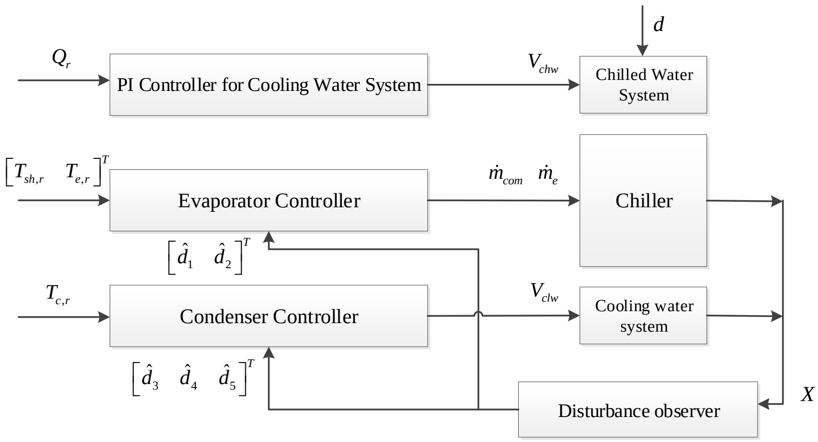


Fig.1 Control scheme for the refrigeration system

According to the physical characteristics of the refrigeration system, the system model is divided into two subsystems, the evaporator H_∞ and the condenser H_∞ :

$$S_e: \begin{cases} \dot{T}_e = c_0(T_{chw} - T_e)l_e + c_1\dot{m}_{com} + d_1 \\ \dot{l}_e = c_2(\dot{m}_e - \dot{m}_{com}) + d_2 \end{cases} \tag{5}$$

$$S_c: \begin{cases} \dot{T}_c = c_3(T_{wa} - T_c)l_c + c_4\dot{m}_{com} + d_3 \\ \dot{T}_{wa} = c_5(T_c - T_{wa}) + c_6(T_{wa} - T_{clw})V_{clw} + d_4 \\ \dot{l}_c = c_7(\dot{m}_{com} - \dot{m}_e) + d_5 \end{cases} \tag{6}$$

Where the system compound disturbance H_∞ mainly includes the following parts: 1. Parameter uncertainty and unmodeled dynamics; 2. System operating point into the unstable region of the evaporator parameter perturbation

3.1. Disturbance observer

Integrate the system models (5) with (6) and write them in the following compact form:

$$\dot{x} = f(x) + g(x)u + d \tag{7}$$

Where H_∞ is a state vector, H_∞ is a control vector, H_∞ is a perturbation vector, and H_∞ is a continuous function of the state vector.

Then the compound disturbance H_∞ can be expressed as:

$$d = \dot{x} - f(x) - g(x)u \tag{8}$$

To estimate the composite disturbance, a perturbation observer of the form:

$$\dot{\hat{d}} = L(x)\tilde{d} = L(x)(d - \hat{d}) \tag{9}$$

Where H_∞ is the gain of the observer designed to be a positive definite diagonal array to ensure that the estimated value of the disturbance observer converges to the actual value.

Bringing the equation (8) into the equation (9) gives the specific form of the disturbance observer:

$$\dot{\hat{d}} = -L(x)\hat{d} + L(x)(\dot{x} - f(x) - g(x)u) \tag{10}$$

It is noted that the above disturbance observer contains the differentiation of the system state values, which greatly increases the influence of the measurement noise on the disturbance estimation. To this end, the differential term in the perturbation observer described above is eliminated by introducing an auxiliary variable.

Design auxiliary variables and observer gain matrices:

$$z = \hat{d} + p \tag{11}$$

$$L(x) = \frac{\partial p(x)}{\partial x} \tag{12}$$

Bringing the formulas (11) and (12) into the formula (10) is:

$$\begin{aligned} \dot{z} &= \dot{\hat{d}} - \dot{p}(x) \\ &= -L(x)z + L(x)(-f(x) - g(x)u - p(x)) \end{aligned} \tag{13}$$

The improved disturbance observer is:

$$\begin{cases} \dot{z} = -L(x)z + L(x)(-f(x) - g(x)u - p(x)) \\ \hat{d} = z + p(x) \end{cases} \tag{14}$$

Where H_∞ , H_∞ and H_∞ are respectively the disturbance estimation value, H_∞ is the auxiliary variable and H_∞ is the observer gain, which are nonlinear functions and are designed as H_∞ .

3.2. Design of sliding mode control method

Define the evaporation temperature tracking error:

$$e_1 = T_{e,r} - T_e \tag{15}$$

Define the tracking error of the two-phase zone length of the evaporator:

$$e_2 = l_{e,r} - l_e \tag{16}$$

Deriving equations (15) and (16) respectively:

$$\dot{e}_1 = \dot{T}_{e,r} - [c_0(T_{chw} - T_e)l_e + c_1\dot{m}_{com} + d_1] \tag{17}$$

$$\dot{e}_2 = \dot{l}_{e,r} - [c_2(\dot{m}_e - \dot{m}_{com}) + d_2] \tag{18}$$

The compressor mass flow rate is designed according to equation (17).

Design the sliding surface:

$$\sigma_1 = e_1 \tag{19}$$

Deriving the equation (19), the dynamic characteristics of the open-loop sliding mode are obtained:

$$\dot{\sigma}_1 = \dot{e}_1 = \dot{T}_{e,r} - c_0(T_{chw} - T_e)l_e - c_1\dot{m}_{com} - d_1 \tag{20}$$

Design control variable:

$$\dot{m}_{com} = \frac{1}{c_1}[\dot{T}_{e,r} - c_0(T_{chw} - T_e)l_e - \hat{d}_1 + k_1 sign(\sigma_1)] \tag{21}$$

Substituting the equation (21) into the equation (20), the closed-loop sliding mode dynamic characteristics are obtained:

$$\dot{\sigma}_1 = -k_1 sign(\sigma_1) - \tilde{d}_1 \tag{22}$$

According to the equation (18), the control law is designed as the control variable.

Design the sliding surface:

$$\sigma_2 = e_2 \tag{23}$$

Deriving the equation (23), the dynamic characteristics of the open-loop sliding mode are obtained:

$$\dot{\sigma}_2 = \dot{e}_2 = \dot{l}_{e,r} - c_2(\dot{m}_e - \dot{m}_{com}) - d_2 \tag{24}$$

Design control variable:

$$\dot{m}_e - \dot{m}_{com} = \frac{1}{c_2}\left[\dot{l}_{e,r} - \hat{d}_2 + k_2 sign(\sigma_2)\right] \tag{25}$$

Substituting the equation (25) into the equation (24), the closed-loop sliding mode dynamic characteristics are obtained:

$$\dot{\sigma}_2 = -k_2 sign(\sigma_2) - \tilde{d}_2 \tag{26}$$

According to the equation (21) and the equation (25), the control variable can be obtained.

The condenser controller design process is similar to that of the evaporator, and is not described here.

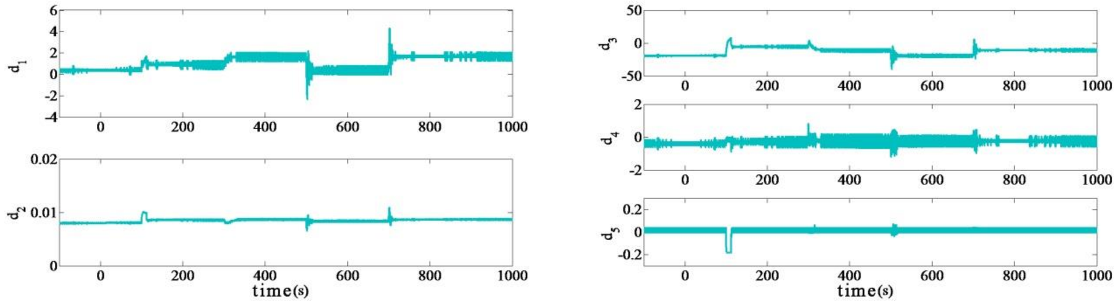
4. Simulation results and analysis

In this section, the high-order nonlinear mechanism model of the refrigeration system developed by Yao Y[7] is used for dynamic simulation of the controlled object. In order to better illustrate the superiority of the robust control method combined with the disturbance observer, the simulation results are compared with the nonlinear method with integral. The evaporator and condenser of the comparison method adopt sliding mode controller and backstepping controller respectively.

Control target: The initial reference value of superheat is 7ºC, the step is reduced to 5ºC at 100s; the reference value of evaporation temperature is 0ºC, which remains unchanged during the simulation; The initial reference value of condensation temperature is 36ºC, and at 300s, the step rises to 38ºC. In addition, the controlled high-order model evaporator changes from a nucleate boiling state to a convection boiling state at 500 s, and it returns to the state of nucleate boiling at 700s.

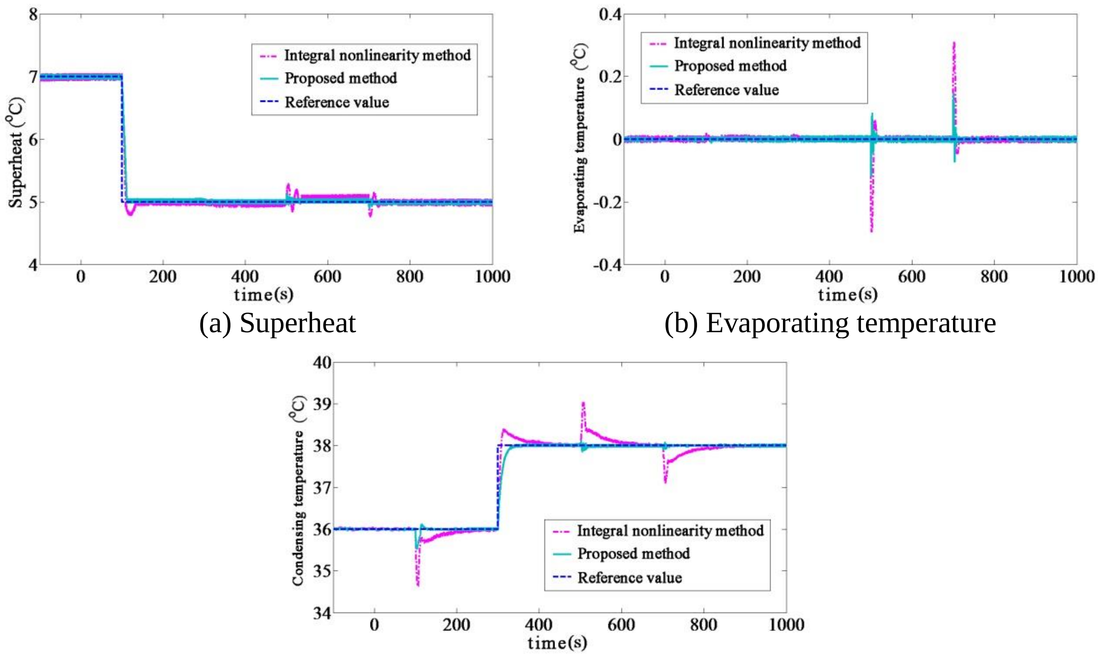
Nonlinear disturbance observer parameters: $C = diag(2.1, 1.7, 2.5, 2.0, 3.1)$; Sliding mode controller parameters: $k_1 = 1.3, k_2 = 1.95$; Backstepping controller parameters: $k_3 = 1.5, k_4 = 0.5$.

The simulation results are shown in figure 2-4, where figure 2 shows the estimates of the composite disturbances during the simulation process. It can be seen from the figure that the estimated values of the disturbance observer output at each steady state are not zero, which better reflects the model mismatch between the simplified nonlinear model and the high-order mechanism model. At 500s and 700s, the high-order models are subjected to parameter perturbation, the observer estimates the disturbances quickly, which also provides a powerful guarantee for the control algorithm to quickly compensate the disturbance.



(a) Perturbation estimation of evaporator (b) Perturbation estimation of condenser

Fig. 2 Perturbation estimation



(a) Superheat (b) Evaporating temperature

(c) Condensing temperature

Fig. 3 Dynamic response comparison

Figure 3 shows the dynamic response of superheat, evaporation temperature and condensation temperature of refrigeration system under two control methods. It can be seen that although there is a model mismatch between the simplified model and the controlled high-order model, the robust control method designed in this paper can eliminate the steady-state error better based on the disturbance observer's estimation of the disturbance.

Figure 4 shows the control input of the refrigeration system of the proposed method. It can be seen that the control inputs are kept within a reasonable range, which can form a good protection for the actuator. Based on the above analysis, the robust control method designed in this chapter combined with the disturbance observer is better than the integral nonlinearity method, and it has better robustness to the system disturbance, especially the non-matching disturbance, and better control of the refrigeration system

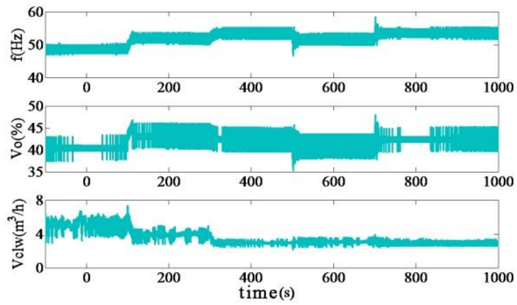


Fig. 4 Control inputs of the proposed method

5. Conclusion

This paper presents a robust control method of the refrigeration system, combining the disturbance observer and nonlinear control method. Firstly, in order to facilitate the design of nonlinear controller, the refrigeration system simplified nonlinear model is deduced based on a series of simplifications. Secondly, the nonlinear disturbance observer is designed to estimate the system composite disturbance, which primarily contains the system parameter perturbation and model mismatch. The sliding mode control algorithm is designed to compensate the disturbance in the evaporator and condenser subsystem. Finally, the simulation and experimental results show that the robust control algorithm with disturbance observer has excellent disturbance rejection ability, which can effectively guarantee the efficient and stable operation of refrigeration system.

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