Hybrid PID-Cascade Control for HVAC System

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Abstract—The primitive controller used in the early version for HVAC systems, like the on-off (Bang-Bang) controller, is inefficient, inaccurate, instable, and suffers from high-level mechanical wear. On the other hand, other controllers like PID, compensator, and cascade controllers overcome these disadvantages but when an offset response (inaccurate response) occurs, power consumption will increase. In order to acquire better performance in the central air-conditioning system, a hybrid PID-cascade control is investigated in this paper and compared to the traditional PID, industrial PID (Ziegler-Nichols tuning) and compensator controllers in simulation and experiments. The output of the system is predicted through indoor and outdoor disturbances. Based on the mathematical model of air-conditioning space, the simulations in this paper have found that the proposed hybrid PID-cascade controller has the capability of self-adapting to system changes and results in faster response and better performance.

Keywords: PID, Lag and Lead compensators, Cascade control, HVAC system, Disturbance rejection, hybrid control.

NOMENCLATURE

Symbols

M_h	The quality of heat exchanger, (kg)
AHU	Air handling unit
M_r	Air quality of air-conditioning room, (kg)
G_a	Supply air flows ,(kg/s)
G_w	Cold water (or hot water) flows,(kg/s)
C_a	Specific heat capacities of air, (kj/kg C^o)
C_w	Specific heat capacities of water(kj/kg C^o)
C_h	Specific heat capacities of heat exchanger (kj/kg C^o)
T_m	The temperature of mix fresh air, (C^o)
T_l	The temperature after heat exchanger, (C^o)
T_h	Surface temperature of heat exchanger,(C^o)
T_{win}	The temperature of supply water, (C^o)
T_{wout}	Back-water from heat exchanger, (C^o)
Q_{room}	Perturbations inside thermal load, (kj)
T_{out}	Uncontrolled outside temperature (C^o)
K_1	The amplify coefficient of heat exchanger, (C^o .s /kg)
T_1	Heat exchanger time constant, (s)
$G_r(s)$	The disturbance of heat exchanger, (kg/s)
K_2	The amplify coefficient of room, (C^o.s/kg)
T_2	Conditioning space time constant, (s)
T_f (s)	The disturbance to room, which include the disturbances from outdoor and indoor, (C^o).

Subscripts

h	Heat exchanger
r	Room
a	Air
w	Water
m	Mixed fresh and return air
l	Leaving
W_{in}	Water input
W_{out}	Water output
room	Inside room
out	Outside room
1,2	Heat exchanger, room region
f	Indoor and outdoor

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1. INTRODUCTION

Heating, Ventilation and Air-Conditioning (HVAC) systems are widely used in different environments. HVAC systems are composed of a large number of subsystems, each of which may exhibit time-varying and/or nonlinear characteristics as in Jiangjiang et al. (2006). Furthermore, increase in capacitance of building structure raises thermal inertia (Szymon (2010)). It would be very difficult to dedicate control system to a specific building due to great variety of building technologies and its dynamical properties. Such these complexities indicated that the use of some simple control schemes (like on-off control that many HVAC systems are using) may not be appropriate for some of the new load-management technologies and systems. Traditional PID controller sometimes doesn't satisfy the control purpose for the object, which has large inertia, delay and nonlinear characteristic and uncertain disturbance factor, like the tall and big space, because of the dissatisfaction of tuning parameters, the effect of dissatisfying performance and the adaptability to different running medium (Wang et al. (2008) and Servet et al. (2009)). To overcoming the failing of traditional PID control, we will add cascade (Cheng (2006)).

Cascade control is especially useful in reducing the effect of a load disturbance that moves through the control system slowly (Riccardo (2009)). The inner loop has the effect of reducing the lag in the outer loop, resulting in the cascade system responds more quickly with a higher frequency of oscillation. Simulations have illustrated this effect of cascade control (Donald (1991) and Antonio (2006)).

In this study, the heat exchanger and air-conditioning space in the HVAC system are modeled. Then the hybrid PID-cascade control system is presented, which combines the traditional PID control and cascade. The cascade internal loop control is designed in terms of the robust control H2 optimal performance specification as in Vojtech et al. (2009).

Through the simulation in the HVAC system, it is found that the PID controller enhances the stability and rejects the disturbance and the cascade control increases the response speed and control precision in the HVAC system.

2. MATHEMATICAL MODEL FOR HVAC SYSTEM

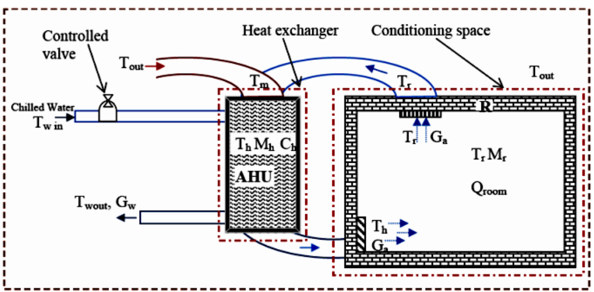

Figure (1) Room-air conditioning system

There are basically two ways of determining a mathematical model of a system: by implementing known laws of nature or through experimentation on the process (Xiaosh (2001) and (2002)).

The first one will take to obtain the mathematical model for HVAC which contain from the first part is the air-processing unit (heating/cooling system) and the second part is the air-conditioning room. These two parts are illustrated in Figure 1.

2.1 Heat Exchanger model

Based on the law conservation of energy, the thermal balance equation (Wang et al. (2007a)) is shown as follows:

$$M_h C_h \frac{dT_h}{dt} = G_a C_a (T_m - T_l) - G_w C_w (T_{wout} - T_{w\,in}) \qquad (1)$$

The transfer function of handled air temperature can be derived from (1) :

$$\frac{T_l(s)}{\dfrac{G_a C_a T_m}{C_w (T_{wout} + T_{w\,in})} + G_w(s)} = \frac{C_w (T_{wout} + T_{w\,in})}{G_a C_a \left(\dfrac{M_h C_h S}{G_a C_a} + 1\right)}$$

$$G_1(S) = \frac{T_l(s)}{G_r(s) + G_w(s)} = \frac{K_1}{T_1 S + 1} e^{-\tau_1 s} \qquad (2)$$

where K_1 is the coefficient of amplify, (C^o .s /kg); T_1 is time constant, (s); τ_1 is the pure time delay of the controlled object, (s); G_r is the disturbance of heat exchanger, (kg/s) where:

$$T_1 = \frac{M_h C_h}{G_a C_a}, \quad K_1 = \frac{C_w (T_{wout} + T_{w\,in})}{G_a C_a},$$

$$G_r(s) = \frac{G_a C_a T_m}{C_w (T_{wout} + T_{w\,in})}$$

2.2 Conditioning space model

The supposition for the temperature control model of air-conditioning space is described as follows: firstly the space is airtight and there is not the direct heat exchange between indoors and outdoors; secondly the temperature in the space is almost equal; thirdly the heat capacity of the door, windows and the goods in the space is ignored. The thermal balance equation is similar to the heat exchanger by Wang et al. (2007a).

$$M_r C_r \frac{dT_r}{dt} = G_a C_a (T_s - T_r) + \frac{T_{ou} - T_r}{R} + Q_{room} \qquad (3)$$

Here, the supply air temperature to room is the handling air temperature after AHU and T_s= T_l. Assumption is that the specific heat capacity of air in air-conditioning room is equal to specific heat capacity of supply air and C_r = C_a. The transfer function of conditioning space temperature can be derived from (3).

$$\frac{T_r(s)}{T_s(s)+\frac{\frac{T_{ou}}{R}+Q_{room}}{G_aC_a}} = \frac{G_aC_a}{\left((G)_aC_a+\frac{1}{R}\right)\left(\frac{M_rC_aS}{G_aC_a+\frac{1}{R}}+1\right)}$$

$$G_2(S) = \frac{T_r(s)}{T_s(s)+T_f(s)} = \frac{K_2}{T_2S+1}e^{-\tau_2 s} \tag{4}$$

where K_2 is the amplify coefficient of room, (C°. s /kg); T_2 is time constant, (s); τ_2 is the pure time delay of the controlled object, T_f (s) is the disturbance to room, which include the disturbances from outdoor and indoor, (C°). Here,

$$T_2 = \frac{M_rC_a}{G_aC_a+\frac{1}{R}}, \quad K_2 = \frac{G_aC_a}{G_aC_a+\frac{1}{R}},$$

$$T_f(s) = \frac{\frac{T_{ou}}{R}+Q_{room}}{G_aC_a}$$

The following transfer function of heat exchanger and conditioning space can be derived from (2) and (4).

$$G(s) = G_1(S)\times G_2(S) = \frac{K_1K_2}{((T)_2S+1)((T)_1S+1)}e^{-\tau s} \tag{5}$$

3. CONTROL SYSTEM DESIGN

An air-handling unit for a commercial building consists of a fan, heating-coil and discharge air duct to deliver heated/cooled air into the space. In a conditioned space, a throttling-type thermostat (not the on/off type common in residential applications) senses the temperature and controls a steam (or chilled water) valve on the heating coil. Return air from the space is mixed with outdoor air at the inlet to the fan. The fixed ratio of outdoor air to return air is used to meet the requirements for ventilation as shown in Figure 2.

In normal operation, the thermostat does an adequate job of maintaining a stable space temperature. However, in the particular climate of this application, the outside air temperature sometimes drops very rapidly. When it does the mixed air temperature drops, as does the discharge air temperature. This eventually causes a drop in the space temperature. The thermostat senses and corrects for this, but because of the large volume of space, it takes an excessively long time to recover to the desired temperature.

We can improve the control system to alleviate this problem, by adding an intermediate measurement which is used in an inner feedback loop that will encompass the disturbance (Riccardo (2009)).

Therefore the possible solution is to measure the steam flow and cascade the output of the thermostat to a steam-flow controller. But the flaw in the inner loop (steam flow) does not encompass the disturbance (change in outdoor air temperature).

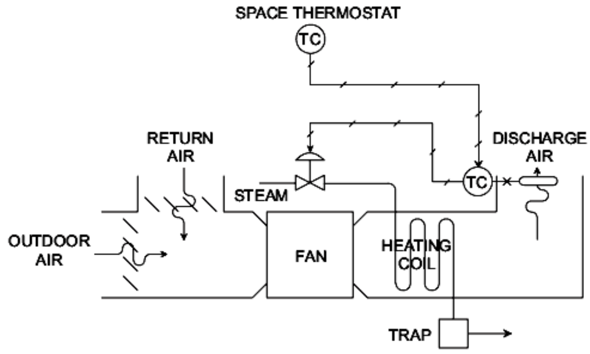

Figure (2) A Cascade Control Application in the HVAC

The possible solution is to measure the discharge temperature and let that control the steam valve, as shown in Figure 2. When the outside air temperature drops, and consequently the mixed air temperature, the discharge temperature controller will sense this. The discharge temperature control loop will be rapid, in comparison with the space temperature control loop. Hence, the discharge temperature will be maintained approximately constant at its set point (Harold (2004), Jairo et al. (2005) and Riccardo (2009)).

3.1 Architecture of the hybrid PID-cascade

There are researches who studied simple controller design and tuning rule of traditional cascade control system such as Wang et al. (2007a).

Here we designed the cascade control system for the air-conditioning system, which is shown in Figure (3). G_1 and G_2 are respectively the controlled process of heat exchanger and air-conditioning room; G_{1m} and G_{2m} are respectively identification models; PID is the setpoint response controller and F (s) is the internal loop controller; Q_{room} is the internal load disturbance; and T_a is the outside load disturbance.

3.2 PID tuning

There are a large number of methods of tuning a PID controller.

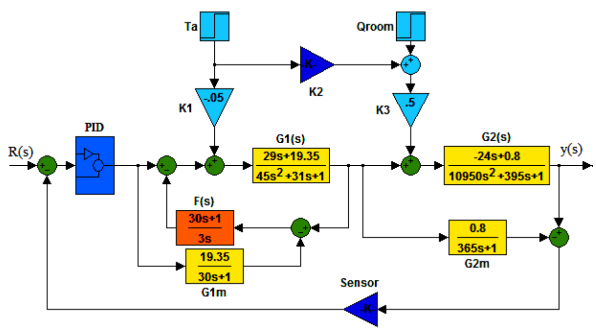

Figure (3) Block diagram of proposed model based on Cascade control

The most popular ones amongst them are the *reaction curve method* and *instability method*. Both are referred to as the *Ziegler–Nichols tuning* method.

From the Ziegler-Nichols rules the parameters of the traditional single PID control are tuned as follows: K_p=0.199, K_i= 0.0006 and K_d= 7.063 (Graham (2000)).

3.2 Internal loop controller

As Figure 3 shows, the disturbance transfer function of middle process is shown as follows:

$$H_d(s) = \frac{T_a}{T_s} = \frac{G_1(s)}{1 + F(s)G_1(s)} \tag{6}$$

where T_s is the output of $G_1(s)$.
Then the closed-loop complementary sensitivity function is obtained as follows:

$$T_d(s) = \frac{f}{T_a} = \frac{F(s)G_1(s)}{1 + F(s)G_1(s)} \tag{7}$$

where *f* is the inner feedback.

In the ideal condition, $T_d(s)$ should be in the form: $Td(s) = e^{-\tau_1 s}$, which means that when the T_a disturbance inputs to the middle process, the internal loop controller F should detect the error of T_s after τ_1 time delay. Then the controller F outputs a reverse isoperimetric signal to reject the disturbance. Actually it is considered that the output of controller is limited and the error is gradually offset. Here based on the robust control H2 optimal performance objective, the actual desired closed-loop complementary sensitivity function is designed as the follows:

$$T_d(s) = \frac{1}{\lambda_f s + 1} e^{-\tau_1 s} \tag{8}$$

where λ_f is a control parameter.

λ_f is needed to tune in the control system. In practical application, λ_f can be initialized to τ_1 and then be tuned monotonically on-line to obtain better control performance.
The F(s) can be solved from (7) and is shown in (9).

$$F(s) = \frac{T_d(s)}{1 - T_d(s)} \cdot \frac{1}{G_1(s)} \tag{9}$$

To carry out conveniently, (9) is transferred and F(s) can be shown as follows:

$$F(s) = \frac{F_1(s)}{1 - F_1(s)G_1(s)} \tag{10}$$

where $F_1(s)$ can be expressed as follows:

$$F_1(s) = \frac{T_d(s)}{G_1(s)} = \frac{(T_1 s + 1)}{K_1(\lambda_f s + 1)} \tag{11}$$

The configuration of observer F is shown in Figure 4.

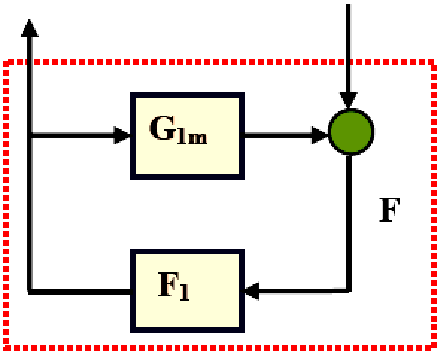

Figure (4) Configuration of observer F Wang et al. (2007a)

When λ_f is relatively large, the robust of control system is relatively strong, but in the meantime the disturbance rejection will be weak. When λ_f is small, robustness of control system is relatively weak and the disturbance rejection will be relatively strong. So the robustness and disturbance rejection should be both considered during the tuning of λ_f. In practical application, λ_f can be initialized near the time delay of P_1, τ_1. If the process P_1 has no pure time delay, λ_f can also be initialized near to the time constant of P_1. Then λ_f is tuned automatically online to obtain better control performance.

4. SIMULATION

To evaluate the advantages of the proposed techniques implemented on the HVAC system, the following simulation has been conducted.

The PID parameters are tuned in using the robust PID tuning method (Li et al. (2005)) while PID industrial tuning used Ziegler-Nichols method (Jairo et al. (2005)).

Bode plot and root-locus are used to design the compensator (Donald (1991) and Brian et al. (2006)). The bode plot is a plotting of the frequency transfer function that can be systematized and simplified by using logarithmic plots.

The parameters of the HVAC system are as follows: the volume of air conditioning room is 10m length, 8m width and 4.5m height; the air specific heat capacity is Ca=1.0 kJ/kg.C^o; and the air density is 1.2 kg/m^3. Based on the criterion in the heating, ventilation and air-conditioning field, and the number of taking a breath in air-conditioned room, the calculation of the supply air is Ga=1.08kg/s, the heat resistance of wall is 1/R=0.2kw/C^o and the temperature error of supply cold-water and back-water to the heat exchanger in summer is T_{win}-T_{wout}= - 5C^o. Lastly, the parameters of the middle process, heat exchanger, are calculated as following: K_1= -19.35C^o. s /kg, T_1=30s.

The parameters of air-conditioned room are calculated as follows: K_2=0.4C^o s /kg, T_2=338s. So the controlled object $G_1(s)$ and $G_2(s)$ can be expressed as: (Wang et al. (2007b))

$$G_1(S) = \frac{19.35}{30S + 1} e^{-\tau_1 s} \approx \frac{29s + 19.35}{45s^2 + 31s + 1} \tag{12}$$

$$G_2(S) = \frac{0.8}{365S+1} e^{-\tau_2 s} \approx \frac{-24s+0.8}{10950s^2+395s+1} \quad (13)$$

$$G(s) = G_1(S) \times G_2(S) = \frac{K}{(T_2 S+1)(T_1 S+1)} e^{-\tau s} \quad (14)$$

and the disturbances from outdoor and indoor can be expressed as:

$$G_r(s) = 0.052 T_a$$

$$T_f(s) = 0.81 T_a + 0.926 Q_{room}$$

Simulation (1): The comparison curves of four controllers are shown in Figure 5. It can be seen that the setpoint response of the hybrid PID-cascade control system has accurate response while the others all have offset and the hybrid PID-cascade control responses quicker.

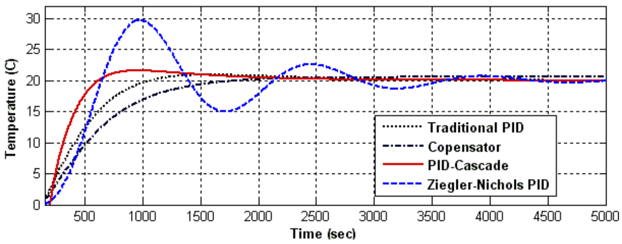

Figure (5) Simulation results of four controllers

Simulation (2): To validate the rejection to the disturbances of hybrid PID-cascade control in the HVAC system, the input of disturbance is increased from 0 to 50 in step of 4000th per second while the controllers' parameters are kept the same as in *Simulation (1)*.

The simulation results of traditional PID, compensator, Ziegler-Nichols PID tune and hybrid PID-cascade controllers are shown in Figure 6. The overshoot of hybrid PID-cascade control is lesser that other controllers while its governing time is near to the traditional PID control. The rejection to the disturbances of traditional PID and hybrid PID-cascade controllers is similar. Moreover, the hybrid PID-cascade controller's response was quicker.

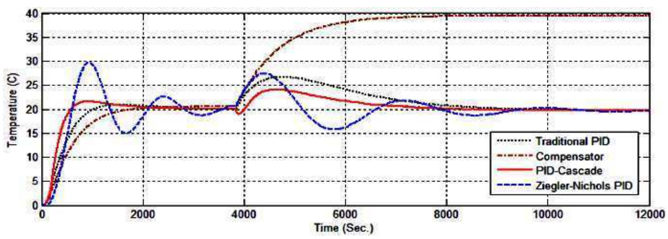

Figure (6) Response for four controllers when the disturbance is inserted at 4000 seconds

Simulation (3): To certify the robustness of hybrid PID-Cascade control in the HVAC system, the controlled process parameters are changed during simulation while the controllers are kept unchanged. The gains of the controlled process, K_1=-19.35 C^{O}s/Kg, K_2=0.8 C^{O}s/Kg are increased by 10% to K_1= -21.3 C^{O}s/Kg, K_2=0.924 C^{O}s/Kg respectively.

The simulation results are shown in Figure 7. It can be seen that the setpoint response is similar to the *Simulation (1).* However, the overshoot of traditional PID control is greater than the *Simulation (1).* Based on this simulation, the robustness of the hybrid PID-Cascade control system has been proven.

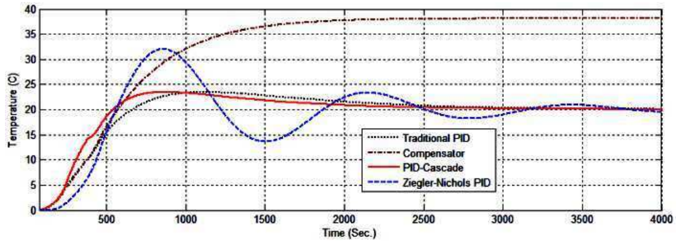

Figure (7) Step response for four controllers when the parameters of process are changed

5. CONCLUSION

The hybrid PID-cascade controller, which is a method for adaptively adjusting the PID gains using cascade feed forward, is adopted the constant temperature central air-conditioning system. It has shown better performance than traditional PID control system. Through simulation, hybrid PID-cascade control has shown better robustness and adaptability for nonlinear object. The central air-conditioning system, which is controlled by the proposed hybrid PID-cascade controller, has faster response and better performance even with the seasonal outdoor heat disturbance and uncertain indoor heat disturbance compared to the conventional system. The hybrid PID-cascade control can also be applied to objects with large inertia, pure lag, nonlinear characteristic and uncertain disturbance factor.

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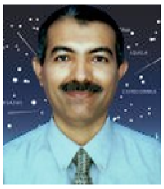

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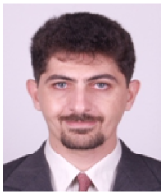

Dr. Haider A. F. Mohamed received his PhD in Electrical Engineering from the University of Malaya, Malaysia in 2006. He worked as a computer engineer for two years and as a researcher for four years before he became a lecturer in the Department of Electrical Engineering in University of Malaya, in 2000. He is currently an associate professor in the University of Nottingham Malaysia Campus. His main research fields are identification and nonlinear intelligent control of various systems such as robot arms, automated guided vehicles, and electric drives.

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