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Decoupled HVAC System via Non-Linear Decoupling Algorithm to Control the Parameters of Humidity and Temperature through the Adaptive Controller

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Abstract- Control methodologies could achieve better comfort conditions of heating, ventilating and air conditioning (HVAC) systems, however, the application of classical controllers is unsatisfactory as HVAC systems are non-linear and the control variables such as temperature and relative humidity (RH) inside the thermal zone are coupled. The objective of this study is to implement and simulate adaptive control for decoupled HVAC to control temperature and RH. A non-linear decoupling algorithm is used to decouple the HVAC system and an adaptive controller is applied to control the temperature and RH, and improve the transient response of system. The simulation results show that the adaptive controller performance is superior, as compared with classical PID controller. In addition, the steady state set points for temperature and RH are better reached in a shorter period of time

Index Terms- HVAC components, HVAC systems, nonlinear decoupling, adaptive controller

I. Introduction

It is well known that the dynamic performance of a Heating, Ventilation and Air Conditioning (HVAC) system has great impact on power and energy consumption, as well as on indoor air quality. In order to study the system performance at the design stage, it is necessary to obtain approximate mathematical models for system components. In addition, efficient control strategies play an essential role in developing improved energy control systems for buildings. The most important criteria for designing HVAC plants are energy efficiency and indoor climate conditions. An adequate combination of these two criteria demand gives the proper control of the plant. The two main functions of any HVAC system are to provide satisfactory indoor comfortable conditions (temperature and relative humidity) for buildings and housing, for humans and equipment, and, concurrently, minimize the overall energy consumption whilst providing comfortable temperatures and humidity for the occupants [1]. Typically, an HVAC system is used to offset the thermal load (Qz) and moisture load (Mz) in a thermal zone and to maintain desirable set-points for comfort condition parameters such as temperature, RH, CO_2 content, and air velocity. Among the different parameters, temperature and RH have more direct influence on the performance of HVAC systems in nearly all the applications [2, 3].The design of successful controllers for HVAC systems primarily depends on the availability of good dynamic models of the systems and mathematical equations that describe its behavior. In recent years, there has been a growing interest in the mathematical modeling of HVAC systems and its components. Many researchers have studied HVAC dynamic models using either theoretical or experimental approach. Clark et al. [4] derived dynamic models for a duct and a hot water coil. Underwood and Crawford [5] developed an empirical nonlinear model of a hot-water-to-air heat exchanger loop that is used in developing nonlinear control law. This model accurately predicted the effect of inlet air temperature, air flow rate, and inlet water temperature during closed loop control of output air temperature using water flow rate as a control input. Maxwell et al. [6] developed an empirical model of chilled water coil and used it to predict the system response to inputs with Proportional (P), Proportional Integral (PI), and Proportional Integral Derivative (PID) control algorithms. The actual response of chilled water was measured to validate the coil model. They found that the coil model effectively predicted the response at different values of gains for each type of control algorithm. Kasahara et al. [7] described a procedure for deriving a dynamic model of an air-conditioned room by applying physical laws. Riederer et al. [8] developed a room model to study the influence of the sensor position in building thermal control. Since the temperature measured by the sensor of a room temperature controller depends on its position in the zone they obtained a detailed list of criteria for the development of zone models. Peng and Paassen [9] presented the modeling and control of an air conditioning system. The process was decomposed into two subsystems connected in series with natural feedback which are obtained very good and satisfactory results in maintaining the room conditions close to the desired values. Recently Parvaresh et al. [10] presented a mathematical dynamic model for HVAC system components based on Matlab.To enhance the performance of HVAC systems by means of controlling temperature and RH, several studies have been conducted based on simple on/off and proportional (P)-integral (I)-derivative (D) control methodologies, and more complex algorithms such as non-linear, multivariable, AI methodologies as well as their combinations [11, 12]. Classical control techniques such as ON/OFF controllers (thermostats) and PID controllers are still very popular, due to their low cost and ease of tuning and operation [13, 14]. Thus, he most widely used control algorithms for HVAC systems are based on PIDs. However, it must be noted that the PID control methodology as a linear controller, is suitable for linear systems and HVAC system is inherently non-linear [10]. The assumption of linear behavior of the system, including the building envelope components and those of the equipment is usually valid and satisfactory control action may result. However, with the advent of the digital technology and high speed computing hardware that can be imbedded in controllers, it is possible to address the non-linear behavior of HVAC systems by means of more sophisticated control algorithms [15].

Nomenclature	
A_R	area of the roof=9 m^2
A_{w1}	area of the wall (East, West)=9 m^2
A_{w2}	area of the wall (South, North)=12 m^2
Cah	overall thermal capacitance of the air handling unit=4.5 kJ/C
Cd	specific heat of the duct material=0.4187 kJ/kg ◦C
Ch	overall thermal capacitance of the humidifier=0.63 kJ/◦C
Cpa	specific heat of air=1.005 kJ/kg ◦C
Cpw	specific heat of water=4.1868 kJ/kg ◦C
CR	overall thermal capacitance of the roof=80 kJ/C
Cw1	overall thermal capacitance of the wall (East, West)=70 kJ/C
Cw2	overall thermal capacitance of the wall (South, North)=60 kJ/C
Cz	overall thermal capacitance of the zone=47.1 kJ/C
e(t)	error
fsa	volume flow rate of the supply air=0.192 m^3/s
fsw	water flow rate=8.02*10^{-5} m^3/s
h(t)	rate of moisture air produced in the humidifier
hi	heat transfer coefficient inside duct=8.33 W/m^2◦c
ho	heat transfer coefficient in the ambient=16.6 W/m^2◦C
M_d	mass of the duct model=6.404 kg/m
m_s	mass flow rate of the air stream=0.24 kg/s
m_m	total mass flow rate of the mixing air=0.24 kg/s
m_o	mass flow rate of the outdoor air=0.12 kg/s
m_r	mass flow rate of the recalculated air=0.12 kg/s
m_t	mass of tube material kg/m
p(t)	evaporation rate of the occupants=0.08 kg/h
q(t)	heat gains from occupants, and light (W)
Tco	temperature of the air out from the coil (◦C)
Th	supply air temperature (in humidifier) in (◦C)
Tin	temperature in to the duct
Tm	temperature of the air out of the mixing box (◦C)
Tme	temperature measured (◦C)
To	temperature outside=32 ◦C (Summer)=5 ◦C (Winter)
Tout	temperature out from the duct
Tr	temperature of the recalculated air (◦C)
Ts	supply temperature from the Heating coil
Tsa	supply air temperature (◦C)
Tse	temperature output from the sensor (◦C)
Tsi	temperature of supply air (to the humidifier) (◦C)
Tt,o	tube surface temperature (◦C)
Two	return water temperature=10 (◦C)
Twi	supply water temperature (◦C)

Tw1	temperature of the wall (East, West) (◦C)
Tw1	temperature of the wall (South, North) (◦C)
Tz	temperature of the zone (◦C)
Uw1	overall heat transfer coefficient of (East, West) walls=2W/m^2 ◦C
Uw2	overall heat transfer coefficient of (South, North) walls=2W/m^2 ◦C
UR	overall heat transfer coefficient of the roof= W/m^2◦C
(UA)ah	overall transmittance area factor of the air handling unit=0.04 kJ/sC
Va	volume of the air handling unit=0.88 m^3
Vh	volume of humidifier=0.44 m^3
Vz	volume of the zoneZ36 m^3
Wco	humidity ratio of the air out from the coil (kg/kg dry air)
Wh	supply air humidity ratio (in humidifier) in kg/kg(dry air)
Wm	humidity ratio of the air out the mixing box (kg/kg dry air)
Wo	humidity ratio outside=0.02744 kg/kg (dry air) (Summer)=0.002 kg/kg(dryair)(Winter)
Wsa	humidity ratio of the supply air in kg/kg (dry air)
Wsi	humidity ratio of the supply air (to the humidifier) in kg/kg (dry air)
Wz	humidity ratio of the zone in kg/kg (dry air)
Subscripts	
a	air
ah	air handling unit
ai	air in
ao	air out
co	out from the coil
d	duct
h	humidifier
in	in
m	mixed
me	measured
o	out
R	roof
r	recirculated
s	supply
sa	supply air
se	sensor
sw	supply water
w	water
W1	East, and West walls
W2	North, South walls
wi	water in
wo	water out
z	zone

Greek letters	
ah	(UA)h overall transmittance area factor of the humidifierZ0.0183 kJ/s ◦C
tse	time constant of the sensor (seconds)
ra	density of airZ1.25 kg/m^3
rw	density of waterZ998 kg/m^3

Temperature and RH controls, on an individual basis, have counter effects on each other [16, 17]. In a numerical study, Cui et al. [16] the simultaneous and separate control of temperature and RH in a thermal zone were investigated. Moreover, in the simultaneous control, setting the controller parameters is difficult and, due to thermal coupling between RH and temperature, the response of the system oscillates. As a result of that study, the coupling effects of RH and temperature should be considered, when it is desired to control such parameters simultaneously. There are two approaches to address the decoupling of temperature and RH. First, the coupling behavior between temperature and RH can be overcome by utilizing a decoupling algorithm, when a control law is developed. For that purpose, control methodologies such as multivariable or intelligent control methods can be used. Becker et al. [18] designed a fuzzy controller to regulate the temperature and RH in a cold store by studying the coupling behavior of temperature and RH in fuzzy controller actions. The rule-based control methodology solved the coupling problem of temperature and RH directly, and the proposed fuzzy controller could control the system under disturbances and changes in setpoints efficiently. To avoid interaction between temperature and moisture content responses, an experimental study Qi and Deng [19] which temperature and moisture content have been controlled simultaneously by varying the speeds of both compressor and supply fan in a direct expansion air conditioning system was carried out. In that study, the controller was constructed by using the linearized model of the direct expansion air conditioning system, and LQG method was used to design the controller. Although the method in that study appeared to ge straightforward for solving the problem, but when the setpoint of temperature in the thermal zone changed, the moisture content of the thermal zone experienced some fluctuations and it nearly settled at setpoint after 3000 s. In the second approach for decoupling, distinct channels for controlling the temperature and RH are developed where they are controlled individually through single input single output (SISO) channels. In addition, non-linear decoupling control algorithm can control the temperature and RH individually [17]. In that study, the error between thermal zone temperature and the setpoint was input to a PD controller for determination of the control law that is used in conjunction with the same for RH by the decoupling algorithm for computing the final values of the controlled variables, namely, flow rate of air (fa) and flow rate of water (fw) in each time step. The controlled variables are then used to solve the differential equations for finding the thermal zone temperature and RH responses. The aim of this study is to apply the adaptive control methodology for the HVAC system introduced by Parvaresh et al. [10] which will be decoupled via decoupling method. The next sections will present the methodology section which includes system model, the non-interactive method, step response and adaptive control is presented. The simulation results and conclusions are provided in sections 3 and section 4 respectively.

II. METHODOLOGY AND CONTROL ALGORITHMS

2.1. System modeling

The system modeled in this study is simply referred to as HVAC system, serving a single thermal zone as shown in Figure1 [10]. The new and complete mathematical dynamic model of HVAC components such as heating/cooling coil, humidifier, mixing box, ducts and sensors is described by Bourhan et al. [20]. All of these components are proposed and simulated in Matlab/Simulink platform are given by Parvaresh et al. [10]. The proposed model is presented in terms of energy mass balance equations for each HVAC component. Two control loops for this model, namely, temperature control loop and humidity ratio control loop are considered. Initially, fresh air enters the system and mixes with 50% of the return air, while the remaining air is passed through the heating coil and humidifier. Next, the mixed air is delivered to a heating coil where it is conditioned according to a desired setpoint by a draw-through fan. After that, the air supplied passes through the humidifier and condition according to desired setpoint through the duct. The system controller simultaneously varies f_{sa} and f_{sw} according to load changes, so that the desired setpoints in temperature and RH, as control variables, are maintained. The variables are defined in the nomenclature. The differential equations formulated based on energy and mass balances for the system of Figure 1 are given by bourhan et al. [20].

$$c_{ah}\frac{dT_{co}}{dt} = f_{sw}\rho_w c_{pw}(T_{wi} - T_{wo}) + (UA)_a(T_o - T_{co}) + f_{sa}\rho_a c_{pa}(T_m - T_{co}) \tag{1}$$

$$c_h\frac{dT_h}{dt} = f_{sa}c_{pa}(T_{si} - T_h) + \alpha_h(T_o - T_h) \tag{2}$$

$$v_h\frac{dw_h}{dt} = f_{sa}(w_{si} - w_h) + \frac{h(t)}{\rho_a} \tag{3}$$

Equations (1) and (3) show that the humidity and heating coil of HAVC system have interconnections. Equations (1)–(3) describe the transient mass and energy balances for the heating coil and humidifier model. Equation (1) indicates that the rate of change of energy in the air passes through the coil is equal to the energy added by the flow rate of water in the heating coil, and the energy transferred by the return air to the surrounding.

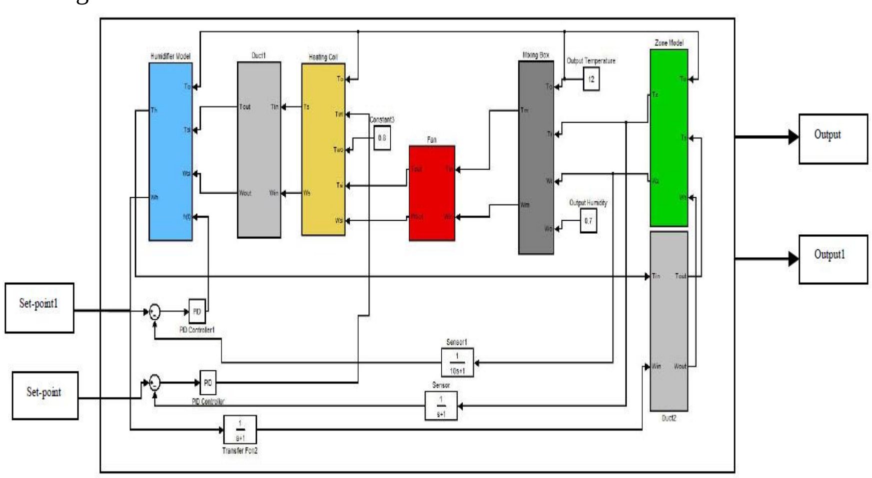

Figure1: HVAC system [10]

In equation (3), h(t) is the rate of humid air that the humidifier can produce and it is a function of humidity ratio. The revised differential equations in the system can be expressed in the state space form [17] as follows:

$$\dot{x}_1 = g_{11}(x).u_1 + g_{21}(x).u_2 \tag{4}$$

$$\dot{x}_2 = g_{12}(x).u_1 \tag{5}$$

$$\dot{x}_3 = f_3(x) + g_{13}(x).u_1 \tag{6}$$

Where $u_1 = f_{sa}$, $u_2 = f_{sw}$, $g_{11}(x) = (\rho_a c_{\rho a} / c_{ah})(T_m - x_1)$, $g_{21}(x) = (\rho w_c \rho w / c_{ah})(T_{wi} - T_{wo})$, $g_{13}(x) = (w_{si} - x_3)/v_h$, $g_{12}(x) = (w_m - x_2)/v_{ah}$, $f_3(x) = h(t)/\rho_a vh$, x_1=T, x_2=ψ_1, x_3=ψ_2

$$\dot{x} = \begin{bmatrix} 0 \\ 0 \\ f_3(x) \end{bmatrix} + \begin{bmatrix} g_{11}(x) \\ g_{12}(x) \\ g_{13}(x) \end{bmatrix} u_1 + \begin{bmatrix} g_{21}(x) \\ 0 \\ 0 \end{bmatrix} u_2 \tag{7}$$

$$y_1 = h_1(x) = x_1$$

$$y_1 = h_2(x) = x_3$$

Note that, the HVAC system model described by [10] is nonlinear as the multiplication of control and controlled variables are presented.

2.2. Non-interactive method

Different methods such as pole placement, linear quadratic, linear quadratic gaussian (LQG), and decoupling or non-interactive method can be used to control the multi-input multi-output (MIMO) systems. In the non-interactive control method, the feedback is used in order to transform the MIMO system, from an input–output point of view to an aggregate of independent single input SISO channels. It is observed from the equations (4)–(6) that when it becomes necessary to make a change in one control variable, namely ψ_2, by means of modulating a controlled variable, namely fa, the other control variables, namely ψ_1 and T, are also forced to change. As a result, achieving desired setpoints for both control variables becomes impossible and implementing a non-linear decoupling procedure is necessary. The non-interacting control law is obtained by non-linear decoupling theory resulting in the following expression [17]. The first step to apply the nonlinear decoupling theory is to determine the vector relative degree:

$$\left.\begin{aligned} L_{g_1} h_1(x) &= g_{11} \\ L_{g_2} h_1(x) &= g_{21} \\ L_{g_1} h_2(x) &= g_{13} \\ L_{g_2} h_2(x) &= 0 \end{aligned}\right. \longrightarrow \quad A(x) = \begin{bmatrix} g_{11} & g_{21} \\ g_{13} & 0 \end{bmatrix} \tag{8}$$

$$V = -A^{-1}(x)B(x) + A^{-1}(x)\bar{V} \tag{9}$$

$$V = A^{-1}(x)\left[\bar{V} - B(x)\right] \tag{10}$$

$$B(x) = \begin{bmatrix} B_1(x) \\ B_2(x) \end{bmatrix} = \begin{bmatrix} L_f{}^2 h_1(x) \\ L_f{}^2 h_2(x) \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix} \tag{11}$$

$$V = (1/-g_{11}g_{13}) \begin{bmatrix} 0 & -g_{13} \\ -g_{21} & g_{11} \end{bmatrix} \begin{bmatrix} \bar{V}_1 \\ \bar{V}_2 \end{bmatrix} \tag{12}$$

The application of the previous control law system in equation (7) results in a closed-loop system with a new set of coordinates set $z(x) \epsilon R^4$ as follows:

$$\dot{z}_1 = z_2 \tag{13}$$

$$\dot{z}_2 = \bar{v}_1 \tag{14}$$

$$\dot{z}_3 = z_4 \tag{15}$$

$$\dot{z}_4 = \bar{v}_2 \tag{16}$$

Where the revised controlled variables $\bar{v}_1$ and $\bar{v}_2$ are capable of controlling ψ_1, and T, respectively, on individual basis, the new states are described below:

$$z_1 = h_1(x) = x_1 \tag{17}$$

$$z_2 = g_{11} \tag{18}$$

$$z_3 = h_2(x) = x_3 \tag{19}$$

$$z_4 = g_{13} \tag{20}$$

The closed-loop system resulting from the application of the decoupling HVAC system is shown in Figure 2.

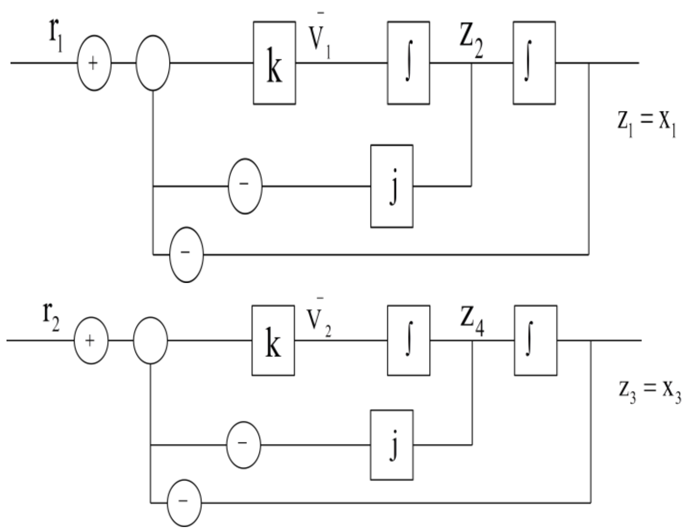

Figure 2: Closed-loop of the linearized decoupled system

2.3. Step response method

The step response of linearized decoupled system and original model of HVAC system is considered in Figures 3 (a-d) and Figures 4 (a-d) below.

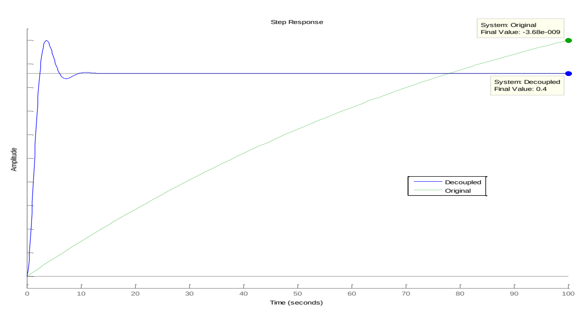

Figure 3(a): Final value of decoupled and original system

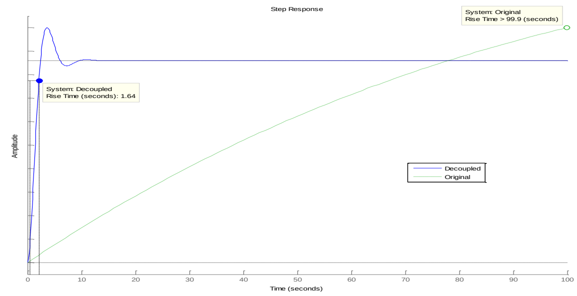

Figure 3(b): Rise time of decoupled and original system

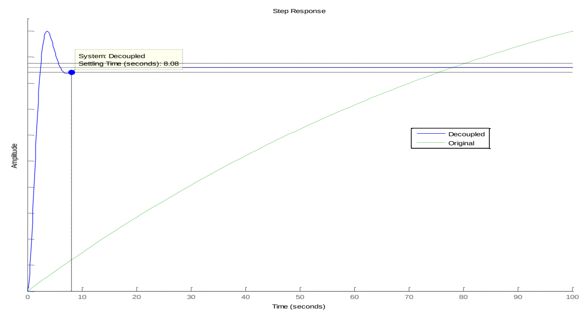

Figure 3(c): Setting time of Original and decoupled system

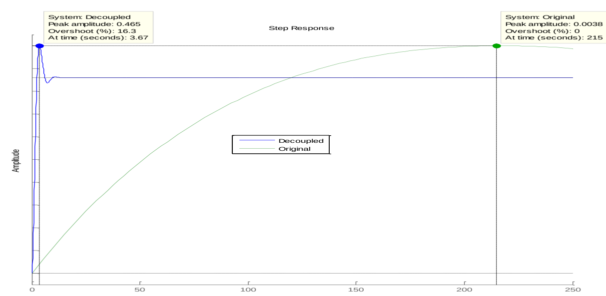

Figure 3(d): Peak time of Original and decoupled system

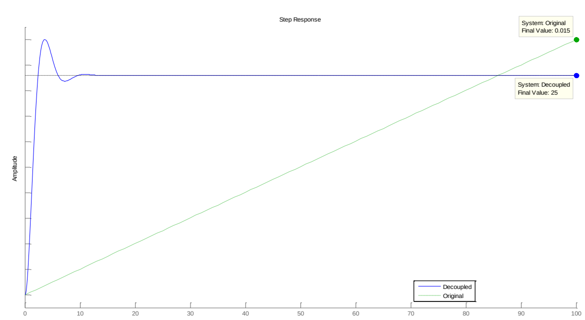

Figure 4(a): Final value of decoupled and original system

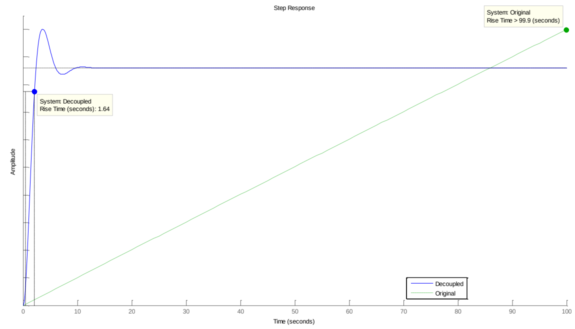

Figure 4(b): Rise time of decoupled and original system

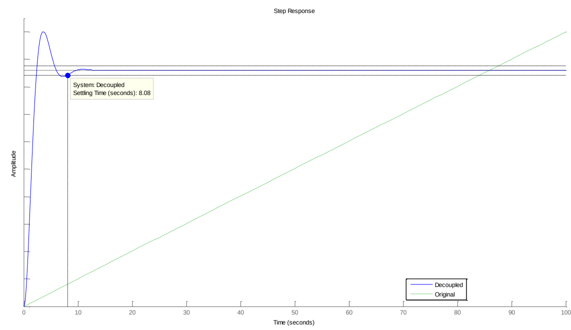

Figure 4(c): Settling time of original and decoupled system

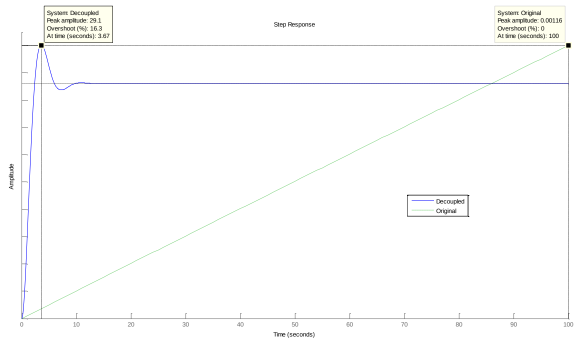

Figure 4(d): Peak time of original and decoupled system

Some characteristics of the normalized step response of humidity and temperature for decoupled and original HVAC system are tabulated in Table I and Table II.

Table I. Step response characteristics of decoupled and original system for humidity

Model	Peak time/amplitude	Rise time	Settling time	Final value	IEA
Original	215/0.003	99.9	-----	-3.6-e009	127.6
Decoupled	3.6/0.46	1.64	8.08	0.4	0.66

Table II. Step response characteristics of decoupled and original system for temperature

Model	Peak time/amplitude	Rise time	Settling time	Final value	IEA
Original	100/0.001	99.9	----	0.015	1.20e004
Decoupled	29.1/3.67	1.64	8.08	25	39.09

By comparing the original and decoupled system, it is clear that the decoupled system is a better option than the use of the original system. In addition it is found that the decoupled system has a better response to the step response which means the decoupled system could be used instead of the original system. A comparison of PID controller for temperature and humidity of decoupled and original HVAC system are illustrated in Figure 5 and Figure 6.

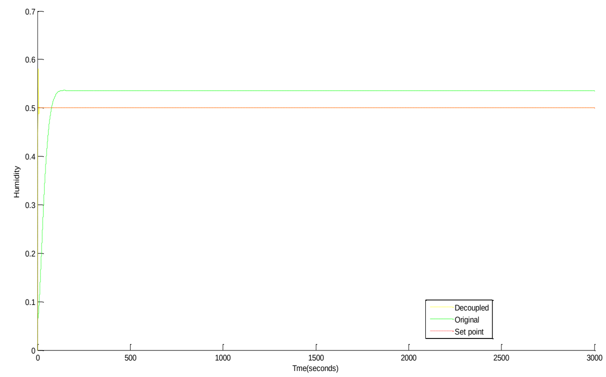

Figure 5: PID comparison of humidity for decoupled and original HVAC system

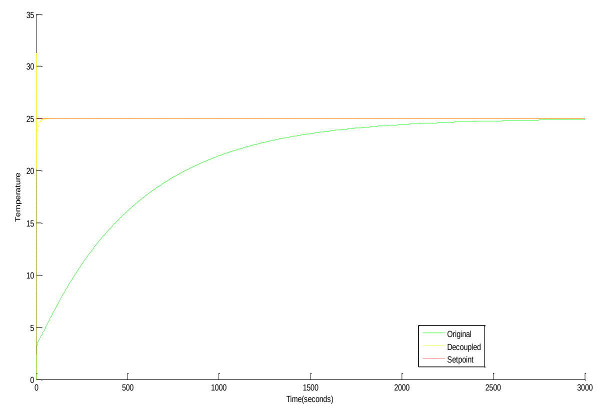

Figure 6: PID comparison of original and decoupled HVAC system

In the control system, the tracking of the system is very important. By checking the step response of original and decoupled system, it is noted that the decoupled system can be used instead of the original one. As discussed in Beghi and Cecchinato [21], adaptive controllers are best suited to meet the challenge of reducing the overall energy consumption heating and cooling of buildings; therefore, adaptive control is used to improve the transient response of the system as well as decrease the amount of error. In the following sections, the adaptive control algorithm is used to control the parameter of the HVAC system.

2.4. Adaptive control algorithm

Advanced control systems that are able to efficiently track the actual cooling/heating power requests from the plant, such as predictive or adaptive controllers, are best suited to meet the challenge of reducing the overall energy consumption for building heating and cooling [21]. One of the earliest works by Farris and McDonald [22] apply adaptive control for HVAC&R systems focused on DDC for solar-heated buildings, with a single-zone air space and room air temperature as the output of the system. In one of the studies, an adaptive optimal control (AOC) strategy was designed using a linearized model of the original nonlinear HVAC&R system and closed-loop optimal obtained via the matrix Riccati equation in [23]. Cao et al. [24] described an adaptive control as a type of controller that has the ability to adjust itself to any parameter variations occurring in a control system. Beghi and Cecchinato [21] has mentioned that predictive or adaptive controllers are best suited to meet the challenge of reducing the overall energy consumption for building heating and cooling. Using zone temperature and hot water temperature as the two state variables, and heat pump input as the control variable, an adaptive control strategy [25] was applied to a discharge air temperature model [26] for the discharge air temperature to track the optimal reference temperature in the presence of disturbances. Temperature and RH controls, on individual basis, have counter effects on each other [16, 17]. Another class of adaptive systems, known as model-following or model-reference adaptive control (MRAC) was applied to a VAV system with zone, coil, and water temperatures as the three state variables; mass flow rate of supply air, mass flow rate of chilled water, and input energy to the chiller as the three control variables; and a second-order model as the reference model for the VAV system. The simulations showed good adaptability of the actual zone temperature with its reference value. Figure7, presents a schematic of the MRAC used in the model. The MIT rule is used to control the parameters of the HVAC system because it is relatively simple and easy to use. The enhancement in efficiency of the HVAC system is accomplished due to improvement of transient response, when the MRAC is used to control the decupled HVAC system. The goal is to minimize the error ($e = y - y_m$) by designing a controller that has one or more adjustable parameters to minimize certain cost functions [$j(\theta) = 1/2e^2$]. Therefore the output of the closed-loop system (y) followed the output of the reference model (y_m).

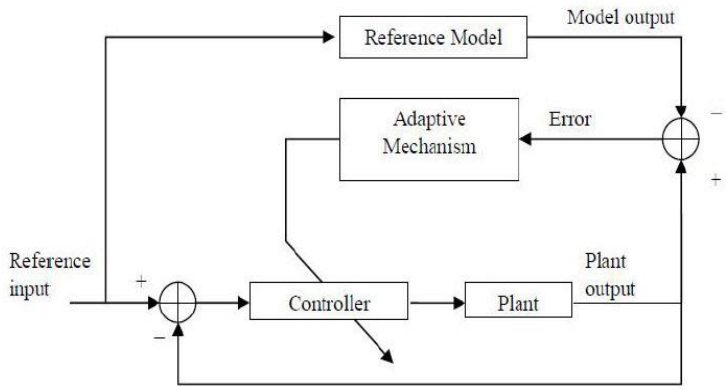

Figure 7: Diagram of MRAC[27]

2.4.1. Adaptive mechanism

The first step is to extract Y_r. The next step is to find the error between Y and Y_r. The controller is described using equation 21:

$$u = \theta_1 r - \theta_2 y \quad (21)$$

The cost function is determined using the following equations:

$$\partial\theta / \partial t = -(\gamma) * \partial j / \partial\theta \quad (22)$$

$$j(\theta) = 1/2 * e^2 \quad (23)$$

The parameter θ is adjusted to enable the loss function to be minimized. Therefore, it is reasonable to change the parameter in the direction of the negative gradient of j, i.e.

$$\partial\theta / \partial t = -(\gamma) * \partial j / \partial e * \partial e / \partial\theta = -(\gamma) * e * \partial e / \partial\theta \quad (24)$$

– Change in γ is proportional to negative gradient of J

The second order system is calculated using equation (24):

$$G_p = y / u = [k_p / s^2 + as + b]u \quad (25)$$

When the first equation (21) is replaced by equation (25):

$$y = [k_p / s^2 + as + b] * (\theta_1 r - \theta_2 y) \quad (26)$$

Then,

$$y[1 + (\theta_2 k_p) / s^2 + a_1 s + a_2] = \theta_1 r (k_p / s^2 + a_1 s + a_2) \quad (27)$$

$$y = \theta_1 (k_p) r / s^2 + a_1 s + (a_2 + \theta_2 k_p) \quad (28)$$

MRAC tries to reduce the error between the model and plant as shown in equations 29 to 37:

$$e = y - y_r \quad (29)$$

$$e = \theta_1 (k_p) r / s^2 + a_1 s + a_2 + \theta_2 k_p - G_m r \quad (30)$$

$$\partial e / \partial\theta_1 = (k_p) r / s^2 + a_1 s + a_2 + \theta_2 k_p \quad (31)$$

$$\partial e / \partial\theta_2 = -\theta_1 (k_p{}^2) r / [s^2 + a_1 s + a_2 + \theta_2 k_p]^2 \quad (32)$$

$$s^2 + a_1 s + (a_2 + \theta_2 k_p) \approx s^2 + A_1 s + A_2 \quad (33)$$

$$\partial e / \partial \theta_1 = [(k_p / k_m)*(k_m)]r / (s^2 + A_1 s + A_2) \quad (34)$$

$$\partial e / \partial \theta_2 = (-[(k_p / k_m)*(k_m)] / s^2 + A_1 s + A_2)*y \quad (35)$$

$$a_2 + \theta_2 k_p = A_2 \rightarrow A_2 - a_2 / k_p \quad (36)$$

$$y = y_m \rightarrow k_m r / s^2 + A_1 s + A_2 = \theta_1 (k_p) r / s^2 + a_1 s + a_2 + \theta_2 k_p \quad (37)$$

Controller parameters are chosen as

$$\begin{cases} k_m r = \theta_1 (k_p) r \rightarrow \theta_1 = k_m / k_p \\ a_2 + \theta_2 k_p = A_2 \rightarrow \theta_2 = A_2 - a_2 / k_p \end{cases}$$

Using MIT rule

So $\partial \theta_1 / \partial t = -(\gamma)*e*\partial e / \partial \theta_1 = -(\gamma)*e*[(k_p / k_m)*(k_m)]r / (s^2 + A_1 s + A_2)$

$$\partial \theta_1 / \partial t = -(\gamma)*e*(k_p / k_m)*y_r = -\gamma^{'} *e*y_r \quad (38)$$

Where $\gamma^{'} = (\gamma)*(k_p / k_m)$ =Adaptation gian (39)

$$\partial \theta_2 / \partial t = -(\gamma)*e*-[(k_p / k_m)]*y_r / r*y \quad (40)$$

$$\partial \theta_2 / \partial t = \gamma^{'} *e*(k_m) / s^2 + A_1 s + A_2 *y \quad (41)$$

Considering a =1, b = 1 and A_1 =1, A_2 = 1

2.4.2. Adaptive control of decoupled HVAC system

The decoupled HVAC system, described by the differential equations in formulation (13) – (16), is controlled by adaptive controller. The adaptive mechanism-1 and adaptive mechanism-2 with training procedure discussed in the previous section are used to control the temperature and humidity of the system respectively. According to the block diagram shown in Figure 8, the decoupled HVAC system is controlled by two distinct adaptive controllers. The adaptive mechanism1 is responsible for ψ and the adaptive mechanism-2 is responsible for T.

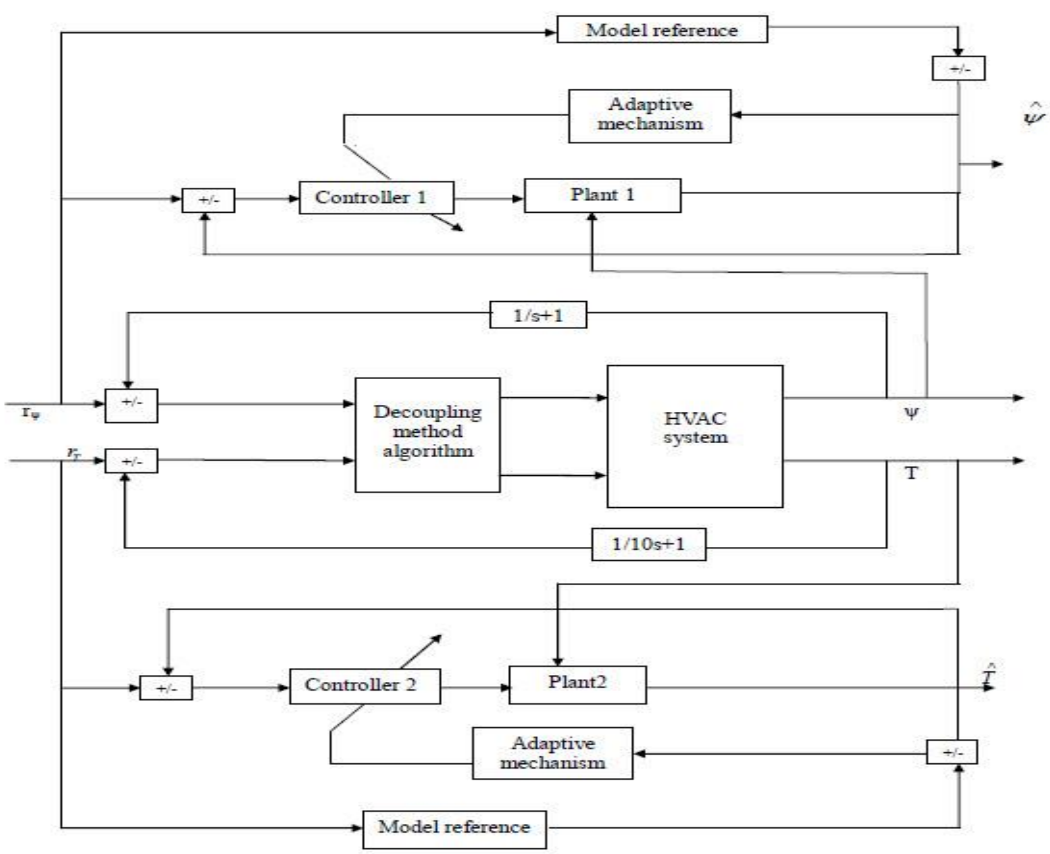

Figure 8: Close loop of decoupled HVAC system

III. SIMULATION RESULT

Figure 9 and Figure 10 show the simulation results for the transient behavior of the temperature and RH in the thermal zone, when classical PID controller and adaptive controller are used. According to the simulation results, adaptive controller provides smooth and fast responses without any large overshoots and undershoots. The target values for the humidity and temperature are 0.4 and 25 C̊, respectively.

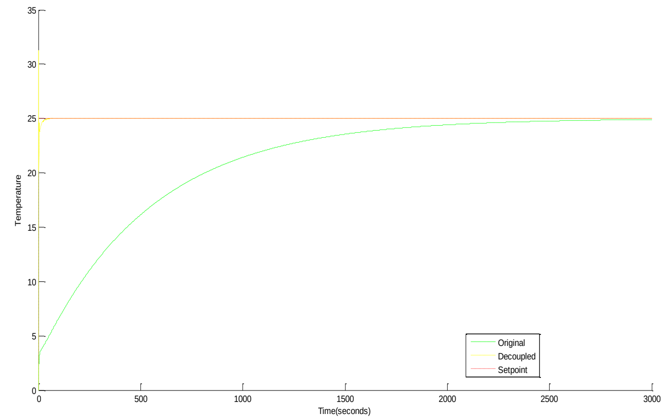

Figure 9: Comparison of PID and adaptive control for temperature

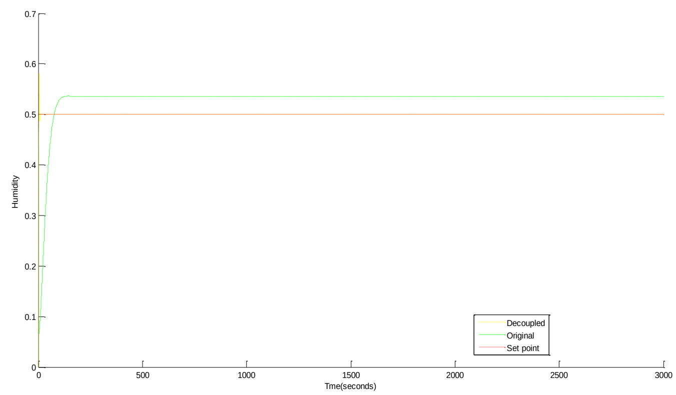

Figure 10: Comparison of PID and adaptive control for humidity

Table III shows the IAE amounts for temperature and humidity of the original and decoupled HVAC system. The amount of IAE indicated that the decoupled model can reach target values and follow them very well. In addition, the enhancement in efficiency of the decoupled HVAC system is accomplished due to improvement of transient operation.

Table III Amount of IAE of humidity and temperature of the simplified and original HVAC system

Controller	IAE of humidity	IAE of Temperature
PID	127.6	1.205+e004
MRAC	2.8	44

IV. CONCLUSION

In this study, a procedure for deriving a dynamic model to control the parameters of HVAC system is investigated. Even though the behavior of the HVAC system is nonlinear, in this study, the use of nonlinear decoupling method for decoupling of the HVAC system is used. It is expected that the decoupled system could improve the humidity and temperature of the HVAC system because it reduces the complexity of the system. The step response of decoupled models was compared with the original system and it is found that the decoupled model could be used to control the parameters of the HVAC system very well. It is clear that the transient response of the decoupled system with the original system is reduced and the amount of error is decreased. For tracking of the setpoint, model reference adaptive controllers have been used to improve the transient behavior of the system. The results obtained show that the system is capable of following the setpoints effectively with minimal error and within a shorter time. The comparison between the reference, existing, and adaptive solutions for the real HVAC system yielded significant improvement of transient behavior of the system. It is concluded that decoupling control law works satisfactorily and as a result, the performance of the decoupled HVAC system is enhanced. Two cases are considered in this study. In case 1, decoupling method is implemented in HVAC system and classical PID controllers are applied compare the parameters of original and decoupled HVAC system. In case 2, for improving the transient response and track the target values MRAC is used and compare with PID controller.

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