www.s2is.org
International Journal on Smart Sensing and Intelligent Systems
ISSN 1178-5608

REVIEW ON MODELING AND CONTROLLER DESIGN IN PNEUMATIC ACTUATOR CONTROL SYSTEM

[1]M. F. Rahmat, [1]N. H Sunar and [2]Sy Najib Sy Salim, [3]Mastura Shafinaz Zainal Abidin , [4]A. A Mohd Fauzi and [4]Z. H. Ismail

[1]Department of Control and Instrumentation Engineering, Faculty of Electrical Engineering,
Universiti Teknologi Malaysia, 81310 Skudai, Malaysia.

[2]Department of Control, Instrumentation and Automation Department, Faculty of Electrical Engineering, Universiti Teknikal Malaysia Melaka, Hang Tuah Jaya, 76100 Durian Tunggal, Melaka, Malaysia

[3]INSEED, Faculty of Electrical Engineering, Universiti Teknologi Malaysia, 81310 Skudai, Johor, Malaysia

[4]Department of Mechatronics and Robotics,
Universiti Teknologi Malaysia, 81310 Skudai Johor, Malaysia

Emails: fuaad@fke.utm.my and noorhazirah.sunar@yahoo.com

Submitted: Sep. 20, 2011 Accepted: Nov. 10, 2011 Published: Dec. 1, 2011

Abstract- Pneumatic actuators are highly nonlinear characteristics and uncertainties make it difficult to achieve high performances. The objective of this paper is to present a brief overview of pneumatic actuators based on modeling and control strategies that has been

proposed by various researchers. Before the main discussion, some background information will be presented in a relation to pneumatic actuators. This review concludes with a short summary and discussion of modeling and control approaches of pneumatic actuators. The implication of this paper is for further improving the performance of existing pneumatic actuators.

Index terms: Pneumatic actuator system, modeling, controller, nonlinearities, uncertainties, position tracking.

I. INTRODUCTION

Nowadays, pneumatic actuators have become an important driving element that extensively used in industrial robotics and automation. Due to their special attributes, pneumatic actuators have become alternate actuator in automated material handling task. The compressibility of air and friction in the pneumatic actuators are the main factors to the nonlinearities in the system that makes the pneumatic actuators difficult to control. A high number of unknown parameters need to be identified in order to achieve a dynamical response closed to real systems. In 1950s, the study on the pneumatic actuators has become vigorous due to the increasing demands of automation in industrial production lines. The first theoretical basis of the pneumatic system dynamic control was initially made by Prof. J. L. Shearer in 1956 [1]. Shearer derived the dynamic of the system according to nonlinear differential equations and then followed by the linear model. Shearer's methodology toward developing the model has been used in most subsequently researches. In the modeling, several types of valves were used in pneumatic actuator system: valve types like on-off solenoid valve [2-3] and proportional valve [4]. The advantages of on-off solenoid valves is simple system structure and low cost [2]. Yet, it also has disadvantages that include lacking in flexibility, very difficult to achieve high positioning and time accuracy to synchronize with other packaging events [2]. Meanwhile, the proportional valve is complicated and is difficult to control [5].

The main body of the review dealing with (II) pneumatic actuator applications, (III) pneumatic actuator contributions, (IV) pneumatic actuator modeling, (V) pneumatic actuator controller design and (VI) in the last section is the short summary and discussion on pneumatic actuators.

II. PNEUMATIC ACTUATOR APPLICATIONS

In 1999, an application of pneumatic actuator was investigated for food packaging in production line [2]. The control strategy was applied in a combination with a modified PID controller to a pusher mechanism in the packaging of confectionery product. In 2005, a pneumatic actuators was developed in construction robot [6]. A construction robot used in the work of attaching the ceramic tile. Other applications of pneumatic actuator was applied in Jack hammer, power drills [4] and blow molding process as a manufacturing applications [4]. Instead of manufacturing application, pneumatic actuators also have been applied to a physical human interface. Distributed physical human interface machine interaction was investigated using intelligent pneumatic cylinder to form an Intelligent Chair Tool in 2008 [7]. In earlier 2010, pneumatic actuators have been applied to a clinical robot assistant. For the enhanced detection and treatment, pneumatic actuator has been applied to an Magnetic Resonance Imaging (MRI) guided prostate biopsy and brachytherapy for accurate needle positioning control [8]. Next, another application of pneumatic actuator has been reported to develop an active 80-faced Polyhedron for haptic physical human interface [9].

III. PNEUMATIC ACTUATOR CONTRIBUTIONS

Lots of attributes offered by pneumatic actuators compared to hydraulic actuators and magnetic actuators. Pneumatic actuators used air as its source of energy. The preferences of using air compare to conventional actuator that used water and magnetic due to air is a free source and is easy availability [10].

Hydraulic actuators need an external source of water supply to run a hydraulic system, while no external source required for pneumatic actuator to run a pneumatic system. These lead to cost effective of pneumatic actuators compare to hydraulic actuators [10].

Pneumatic actuators have a good power-to-weight ratio [3, 10-11]. The density of air much lesser than water caused the decreasing of its weight-to-power ratio [10].

Furthermore, pneumatic actuators offers capability in providing large maximum forces for a longer period of time compare to electrical actuators [11]. Electrical actuators contribute to overheating caused by thermal expansion for a longer period of time caused the increasing error of the performance [11].

Pneumatic actuators are easy to build system design and is very flexible [4]. It can be design in range from lightweight, compact domestic appliances to heavy-duty industrial applications. Pneumatic actuators provide a clean operating conditions [3] and make it safe to be used.

However, pneumatic actuator is a highly nonlinear characteristic due to the air mass flow through the servo valve [11] and the friction force acting on the piston [11]. The best dynamic description is needed in order to develop a high performance position controller for pneumatic actuator system.

IV. PNEUMATIC ACTUATOR MODELING

Pneumatic actuator can be modeled from theoretical mathematical analysis or system identification. Many approaches have been proposed for pneumatic actuators modeling. Most of researchers used theoretical mathematical analyses for modeling the pneumatic actuator. Three major considerations on obtaining the pneumatic actuator system: (1) the dynamic of the load, (2) the pressure, volumes and temperature of the air in cylinder, and (3) the mass flow rate through the valve [12]. Friction force exist in the pneumatic actuator system makes it difficult to control. Several researchers modeled friction force in order to develop friction compensation for high accuracy of controller performance. This modeling review starts with review for theoretical mathematical model. Then the model review continued with linear modeling method and system identification method. Next, the review of friction force modeling and pneumatic actuator model due to the physical or mechanical enhancement.

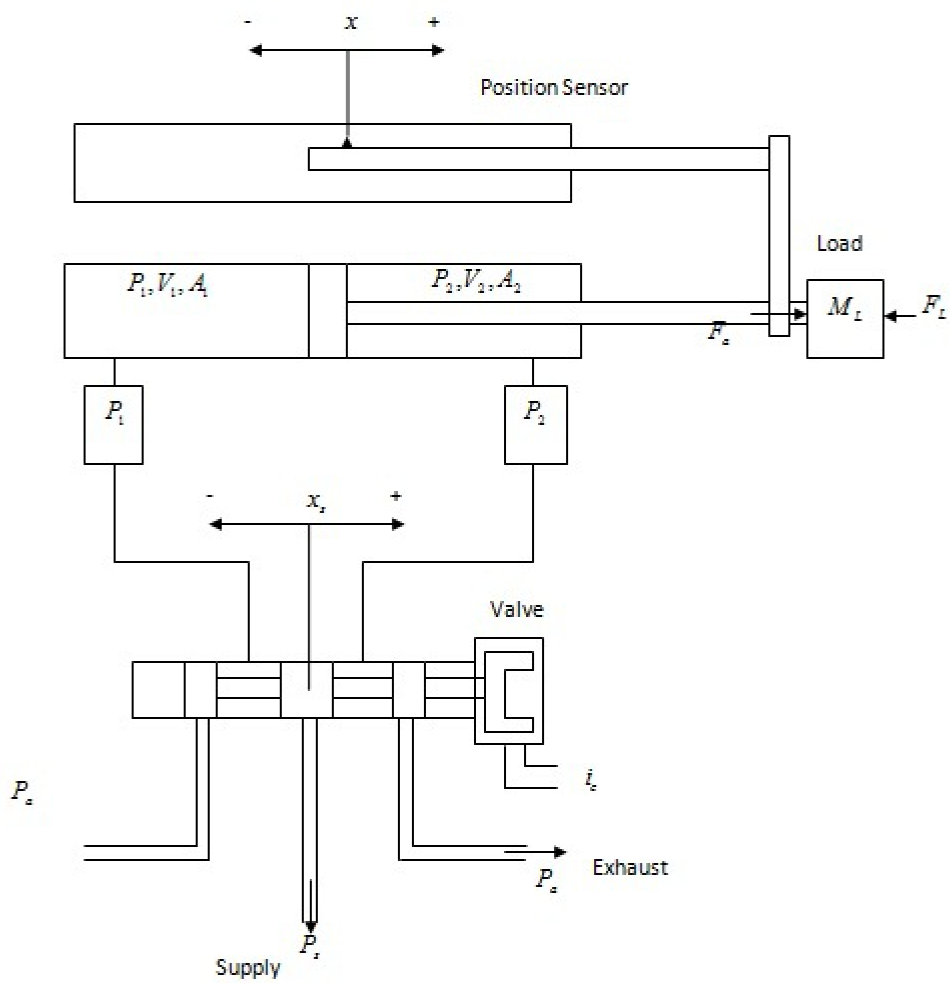

Figure 1. Diagram of double acting pneumatic actuator system

The diagram of double acting pneumatic actuator system can be represented as shown in Figure 1 [12]. The dynamic of the loads was derived from piston load dynamics. The dynamic of the loads was derived based on Newton's second law. The piston-rod-load assembly dynamic equilibrium of motion was represented as [4, 12-16]:

$$(M_L + M_p)\ddot{x} + \beta\dot{x} + F_f + F_L = P_1A_1 - P_2A_2 - P_aA_r \tag{1}$$

where M_L is the external load mass, M_P is the piston and rod assembly mass, x is the piston position, β is viscous friction coefficient, F_f is the coulomb friction force, F_L is the external force, P_1 and P_2 are the absolute pressure in chamber 1 and 2, P_a is the absolute ambient pressure, A_1 and A_2 are the piston effective areas and A_r is the rod cross-sectional area.

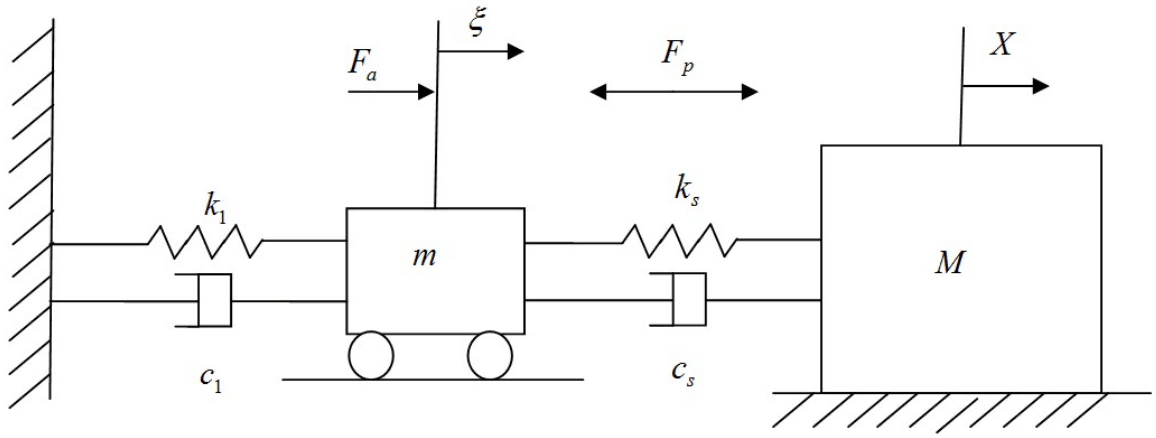

Figure 2. 2-DOF analytic model based on PZT actuation.

An innovation of actuating method, a pneumatic system model with piezoelectric (PZT) actuator based on the two degree-of-freedom analytical (2-DOF) model was presented [17]. Figure 2 shows the diagram of 2-DOF analytic model based on PZT actuation. The 2-DOF analytical model governed the dynamic behavior by two coupled nonlinear differential equation. Two differential equation expressed as [17]:

$$\begin{aligned} m\ddot{\xi} &= -k_1\xi - c_1\dot{\xi} + k_s(X-\xi) + c_s(\dot{X}-\dot{\xi}) + F_a - F_p \\ M\ddot{X} &= -k_s(X-\xi) - c_s(\dot{X}-\dot{\xi}) + F_p - F_t \end{aligned} \tag{2}$$

Whew M is the mass directly connected to PZT actuator, k_s is the spring constant, c_s is the spring damper, m is the pneumatic cylinder with piston rod mass, k_1 is the linear spring, c_1 is the linear damper spring, X is the displacement of the sliding table ξ is the displacement of piston rod, F_a is the net force pressure, F_p is the force generated by the PZT actuator and F_t is the friction force.

The cylinder chamber model was derived based on relationship of the pressure change with respect to mass flow-rate. Based on adiabatic flow condition process, a pressure dynamic equation was presented as [18]:

$$\begin{aligned} \dot{P}_1 &= \frac{k}{V_1}(\dot{m}_{in}RT_s - \dot{m}_{out}RT_1 - P_1\dot{x}A_p) \\ \dot{P}_2 &= \frac{k}{V_2}(\dot{m}_{in}RT_s - \dot{m}_{out}RT_2 - P_2\dot{x}A_p) \end{aligned} \tag{3}$$

with the rate of the change at temperature [18]:

$$\begin{aligned} \dot{T}_1 &= \frac{T_1 \dot{x} A_p}{V_1} + \frac{T_1 \dot{P}_1}{P_1} - \frac{R T_1^{\,2} (\dot{m}_{in} - \dot{m}_{out})}{P_1 V_1} \\ \dot{T}_2 &= \frac{T_2 \dot{x} A_p}{V_2} + \frac{T_2 \dot{P}_2}{P_2} - \frac{R T_2^{\,2} (\dot{m}_{in} - \dot{m}_{out})}{P_2 V_2} \end{aligned} \tag{4}$$

Instead on adiabatic flow in equation (3), the pressure dynamic was expressed based on thermal characteristics of adiabatic and isothermal as [12-15, 19]:

$$\dot{P}_i = \frac{RT}{V_{0i} + A_i\left(\frac{1}{2}L \pm x\right)}\left(\alpha_{in}\dot{m}_{in} - \alpha_{out}\dot{m}_{out}\right) - \alpha \frac{PA_i}{V_{oi} + A_i\left(\frac{1}{2}L \pm x\right)}\dot{x} \tag{5}$$

where $\dot{P}_i$ is pressure dynamic in each chamber, R is universal gas constant, T is the temperature, $\dot{m}_{in}$, $\dot{m}_{out}$ are mass flows entering and leaving the chamber and α, α_{in}, and α_{out} are values between 1 and k depend on the heat transfer during the process.

The volume of each chamber was expressed as [13-14]:

$$V_i = V_{0i} + A_i\left(\frac{1}{2}L \pm x\right) \tag{6}$$

where i is the cylinder chamber index, V_{0i} is the dead volume in each chamber, L is the stroke length of the cylinder, x is the piston position and A_i is the piston effective area.

In 2006, air leakage was considered in the pneumatic actuator modeling. The air leakage was expressed as [20]:

$$G_n = \frac{1}{24}\frac{d^3}{l}\frac{P_1^{\,2} - P_2^{\,2}}{\mu RT}.P \tag{7}$$

where d is an interstice height, l is an interstice length and $P = 2\pi R_c$ which is the perimeter.

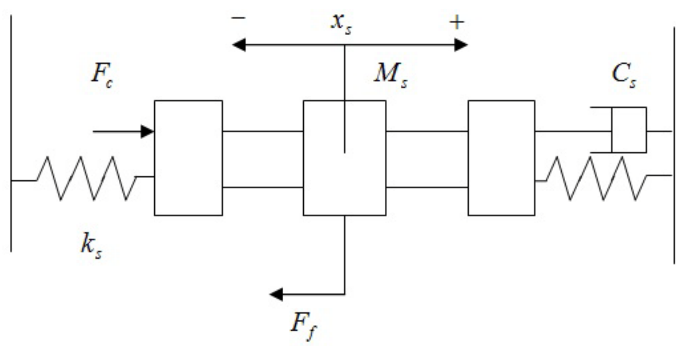

Figure 3. Valve spool dynamic equation

Pneumatic valve is the command element in the pneumatic system. It should provide fast and precisely controlled air flow. Proportional spool valves are commonly used in pneumatic actuator system modeling. Figure 3 shows the valve spool dynamic equation [12]. The modeling of this valve involves two aspects: the dynamic of the valve spool, and the mass flow rate through the valve's orifice. The equation of motion for the valve spool can be written as [13]:

$$M_s\ddot{x}_s + C_s\dot{x}_s + F_f + 2k_s x_s = K_{fc} i_c \tag{8}$$

where M_s is the spool and coil assembly mass, x_s is the spool displacement, C_s is the viscous friction coefficient, F_f is the friction force and can be neglected, k_s is the spool spring constant, K_{fc} is the coil force coefficient, and i_c is the coil current.

The mass flow rate through an orifice of area A_v was represented as [12-13, 19]:

$$\dot{m}_v = \begin{cases} C_f.A_v.C_1.\dfrac{P_u}{\sqrt{T}}, if\ \dfrac{P_d}{P_u} \le P_{cr} \\ C_f.A_v.C_2.\dfrac{P_u}{\sqrt{T}}.\left(\dfrac{P_d}{P_u}\right)^{\frac{1}{k}}.\sqrt{1-\left(\dfrac{P_d}{P_u}\right)^{\frac{k-1}{k}}}, if\ \dfrac{P_d}{P_u} > P_{cr} \end{cases} \tag{9}$$

where $\dot{m}_v$ is mass flow through valve orifice, C_f is non-dimensional discharge coefficient, k is the specific heat ratio, T is the temperature, P_u, P_d are upstream and downstream pressure and

$$C_1 = \sqrt{\frac{k}{R}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}};C_2 = \sqrt{\frac{k}{R}\left(\frac{2}{k-1}\right)};P_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \quad (10)$$

are constant for a given fluid. For air $k = 1.4$.

For charging, the pressure tank considered as upstream pressure and pressure in the chamber is considered as downstream pressure. In contrast, the pressure in the chamber considered as upstream pressure and ambient pressure considered as downstream pressure during discharging process.

The valve effective area for input and exhaust paths represented as [13]:

$$A_{vin} = \begin{cases} 0 \text{.............} if\,(x_s) \le (p_w - R_h) \\ n_h\left[2R_h^{\,2}\arctan\left(\sqrt{\dfrac{R_h - p_w + x_s}{R_h + p_w - x_s}}\right) - (p_w - x_s)\sqrt{R_h^{\,2} - (p_w - x_s)^2}\right] \\ \text{................} if\,(p_w - R_h) < x_s < (p_w + R_h) \\ \pi n_h R_h^{\,2} \text{.....} if\,(x_s) \ge (p_w + R_h) \end{cases} \quad (11)$$

$$A_{vout} = \begin{cases} \pi n_h R_h^{\,2} \text{....} if\,(x_s) \le (-p_w - R_h) \\ n_h\left[2R_h^{\,2}\arctan\left(\sqrt{\dfrac{R_h - p_w + |x_s|}{R_h + p_w - |x_s|}}\right) - (p_w - |x_s|)\sqrt{R_h^{\,2} - (p_w - |x_s|)^2}\right] \\ \text{.............} if\,(-p_w - R_h) < x_s < (R_h - p_w) \\ 0 \text{...........} if\,(x_s) \ge (R_h - p_w) \end{cases} \quad (12)$$

where n_h is the hole for an air path in sleeve, R_h is the hole radius, $2p_w$ is the spool width.

However, a simple mathematical expression for the effective of valve orifice compared to equation (11) and (12) was represented as [21]:

$$A_v = X_{sp}^{\;2}\frac{\pi}{4} \quad (13)$$

where control spool movement (x_s) relationship with voltage input (u) as [21]:

$$x_s = C_v u \quad (14)$$

and C_v is the valve constant.

Valve-cylinder connecting tube is a source for the air flow entering the cylinder chamber. Most researchers neglected the effect of valve-cylinder connecting tube. There are several researchers assumed that the flow throughout the tube is either fully laminar or turbulent flow. The mass flow at the outlet is represented as [13-14]:

$$\dot{m}_t(L_t,t)=\begin{cases}0, if, t<L_t/c\\ e^{-\frac{R_t RT}{2P}\frac{L_t}{c}h\left(t-\frac{L_t}{c}\right)} if, t>L_t/c\end{cases} \tag{15}$$

where c is the sound speed, R_t is the flow resistance and L_t is the tube length.

The tube resistance obtained from the pressure drop along the tube [13-14]:

$$\Delta p = f\frac{L_t}{D}\frac{\rho u^2}{2}=R_t u L_t \tag{16}$$

where f is the attenuator factor and D is the inner diameter of the tube. For fully developed laminar flow:

$$f=\frac{64}{\mathrm{Re}} \tag{17}$$

where Re is the Reynolds number. Then, the tube resistance becomes [13]:

$$R_t=\frac{32\mu}{D^2} \tag{18}$$

where μ is the dynamic viscosity of air.

For wholly turbulent flow in smooth tubes, the friction factor computed using Blasius formula [13]:

$$f=\frac{0.316}{\mathrm{Re}^{1/4}} \tag{19}$$

The tube resistance for wholly turbulent flow becomes [13]:

$$R_t = p.158\frac{\mu}{D^2}\mathrm{Re}^{3/4} \tag{20}$$

For the friction model, an asymmetrical Karnopp model of friction was presented in 2004 [22]. The asymmetrical Karnopp model of friction was expressed as:

$$F_f(t)=\begin{cases}F_{slip}(t)=\begin{cases}F_{cp}sign(\dot{y}(t))+F_{vp}\dot{y}(t), \dot{y}(t)\geq d_{vp}\\ F_{cn}sign(\dot{y}(t))+F_{vn}\dot{y}(t), \dot{y}(t)\geq d_{vn}\end{cases}\\ F_{stick}(t)=\begin{cases}\min(F_c(t),F_{sp}), d_{vn}<y(t)<d_{vp}, F_c(t)\geq 0\\ \max(F_c(t),F_{sn}), d_{vn}<y(t)<d_{vp}, F_c(t)\geq 0\end{cases}\end{cases} \tag{21}$$

where $F_{cp,cn}$ is the Coulomb friction for positive and negative velocity, $d_{vp,vn}$ are the limited velocity in stick-positive slip and stick-negative slip regions, $F_{sp,sn}$ are the maximum and minimum stick friction, $\dot{y}(t)$ is the velocity, $F_c(t)$ is the applied external force and $F_f(t)$ is the friction force.

Instead of model friction based on Karnopp model, static friction model regards Coulomb friction and Stribect effect can be expressed as [11, 19]:

$$F_{Fr,static} = g(\dot{x}_d).sign(\dot{x}_d) + (b_0 - |\dot{x}_d| b_1)\dot{x}_d \qquad (22)$$

where

$$g(\dot{x}) = F_c + (F_s - F_c)e^{-\left(\dot{x}/v_s\right)^2} \qquad (23)$$

and F_c is coulomb friction, F_s is static friction, v_s Stribeck velocity and b_0, b_1 are viscous friction and $\dot{x}_d$ are velocity.

The friction force then was extended modeled under the influence of the piston motion [23]. At the beginning of piston motion, sticking effects of friction was dominant (coulomb friction). During the piston motion, the viscous friction become dominant. The friction model under different piston motion was expressed as [23]:

$$F_{fric} = \begin{cases} |F_p| sign(F_p), v = 0, |F_p| < F_{sc} \\ |F_{sc}| sign(F_p), v = 0, |F_p| \geq F_{sc} \\ |F_{dc}| sign(v) + \beta v, v \neq 0 \end{cases} \qquad (24)$$

F_{sc}, F_{dc} are the static and dynamic friction forces, β is the coefficient of viscous friction, v is the piston velocity.

An improvement based on a physical device innovation was presented. A pneumatic model was extended with bridging of the dual action cylinder chamber [14]. The bridging was model used valve 2/2 with electromagnetic activation. The bridging with proposed valve offers a simple design and easy to built, flexible port design, small internal leakage, fast response time up to $4ms$ and small internal friction. Moreover, the bridging reduced the spending of compressed air which good for energy saving. The mass compressed air was identical with mass flow rate in the equation (15) and (16).

However, A_v was designed to be constant. A_v is the active area of the orifice on the given valve.

Other than bridging, cushioning effects at both ends of the cylinder gives impact the positional accuracy of the piston stroke. An extended pneumatic model with cushioning was presented [23]. The model developed based double-acting cylinder with 3/2 on-off valve to drive the system with Pulse-Width-Modulation (PWM) algorithm. The model included the influence of the effect of cushioning section. The mathematical model of cylinder cushions expressed as [23]:

$$\begin{aligned} F_{pcl} &= P_1 A_{rod} + P_{c1} A_2 - P_2 A_2 - P_a A_{rod} \\ F_{pcr} &= P_1 A_1 - P_2 A_2 - P_r A_r - P_a A_{rod} \end{aligned} \tag{25}$$

where F_{pcl} and F_{pcr} are resultant force exerted on the piston at the cushioning sections, P_l and P_r are the pressures at left and right cushioning sections, A_r is the cross sectional area of the piston in cushioning regions and P_{c1} and P_{c2} are the air pressure in the cylinder chamber 1 and 2.

Volume correction for cushioning represented as [23]:

$$\begin{aligned} V_r &= V_{r0} - A_r (x - L_c) \\ V_l &= V_{l0} - A_r (x + L_c) \end{aligned} \tag{26}$$

where V_{r0}, V_{l0} are the volumes of the right and left cushioning regions, and L_c is the distance between mid-stroke of piston and the start point of cushioning section.

Another approach of modeling is a linear mathematical model. A pneumatic positioning system based on binary solenoid valve with time-delay was modeled [5]. The proposed model based on two-position valve as a switching system. Mass flow rate dependent on the upstream and downstream pressure. The linear pressure dynamic model was obtained to reduce order model of mass flow rate. The reduced model was achieved by manipulating orifice size and simplifies the modeling. By exploiting orifice size, a switching mode operation was commanded valve either completely opened or completely closed. Simplifies modeling was achieved by driving pressure either to be supply pressure or atmospheric pressure. The proposed model was cost-effective since the model was not used pressure sensors and proportional valves.

A static state feedback pneumatic actuator linearized by regular static state feedback and coordinate transformation was presented [24]. Two main themes for this model: (1) the input-state linearization where the full state equation was linearized and (2) the input-output linearization where map from input to output was emphasized. The proposed model was discussed by pneumatic cylinder driven by a single five-port proportional valve and two three-port proportional valves. For five-port valve, the system inlet and outlet ports were not independent input. A coordinate transformation was used to linearized the model with respect to z and V. For three-port valves, the system was a multi-input and single output system. Same method used in five-port valve, a system was linearized with two independent input; V_1 and V_2. For both valves, the input-output map was also linerized. For full state feedback, three parameters required to measure: position, velocity, and 2 chambers pressure. As result, three sensors required to measure those parameters. For cost-effective in industrial, a pressure was replace by constant.

For the system identification, a complex mathematical derive from theoretical analysis is not required. A linear time-invariant model was developed based on system identification approach in 1995 [25]. The proposed model was performed by applying a least-square method to identify the plant. The model parameters were obtained by using the reduced second-order model and PID controller.

A linear process model for pneumatic actuator was obtained from experimental data by using system identification in 1997 [26]. The model data was recorded using open-loop test.

Two transfer functions were developed to represent the dynamic of pneumatic rodless cylinder [27]. The first transfer function developed based on linearization about an off-center load position. The second transfer function derived for small perturbation about the mid-stroke position.

In 2001, a model-based control was presented using pole-placement control structure [28]. The proposed model used proportional control to obtain data. The plant model was calculated using least square parameter estimation.

Usually, model based on system identification is a linear function. An open-loop transfer function as expressed in equation (23) was applied [29]

$$G(s) = \frac{y(s)}{r(s)} = \frac{K_s \omega_n^{\ 2}}{s(s^2 + 2\zeta\omega_n s + \omega_n^{\ 2})} \tag{27}$$

where y is the position outputs of the cylinder, r is the input of the system , K_s is the open-loop gain of the system, ζ is damping coefficient of the system and ω_n is the intrinsic frequency.

Other approaches of system identification were black-box modeling and grey-box modeling. Two local linear model structures were introduced to describe the dynamic plant of servo-pneumatic actuators [30]. The two model structures were (1) a black-box model and (2) a grey-box model. A black-box model was developed without needed of physical model interpretation. A gray-box model was developed by friction force between the piston pressure equations in the actuator chambers. Both model developed based on the continuous-time dynamic model of Takagi-Sugeno model. The developed models then were defined as the numbers of local models, the linear model order and the scheduling vector to generate local linear models. A state feedback controller then developed based on local linear models to keep the accuracy of the position control.

System identification was applied to friction force to develop friction compensation. A model friction force variation using the LuGre friction model was presented [31]. The proposed model was applied to the estimation of friction in pneumatic actuator. All dynamic and static friction parameters were assumed to be uncertain and varying remains limited between upper and lower bounds. The proposed model then used to develop friction compensation.

A multi-model PD-control of pneumatic actuator under variable load presented [32]. A linear transfer functions were identified based on PD-controller under different variation load. The PD-controller gain obtained experimentally based on real response of the actual system. The obtained models then were used to develop a multi-model controller. Based on mathematical modeling of pneumatic actuator, a position control can be designed to perform an accurate position control and high performance pneumatic actuator system. The next section is the review of controller design for position controller of pneumatic actuator system.

V. PNEUMATIC ACTUATOR CONTROLLER DESIGN

Pneumatic position control research has become significantly growth in 1990s. Many approaches have been investigated and developed in attempt to overcome difficulties in pneumatic actuators. Some classical control was investigated such as Proportional-Integral-Derivative (PID) controller by [26-27, 33-34]. Besides, there are also some reports on the advanced control such as sliding mode controller (SMC) [15, 35-37], adaptive controller [11, 38] and a combination of controller [39-41].

a. PID Controller

A combination of conventional controller with friction compensation and with another controller method was investigated by several researchers. In 1997, a combination of PID controller, friction compensation, boundary integral action and position feed-forward were implemented [26]. The proposed controller used PWM on-off solenoid valve. The controller coefficient for PID controller was selected based on Isermann's method. The friction compensation was designed based on a simple Coulomb friction model. At zero velocity, the friction force responsible for any steady-state error and the small dead-band. Friction compensation disabled once the steady-state error is within the specified tolerance. The bounded integral action added to control algorithm to accommodate the variations of stiction friction force along the piston stroke. Position feed-forward was applied to reduce following error. Result analysis demonstrated that without affect of steady-state accuracy, the proposed controller robust to varying mass. The designed friction compensation reduced average of steady state error by 40% and following errors to 64mm.

A modified PID controller with acceleration feedback and two nonlinear compensations: (1) time-delay minimization and (2) null offset compensation were presented in 1999 [33]. The two nonlinear compensations were used to solve time-delay problem and dead-zone respectively.

Another friction compensation method developed by identified the differences behavior of real and virtual loads. The proposed method was combined with a PID velocity feed-forward and feedback (PID-VF) controller [42]. The results show that the proposed method was high degree of accuracy.

Instead of nonlinearities, several limitations exist in the derivative of the references signal and disturbance of piston velocity. A proportional feedback force controller with saturation was presented to solve the limitations in 2006 [43].

A fuzzy PD-controller by using a trapezoid type 25 rules adopt by Mamdani model was designed [44]. Results demonstrate that system is stable resistance to disturbance and can be applied on any configurations of pneumatic servo drive.

A P, PI, PD and PID controller were study and simulation experiment were performed [45]. Three controller gains K_p, K_i and K_d for P-I-D terms respectively estimated by simulation controller design. From the simulation result, target position for P-controller reached permanent oscillation. Permanent oscillation was unacceptable for positioning system. In order to eliminate oscillation, PD-controller was introduced. PD-controller improved the system performance and eliminated oscillation. PD-controller also minimized time rise from $0.15s$ to $0.1s$. PI-controller was simulated to study the performance of the system. From the result, the rise time of system response worse than the PD-controller. However, the systems had a constant position error on both target position. The result leaded to the implementing PID-controller. The rise time closed to $0.15s$ and steady state error was below than $0.1mm$. The PID-controller however increased overshoots as time increased. An alternate classical controller was introduced to eliminate this behavior. An auto-selective classical PID consists of combinational of PID-controller (t-PID). From the simulation results, time increased and steady state error were eliminated. The proposed controller however was difficult to tune. It must first tune via simulation before implemented to the real plant.

Several researchers improved the PID controller by enhanced the gain tuning method. In 1999, a PD controller was developed for repeatability of pneumatic rodless cylinder system [27]. Conventional PD controller, the gain K_p and K_d was determined by using the root locus techniques. Instead of using rood locus technique, a pragmatic method was proposed to determine controller gains. The proposed gain tuning method achieved satisfactory nominal transient characteristics over the range of the operating requirement. The proposed controller was test under three effects of repeatability: (1) loading conditions, (2) start-stop position in the work envelope and (3) changing the proportional gain K_p. The repeatability of the system under mentioned condition less than $0.3mm$.

Another gain tuning method were neural network and nonlinear observer that presented in 2002 [34]. The proposed controller consists of outer controller and inner controller. For inner controller, a PID feedback linearization was designed to nullify the nonlinearities due to compressibility of air while. For outer controller, a PID position control augment with friction compensation was developed. Two friction compensations were developed using neural network and nonlinear observer. Experimental and simulation results show that friction compensation with neural network reduced 75% of RMS error compared to without compensation. However, nonlinear observer is more stable than neural network for friction compensation.

Other controller approach that used neural network as gain tuning method was proposed in 2008 where a Proportional-Integral (PI) feedback controller was developed to get a suitable gain for each feedback [21]. Neural network provides suitable gains for PI controller depending on feedback representing changes in position error and changes in external load force. Proposed methods keep the response of position with steady-state error and rise time. Result show that jittering was eliminated.

Further study of gain tuning method was proposed based on Ziegler-Nicholas method [46]. The result shows that good performance tracking is achieved with minimum sustained oscillation. From the statistical analysis, fourth-order model show the best performance with one-step prediction. Proposed controller has high accuracy in the identification system, fast tuning control parameter and low cost.

In 2010, a PID controller tuning method based on optimization technique was proposed [47]. Sequential Quadratic Programming (SQP) was used as a base of optimal PID tuning method. Sub problem of Quadratic Programming (QP) solve via Infeasible Interior Point Method (IIPM) and step length by linear search. Experimental evaluation demonstrates that systems are consistently reduced response time and overshoot. Proposed method improves the robustness and effectiveness of PID controller numerical optimization of the systems.

Based on several fixed Proportional-Derivative (PD) controller, a multi-model controller was presented [32]. A multi-model controller solved the disadvantages of the classical robust and adaptive controller with enhanced the large limitation bounds in the system. The switching algorithm applied to determine the best model and selected the corresponding controller for any load condition. The proposed controller used velocity as the feedback to the control loop. The velocity in the controller was estimated by using Kalman filter. Kalman filter minimized the measurement noise and improved time-delay compared to Gaussian filters. Experimentally results shows that the proposed controller performed well over the full range of load variations.

b. SMC Controller

A robust SMC based on uncertain bound was investigated in 1997 [35]. Any bounded uncertainties in parameters of pneumatic servo system were investigated instead of lower and upper bound. A proposed controller was designed so that the error dynamic can converges to zero as time $t \to \infty$. However, undesired frequencies dynamics was excited due to the discontinuity from the proposed controller. The undesired frequencies were eliminated by smoothing the control law in the neighborhood of boundary layer. Simulation and practical applied to the servo system has proven that the proposed controller can guarantee the tracking error can converges to zero as time $t \to \infty$. Nevertheless, the controller scheme can only be used in a second-order pneumatic servo system.

There are several researchers who investigated the using of SMC. In 2000, an investigate to the effect of reducing order model-based controller was reported by using two force SMC controller [36]. First controller was developed based on detailed mathematical model of pneumatic system that addressed the dynamic of the valves and the effect of connecting tube. Second controller was developed by reduced order controller. For second controller, the valve dynamics and the time delay due to the tube was neglected. However, for first controller, a complex online computation was needed for control law. Both controllers were validated by numerical simulations and experiments. First controller provided slower performance at higher frequencies and when long connecting tube was used.

Next, another SMC controller effect was investigated in 2007 for linear plant and nonlinear plant [48]. Results demonstrated that, if payload is bigger than nominal, SMC linearized (SMCL) is more robust than SMC nonlinear (SMCN) if payload is bigger than nominal. On the other hand, if payload is smaller than nominal, SMCN is more robust than SMCL. Root-mean-square-error (RMSE) for SMCL is 0.51mm while SMCN is 0.42mm. Tracking error for SMCN is better 18% than SMCL.

In 2005, an investigation to a SMC with fuzzy logic approach for gain tuning was reported where giving advantages on energy efficiency performance [15]. The proposed controller solved the difficulties in knowledge of the cylinder dynamic pressure and disturbance friction. The fuzzy gain tuning was developed based on Mamdani-type model to tune these gains adaptively

Further enhancement was introduced with double line and modified saturation function was to eliminate chattering effect in 2006 [49]. By evaluate with external disturbances and variation of system parameters, system shows good performance, robust and chattering effect was eliminated. This method does not require an accurate modeling to achieve good performance.

A multiple surface Sliding Controller (MSSC) was then proposed in 2008[37]. The proposed control algorithm performed by closed-loop based on Lyapunov's stability method. The benefits of this controller include that regardless the uncertainties and time-varying payload, the system is good in tracking performance. However, controller used SMC caused a chattering effect.

An improvement to reduce chattering phenomenon on the switching control signal was realized with a high- order SMC with robust differentiator that was presented in 2008 [50]. The controller used third-order SMC based on optimal linear quadratic control by using acceleration feedback. The acceleration was estimated by using differentiator via second-order SMC. Experimental and simulation results show that the maximum position error was less than $1mm$ and the steady-state position error was about $1\mu m$.

Next, another improvement on reducing chattering effect on the switching in SMC was introduced by a discrete fuzzy SMC for electro-pneumatic proportional system [29]. The proportional coefficient and differential coefficient in SMC switch was designed by pole assignment method. The control law was based on exponential reaching law. Based on Fuzzy Mamdani model, two inputs and two outputs fuzzy control was designed where switch function and its error as input while gain parameter of the symbol function and speed parameter in the exponential control system as outputs. The fuzzy control softening the control signals and weakened the chattering effect. Results show that interference was eliminated.

Also reduced chattering effect, in 2010, a chattering-free robust variable structure controller (CRVC) was reported for position control [51]. CRVC was designed by combined the conventional SMC with the contriving feedback gain function. Experimental and simulation result show that the advantages of conventional SMC retain while contriving feedback gain function reduced the chattering effect.

c. Adaptive Controller

For alternate control, in 2007 an adaptive control of pressure in chamber for pneumatic-pressure-load system was designed by considering the state-parameters [52]. The designed used linear quadratic Gaussian self-tuning pressure regulator along with the unknown parameter that were estimated by using recursive forgetting factor least-squares (RFFLS) and state parameter by Kalman's filter. Simulation and experimental results show that overshoot, steady-state error and response time have been effectively improved. Designed controller was tested to parametric variation and it shows that an adaptive control for pneumatic-pressure-loads systems successfully achieved.

Next, a self-tuning control of low-friction pneumatic cylinder actuator under influence of gravity was investigated in 2001 [28] where low-friction pneumatic actuator provides small and predictable friction characteristics. A self-tuning of balance pressure control was performed by pole-placement control method to counteract the external force including gravity force in the pneumatic system. It is proven that the proposed controller was able to adapt changeable plant parameters including balancing term. The parameter vectors were calculated by recursive least squares identification equation.

Another improvement was to the self-tuning adaptive control by combining with discrete variable structure control (ADVSC) was proposed to reduce the chattering effect [39]. Both controllers worked in parallel and position output was added in cascade. A self-tuning adaptive control was designed based on autoregressive moving average model and least square parameter estimation algorithm. Online parameter identification scheme of self-tuning adaptive control can keep optimum sliding surface at anytime.

In 2009, the increasing of friction force at the seals based on LuGre model then was taken into account for an adaptive friction compensation [11]. The proposed controller consists of two back-stepping control loops in cascade: (1) underlying control loop and (2) outer control loop. Underlying control loop controlled internal pressure of the left and right chamber of the cylinder. For outer loop, the carriage position and the mean pressure of the two chambers were treated as a control variable. Static friction was compensated in a feed-forward manner. Remaining uncertainties was compensated by LuGre observer which its parameters were estimated by adaptive back-stepping control. Back-stepping was design based on Lyapunov stability theory. Results demonstrate that maximum steady-state position error becomes below $1mm$ and maximum pressure error below $0.2bar$.

d. Hybrid Controller

Another alternate control was a combination of controllers. In 2004, investigation on hybrid iterative learning and SMC for an electro-pneumatic was reported [53]. The iterative learning controller solved the problem of the repeatability of pneumatic actuators at low velocity. Meanwhile, the SMC makes the system robust.

Next, another improvement was introduced in 2001. A hybrid fuzzy with PID controller based on kinematics control of pneumatic actuator has been proven to be suitable for high accuracy position control [54]. The proposed kinematics based of fuzzy logic was to control the piston when the pistons located far away from the desired position and then switched to PID controller when the piston position located near the desired position.

In 2005, two algorithms for comparison of pneumatic servo position control tracking accuracy was performed [40]. First algorithm was a combination of Position-Velocity-Acceleration (PVA), feed-forward and dead zone compensation. The second algorithm was SMC. PVA in first algorithm designed based on pole-placement method. Based on the experiment, the second algorithm is shown as less robust than first algorithm but shows a good tracking performance. RMSE for first algorithm and second algorithm was $1.24mm$ and $0.51mm$ respectively. Both steady state errors were not exceeding $\pm 0.01mm$. Results show that the tracking error for second algorithm was 59% and $0.4mm$ less than first algorithm.

Further investigation was employed for accurate position control. A hybrid controller for positioning control of pneumatic system with PZT actuator has been presented [17] that consist of two controller: (1) the PI controller to get positioning accuracy about $10\mu m$ and (2) the PZT actuator coupled with differential pressure to get positioning accuracy about $10nm$. Results demonstrated that sliding table were successfully positioned form references origin to the target position of $4mm$ in $0.924s$.

e. Iterative Learning Controller

An iterative learning control algorithm combine with feedback loop control was presented [4]. Pneumatic actuators application such as blow molding and glass forming required a repeated on a particular job. The previous tracking information is used to improved present tracking performance. Based on simulation result, the proposed controller position tracking error converged near to zero after iteration. For the new iteration, the continuous tracking error was improved.

In 2008, a practical hybrid piecewise controller with combination of bang-bang controller, PD and fuzzy logic proportional plus conventional integral-derivatives (FP+ID) algorithm have been introduced [55]. Bang-bang controller was used when output pressure faraway from set point and the controller can accelerate output response to the desired error bound quickly. PD controller was used to increase the stability of overall feedback control system while FP+ID controller was designed to eliminate the steady-state error. Pressure error used as a switching condition for the chosen controller. Experiment and simulation shows that the overall system with the practical hybrid piecewise controller has a good performance to the pneumatic system.

Next, another investigation was performed to achieve fast transient response that is good for engine control performance. An iterative model-based predictive control strategy was proposed which consists of combination of proportional and integral control with combination of predictive lift control in a feed-forward manner [41]. The proposed controller was divided into two portions: (1) the state estimation of displacement and velocity based on displacement feedback and (2) iterative model-based displacement prediction. Results show that estimated and actual displacement was around $9mm$.

f. Feedback Linearization Controller

A linear constant state feedback combined with feedback linearization with disturbance rejection was proposed [56]. A single-input nonlinear system was linearized by applying the coordinate and input transformation. The disturbance rejection was designed by measuring disturbance that useful for pneumatic with static friction. The proposed controller performance was compared with linear constant state feedback without linearization controller and the linear constant state feedback with feedback linearization controller. Results show that, the linear constant state feedback without linearization controller performed large differences between simulation and experiment result. For the linear constant state feedback with feedback linearization, the linearity was improved and the experimental result founds similar to the simulation. However, the steady-state error was large due to the static friction. The proposed controller improved all responses performance compared to the linear constant state feedback and reduced steady-state error caused by hysteresis.

In 2002, a tracking position control based on feedback linearization control method was proposed for linear positioning system [34]. The proposed controller consists of two control loops: (1) an inner pressure control loop and (2) an outer position control loop. Pressure controller which is using PID controller with feedback linearization is to nullify the nonlinearity due to compressibility of air. Otherwise, the position control was designed with the friction compensation using either: (1) the multi-layered perception type neural network, or (2) reduced-order nonlinear observer. For neural network, a proper input was calculated using neural network to counteract the friction and linearize the plant dynamics. For nonlinear observer, the friction was estimated by an observer and was assumed to be proportional to the sign of the velocity.

Another improvement was introduced for a position control of an electro-pneumatic system using back-stepping design based on Lyapunov function to guarantee the stability of controller performance [38]. Back-stepping offer a more flexible way of dealing with uncertainties compared to the conventional feedback linearization. The proposed controller was classes to strict-feedback system. A linear control strategy then was used to validate the proposed controller. Experimental results show that, in the dynamic stage, the maximum position error and mean standard deviation was $1.62mm$ and $80\mu m$, while linear controller was $4.26mm$ and $135\mu m$ respectively. In other hand, the mean position error and mean standard deviation of proposed controller was $100\mu m$ and $40\mu m$ while for linear controller was $153\mu m$ and $40\mu m$ respectively for static stage. From the results, the position tracking of proposed controller was improved.

Further improvement was proposed based on a static state feedback linearization of input-output map model [24]. The proposed controller was developed to drive the error state to zero. The problem for feedback linearization was the exact feedback cancellation. This problem was solved by classified nonlinear dynamic as uncertainties. Two aspects were considered in this control strategy: (1) reduce the time delay caused by static friction and (2) reduce the tracking error caused by the friction term of F_c. Time delay was reduced by leads the valves to fully open that gives a maximum compressed air flow rate that leads to fast starting response. F_c was obtained by estimated decreased acceleration.

Next, a feedback linearization combined with a linear feedback control was developed to compensate the nonlinear effect due to the characteristics of pressure dynamic in chamber [57]. The proposed control law employs the position, velocity and acceleration of the piston state variables. The values of feedback gains were calculated by using pole-placement method. The proposed controller was validated by comparison with classical linear state-feedback controller based on the feedback of the signal of position, velocity and acceleration controller. It has been proven that the proposed controller reduced amplitude of trajectory error compared to PVA controller. From simulation result, the proposed RMS value of the position tracking error was at least 50% inferior to PVA controller.

In 2003, a Quantitative-Feedback-Theory (QFT) control for linearized pneumatic positioning system was investigated to meet limited bandwidth [58]. This controller consists of QFT and outer loop control. The QFT control was designed to control with no limit cycle caused, whereas the outer loop control was designed to control position accuracy.

A fuzzy gain scheduler model-based controller design based on a combination of stabilizing local state feedback controller was proposed to control pneumatic actuator motion in 2004 [30]. A local linear model was derived from a black-box and grey-box modeling based on Takagi-Sugeno model that was discussed in the modeling section [30]. The proposed controller was formulated as an interpolation of local linear state feedback controller which designed by pole-placement based on a state space model. The proposed controller has been proven matched the closed-loop requirement at discrete variation of the payload and variation of piston position displacement.

Further improvement to the feedback linearization technique has been employ with friction compensator control strategy that was presented in 2006 [31]. The proposed controller consists of inner force controller, friction compensation and outer position control. For inner force controller, feedback linearization technique based on PI-controller was developed where the nonlinear structure was linearized depending on the parameter uncertainties. The friction compensation then was developed to track position trajectory under influence of friction based on LuGre model. After that, the outer position control was developed by formulating as interconnected linear and passive nonlinear subsystems. The proposed friction compensator was validated experimentally. The result was compared with adaptive coulomb friction estimator and static friction estimator. The results show that, by using static friction estimator, the systems lack of dynamic friction compensation especially when actual force is small. The adaptive coulomb friction is proved improved the static friction estimator. However, the adaptive Coulomb friction had performed bad tracking. On the other hand, the friction compensated using LuGre model demonstrated the best performances of the friction compensation.

g. Others Controller

In 2002, a similar control law of traditional SMC was presented for LTI switching system with time delay of pneumatic actuators [5]. The valve used was on on-off solenoid type. The proposed controller was designed with no equivalent term in the traditional SMC. Based on Lyapunov function, stability and error convergence can be directly enforced. Multiple valves with multiple discrete positions incorporated into control law by forming all possible permutation of the discrete position of the valves. On-off solenoid valves caused time-delay in the systems. To apply proposed controller with time-delay, future state values are required in order to select preset input. Also, a model-based predictor is used to estimate future state values.

Another approach to improve the performance of pneumatic actuator is PWM technique that transforms a discontinuous switching model to a linear continuous was presented [59]. The linearization utilizing loop-shaping methods is used to stabilize the robustness of the system and avoid saturation limit. Experiments demonstrated the proposed method produces a good tracking performance.

In 2005, a PWM-driven solenoid valve was introduced to control the exhaust flow from the cylinder [60]. Duty cycles of PWM were determined from modified PD-controller. The methods giving advantages of reducing the system cost.

Next, a high speed solenoid valve based on PWM algorithm was proposed to improved the reliability of the system [61]. Result shows that response speed and position accuracy were meet by using fast switching values.

Other control strategy has been investigated in 2004. A Genetic-Algorithm based fixed-structure robust $H\infty$ loop-shaping control has been proven successfully achieved good robust performance [62]. The designed controller was compatible with the specified open loop shape.

A Generalized Predictive Controller (GPC) was then proposed in 2008 for pneumatic-pressure-load system [63]. System parameters were estimated using RFFLS. Thus, this control strategy had reduced the dead-zone and eliminated the overshoot but long time response.

Also in 2008, a Multiple Input Single Output (MISO) nonlinear position control law was developed by using back-stepping method [64]. The controller was design based on Lyapunov’s function included the effects of friction modeling error and valve modeling error. As a result, the overall steady-state error was approximated to $0.05mm$. Steady-state error remains $\pm 0.05mm$ although the diameter of cylinders is different.

Another controller strategy has been employed in 2010. An Exact Linearization Control (ELC) based on dynamic model of valve was proven to be suitable for improved then dynamic pressure step response [65]. The unknown parameters were estimated by using reference volume. The differences of expected and real mass flow rate were assume as disturbance and used as input for feed-forward transfer function. The proposed ELC improved the dynamic pressure step response.

VI. SUMMARY

The dynamic model widely describe by using nonlinear theoretical mathematical modeling. The major nonlinear mathematical modeling is based on piston load dynamics, the pressure, volumes, and temperature of air in the cylinder, and the mass flow rate through the valve. Friction makes the system difficult to control. Based on the review of nonlinear mathematical modeling, the enhancements were done to the air flow condition and friction model. For linear model, most linear model is derived by simplified the nonlinear mathematical modeling and using static-state feedback and coordinate transformation. Meanwhile, the system identification modeling method was developed based on experimental data by identifing the plant using pole-placement and least-square method. Complete mathematical modeling however provides the best fits of dynamic description of pneumatic system. However, complete modeling faces complexity mathematical equation. The improvement can be done to reduce the complexity of the modeling. Physical elements such as long connecting tube effect and cushioning effect were taken into account for some other enhancement on pneumatic model.

For controller design, there are two major improvement areas were investigated to the PID controller method: (1) the PID gain tuning method and (2) the PID controller augmented with nonlinear compensation. Instead of this two major area, a PID controller was investigated to improve the limitation bounds in the classical robust controller. Applied conventional SMC method to pneumatic system caused chattering effects. Based on the review, the trends of improving the SMC have enhanced the tuning gain and developed a high-order SMC. Thus, the chattering effect can be reduced. Boundary is one of SMC limitation. For further improvement, the wide range of boundary and other approaches of gain tuning algorithm can be investigated. Another control method applied to pneumatic actuator was feedback linearization controller method. The trend of the controller development was the combination with the nonlinear compensation. The objective of the feedback linearization controller was to reduce the steady state error of the system. A combination of controllers was investigated to be applied to pneumatic actuator system. The combination was applied to eliminate one controller limitation while keeping its advantages.

REFERENCES

[1] J. L. Shearer, "Study of Pneumatic Process in the Continuous Control of Motion With Compressed Air," *Transactions of the ASME,* pp. 233-249, 1956.
[2] J. Wang*, et al.*, "Accurate position control of servo pneumatic actuator systems: an application to food packaging," *Control Engineering Practice,* vol. 7, pp. 699-706, 1999.
[3] A. Messina*, et al.*, "Experimenting and modelling the dynamics of pneumatic actuators controlled by the pulse width modulation (PWM) technique," *Mechatronics,* vol. 15, pp. 859-881, 2005.
[4] Y. Shaojuan*, et al.*, "A New Iterative Learning Controller for Electro-Pneumatic Servo System," in *Intelligent Systems Design and Applications, 2008. ISDA '08. Eighth International Conference on*, 2008, pp. 101-105.
[5] E. J. Barth and M. Goldfarb, "A Control Design Method for Switching Systems With Application to Pneumatic Servo Systems," *ASME Conference Proceedings,* vol. 2002, pp. 463-469, 2002.
[6] H.-S. Choi*, et al.*, "Development of hybrid robot for construction works with pneumatic actuator," *Automation in Construction*, vol. 14, pp. 452-459, 2005.
[7] A. A. M. Faudzi*, et al.*, "Distributed Physical Human Machine Interaction Using Intelligent Pneumatic Cylinders," in *Micro-NanoMechatronics and Human Science, 2008. MHS 2008. International Symposium on*, 2008, pp. 249-254.
[8] S. Sang-Eun*, et al.*, "Development of a pneumatic robot for MRI-guided transperineal prostate biopsy and brachytherapy: New approaches," in *Robotics and Automation (ICRA), 2010 IEEE International Conference on*, 2010, pp. 2580-2585.
[9] T. Kubo*, et al.*, "Development of Active 80-faced Polyhedron for haptic physical human-machine interface," in *Intelligent Robots and Systems, 2009. IROS 2009. IEEE/RSJ International Conference on*, 2009, pp. 1259-1264.
[10] S. Rajendran and P. S. Nanda, "Electro-Pneumatic Servo System," presented at the 14th National Conference on Machines and Mechanisms (NaCoMM09), NIT, Durgapur, India, 2009.
[11] D. Schindele and H. Aschemann, "Adaptive friction compensation based on the LuGre model for a pneumatic rodless cylinder," in *Industrial Electronics, 2009. IECON '09. 35th Annual Conference of IEEE*, 2009, pp. 1432-1437.
[12] H. I. Ali*, et al.*, "Mathematical and Intelligent Modeling of Electropneumatic Servo Actuator Systems," *Australian Journal of Basic and Applied Sciences*, vol. 3, pp. 3662-3670, 2009.

[13] E. Richer and Y. Hurmuzlu, "A High Performance Pneumatic Force Actuator System: Part I---Nonlinear Mathematical Model," *Journal of dynamic systems, measurement, and control,* vol. 122, pp. 416-425, 2000.
[14] V. Blagojevi and M. Stojiljkovi "Mathematical and simulink model of the pneumatic system with bridging of the dual action cylinder chambers," *Facta universitatis-series: Mechanical Engineering,* vol. 5, pp. 23-31, 2007.
[15] B. XU, *et al.*, "A SLIDING MODE FUZZY FORCE TRACKING CONTROLLER FOR PNEUMATIC CYLINDERS," 2005.
[16] E. A.Perondi, *et al.*, "Feedback linearization control applied to a pneumatic actuator system."
[17] Y. T. Liu and C. C. Jiang, "Pneumatic actuating device with nanopositioning ability utilizing PZT impact force coupled with differential pressure," *Precision engineering,* vol. 31, pp. 293-303, 2007.
[18] I. L. Krivts, "Optimization of performance characteristics of electropneumatic (two-stage) servo valve," *Journal of dynamic systems, measurement, and control,* vol. 126, p. 416, 2004.
[19] Ž. Šitum, *et al.*, "Nonlinear mathematical model of a servo pneumatic system."
[20] V. Geleževi ius and A. Grigaitis, "Investigation of dynamic behaviour of nonlinear electropneumatic acting system," *Electronics and Electrical Engineering.– Kaunas: Technologija*, pp. 35–40, 2006.
[21] G. Kothapalli and M. Y. Hassan, "Design of a Neural Network Based Intelligent PI Controller for a Pneumatic System," *IAENG International Journal of Computer Science,* vol. 35, pp. 217-222, 2008.
[22] A. Voda and L. Ravanbod-Shirazi, "High Performance Position Tracking With Friction Compensation For An Electro-Pneumatical Actuator," *Journal of Control Engineering and Applied Informatics,* vol. 6, pp. 15-33, 2004.
[23] F. Najafi, *et al.*, "Dynamic modelling of servo pneumatic actuators with cushioning," *The International Journal of Advanced Manufacturing Technology,* vol. 42, pp. 757-765, 2009.
[24] J. Wang, *et al.*, "Tracking control of nonlinear pneumatic actuator systems using static state feedback linearization of the input-output map," in *Proceedings of the Estonian Academy of Sciences, Physics and Mathematic*, 2007, p. 47.
[25] M.-C. Shih and S.-I. Tseng, "Identification and position control of a servo pneumatic cylinder," *Control Engineering Practice,* vol. 3, pp. 1285-1290, 1995.
[26] R. B. Van Varseveld and G. M. Bone, "Accurate position control of a pneumatic actuator using on/off solenoid valves," *Mechatronics, IEEE/ASME Transactions on,* vol. 2, pp. 195-204, 1997.
[27] S. Fok and E. Ong, "Position control and repeatability of a pneumatic rodless cylinder system for continuous positioning," *Robotics and Computer-Integrated Manufacturing,* vol. 15, pp. 365-371, 1999.
[28] R. Richardson, *et al.*, "Self-tuning control of a low-friction pneumatic actuator under the influence of gravity," *IEEE Transactions on Control Systems Technology,* vol. 9, pp. 330-334, 2001.
[29] Y. Luo, *et al.*, "Research on the Fuzzy Sliding-Mode Control for the Electro-Pneumatic Proportional System," in *Fuzzy Systems and Knowledge Discovery, 2009. FSKD '09. Sixth International Conference on,* 2009, pp. 156-159.

[30] H. Schulte and H. Hahn, "Fuzzy state feedback gain scheduling control of servo-pneumatic actuators," *Control Engineering Practice,* vol. 12, pp. 639-650, 2004.
[31] G. Carducci, *et al.*, "Identification of viscous friction coefficients for a pneumatic system model using optimization methods," *Mathematics and Computers in Simulation,* vol. 71, pp. 385-394, 2006.
[32] M. Taghizadeh, *et al.*, "Multimodel PD-control of a pneumatic actuator under variable loads," *The International Journal of Advanced Manufacturing Technology,* vol. 48, pp. 655-662, 2010.
[33] J. Wang, *et al.*, "A practical control strategy for servo-pneumatic actuator systems," *Control Engineering Practice,* vol. 7, pp. 1483-1488, 1999.
[34] H. K. Lee, *et al.*, "A study on tracking position control of pneumatic actuators," *Mechatronics,* vol. 12, pp. 813-831, 2002.
[35] J. Song and Y. Ishida, "A robust sliding mode control for pneumatic servo systems," *International journal of engineering science*, vol. 35, pp. 711-723, 1997.
[36] E. Richer and Y. Hurmuzlu, "A High Performance Pneumatic Force Actuator System: Part II---Nonlinear Controller Design," *Journal of dynamic systems, measurement, and control*, vol. 122, pp. 426-434, 2000.
[37] Y. C. Tsai and A. C. Huang, "Multiple-surface sliding controller design for pneumatic servo systems," *Mechatronics,* vol. 18, pp. 506-512, 2008.
[38] M. Smaoui, *et al.*, "A study on tracking position control of an electropneumatic system using backstepping design," *Control Engineering Practice*, vol. 14, pp. 923-933, 2006.
[39] M. H. Chiang, *et al.*, "Large stroke and high precision pneumatic-piezoelectric hybrid positioning control using adaptive discrete variable structure control," *Mechatronics,* vol. 15, pp. 523-545, 2005.
[40] N. Shu and G. M. Bone, "Experimental comparison of two pneumatic servo position control algorithms," in *Mechatronics and Automation, 2005 IEEE International Conference*, 2005, pp. 37-42 Vol. 1.
[41] G. G. Zhu, *et al.*, "An iterative algorithm for model-based predictive control of an Electro-Pneumatic Valve Actuator," in *American Control Conference, 2009. ACC '09.*, 2009, pp. 2684-2689.
[42] A. Saleem, *et al.*, "Mixed-reality environment for frictional parameters identification in servo-pneumatic system," *Simulation Modelling Practice and Theory,* vol. 17, pp. 1575-1586, 2009.
[43] A. Ilchmanna, *et al.*, "Pneumatic cylinders: modelling and feedback force-control," *International Journal of Control,* vol. 79, pp. 650-661, 2006.
[44] J. E. Takosoglu, *et al.*, "Rapid prototyping of fuzzy controller pneumatic servo-system," *The International Journal of Advanced Manufacturing Technology*, vol. 40, pp. 349-361, 2009.
[45] M. Papoutsidakis, *et al.*, "Modeling and simulated control of non-linear switching actuation systems," presented at the Proceedings of the 8th WSEAS international conference on System science and simulation in engineering, Genova, Italy, 2009.
[46] A. Saleem, *et al.*, "On-line identification and control of pneumatic servo drives via a mixed-reality environment," *The International Journal of Advanced Manufacturing Technology*, vol. 40, pp. 518-530, 2009.

[47] X. Liang, *et al.*, "A novel PID controller tuning method based on optimization technique," *Journal of Central South University of Technology,* vol. 17, pp. 1036-1042, 2010.
[48] G. M. Bone and S. Ning, "Experimental comparison of position tracking control algorithms for pneumatic cylinder actuators," *Mechatronics, IEEE/ASME Transactions on,* vol. 12, pp. 557-561, 2007.
[49] P. Korondi and J. Gyeviki, "Robust Position Control for a Pneumatic Cylinder," in *Power Electronics and Motion Control Conference, 2006. EPE-PEMC 2006. 12th International*, 2006, pp. 513-518.
[50] M. Smaoui, *et al.*, "High order sliding mode for an electropneumatic system: A robust differentiator–controller design," *International Journal of Robust and Nonlinear Control,* vol. 18, pp. 481-501, 2008.
[51] T. Y. Choi and J. J. Lee, "Control of Manipulator Using Pneumatic Muscles for Enhanced Safety," *Industrial Electronics, IEEE Transactions on,* vol. 57, pp. 2815-2825, 2010.
[52] X.-S. Wang, *et al.*, "Modeling and self-tuning pressure regulator design for pneumatic-pressure-load systems," *Control Engineering Practice,* vol. 15, pp. 1161-1168, 2007.
[53] S. J. Yu, *et al.*, "Practical Design of an Iterative Learning-Sliding Mode Controller for Electro-Pneumatic," *International Journal of Information Technology,* vol. 11, 2004.
[54] M. Parnichkun and C. Ngaecharoenkul, "Kinematics control of a pneumatic system by hybrid fuzzy PID," *Mechatronics,* vol. 11, pp. 1001-1023, 2001.
[55] L. Bo, *et al.*, "Modeling and control of the pneumatic constant pressure system for zero gravity simulation," in *Advanced Intelligent Mechatronics, 2008. AIM 2008. IEEE/ASME International Conference on*, 2008, pp. 688-693.
[56] T. Kimura, *et al.*, "Feedback linearization for pneumatic actuator systems with static friction," *Control Engineering Practice,* vol. 5, pp. 1385-1394, 1997.
[57] E. A.Perondi, *et al.*, "Feedback linearization control applied to a pneumatic actuator system," presented at the XVIII Congresso Brasileiro de Automática, Bonito-MS, 2010.
[58] F. Xiang and J. Wikander, "QFT control design for an approximately linearized pneumatic positioning system," *International Journal of Robust and Nonlinear Control,* vol. 13, pp. 675-688, 2003.
[59] E. J. Barth, *et al.*, "Control design for relative stability in a PWM-controlled pneumatic system," *Journal of dynamic systems, measurement, and control,* vol. 125, p. 504, 2003.
[60] M. B. Thomas, *et al.*, "A novel, low-cost pneumatic positioning system," *Journal of manufacturing systems*, vol. 24, pp. 377-387, 2005.
[61] Z. Situm, *et al.*, "High speed solenoid valves in pneumatic servo applications," in *Control & Automation, 2007. MED '07. Mediterranean Conference on*, 2007, pp. 1-6.
[62] S. Kaitwanidvilai and M. Parnichkun, "Genetic-Algorithm-Based Fixed-Structure Robust H Loop-Shaping Control of a Pneumatic Servosystem," *Journal of robotics and mechatronics,* vol. 16, pp. 362-373, 2004.

[63] P. Chaewieang, *et al.*, "Pressure control of pneumatic-pressure-load system using generalized predictive controller," in *Mechatronics and Automation, 2008. ICMA 2008. IEEE International Conference on*, 2008, pp. 788-791.
[64] Z. Rao and G. M. Bone, "Nonlinear modeling and control of servo pneumatic actuators," *Control Systems Technology, IEEE Transactions on,* vol. 16, pp. 562-569, 2008.
[65] S. Lambeck and C. Busch, "Exact Linearization Control for a pneumatic proportional pressure control valve," in *Control and Automation (ICCA), 2010 8th IEEE International Conference on*, 2010, pp. 22-27.