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Nonlinear dynamics, bifurcation and performance analysis of an air-handling unit: Disturbance rejection via feedback linearization

Hamed Moradi [a,*], Majid Saffar-Avval [a], Aria Alasty [b]

[a] Energy and Control Centre of Excellence, Department of Mechanical Engineering, Amirkabir University of Technology, Tehran, Iran
[b] Department of Mechanical Engineering, Sharif University of Technology, Tehran, Iran

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ABSTRACT

Nowadays, dynamic analysis of air-conditioner units is essential to achieve satisfactory comfort conditions in buildings with low energy consumption and operation cost. In this paper, a nonlinear multi input–multi output model (MIMO) of an air-handling unit (AHU) is considered. In the presence of realistic harmonic disturbances, nonlinear dynamics of AHU is investigated. The effect of various thermodynamics and geometrical parameters on limit cycles behaviour of the indoor temperature is investigated. It is observed that the indoor space volume plays as the bifurcation parameter of the system. Decreasing the indoor space volume leads to the occurrence of secondary Hopf (Neimark) bifurcation and consequently the unstable quasi-periodic solution of the indoor temperature. To overcome this problem, a multivariable control strategy based on feedback linearization approach is implemented, in which the air and cold water flow rates are the control inputs while the indoor temperature and relative humidity are the control outputs. It is shown that the designed controller guarantees the comfort indoor conditions by preventing the unstable quasi-periodic responses and improving the limit cycles behaviour.

1. Introduction

To provide comfortable environment and acceptable indoor air quality (IAQ) for building occupants, air-conditioning systems are extensively used in commercial and office buildings. In order to optimize the performance of heating, ventilation and air conditioning (HVAC) systems while maintaining the comfortable indoor environmental conditions, most of the modern buildings are equipped with energy management and control systems (EMCS). Since energy and operation costs of buildings are directly influenced by how well an air-conditioning system performs, effective thermal management is of great importance. It is estimated that more than 50% of the building energy consumption is accounted by HVAC systems [1]. A comprehensive review on air-conditioning systems and control of the indoor air quality to preserve human health has been presented [2].

Development of accurate and simple dynamic models for HVAC systems is the most important factor for the efficient EMCS design. Dynamics models may be varied in structure (simple or complex), depending on the type of energy management functions and the accuracy demanded. However from practical point of view, deriving simple and accurate while reliable models, in coincidence with the real dynamic behaviour of the whole HVAC system and its components, is of great importance [3].

Among HVAC components, air-handling units (AHU) have an essential role for providing supply air with specific temperature and humidity. Due to the complex nonlinear nature of an AHU with multivariable parameters and time varying characteristics of its components, finding an exact mathematical model is difficult [4]. Therefore, dynamic modelling and simulation of HVAC systems and its components have been accomplished through several investigations.

Self-tuning and classical dynamic models for HVAC system components [3,5], dynamic model and transient response for space heating and cooling zones [6] and dynamic simulation and evaluation of EMCS on-line for variable air volume (VAV) air-conditioning systems have been presented [7,8]. Using automatic data acquisition system for on-line training and artificial neural network [9] and grey-box identification approach [10], air-handling unit has been modelled. Dynamic simulation of energy management control functions for HVAC systems [11], simulation of a VAV air-conditioning system for the cooling mode [12] and adaptive HVAC zone modelling for sustainable buildings [13] have been carried out. Also, an overview of current approaches used for modelling and simulation of HVAC systems including control aspects has been presented [14].

To achieve optimal performance of HVAC systems, control of the energy flows within a building is essential. Stability analysis and tuning of PID controller in the VAV systems [15], model

* Corresponding author. Tel.: +98 21 66165513; fax: +98 21 66000021.
E-mail addresses: hamedmoradi@asme.org, hamed.moradi1@gmail.com (H. Moradi).

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based analysis and simulation of airflow control of AHU units using PI controllers [16] and control tuning of a simplified VAV system have been studied [17]. Also, cascade control algorithm and gain scheduling [18], model predictive control [19], normalized decoupling control [20] and analysis of different control schedules on EMCS [21] of air-handling units have been investigated.

Rule development and adjustment of a fuzzy controller [22], fuzzy control optimized by genetic algorithms (GA) [23], developing an adaptive fuzzy controller based on GA [24] have been implemented on air-handling units, as other control approaches. In addition, optimal control [25–27], data-driven optimization approach [28] and adaptive self-tuning PI control [29] are other control approaches used for HVAC systems.

However, since tuned control parameters cannot cover all the operating ranges of the AHU, using traditional control approaches may result in poor time responses or increase in the energy consumption. Also, for a constant provided ventilation flow rate, when the occupancy is lower or higher than the expected maximum occupancy, they may cause over ventilation or insufficient ventilation [30]. In addition, due to non-stationary and nonlinear characteristics of AHU components and the coupling of the controlled variables, its control is a non-trivial problem [4,10]. Therefore, recently some investigations have been devoted to implement the nonlinear control [31] and robust control approaches [32–34] on a multivariable model of AHU.

Although many investigations have been accomplished for dynamic modelling and control of AHU, its nonlinear dynamic analysis has not been studied in the previous researches. In the presence of nonlinear sources and multivariable model of AHU, where different parameters are involved in a complex interaction, linear theories fail to predict interesting phenomena. Moreover, without an extensive pre-knowledge of AHU behaviour against possible disturbances, applying the designed controllers may lead to aggressive response of output variables and also increase in energy consumption.

In this paper, nonlinear dynamics of an air-handling unit is investigated in the presence of realistic harmonic disturbances in dynamic state variables. In the considered nonlinear model of AHU, the indoor temperature and relative humidity are the output variables which are controlled by manipulation of the air and cold water flow rates, as the control inputs. For disturbance rejection, a multivariable control strategy based on feedback linearization approach is designed. Under steady-state condition, a parametric study is performed to investigate the effect of various thermodynamics and geometrical parameters on limit cycles behaviour of the indoor temperature. For various operating conditions, the location of fixed points for indoor temperature is determined. For the sake of brevity, similar analysis for the indoor relative humidity is not presented. It is shown that decreasing the indoor space volume leads to the occurrence of secondary Hopf (Neimark) bifurcation and consequently the unstable quasi-periodic solution for the indoor temperature. However, implementation of the nonlinear controller improves the limit cycles behaviour, prevents the occurrence of unstable quasi-periodic responses, and consequently leads to the comfort indoor conditions.

2. Nonlinear dynamic model of the air-handling unit

Components of an air-handling unit having one zone (indoor) in VAV system are shown in Fig. 1 [31,34,35]. As it is shown, this unit is constituted of supply and return air fans, cooling coil, filter, ductwork, humidifier and dehumidifying coil (not shown). Since in this research, AHU is essentially designed for operation in summer, chilled water and air loops exist. After the entrance and passing of the hot and humid air through the cooling and dehumidification coil, its temperature and humidity ratio decrease. A mixture of 25% of fresh air with 75% of returned air passes through cooling unit. Finally desired supply air is provided and delivered to the ventilated space through output channel.

Table 1
Thermo-fluid and geometrical parameters of the air-handling unit.

w_o	Environment humidity ratio
w_s	Supply air humidity ratio
w_t	Indoor humidity ratio (zone)
T_o	Environment temperature
T_s	Supply air temperature
T_t	Indoor temperature (zone)
ΔT_c	Temperature gradient in cooling unit
$\dot{M}_o$	Strength of the humidity source
$\dot{Q}_o$	Heat load
$\dot{f}_a$	Air flow rate
$\dot{f}_w$	Cooling water flow rate
C_{pa}	Specific heat of the air
C_{pw}	Specific heat of the water
h_w	Enthalpy of the saturated water
h_{fg}	Enthalpy of the vaporization
ρ_a	Air mass density
ρ_w	Water mass density
V_c	Volume of the cooling unit
V_t	Volume of the indoor space (zone)

For the formulation of the problem, it is assumed that gases are ideal and mixed completely; air flow is homogeneous; the effect of air speed variations on the zone pressure is negligible and there is no air leakage except in the exhaust valves of the zone [36]. Using thermodynamics, heat and mass transfer laws, differential equations describing dynamic behaviour of the air-handling unit are determined as follows [31,36–38]:

$$\begin{aligned}
\dot{T}_s &= \frac{\dot{f}_a}{V_c}(T_t - T_s) + \frac{0.25\dot{f}_a}{V_c}(T_o - T_t) - \frac{\dot{f}_a h_w}{C_{pa} V_c} \\
&\qquad \times (0.25 w_o + 0.75 w_t - w_s) - \dot{f}_w \left(\frac{\rho_w C_{pw} \Delta T_c}{\rho_a C_{pa} V_c} \right) \\
\dot{T}_t &= \frac{1}{\rho_t C_{pa} V_t}(\dot{Q}_o - h_{fg}\dot{M}_o) + \frac{\dot{f}_a h_{fg}}{C_{pa} V_t}(w_t - w_s) - \frac{\dot{f}_a}{V_t}(T_t - T_s) \\
\dot{w}_t &= \frac{\dot{M}_o}{\rho_a V_t} - \frac{\dot{f}_a}{V_t}(w_t - w_s)
\end{aligned} \tag{1}$$

where T_s/w_s, T_t/w_t and T_o/w_o are the temperature/humidity ratio of the supply air, indoor air (zone) and environment, respectively; ΔT_c is the temperature gradient in cooling unit; $\dot{f}_a$ and $\dot{f}_w$ are the air and cold water flow rates; V_c and V_t are the volume of the cold unit and indoor space (zone); $\dot{Q}_o$ and $\dot{M}_o$ are the strength of heat load and humidity load; ρ_a/C_{pa}, ρ_w/C_{pw} are the mass density/specific heat of the air and cold water. h_w and h_{fg} are the enthalpy of saturated water and vaporization. For convenience, the list of thermo-fluid parameters is given in Table 1. To simplify Eq. (1), following terms are defined:

$$\begin{aligned}
&\alpha_1 = \frac{1}{V_t}, \quad \alpha_2 = \frac{1}{\rho_a V_t}, \quad \alpha_3 = \frac{1}{V_c}, \quad \beta_1 = \frac{h_{fg}}{C_{pa} V_t}, \quad \beta_2 = \frac{\rho_w C_{pw} \Delta T_c}{\rho_a C_{pa} V_c} \\
&\gamma_1 = \frac{1}{\rho_t C_{pa} V_t}, \quad \gamma_2 = \frac{h_w}{C_{pa} V_c}
\end{aligned} \tag{2}$$

To formulate the problem in the state space representation, input (u_i), output (y_i) and state (x_i) variables are defined as:

$$\begin{aligned}
&u_1 = \dot{f}_a, \quad u_2 = \dot{f}_w, \\
&y_1 = w_t, \quad y_2 = T_t, \\
&x_1 = T_t, \quad x_2 = w_t, \quad x_3 = T_s
\end{aligned} \tag{3}$$