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ENHANCEMENT OF CONTROL'S PARAMETER OF DECOUPLED HVAC SYSTEM VIA ADAPTIVE CONTROLLER THROUGH THE SYSTEM IDENTIFICATION TOOL BOX

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Seyed Mohammad Attaran[a], Rubiyah Yusof[b*], Hazlina Selamat[a]

[a]Center for Artificial Intelligence & Robotics, Electrical Engineering Faculty, Universiti Teknologi Malaysia, 54100 Kuala Lumpur, Malaysia
[b]Universiti Teknologi Malaysia Jalan Semarak 54100, Bangunan-Malaysia Japan International Institute of Technology (MJIIT), Malaysia

*Corresponding author
rubiyah.kl@utm.my

Abstract

Heating, Ventilating and Air Conditioning (HVAC) systems have nonlinear character and nature. Current models for control components and the optimization of HVAC system parameters can be linear approximations based on an operating or activation point, or alternatively, highly complex nonlinear estimations. This duality creates problems when the systems are used with real time applications. The two parameters temperature and relative humidity (RH) have a more direct effect in most applications of HVAC systems than the execution. This study's objective is to implement and simulate an adaptive controller for decoupled bi-linear HVAC systems for the purpose of controlling the temperature and RH in a thermal zone. The contribution of this study is to apply the adaptive controller for the decoupled bi linear HVAC system via relative gain array (RGA). To achieve this objective, we used a system identification toolbox to increase the speed and accuracy of the identification of system dynamics, as was required for simplification and decoupled HVAC systems. The method of decoupling is relative gain array. The results of the simulation show that when compared with a classical PID controller, the adaptive controller performance is superior, owing to the high efficiency with which the steady state set points for temperature and RH are reached.

Keywords: HVAC system, PID controller, RGA method, decoupling method

Abstrak

Sistem pemanasan, pengalihudaraan dan penyaman udara atau dikenali sebagai "HVAC" adalah satu sistem yang mempunyai sifat yang tidak linear. Di dalam model yang terkini, terdiri daripada komponen untuk kawalan dan juga mengoptimumkan parameter dalam sistem "HVAC" ia dapat dilinearkan melalui proses pengoperasian, titik pengaktifan ataupun melalui proses penganggaran sistem tidak linear yang kompleks. Oleh itu, masalah yang timbul dari sini berlaku apabila sistem ini digunakan di dalam aplikasi sebenar. Terdapat 2 jenis parameter iaitu suhu dan kelembapan relatif yang dipengaruhi didalam setiap aplikasi sistem "HVAC". Sehubungan dengan itu, objektif didalam kajian ini adalah dengan melaksanakan dan mensimulasikan satu alat kawalan penyesuaian bagi pengasingan bilinear sistem "HVAC" bagi tujuan mengawal suhu dan juga kelembapan relatif didalam zon terma. Makanya, sumbangan didalam kajian ini adalah dengan mengaplikasikan alat kawalan penyesuaian bagi pengasingan bilinear "HVAC" sistem melalui tatasusunan ganda relatif atau dikenali sebagai "RGA". Bagi mencapai objektif di atas, kami menggunakan satu kotak alat sistem pengecaman untuk meningkatkan kadar kecepatan dan kejituan pengecaman sistem dinamik, sebagai salah satu keperluan untuk pemudahan dan pengasingan sistem "HVAC". Kaedah bagi pengasingan ini dikenali sebagai tatasusunan ganda relatif. Keputusan yang dihasilkan akan dibandingkan dengan menggunakan kawalan PID yang lazim, manakala prestasi bagi menggunakan alat kawalan penyesuaian adalah lagi bermutu dan baik, dengan menghasilkan kecekapan yang tinggi dalam mencapai titik set dalam keadaan seimbang bagi suhu dan kelembapan relatif.

Kata kunci: HVAC system, pengawal PID, kaedah RGA, kaedah nyahgandingan

1.0 INTRODUCTION

These days control of the energy consumption and energy efficiency are the hottest topics in many research areas. Energy efficiency is used in different application such as building design, transportation and power system [1-3]. Among of different applications HVAC systems are recognized as the greatest energy consumers in commercial and institutional buildings. Therefore, HVAC system modeling tries to the modeling of building, indoor, outdoor, and air handling unit (AHU) equipments to release the energy consumption of the system.

It is normally difficult for one HVAC system model to be completely comprehensive. Therefore, it is possible to divide the comprehensive model into sub models which may be appropriate in some instances [4]. The two main requirements of any HVAC system are: 1) to provide satisfactory indoor conditions within the building, for both humans and equipment (through regulation of temperature and relative humidity) and 2) minimize the overall energy consumption without compromising on performance [5]. Throughout the majority of applications, temperature and RH, above other parameters, have a more direct influence on the performance of HVAC systems [6, 7]. Several studies have been carried out based on on/off and proportional (P)-integral (I)-derivative (D) control methodologies, with the goal of enhancing the performance of HVAC systems through controlling the temperature and RH, using more complex algorithms such as non-linear, multivariable, artificial intelligence (AI) methodologies. Combinations of the algorithms were also tried [8-10].

The most broadly used control algorithms for HVAC systems are based on PIDs. However, more traditional control techniques (e.g. ON/OFF controllers) (thermostats) and PID controllers remain very popular due to their competitive pricing and ease of operation and tuning [11, 12]. However, [13] [13][13] and [14] have shown that the process of adjustment of PID controller coefficients could be a lengthy process, and could be both hard and costly work.

A control algorithm based on PID is now the most widely used control algorithm for HVAC systems, and has remained the focal point of several studies [15, 16]. It must be noted however that this PID-based control methodology is suitable solely for linear systems, as it itself is constructed as a linear algorithm.

One of the earliest works that applied adaptive control to HVAC&R systems, with a focus on DDC for solar-heated buildings, with a single-zone air space and room air temperature as the system output is discovered by [17]. In particular, a linearized model of the original nonlinear HVAC&R system was used to design an adaptive optimal control (AOC) strategy, and an optimal closed-loop obtained via the matrix Riccati equation by [18]. [19] described an adaptive control system as a type of controller that has the ability to adjust itself in response to any parameter variations occurring within a control system. With the factors of zone temperature and hot water temperature used as the two state variables, and heat pump input given as the control variable, an adaptive control strategy [20] was deployed to the 'discharge air temperature' model [21] for the discharge air temperature to track the optimal reference temperature in the presence of disturbances. Model-following or model-reference adaptive control (MRAC), which together constitute another class of adaptive system, was applied to a VAV system with the three state variables set as zone, coil, and water temperatures; the three control variables defined as mass flow rate of supply air, mass flow rate of chilled water, and input energy to the chiller; and a second-order model as the reference model for the VAV system. The simulations showed good adaptability of the actual zone temperature with regard to its reference value.

This results in incompatibilities with the HVAC system, which is inherently non-linear [22]. The assumption of linear system behavior such as those of the equipment and the building envelope components is usually valid, and acceptable control action may ensue. However, it is possible to manage the non-linear behavior of HVAC systems using more sophisticated control algorithms. This has only been made possible with the recent advance of high speed computing hardware and other digital technologies, which can be imbedded in controllers [23]. The design of functional HVAC system controllers depends primarily on the availability of appropriate dynamic models of the systems in equation, as well as mathematical equations describing its behavior. However, HVAC systems are often very complex with a range of distributed parameters, interactions, and multivariable, often making it difficult to obtain an exact mathematical model by which control quality may be improved.

Recently there has been increasing interest in mathematically modeling HVAC systems and their components. Many researchers have studied HVAC dynamic models using either an experimental or theoretical approach. For example, [24] developed an empirical nonlinear model of a hot-water-to-air heat exchanger loop used to develop nonlinear control law, [25] derived dynamic models for a duct and a hot water coil. Additionally, [26] developed an empirical model of a chilled water coil, which they used to predict the system's response to inputs with Proportional (P), Proportional Integral (PI), and Proportional Integral Derivative (PID) control algorithms. They measured the actual response of chilled water for the purpose of validating the coil model, and found that it was able to effectively predict the response at a range of values of gains, for each type of control algorithm. [27] described a procedure to derive a dynamic model of an air-conditioned room by applying physical laws to an air-conditioned room. [28] used the fact that the temperature measured by a sensor in a room temperature controller depends on its position in the zone to develop a room model aimed at studying the influence of the role of sensor position in building thermal control. They used a detailed list of criteria for the development of zone models. Further study by [29] presented a mode and control methodology for an air conditioning system, which was decomposed into two subsystems connected in series, incorporating natural feedback.

They achieved very good and valid results in maintaining room conditions close to desirable values. Dynamic model response and the transient response for space heating and cooling zones has been studied by several authors [30, 31]. [32] created a linear model represented a nonlinear cooling coil principles using principles of heat transfer and energy balance. [33] presented a transient HVAC system including a humidifier and mixing box (among other components), however no specific model for heating or cooling coils was given. [34] proposed a model for cooling coils using empirical parameters, assuming that there would be a constant flow of air and water. [35] and [36] offered two complex cooling coil models containing many iterative computations. [37] presented a mixing box, cooling coil and fan for a variable air volume (VAV) system. [22] recently presented a mathematical dynamic model for HVAC system components which they based on MATLAB. Despite its basis on MATLAB, the model consists of complexity, interconnection and nonlinearity.

There are a couple of ways to deal with the decoupling of RH and temperature. First, the coupling behavior observed between the parameters may be overcome through utilization of a decoupling algorithm, when the control law is developed-for example, intelligent or multivariable control methods may be utilized. [38] created a fuzzy controller designed to accounted for the coupling of RH and temperature for use in a cold store. The control methodology used here, based on rules to solve the coupling problem of temperature and RH directly, proposes a fuzzy controller which could efficiently control the system under disturbances and changes in set points. [39] investigated techniques by which they aimed to avoid interaction between moisture content and temperature. To achieve this they conducted an experimental study in which the two factors were simultaneously controlled by varying the speeds of both supply fan and compressor in a direct-expansion air conditioning system. In their study, Qi and Deng designed the controller using the linear quadratic gaussian (LQG) method with a linearized model of direct expansion air conditioning system. Although the method utilized in the study is straightforward for solving the problem in question, when there are changes in the set points of temperature in the thermal zones, some fluctuations in the moisture content of the thermal zone are observed before almost settling at a set point after 3000 s.

In the alternative approach for decoupling, separate channels are developed such that through single input single output (SISO) channels and non-linear decoupling control algorithms they are able to individually control the RH and temperature [40]. The error between the setpoint and thermal zone temperature in that study provides input to a PD controller to determine which control law 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 differential equations then use the controlled variables to find the RH response and thermal zone temperature. The contribution of this study is to apply the adaptive control methodology to one of the efficient tracks to supply actual cooling/heating power requests from the plant beside the best suited controllers with the purpose of meeting the challenge to reduce overall energy consumption, using an HVAC system introduced by [22]. It is worth noting that the decoupling of HVAC systems was not examined by [22]. Next, the methodology section is presented, which includes system models, the non-interactive method and adaptive control. The simulation results, and conclusions and recommendations are provided in sections 3 and 4 respectively.

2.0 METHODOLOGIES AND CONTROL ALGORITHMS

2.1 System Modeling

The system modeled discussed in this study is simply referred to as components of an HVAC system, serving a single thermal zone, as shown in Figure1, which are proposed and simulated in MATLAB/Simulink platform as used by [22]. It is noted that the new and comprehensive mathematic dynamic model of HVAC components, including the heating/cooling coil, humidifier, mixing box, ducts and sensors is described by [41]. The model proposed in this paper is presented in terms of energy mass balance equations for each of the HVAC components. Two control loops for this model, namely temperature and humidity ratio, are considered. Initially the system intakes fresh air, which it mixes with 50% of the return air, while the remaining half of the returned air passes through the heating coil and humidifier. Next, mixed air is delivered to the heating coil, where it is conditioned according to the specified setpoint by a draw-through fan. After that, supply air passes to the humidifier and conditioned according to the desired setpoint through the duct. The system controller simultaneously and constantly varies f_{sa} and f_{sw} according to load changes, so that the desired setpoints in temperature and RH, as control variables (as defined in the nomenclature[41]), are maintained. The variables are defined in the nomenclature. The differential equations for the system of Figure 1, as formulated by [41], are based on energy and mass balances. A complete theoretical approach of formulating the model is impractical owing to the fact that the zone is a complex thermal system. Three state variables define the zone model: the zone humidity ratio (W_z), zone temperature (T_z), and inner walls temperature (T_{wi}). It is assumed that the air is fully mixed, so that the zone temperature distribution is uniform and the dynamics of the zone can be expressed in a lumped capacity model.

$$c_z \frac{dT_z}{dt} = f_{sa}\rho_a c_{pa}(T_{sa} - T_z) + 2u_{w1}A_{w1}(T_w - T_z) + U_R A_R(T_R - T_z) + 2U_{w2}A_{w2}(T_{w2} - T_z) + q(t) \quad (1)$$

$$c_{w1}\frac{dT_{w1}}{dt} = u_{w1}A_{w1}(T_z - T_{w1}) + U_{w1}A_{w1}(T_0 - T_{w1}) \quad (2)$$

$$c_{w2}\frac{dT_{w2}}{dt} = u_{w2}A_{w2}(T_z - T_{w2}) + U_{w2}A_{w2}(T_0 - T_{w2}) \quad (3)$$

$$c_R \frac{dT_r}{dt} = u_R A_R (T_z - T_R) + U_R A_R (T_0 - T_R) \quad (4)$$

$$v_z \frac{dw_z}{dt} = f_s(w_s - w_z) + \frac{p(t)}{\rho_a} \quad (5)$$

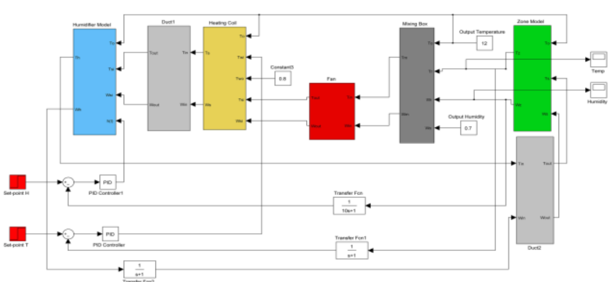

Figure1 HVAC system diagram [22]

Equation (1) states that the energy transferred into the zone by either convection or conduction, in combination with the energy removed from the zone, is equal to the rate change of energy in the zone. In equations (2)-(4) the rate change of energy is equal to the transferred energy through the walls because of temperature differences between indoor and outdoor air. Similarly, in equation (5), the difference between the vapor removed and added to the zone is equal to the rate change of moisture content.

2.1.2 The Heating Coil Model

The heating coil is a water-to-air heat exchanger, which provides the conditioned air required for ventilation purposes in buildings. The energy balance between cold air and hot water can be expressed by:

$$c_{ah}\frac{dT_{co}}{dt} = f_{sw}\rho_w c_{pw}(T_{wi} - T_{wo}) + (UA)_a(T_o - T_{co}) + f_{sa}\rho_a c_{pa}(T_m - T_{co}) \quad (6)$$

The mass balance is:

$$v_{ah}\frac{\partial w_{co}}{\partial t} = f_{sa}(w_m - w_{co}) \quad (7)$$

Equation (6) indicates that the energy transferred by the return air to the surrounding and the energy added by the flow rate of water in the heating coil is equal to the rate change of energy in the air which passes through the coil.

2.1.3 The Humidifier Model

Humidification is a process by which water vapor is transferred to atmospheric air, resulting in an increase of water vapor in the mixture. The following equations express the energy and mass balance for the humidifier model:

$$c_h \frac{dT_h}{dt} = f_{sa}c_{pa}(T_{si} - T_h) + \alpha_h(T_o - T_h) \quad (8)$$

$$v_h \frac{dw_h}{dt} = f_{sa}(w_{si} - w_h) + \frac{h(t)}{\rho_a} \quad (9)$$

In equation (9), h(t) represents the rate at which the humidifier can produce humid air. It is a function of the humidity ratio.

2.1.4 The Sensor Model

The function of sensors is to monitor the relative humidity and temperature, by which the system may use feedback signals to enhance the performance of the system.

$$T_{se}(s) = \frac{\tau_{se}}{\tau_{se}s + 1} = T_{me}(s) \quad (10)$$

2.1.5 The Fan Model

The first order fan model is chosen. The assumption is made that the physical properties of air are negligibly affected by air temperature.

$$T_{out} = (1/s + 1)T_{in} \quad (11)$$

2.1.6 Mixing Box

Air-conditioning systems will commonly mix air streams, which usually occurs under steady and adiabatic conditions.

$$m_r C_{pa} T_r + m_o c_{pa} T_o = m_m C_{pa} T_m \quad (12)$$

$$m_r + m_o = m_m \quad (13)$$

Where

$$T_m = \frac{m_r T_r + m_o T_o}{m_r + m_o} \quad (14)$$

$$w_m = \frac{m_r w_r + m_o w_o}{m_r + m_o} \tag{15}$$

2.1.7 The Duct Model

The exit air temperature is T_{out} and the inlet air temperature is T_{in}.

It is worth noting that the HVAC system model described by [22] is nonlinear, as the multiplication of control and controlled variables are presented.

2.2 Non-Interactive Method

[42] mentioned in their paper that if a plant transfer matrix is diagonally dominant, it may be feasible to design an efficient controller by regarding each input-output pair as a separate loop, forming an approach sometimes known as 'decentralized control'. This approach, although a choice of limited flexibility, has several well-known advantages, including sequencing of loop closing, tuning, and the chance to use the knowledge and intuition built upon these control designs of single input–single output systems. A key issue in the decentralized diagonal architecture is the pairing method of inputs and outputs [43]. Over the last four decades, this issue has received increasing amounts of attention inspiring many pieces of research, the most significant of which is the development of the idea of RGA, an original work by [44]. This methodology is a screening tool, widely used to determine whether a particular input/output pair (say, yi and uj) is a good choice for implementation of a SISO control loop, in an effort to minimize coupling and interactions with other loops, refining the Niederlinski result [45]. In the RGA, the channel interaction measurement is based on the D.C. gain of the multi input multi output (MIMO) process. Coupling of control loops invalidates conventional single loop controllers in many complex and complicated industrial processes.

A topic that has recently become of considerable importance in the field of control engineering is troubleshooting the decoupling of control systems [46]. Many researchers have tried to decouple systems to allow better control of the MIMO system. [47] used multiple variable system control theory to design a state feedback decoupling control system, in an effort to significantly improve the stability of the system. [48] proposed a decentralized control system consisting of independent SISO-controllers based on the diagonal elements of the system to control the MIMO system, resulting in higher close loop

$$\frac{dT_{out}}{dt} = \frac{(h_i + h_o) m_a C_p}{h_i M_c C_c} (T_{in} - T_{out}) \tag{16}$$

performance. [49] explained that the main advantage of two point control decoupling to provide the possibility of tuning and treating the multivariable system as not one but two single composition loops. Moreover, decoupling multivariable systems may have positive aspects, such as easier operation of a decoupled system compared to that of an interacting one. In this study, all dynamic mathematical components of the HVAC system are considered. According to the system identification toolbox, the transfer function of full mathematical dynamic components of HVAC system is illustrated. Process model is used to estimate mathematical components of HVAC system. The size of the model is 2*2. Table 1 shows the transfer function of the model.

Table 1 Transfer function of the model

Transfer function	Model process (Model transfer function)
$G_{11}=y_1/u_1$	$2.7*10^{-3}\exp(-1.69s)/(2.68*10^{-4}+0.109s+s\hat{}2)$
$G_{12}=y_2/u_1$	$2.844*10^{-6}*(1+10049s)*\exp(0.199s)/(1+305.16s)$
$G_{21}=y_1/u_2$	$3.395*10^{-3}\exp(-7.42s)/(4.23*10^{-4}+0.05s+s\hat{}2)$
$G_{22}=y_2/u_2$	$4.86*10^{-5}/(0.026+3.27*10^{-4}s+s\hat{}2)$

We use the following steps to find the RGA method:

1) Invest the transfer function's matrix of the system (shown in Table 1)

2) Evaluate the steady state of the matrix which consists of G (0) (equation 17)

$$G(0) = \begin{bmatrix} 10.1330 & 2.844e^{-006} \\ 8.0196 & 0.0018 \end{bmatrix} \tag{17}$$

$$\hat{G} = invG(0)' = \begin{bmatrix} 0.0969 & -431.3118 \\ -0.0002 & 556.2361 \end{bmatrix} \tag{18}$$

$$\Lambda(G) = \text{RGA} = G(0) * \text{inv}(G(0))' \tag{19}$$

RGA= [1.0012,-0.0012;-0.0012, 1.0012] (20)

NI= detG (0) /∏Gii=0. 0018 (21)

3) Finding $\hat{G}$ and $\Lambda(G)$ (equation 18)

The information of the RGA is shown in Table 2.

Table 2 Information about the RGA method

Pairing of input-output	λ	NI
(U1,Y1),(U2,Y2)	1.0012	0.0018

Instead of using the nonlinear and complex mathematical model of the HVAC system, pairing of (U1, Y1) and (U2, Y2) are considered as models of a decoupled HVAC system. The characteristic step responses of a decoupled and original HVAC system for temperature and humidity are considered in Table 3 and Table 4. It is clear that in Tables 3 and Table 4, the rise time for decoupled HVAC system is decreased rather than original system. In addition, the amount of peak time and peak amplitude are decreased in decoupled HVAC system rather than original system respectively which means the decoupled HVAC system can get better response to the step response.

Table 3 Step response of original and decoupled HVAC system for humidity with amplitude 0.46

Model	Peak time/amplitude	Rise time	Settling time	Final value	IAE
Original	1463/0.5	597.02	0.5	0.5	127.6
Decoupled	32/0.44	6.2	36.38	0.418	136.1

Table 4 Step response of original and decoupled HVAC system for temperature with amplitude 25°c

Model	Peak time/amplitude	Rise time	Settling time	IAE
Original	30001/24.9	1.1759e+004	1.9898e+004	1.205e004
Decoupled	19.3918/0.0036	6.5880	2.3726e+004	1.71e004

2.3 PID Control of Decoupled HVAC System

The decoupled HVAC system which is produced by the decoupling algorithm is controlled by a PID controller. Because a clear method for tuning the PID controller to control the humidity and temperature of HVAC systems is not mentioned, different types of tuning PID controllers are considered and compared with original system in Table 5.

Table 5 comparison of different tuning of PID controllers

Model	Controller	Method of tuning	IEA's Temperature simplifying/original	IEA's Humidity simplifying/original
Simplify/Original	PID	Trial and error	826.6/1257	29.9/31.83
	PID	Robust time	849.2/761	84.2/242.8
	PID	Ziegler-Nichols	1.69e004/1.49e+004	9.9/203.2
	PID	C-H-R	1.20e004/1.5e+004	11.6/85.59
	PID	----------	1.71e004/1.20e004	135.5/127.6
	PID	PSO	22.36/1.5e0023	10.42/1.5e0023

Different type of PID tuning such as Try and error, Robust time, Ziegler-Nichols, C-H-R and PSO are used to control the parameters of simplify and original HVAC system. In trial and error tuning method is based on guess-and-check. In this method, the proportional action is the main control, while the integral and derivative actions refine it. The controller gain, Kp, is adjusted with the integral and derivative actions held at a minimum, until a desired output is obtained. Robust time tuning method tunes the PID gains to maximize bandwidth and optimize phase margin. The Ziegler-Nichols' open loop method is based on the process step response. The Ziegler-Nichols tunings result in a very good disturbance response for integrating processes, but are otherwise known to result in rather aggressive tunings, and also give poor performance for processes with a dominant delay. CHR method is the modified version of the Ziegler-Nichols method which provides a better way to select a compensator for process control applications. PSO is one of the optimization techniques and a kind of evolutionary computation technique. The method has been found to be robust in solving problems featuring nonlinearity and non differentiability, multiple optima, and high dimensionality through adaptation, which is derived from the social-psychological theory. By observation of the different types of PID tuning it is found that the PSO method can be decreased by the amount of error among of all other types. However the PSO method is not suitable for tuning the parameters to control the humidity and temperature of the HVAC system, due to its time consuming nature. Therefore, it is suggested to use an adaptive controller to control the parameters of the HVAC system.

2.4 Adaptive Control Algorithm

Figure 2 presents a schematic of the MRAC used in the model. The Massachusetts institute of technology (MIT) rule is used to control the parameters of the HVAC system because; it is relatively simple and easy to use. When designing the MRAC controller developed for this paper, the output of the closed-loop system (y) followed the output of the reference model (y_m). The goal was to minimize the error ($e = y-y_m$) by designing a controller that had one or more adjustable parameters to minimize certain cost functions [$j(\theta) = 1/2e2$].

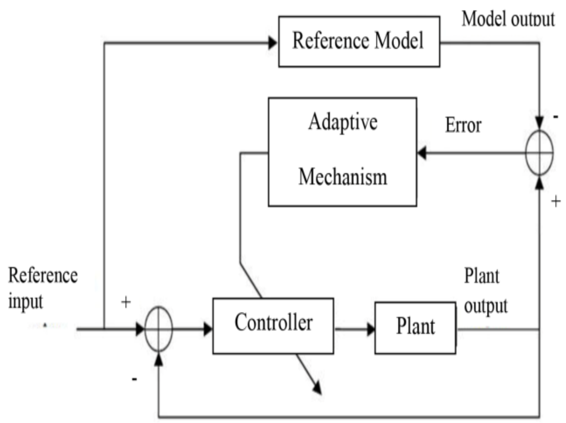

Figure 2 Diagram of MRAC [50]

2.4.1 *Adaptive Mechanism (G11 and G22)*

To investigate the value of G_{11} and G_{22} (decoupled model of HVAC system), first Yr has to be extracted. Note that Yr is the output of reference model. The next step is to find the error between Y and Yr. The controller is described by using equation 22:

$$u = \theta_1 r - \theta_2 y \tag{22}$$

The cost function is determined via using the following equations (23-24)

$$\partial\theta / \partial t = -(\gamma) * \partial j / \partial\theta \tag{23}$$

$$j(\theta) = 1/2 * e^2 \tag{24}$$

The parameter θ is adjusted in an effort to minimize the loss function. Therefore, it is reasonable to change the parameter in the direction of the negative gradient of j, i.e.

$$\partial\theta / \partial t = -(\gamma) * (\partial j / \partial e) * (\partial e / \partial\theta) =$$
$$-(\gamma) * e * (\partial e / \partial\theta) \tag{25}$$

Change in γ is proportional to negative gradient of J

The second order system is calculated by using equation (26):

$$G_p = y / u = [k_p / (s^2 + as + b)]u \tag{26}$$

When the first equation (22) is replaced by equation (26):

$$y = [k_p / (s^2 + as + b)] * (\theta_1 r - \theta_2 y) \tag{27}$$

Then

$$y[1 + (\theta_2 k_p) / s^2 + a_1 s + a_2] = \theta_1 r (k_p / s^2 + a_1 s + a_2) \tag{28}$$

$$y = \theta_1 (k_p) r / [s^2 + a_1 s + (a_2 + \theta_2 k_p)] \tag{29}$$

The MRAC tries to reduce the error between the model and plant as shown below in equations (30-38):

$$e = y - y_r \tag{30}$$

$$e = [\theta_1 (k_p) r / (s^2 + a_1 s + a_2 + \theta_2 k_p)] - G_m r \tag{31}$$

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

$$\partial e / \partial\theta_2 = -\theta_1 (k_p{}^2) r / [s^2 + a_1 s + a_2 + \theta_2 k_p]^2 \tag{33}$$

$$s^2 + a_1 s + (a_2 + \theta_2 k_p) \approx s^2 + A_1 s + A_2 \tag{34}$$

$$\partial e / \partial\theta_1 = [(k_p / k_m) * (k_m)] r / (s^2 + A_1 s + A_2) \tag{35}$$

$$\partial e / \partial\theta_2 = (-[(k_p / k_m) * (k_m)] / (s^2 + A_1 s + A_2)) * y \tag{36}$$

$$a_2 + \theta_2 k_p = A_2 \rightarrow A_2 - a_2 / k_p = \theta_2 \tag{37}$$

$$y = y_m \rightarrow k_m r / s^2 + A_1 s + A_2 =$$
$$\theta_1 (k_p) r / (s^2 + a_1 s + a_2 + \theta_2 k_p) \tag{38}$$

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 the MIT rule

$$\partial\theta_1 / \partial t = -(\gamma) * e * (\partial e / \partial\theta_1) =$$

So,

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

$$\partial\theta_1 / \partial t = -(\gamma) * e * (k_p / k_m) * y_r = -\gamma' * e * y_r \tag{39}$$

Where, $\gamma' = (\gamma) * (k_p / k_m)$ =Adaptation gain (40)

$$\partial\theta_2 / \partial t = -(\gamma) * e * -[(k_p / k_m)] * y_r / r * y \tag{41}$$

$$\partial\theta_2 / \partial t = \gamma' * e * (k_m) / s^2 + A_1 s + A_2 * y \tag{42}$$

Considering a =1.3, b = 6 and A1 =1.3, A2 = 3

3.0 SIMULATION RESULTS

3.1 Adaptive Control of Decoupled HVAC System

The decoupled HVAC system is described by the differential equations in (17-21), and controlled by adaptive controller in the on-line mode. The adaptive mechanism-1 which is used for G11 and adaptive mechanism-2 which is used for G22 with training procedure discussed in the previous section are used to control the temperature and humidity of the system respectively. A diagram of a closed loop block for adaptive control of decoupled HVAC systems is shown in Figure 3.

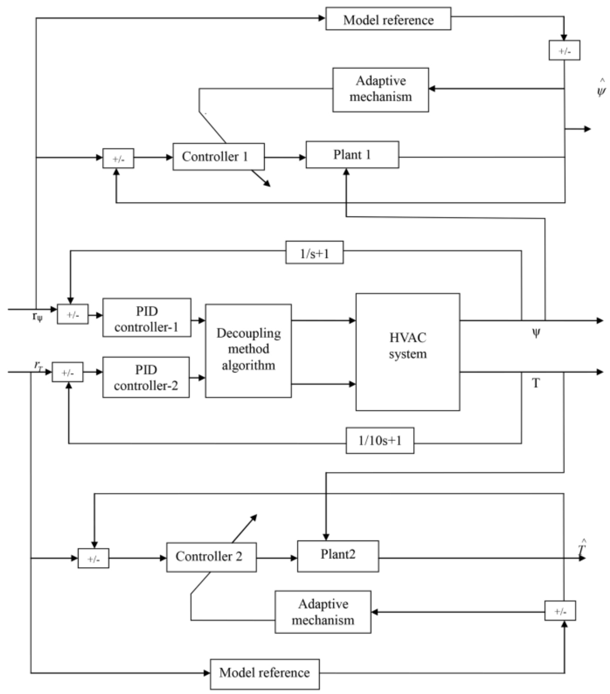

Figure 3 Close loop of decoupled HVAC system

The simulation results for the transient behavior of the temperature and RH in the thermal zone are shown in Figure 4 and Figure 5, in which a classical PID and an adaptive controller are used. According to the simulation results, the adaptive controller provides fast, smooth responses, avoiding any larger overshoots and undershoots. Moreover, the simulation shows good adaptability of the actual zone temperature with regard to its reference value. The target values for the humidity and temperature are 0.46 and 25 ℃, respectively.

Table 6 shows the integrated absolute error (IAE) amounts. The amount of IAE indicates that the decoupled model can reach target values and follow them efficiently. It is clear that the adaptive controller can be get track the input value with the small amount of error for both humidity and temperature rather than PID controller.

Table 6 Amount of IAE of humidity and temperature of the simplified HVAC system

Controller	IAE of humidity	IAE of Temperature
MRAC	3.2	18.79
PID	127.6	1.20e004

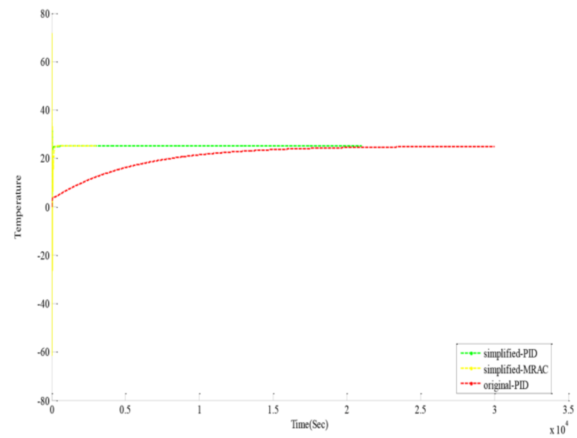

Figure 4 Comparison of PID and adaptive control for temperature (setpoint T=25°c)

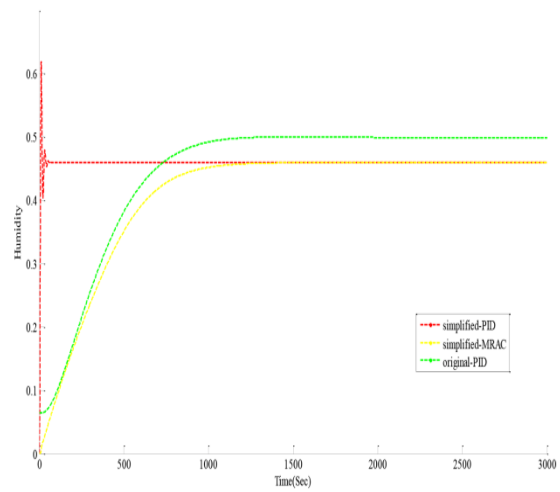

Figure 5 Comparison of PID and adaptive control for humidity (setpoint RH=0.46)

4.0 CONCLUSIONS AND RECOMMENDATIONS

In this study, we investigate a procedure to derive a dynamic model designed to control the parameters of an HVAC system. As the PID controller is suitable for the linear model, and the HVAC system is a nonlinear model, we therefore consider the linearization of the HVAC system. The RGA method is used to decouple the HVAC system to convert the MIMO model into SISO systems. One of the advantages of using this methodology is for better control of the parameters of the HVAC system. Model reference adaptive controllers are utilized to for the benefit of the transient behavior of the system. The results obtained demonstrate that the system has the capability to follow the set points effectively, with low error and with a short time period. The comparison between reference, existing, and adaptive solutions for the real HVAC system yielded significant improvement of steady state error behavior of the system. Comparison with the IAE found that adaptive control can achieve the target value more efficiently than the PID controller, which results in improved energy efficiency of the system. These improvements may be put down to the lower transient response of the system, and the trace set points of the parameters. This study looks specifically at two cases. In case 1, a decoupling method is implemented in HVAC systems, and classical PID controllers are applied to compare the parameters of original and decoupled HVAC system. In case 2, MRAC is used and compared with the PID controller to improve the transient response and to track the target values. The thermal zone temperature is held constant as a set-point change, and thermal zone RH is employed. The results for both cases 1 and 2 determine that using a decoupling control law and adaptive control algorithm, in the place of a classical PID, enhances the performance of the HVAC system, as well as target values of the setpoints. In this study, the decoupling control law and adaptive control algorithm are obtained at a given system. Therefore, the optimized control law and control methodology can be investigated in future works.

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
C_{ah}	overall thermal capacitance of the air handling unit=4.5 kJ/C
C_d	specific heat of the duct material=0.4187 kJ/kg ◦C
C_h	overall thermal capacitance of the humidifier=0.63 kJ/◦C
C_{pa}	specific heat of air=1.005 kJ/kg ◦C
C_{pw}	specific heat of water=4.1868 kJ/kg ◦C
C_R	overall thermal capacitance of the roof=80 kJ/C
C_{w1}	overall thermal capacitance of the wall (East, West)=70 kJ/C
C_{w2}	overall thermal capacitance of the wall (South, North)=60 kJ/C
C_z	overall thermal capacitance of the zone=47.1 kJ/C
e(t)	error
f_{sa}	volume flow rate of the supply air=0.192 m^3/s
f_{sw}	water flow rate=8.02*10^{-5} m^3/s
h(t)	rate of moisture air produced in the humidifier
h_i	heat transfer coefficient inside duct=8.33 W/m^2◦c
h_o	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)
T_{co}	temperature of the air out from the coil (◦C)
T_h	supply air temperature (in humidifier) in (◦C)
T_{in}	temperature in to the duct
T_m	temperature of the air out of the mixing box (◦C)
T_{me}	temperature measured (◦C)
To	temperature outside=32 ◦C (Summer)=5 ◦C (Winter)
T_{out}	temperature out from the duct
T_r	temperature of the recalculated air (◦C)
T_s	supply temperature from the Heating coil
T_{sa}	supply air temperature (◦C)
T_{se}	temperature output from the sensor (◦C)
T_{si}	temperature of supply air (to the humidifier) (◦C)
$T_{t,o}$	tube surface temperature (◦C)

T_{wo}	return water temperature=10 (◦C)
T_{wi}	supply water temperature (◦C)

T_{w1}	temperature of the wall (East, West) (◦C)
T_{w2}	temperature of the wall (South, North) (◦C)
Tz	temperature of the zone (◦C)
U_{w1}	overall heat transfer coefficient of (East, West) walls=2W/m^2 ◦C
U_{w2}	overall heat transfer coefficient of (South, North) walls=2W/m^2 ◦C
U_R	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
V_a	volume of the air handling unit=0.88 m^3
V_h	volume of humidifier=0.44 m^3
V_z	volume of the zoneZ36 m^3
W_{co}	humidity ratio of the air out from the coil (kg/kg dry air)
W_h	supply air humidity ratio (in humidifier) in kg/kg(dry air)
W_m	humidity ratio of the air out the mixing box (kg/kg dry air)
W_o	humidity ratio outside=0.02744 kg/kg (dry air) (Summer)=0.002 kg/kg(dryair)(Winter)
W_{sa}	humidity ratio of the supply air in kg/kg (dry air)
W_{si}	humidity ratio of the supply air (to the humidifier) in kg/kg (dry air)
W_z	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
U1	first input
U2	second input
Y1	first output
Y2	second output
w	water
W1	East, and West walls
W2	North, South walls
wi	water in
wo	water out
z	zone

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