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Design and simulation of self-tuning PID-type fuzzy adaptive control for an expert HVAC system

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ABSTRACT

The modelling, numerical simulation and intelligent control of an expert HVAC (heating, ventilating and air-conditioning) system having two different zones with variable flow-rate were performed by considering the ambient temperature in this study. The sub-models of the system were obtained by deriving heat transfer equations of heat loss of two zones by conduction and convection, cooling unit and fan. All models of the variable flow-rate HVAC system were generated by using MATLAB/SIMULINK, and proportional-integral-derivative (PID) parameters were obtained by using Fuzzy sets. For comfortable of people the temperatures of the two different zones were decreased to 5 °C from the ambient temperature. The successful results were obtained by applying self-tuning proportional-integral-derivative (PID)-type fuzzy adaptive controller if comparing with the fuzzy PD-type and the classical PID controller. The obtained results were presented in a graphical form.

1. Introduction

The environmental conditions of human beings always been affect by their work performances. In addition, it is well known that the importance of the comfort of human beings gradually increases by the development of technology (Aereboe, 1995; Zheng & Zaheer-Uddin, 1996). That is why, humans try to design a more comfortable living environment. As a result, the studies related to the design and control of ambient conditions of buildings have been attracting interest in the last years. In a multipurpose building having shopping-center, offices and different usage areas, their desired temperatures and air conditions may be different. For this reason, flexible design of HVAC systems supplying different demands is substantially to decrease both the first investment cost and the operational cost. Further, the analysis of the performance and operational strategies of HVAC systems becomes very important for the effective usage of energy (Ashrae handbook, 1992). The studies on the parameters of HVAC systems as temperature, volume and control strategies in the last 50 years were shown that the high performance of HVAC systems could be obtained by minimizing energy consumption (Stanfort, 1998). Technological developments lead to obtain system models of HVAC systems by simulation programmes in PC media (Ellis, 1996; Geller, 1998).

In this study, it was shown that the usage of modelling and simulation methods for analyzing, testing and developing of HVAC systems decreased the design cost as well as the design process. Furthermore, obtaining better performances of the simulated systems became easier. Soyguder and Alli studied the classical PID control of HVAC system having two zones with different properties (Soyguder & Alli, 2006). k_p, k_i and k_d parameters of PID were obtained to minimize the system error in their study; however, the steady-state error was not totally eliminated.

In addition to PID control of HVAC systems, fuzzy logic control (FLC) of HVAC systems was studied by many authors (Huang & Nelson, 1994a, 1994b). The obtained results were compared with those of PID control and these studies indicated that FLC had better results. FLC is extensively used in processes where systems dynamics is either very complex or exhibit a highly nonlinear characters and FLC is one of the useful control schemes for plants having difficulties in deriving mathematical models or having performance limitations with conventional linear control methods. FLC is designed on the basis of human experience, which means that a mathematical model is not required for controlling a system. Because of this advantage, fuzzy logic-based control schemes were implemented for many industrial applications (Hung, Lin, & Chung, 2007). FLCs were successfully applied to many complex industrial processes and domestic appliances in the recent years (Tsang, 2001). The first FLC algorithm implemented by Mamdani was designed to synthesize the linguistic control protocol of an experienced operator (Mamdani, 1974). Although this type of FLC application has been successful compared to the classical controllers, the design

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procedure depends on the experience and knowledge of the operator and it is limited by the elucidation of the heuristic rules of control. To avoid this major disadvantage of depending on the control experience of the operator, MacVicar-Whelan first proposed some general rules for the structure of fuzzy controllers (Macvicar-Whelan, 1976). These fuzzy rules devised by MacVicar-Whelan approach to a deterministic (PI) or (PD) controller in the limit as quantization levels of control and measurement variables become infinitely fine. It was shown that better results for the same system were obtained by using FLC with respect to PID control (Tang & Mulholland, 1987). However, control rule sets for FLC are quite difficult to redesign.

To eliminate this negative condition, self-tuning FLC can be designed and applied to HVAC systems. The obtained results showed that the self-tuning advanced FLC reached the optimal solution according to the described performance criteria (Huang & Nelson, 1999). In addition to PID control and FLC of HVAC systems, the studies on combining PID and FLC for HVAC systems were performed (Haissig & Woessner, 2000; Huang & Nelson, 1991). Various types of fuzzy PID (including PI and PD) controllers have been proposed. In the literature, the application of FL to PID controller design can be generally classified into two major categories according to design type (Xu, Hung, & Liu, 2000):

(i) The gains of the classical PID controller are tuned on-line in terms of the knowledge base and fuzzy inference, and then the classical PID controller generates the control signal (Zhao & Tomizuka, 1993).
(ii) FLC is designed as a set of heuristic control rules, and the control signal is directly deduced from the knowledge base and the fuzzy inference as it is done in MacVicar-Whelan or diagonal rule-base generation approaches (Xu, Hung, & Liu, 1998).

The second type controllers are referred to as PID-type FLC's because, from the input–output relationship point of view, their structures are analogous to that of the classical PID controller. The equivalence of PD-type FLC's and classical PD controllers has been established under the special conditions (Qiao & Mizumoto, 1996). Different control tuning methodologies have been proposed in the literature such as auto-tuning, self-tuning, pattern recognition, artificial intelligence, and optimization methods (Santos Coelho, 2007). A new approach has been proposed for tuning the coefficients of PID-type fuzzy logic controllers (FLCs) (Güzelkaya, Eksin, & Yeşil, 2003). In addition, Wei proposed a self-tuning method for a class of nonlinear PID control systems based on Lyapunov approach (Changa, Hwangb, & Hsieha, 2002). The self-organizing fuzzy controller has an extension of the rule based fuzzy controller with an additional learning capability (Kazemian, 2001). By relating to the conventional PID control theory, Zhi-Wei proposed a new fuzzy controller structure, called PID type fuzzy controller (Woo, Chung, & Lin, 2000). A fuzzy auto-tuning proportional-integral-derivative (PID) controller for the robot has been experimented, in which a simple method to tune parameters of the PID type fuzzy controller on-line has been developed(Sun, Xing, Zhao, & Huang, in press).

Based on these studies, the modelling of an expert HVAC systems with variable flow-rate has been established by using MATLAB/SIMULINK, and k_p, k_i and k_d parameters of PID have been determined by using self-tuning PID type fuzzy adaptive controller. The performance of the proposed control algorithm has been compared with those of the classical PID and fuzzy-PD type controllers. As a result of the simulations, it has been shown that the performance of the proposed controller is better among the others. The instantaneous time-dependent solution of the system has been obtained for each zone and model by considering the input and output values of each device and the desired comfortable conditions. The temperatures of two different zones for each time-step and the required damper gap rates for supplying the desired comfortable conditions have been found in the result of the numerical simulations. The obtained results have been presented in a graphical form.

The outline of this paper can be summarized as follows: Section 2 describes the elements and structure of an expert HVAC systems. The heat transfer equations used for the numerical model of the HVAC system are obtained in Section 3. Then, Section 4 includes the design of the classical PID, fuzzy PD-type and self-tuning PID-Fuzzy Adaptive control of the considered system. MATLAB/SIMULINK numerical simulations of the HVAC system are presented in Section 5. Finally, conclusions are given in Section 6.

2. Design of an expert HVAC system having two different zones

The cooling process has been performed for two zones having different properties as shown in Fig. 1. The volume of each zone is 0.5 m^3. All surfaces of Zone-1 were covered by the isolation materials (strafor); however, those of Zone-2 were not covered. The purpose of this type of design is to see and determine clearly the steady-state differences to obtain the reference temperatures for two zones having different properties. The cooled air transfer has been realized from the main channel having the supply fun to the regions of Zone-1 and Zone-2. The temperature controls of the two zones were realized by the proposed controllers by regulating the damper gap rates. The air supply fan first absorbs 5 °C air from the evaporator, then sends to the zones as shown in Fig. 1.

The mass flow-rate $(\dot{m}_{ca})$ absorbed from the cooling unit does not change because the supply fun has the constant number of revolution. However, the mass flow-rate of the air entering to the zones changes depending on the temperatures of the zones. The continuous variations of the input mass flow-rate $(\dot{m}_{z1a,in})$ Zone-1 and $(\dot{m}_{z2a,in})$ Zone-2 are realized by regulating the gap rates of dampers into the entrances of zone-channels, depending on the control output signals.

The continuity equation of the controlled system can be formed as

$$\dot{m}_{ca} = \dot{m}_{z1a,in} + \dot{m}_{z2a,in} + \dot{m}_{sva,out}. \quad (1)$$

The mass flow-rate $\dot{m}_{sva,out}$ in Eq. (1) belongs to the safety valve discharging the excessive air coming from the zones.

The variable-input mass flow-rates of the zones are provided by the damper motors. The damper gap rates are changed between 0° and 90°, depending on the zone temperatures. In the beginning of the control process, the damper gap rate for each zone is 90° as a maximum value. The damper gap rates decreasingly become 0° depending on the control parameters when the zone temperatures reach the desired temperatures. The designers should take account that the flow-speed of the air in the zones does not exceed 0.15 m/s because of the comfortable conditions. This condition has been realized by using the exhaust valve mounted to the exit of the zones. The exhaust mass-flow-rate $(\dot{m}_{exha,out})$ has been manually controlled in this study.

3. The model of the HVAC system

The transient and steady-state behaviors, controllability, control performances, design of energy-effective systems and analysis of HVAC systems can be performed by obtaining system models and using simulation tools. Furthermore, the effects of controllers on the controlled system can be examined. In addition to this, system modelling leads to minimize the design process of mechanical systems (Mathews, Arndt, & Geyser, 2002). Fig. 1 shows the

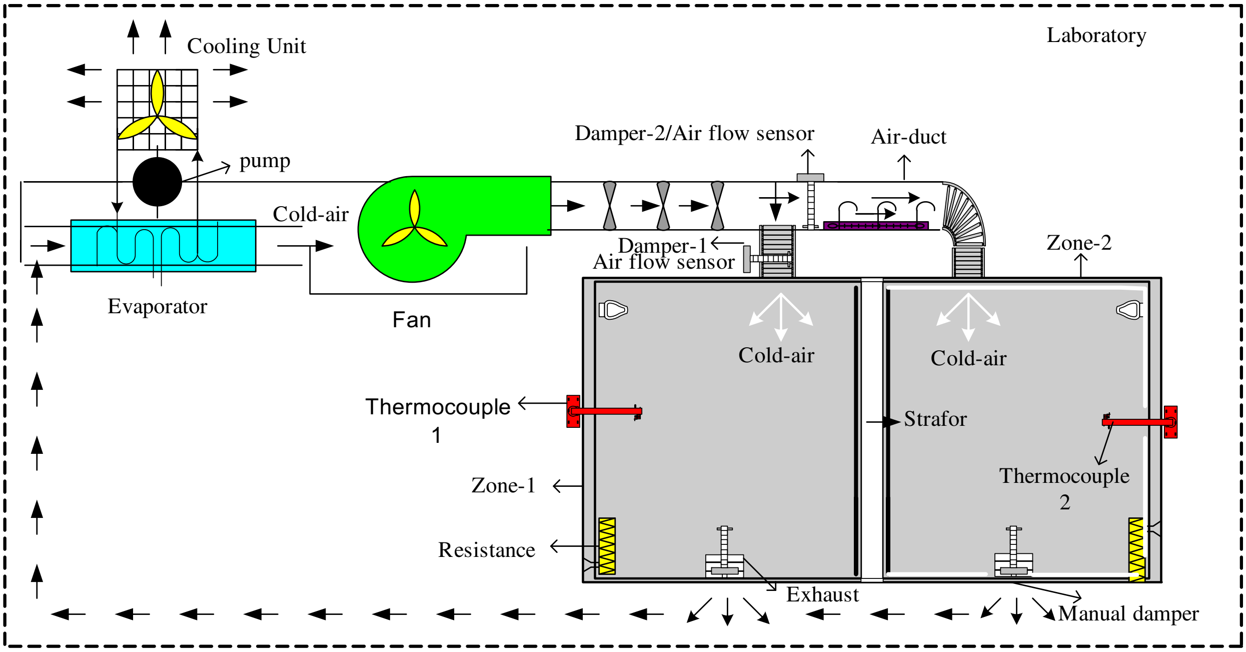

Fig. 1. The schematic view of the HVAC system having two zones.

schematic diagram of the modelled system in this study. The main elements of the system are the cooling zone-areas, evaporator, cooling unit, fan, dampers motors, channels, and thermocouples. We define the symbol list indicated in Table 1.

Obtaining the mathematical model of the cooled zone by considering all parameters is quite difficult. For this reason, we consider the following assumptions:

(1) The effect of the instantaneous variations of air speed in the zones on the pressure is neglected.
(2) There is no air leakage except the exhaust valves of the zones.
(3) The air flow in the zones is homogeneous.

Table 1
Symbol list

A	Area (m^2)
$\dot{m}_{ca}$	The mass flow-rate in fan channel (kg/s)
$\dot{m}_{z1a,in}$	The mass flow-rate entered to Zone-1 (kg/s)
$\dot{m}_{z2a,in}$	The mass flow-rate entered to Zone-2 (kg/s)
$\dot{m}_{sva,out}$	The mass flow-rate exist from safety valve (kg/s)
$\dot{m}_{exha,out}$	The mass flow-rate exist from exhaust (kg/s)
Q	Convection and transmission heat (J)
W	Work (J)
$\dot{m}_{z1a,in} = \dot{m}_{za}$	The mass flow-rate entered to Zone-1 (kg/s)
h_{in}	Specific enthalpy(J/kg)
h_{out}	Specific enthalpy (J/kg)
U	The internal energy (J)
C_v	Constant heat (kJ/kgK)
C_p	Constant pressure (kJ/kgK)
T	İnner temperature (°C)
T_n	İnstant temperature (°C)
T_{n-1}	Vicious circle temperature (°C)
$T_{sh,gir}$	Cool air temperature (°C)
T_{out}	Outside temperature (°C)
h_{out}	Convection coefficient for outside-surface (J/m^{2}K)
h_{in}	Convection coefficient for inner-surface (J/m^{2}K)
k	Transmission coefficient (J/mK)
L_1	Thickness for Zone-1 (m)
L_2	Thickness for Zone-2 (m)

There is no change in the flow-rates of the zones since the input flow-rate equals the output flow-rate. That is why we can write the continuity equation as

$$\dot{m}_{za,in} = \dot{m}_{exha,out} = \dot{m}_z. \tag{2}$$

According to thermodynamic first law, the internal energy equation can be stated as follows:

$$Q - W + \sum \dot{m}_{za,in} \cdot h_{in} - \sum \dot{m}_{exha,out} \cdot h_{out} = \frac{du}{dt}, \tag{3}$$

where u represents time-dependent variation of heat. Furthermore, Eq. (3) can be re-written as the following form, assuming that there is no work in the system:

$$Q + \dot{m}_{za} \cdot (h_{in} - h_{out}) = \frac{du}{dt} = \frac{m_{za} \cdot C_v \cdot (T_{n-1} - T_n)}{dt}, \tag{4}$$

$$h_{in} - h_{out} = C_p \cdot (T_{ca,in} - T_n). \tag{5}$$

If Eq. (4) is rearranged, we get

$$Q + \dot{m}_{za} \cdot C_p . (T_{ca,in} - T_n) = \frac{m_{za} \cdot C_v \cdot (T_{n-1} - T_n)}{dt}, \tag{6}$$

$$Q + \dot{m}_{za} \cdot C_p . (T_{ca,in} - T_n) = m_{za} \cdot C_v \cdot \frac{dT}{dt}, \tag{7}$$

where T represents the instantaneous temperature variation. The heat transfer from the outside to the system can be stated as

$$Q = \frac{T_{out} - T_n}{R} \tag{8}$$

or

$$Q = \frac{T_{out} - T_n}{\frac{1}{h_{out} \cdot A} + \frac{L_1}{k_1 \cdot A} + \frac{L_2}{k_2 \cdot A} + \frac{1}{h_{in} \cdot A}}. \tag{9}$$

If Eq. (9) is substituted in Eq. (8), we get:

$$\frac{dT}{dt} = \frac{Q + \dot{m}_{za} \cdot C_p (T_{ca,in} - T_n)}{m_{za} \cdot C_v}. \tag{10}$$

4. Self-tuning PID-type fuzzy adaptive control

The self-tuning PID-type fuzzy controller is an auto-adaptive controller that is designed by using an incremental fuzzy logic controller to tune the parameters of PID controller on-line by fuzzy control rules. The controller uses the error and the rate of change of error as its inputs and can meet the desire of self-tuning parameters based on time-varying e and $\dot{e}$

$$u(k) = k_p e(k) + k_i \sum^{k} e(i) + k_d[e(k) - e(k-1)], \tag{11}$$

where k_p is the controller gain; $k_d = k_p T / T_i$;

$k_d = k_p T_D / T$; T is the sample time; T_i is the integral time parameter; T_D is the derivative time parameter (Tao, Yin, & Ge, 2001).

Because the proposed fuzzy self-tuning PID controller aims to improve the control performance yielded by a PID controller, it keeps the simple structure of the PID controller and it is not necessary to modify any hardware parts of the original control system for implementation. Fuzzy self-tuning of PID parameters is to find out the fuzzy relation between three parameters of PID and e, $\dot{e}$. It incessantly examines e and $\dot{e}$ in work, then tunes three parameters by fuzzy control rules on-line so that controlled objects achieve better dynamic steady performance.

The self-tuning adaptive control method has been applied to HVAC systems as the other systems. There are many studies on determining the parameters of controllers and finding new values of controller parameters according to changing situations. The optimum control method has been used to find the required-control parameter values that the heater provides the designed temperatures and humidities in HVAC systems (Kasahara et al., 2001).

Table 2
The rule base for fuzzy PD-type control

u (e \ $\dot{e}$)	NB	NS	Z	PS	PB
NB	NB	NB	NS	NS	Z
NS	NB	NS	NS	Z	PS
Z	NS	NS	Z	PS	PS
PS	NS	Z	PS	PS	PB
PB	Z	PS	PS	PB	PB

Tref<0

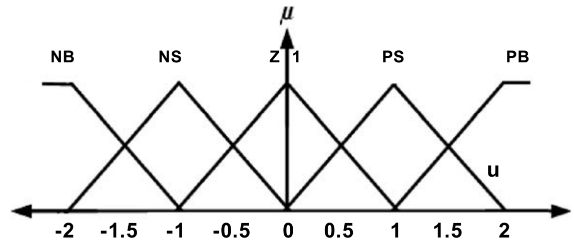

Fig. 4. The membership functions for the output u.

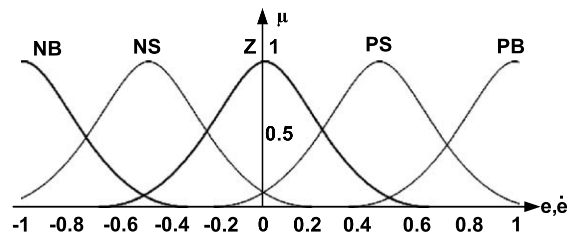

Fig. 5. The membership functions for the inputs e and $\dot{e}$

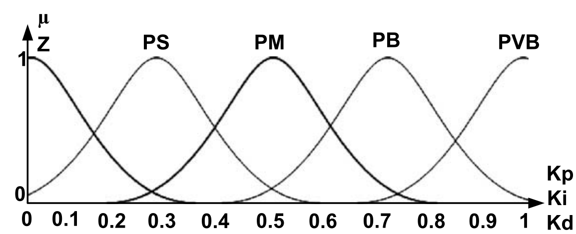

Fig. 6. The membership functions for the outputs k_p–k_d–k_i.

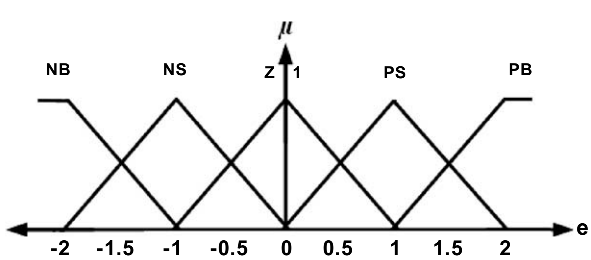

Fig. 2. The membership functions for the input e.

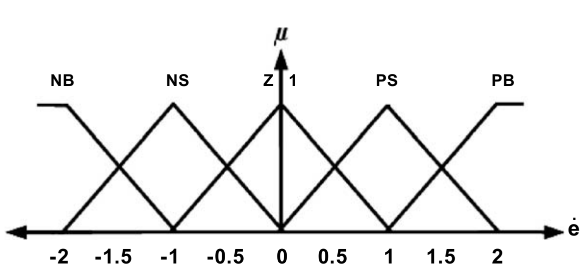

Fig. 3. The membership functions for the input $\dot{e}$.

Table 3
The rule base for k_p

Kp (e \ $\dot{e}$)	NB	NS	Z	PS	PB
NB	PVB	PVB	PVB	PB	PM
NS	PVB	PVB	PB	PB	PM
Z	PB	PB	PM	PS	PS
PS	PM	PS	PS	PS	PS
PB	PS	PS	Z	Z	Z

Tref<0

Table 4
The rule base for k_d

Kd (e \ $\dot{e}$)	NB	NS	Z	PS	PB
NB	Z	Z	PS	PS	PB
NS	Z	Z	Z	Z	PS
Z	Z	Z	Z	PS	PB
PS	PS	PS	PS	PB	Z
PB	Z	Z	Z	PS	PB

Tref<0

Table 5
The rule base for k_i

Ki (e \ $\dot{e}$)	NB	NS	Z	PS	PB
NB	PVB	PB	PM	PM	PM
NS	PVB	PB	PB	PM	PS
Z	PM	PS	Z	Z	Z
PS	PM	PM	PS	Z	Z
PB	PS	Z	Z	Z	Z

Tref<0

First of all, the error and the variation of error depending on the measured T1 and T2 temperatures, which belong to two zones having different properties, are taken as the inputs of fuzzy-PD type controllers. Mamdani's fuzzy inference method is used in the system with two inputs and one output. The output (the control variable), which is the mass flow-rate of the cooled air entering the zones, is determined depending on the defined rule base of e and ($\dot{e}$), with the aim of minimizing the error.

Once we define the rule base indicated in Table 2, we now need to determine the membership functions for e, ($\dot{e}$) and u shown in Figs. 2–4, respectively.

A membership function is a curve that defines how each point in the input space is mapped to a membership value (on degree of membership) between 0 and 1. In this case, the triangular membership functions are used for all variables and {−2, −1.5, −1, −0.5, 0, 0.5, 1, 1.5, 2} physical domain is selected for all variables based on trial and error method.

The fuzzy variables are defined for the rule base as: e, ($\dot{e}$), u = {the error, the variation of error, the control variable {NB (Negative Big), NS (Negative Small), Z (Zero), PS (Positive Small), PB (Positive Big)}, [−2, 2], μ}.

The fuzzy rule base is constructed by using several if-then statements and premise and consequent of each statement which are fuzzy propositions. Table 2 indicates that 25 rules define the rule base for the fuzzy-PD type controller.

For the aim of the comparison of the control performance, the control of the same HVAC system has been realized by applying the self-tuning PID-type fuzzy adaptive controller. We combine the classical PID and FLC theories in this study. The k_p, k_i, k_d values of PID parameters have adaptively been determined by using the dynamic FLC for each time-step. In this case, FLC has two inputs (e, $\dot{e}$) and three outputs (k_p, k_i, k_d).

The gauss membership function is used for all variables shown in Figs. 5 and 6. The physical domain of the inputs (e, $\dot{e}$) is {−1, −0.8, −0.6, −0.4, −0.2, 0, 0.2, 0.4, 0.6, 0.8, 1} and that of the outputs (k_p, k_i, k_d) is {0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9, 1}, selected again based on trial and error. The fuzzy variables are defined for the rule base as: e, ($\dot{e}$) = {the error, the variation of error, {NB (Negative Big), NS (Negative Small), Z (Zero), PS (Positive Small), PB (Positive Big)}, [−1, 1], μ}.

k_p, k_d, k_i = {the control parameters, {Z (Zero), PS (Positive Small), PM (Positive Medium), PB (Positive Big), PVB (Positive Very Big), [0, 1], μ}.

Tables 3–5 indicate the rule bases for k_p, k_d and k_i, respectively.

5. The numerical simulation of the HVAC system

The model, control and numerical simulation of the HVAC system having two zones with different properties have been realized by using MATLAB/SIMULINK package programme. The instantaneous time-dependent solution of the system has been obtained for each zone and model by considering the input and output values of each device and the desired comfortable conditions. The temperatures of two considered zones for each time-step and the required damper gap rates for supplying the desired comfortable conditions have been found in the result of the numerical simulations. The obtained results have been presented in a graphical form.

Figs. 7 and 8 show the block diagrams of the considered expert HVAC system controlled by the fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller, respectively. In these figures, $T1_{ref}$ and $T2_{ref}$ are the desired reference temperatures of Zone-1 and Zone-2, respectively. FLC provides the control signals ($u1$, $u2$) to minimize error depending on e and ($\dot{e}$). The obtained control sig-

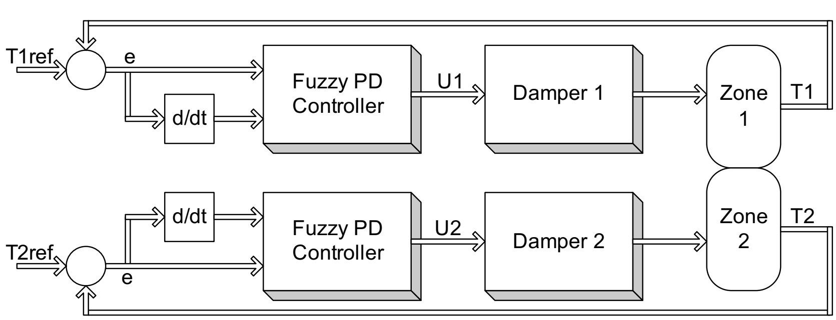

Fig. 7. The block diagram of the considered HVAC system controlled by the fuzzy-PD type controller.

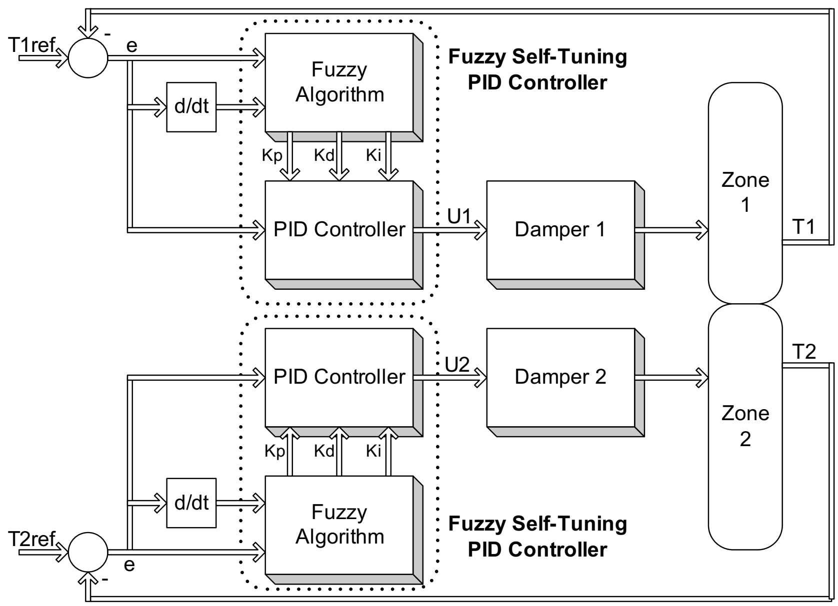

Fig. 8. The block diagram of the self-tuning PID-type fuzzy adaptive controller.

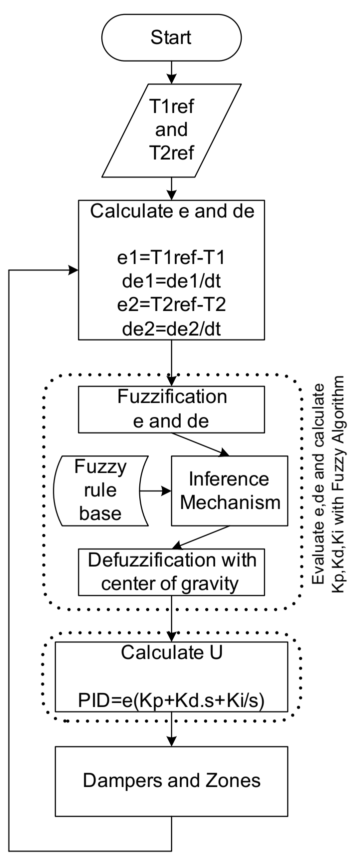

Fig. 9. The flow-chart of the self-tuning PID-type fuzzy adaptive controller.

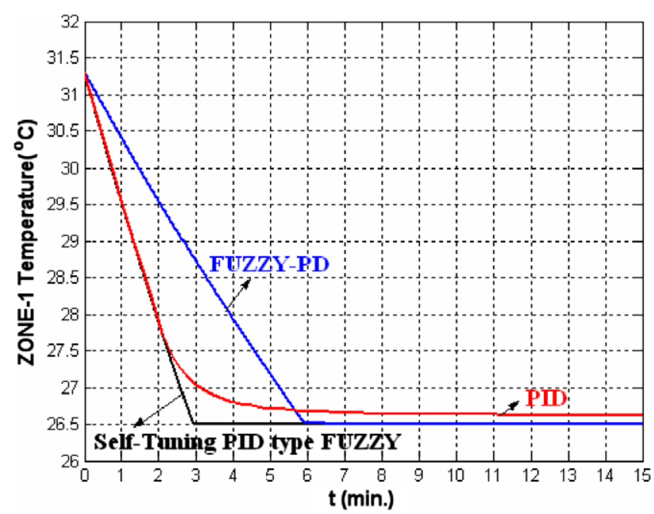

Fig. 10. The temperature variation of Zone-1 when the classical PID, fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller are applied.

nals change the damper gap rates and the air mass flow-rates entered that the zones to reach the desired reference temperatures. Fig. 9 shows the flow-chart of the considered expert HVAC system controlled by the self-tuning PID-type fuzzy adaptive controller.

For numerical simulation, the ambience temperature has been taken as 31.3 °C. The desired reference temperature has been selected as 26.5 °C for Zone-1. Fig. 10 shows the temperature control of Zone-1 when the classical PID, fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller are applied. The performance of the self-tuning PID-type fuzzy adaptive controller is the best among the others in terms of both the steady-state error and the settling time. There is no steady-state error and the system reaches the desired reference temperature with the minimizing settling time. Although the fuzzy-PD type controller has no steady-state error, it has longer settling time. The worst performance belongs to the classical PID controller.

Fig. 11 shows the variation of the damper gap rates for Zone-1 for the considered three controllers. The on and off positions of damper are also shown in this figure. The channel area is 0.02 m^2. The 90° position of the damper is the full open position and the system has the maximum mass flow-rate. The 0° position of the damper is the closed position of the damper and the cooled air cannot pass through the Zone-1.

The desired reference temperature for Zone-2 is 27.5 °C. Fig. 12 shows the temperature variations of Zone-2 when the considered

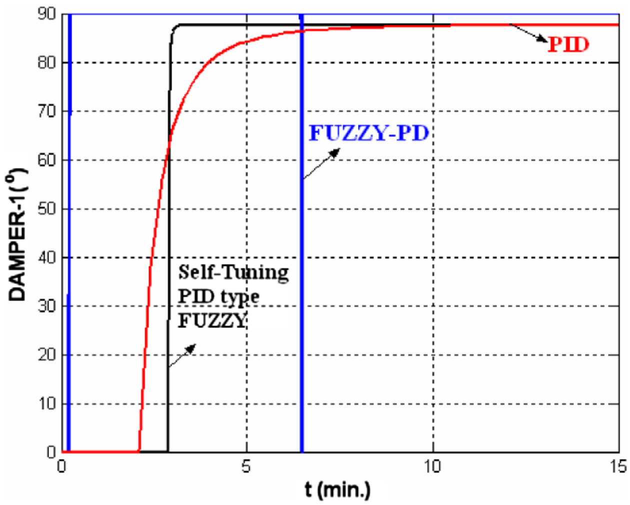

Fig. 11. The damper gap rate variation for Zone-1 when the classical PID, fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller are applied.

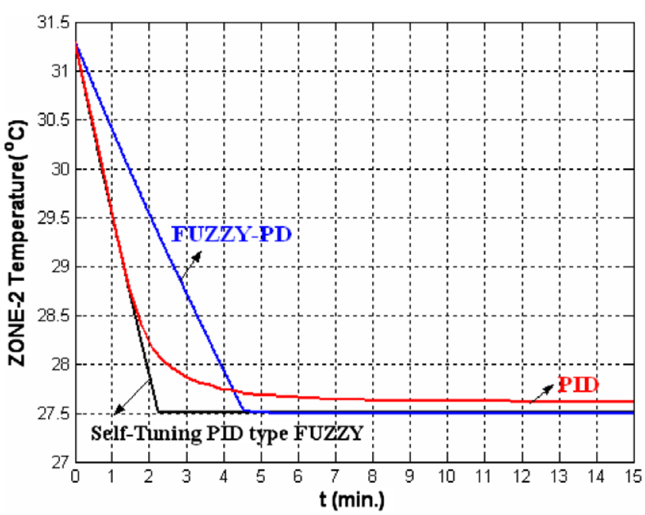

Fig. 12. The temperature variation of Zone-2 when the classical PID, fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller are applied.

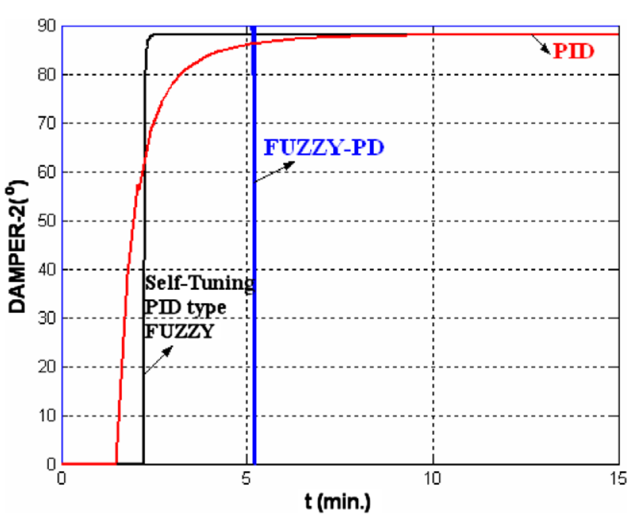

Fig. 13. The damper gap rate variation for Zone-2 when the classical PID, fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller are applied.

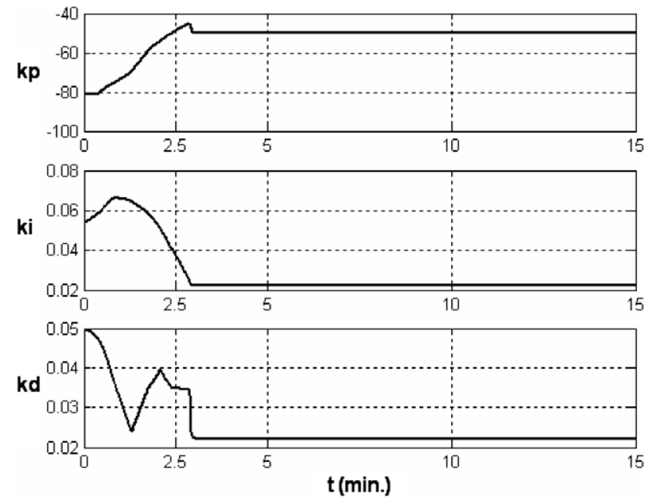

Fig. 14. The variations of k_p–k_i–k_d for Zone-1 when the self-tuning PID-type fuzzy adaptive controller is applied.

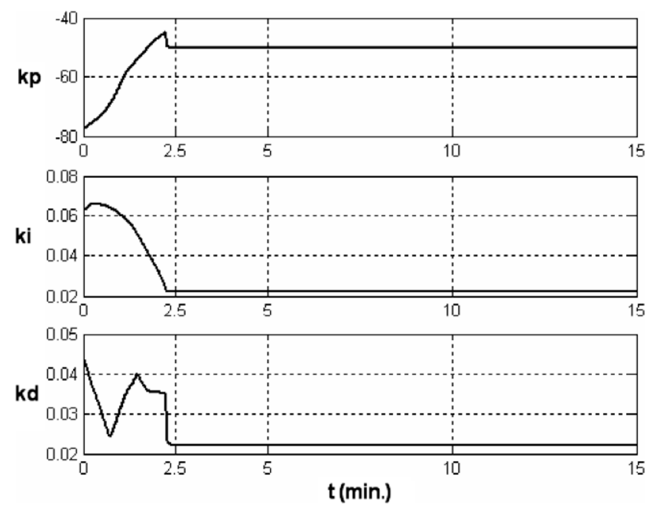

Fig. 15. The variations of k_p–k_i–k_d for Zone2 when the self-tuning PID-type fuzzy adaptive controller is applied.

three controllers are applied. The system has been cooled from 31.3 °C to the desired temperature 27.5 °C. Again, the self-tuning PID-type fuzzy adaptive controller performs the best without the steady-state error and having the minimum settling time if we compare with the others. Fig. 13 indicates the damper gap rate variations for Zone-2. The time histories of the damper positions are clearly seen in this figure. Fig. 14 shows k_p–k_i–k_d variation of Zone-1 when the self-tuning PID-type fuzzy adaptive controller is applied. Further, Fig. 15 shows k_p–k_i–k_d variation of Zone-2 when the self-tuning PID-type fuzzy adaptive controller is applied.

As a result, the self-tuning PID-type fuzzy adaptive controller is more applicable for HVAC systems among the others.

6. Conclusions

The numerical simulation of the cooling process from the ambient temperature to the desired reference temperatures of the HVAC system having two zones with different properties has been realized. The instantaneous time-dependent solution of the considered HVAC system has been found for each zone and model by considering the input and output values of each device and the desired comfortable conditions. The temperatures of two zones and the required damper gap rates for providing the desired conditions for each time-step have been obtained by using the numerical simulations. The numerical simulation of the system applying the classical PID [7], fuzzy-PD type and the self-tuning PID-type fuzzy adaptive controller has been realized by using MATLAB/SIMULINK package programme. The obtained figures indicate that the performance of the self-tuning PID-type fuzzy adaptive controller is the best among the others, in terms of both the steady-state error and the settling time. The proposed controller leads to the considered expert HVAC system having the minimum settlings time and without the steady-state errors. The obtained results make the proposed controller more effective among the others.

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