Dynamical Adaptive Backstepping-Sliding Mode Control for Servo-pneumatic Positioning Applications: Controller Design and Experimental Evaluation

by
Ramhuizaini Abd. Rahman

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Department of Mechanical Engineering University of Manitoba Winnipeg, Manitoba

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Abstract

Servo control of pneumatic actuators is difficult due to the high compressibility and non-linear flow of air. Friction as well as uncertainties in the parameters and model characterizing dynamics of the pneumatic systems further contribute to control challenges. These drawbacks cause stick-slip motion, larger tracking error and limit cycles, which degrades the control performances. Selection of a controller that satisfies requirements of the performing tasks is thus crucial in servo-pneumatic applications. This thesis focuses on the design and experimental evaluation of a model-based, nonlinear controller known as Dynamical Adaptive Backstepping-Sliding Mode Control (DAB-SMC). Originally designed for chemical process control and applied only in simulations, the DAB-SMC is adopted in this thesis and applied to the new area of servo-pneumatic control of a single-rod, double acting pneumatic cylinder and antagonistic pneumatic artificial muscles (PAMs). The controller is further enhanced by augmenting it with LuGre-based friction observers to compensate the adverse frictional effect presents in both actuators. Unlike other research works, the actuators are subject to a varying load that influences control operations in two different modes: motion assisting or resisting. The implementation of DAB-SMC for such servo-pneumatic control application is novel. The mass flow rates of compressed air into and out of the actuators are regulated using one of the following valve configurations: a 5/3-way proportional directional valve, two 3/2-way or four 2/2-way Pulse Width Modulation (PWM)-controlled valves. Over the entire range of experiments which involve various operating conditions, the DAB-SMC is observed to track and regulate the reference input trajectories successfully and in a stable manner. Average root mean square error (RMSE) values of tracking for cylinder and PAMs when the compressed air is regulated using the 5/3-way proportional valve are 1.73mm and 0.10°, respectively. In case of regulation, the average steady-state error (SSE) values are 0.71mm and 0.04°, respectively. The DAB-SMC exhibits better control performance than the standard PID and
classical SMC by at least 33%. The DAB-SMC also demonstrates robustness for up to 78% in uncertainty of load parameter. When the control valve is replaced by the PWM-controlled valves of 3/2-way and 2/2-way configurations, performance is slightly compromised.
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Last but not least, I would also like to place on record, my sense of gratitude to one and all, who directly or indirectly, have lent their hand in this venture.
Dedication

This thesis is especially dedicated to my parents, Hozaimah Zakaria and Abd. Rahman Ismail, who have brought me up with unconditional love and educates me to be a person who benefits others. I am truly thankful for having both of you in my life.
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List of Symbols

Parameters and variables

A
amplitude of sinusoidal reference input trajectory

\(A_{i(=1,2)}\)
piston annulus areas

\(A_r\)
cross-sectional area of piston rod

\(A_v\)
valve orifice area

b
thread length of pneumatic artificial muscle

\(b(\cdot)\)
function describing system’s control gain

\(c_p\)
specific heat at constant pressure

\(c_v\)
specific heat at constant volume

\(C_d\)
discharge coefficient of valve

D
diameter of pneumatic artificial muscle

e
error variable

\(f(\cdot)\)
function describing system dynamic

\(F(\cdot)\)
boundary function of system dynamic

\(F_a\)
actuating force

\(F_C\)
Coulomb friction

\(F_f\)
friction force

\(F_i(=1,2)\)
force generated by each of the pneumatic artificial muscles

\(F_L\)
external load

\(F_S\)
static friction

g
gravity

\(g(\cdot)\)
function describing Striebeck effect in LuGre friction model

G
robustness gain

I
total moment of inertia

\(k_i\)
controller gains

\(k_{sm}\)
smooth function gain

\(k_p\)
control gain of the valve spool

\(K_P\)
proportional control gain

\(K_I\)
integral control gain

\(K_D\)
derivative control gain
\( l \)  
length of full cylinder stroke or pneumatic artificial muscle

\( l_{SF} \)  
scale factor in multiple-step polynomial reference input

\( m \)  
mass of air in pneumatic cylinder chamber

\( m_{t(\pm1,2)} \)  
mass flow rates through each orifice

\( M \)  
total mass

\( M_e \)  
external mass

\( n \)  
number of thread turns of pneumatic artificial muscle

\( OS \)  
offset of sinusoidal reference input trajectory

\( P \)  
pressure

\( P_{atm} \)  
atmospheric pressure

\( P_{cr} \)  
critical pressure

\( P_{t(\pm1,2)} \)  
pressures at cylinder chambers

\( P_u, P_d \)  
up-stream and down-stream pressures with respect to valve orifice

\( P_s \)  
supply pressure

\( Q \)  
heat transfer through cylinder wall

\( r_a \)  
length of the beam from external load to cylinder rod

\( r_b \)  
length of the beam from cylinder rod to shaft

\( r_s \)  
radius of the shaft

\( R \)  
universal gas constant

\( s \)  
sliding surface

\( \text{sat}(\cdot) \)  
saturation function

\( \text{sign}(\cdot) \)  
sign function

\( t \)  
time

\( T \)  
absolute temperature

\( T_a \)  
actuating torque

\( T_e \)  
external torque

\( u \)  
controller output

\( u_{cw} \)  
saw-tooth carrier wave

\( u_p \)  
amplitude of PWM signal

\( u_v \)  
control valve command

\( U \)  
energy potential

\( U_{PWM} \)  
PWM signal

\( v \)  
velocity

\( v_s \)  
Stribeck velocity

\( V_p \)  
amplitude of saw-tooth carrier wave

\( V(\cdot) \)  
Control Lyapunov function (or candidate)

\( V_1, V_2 \)  
volumes of pneumatic chamber 1 and 2

\( V_{\theta 1, \theta 2} \)  
fixed volume of pneumatic chambers 1 and 2

\( W \)  
virtual work
\( w \)  
valve orifice area gradient

\( x_p \)  
piston displacement

\( x_v \)  
valve spool displacement

\( x \)  
column vector in \( R^n \)

\( z \)  
average bristle deflection

\( \alpha_i \)  
virtual control laws

\( \beta \)  
gain margin

\( \gamma \)  
ratio of specific heat

\( \gamma_i \)  
adaptation gains

\( \Gamma \)  
controller bandwidth (DAB-SMC scheme)

\( \eta \)  
convergence rate (or eigenvalue)

\( \theta \)  
pneumatic muscle thread angle

\( \theta \)  
column vector in \( R^n \)

\( \theta_s \)  
shaft displacement

\( \lambda \)  
controller bandwidth (SMC scheme)

\( \rho \)  
air density

\( \sigma_0 \)  
equivalent spring constant of bristle

\( \sigma_1 \)  
equivalent damping coefficient of bristle

\( \sigma_2 \)  
viscous coefficient in LuGre friction model

\( \tau \)  
time constant of valve spool

\( \tau_{SF} \)  
time shift factor in multiple-step polynomial reference input

\( \phi_p \)  
inclination angle of pneumatic cylinder

\( \phi_s \)  
inclination angle of pneumatic artificial muscle

\( \Phi \)  
boundary layer thickness

\( \psi(\cdot,\cdot) \)  
normalized mass flow rate

\( \omega \)  
frequency of sinusoidal reference input trajectory

**Subscripts**

\( d \)  
desired (or reference) value

\( p \)  
piston

\( r \)  
relative motion

\( s \)  
shaft

\( v \)  
valve spool

**Accents**

\( \hat{\cdot} \)  
estimate value

\( \tilde{\cdot} \)  
estimation error

\( \dot{\cdot}, \ddot{\cdot}, \dddot{\cdot} \)  
time derivative (1\text{st}, 2\text{nd} and 3\text{rd})
**Acronyms**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>CLF</td>
<td>Control Lyapunov Function</td>
</tr>
<tr>
<td>CPU</td>
<td>Central Processing Unit</td>
</tr>
<tr>
<td>DAB-SMC</td>
<td>Dynamical Adaptive Backstepping-Sliding Mode Control</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data Acquisition</td>
</tr>
<tr>
<td>GUI</td>
<td>Graphic User Interface</td>
</tr>
<tr>
<td>HIL</td>
<td>Hardware-in-the-Loop</td>
</tr>
<tr>
<td>LAN</td>
<td>Local Area Network</td>
</tr>
<tr>
<td>MEX</td>
<td>MATLAB Executable</td>
</tr>
<tr>
<td>MRAC</td>
<td>Model Reference Adaptive Controller</td>
</tr>
<tr>
<td>PAM</td>
<td>Pneumatic Artificial Muscle</td>
</tr>
<tr>
<td>PC</td>
<td>Personal Computer</td>
</tr>
<tr>
<td>PCI</td>
<td>Peripheral Component Interconnect</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional + Integral + Derivative</td>
</tr>
<tr>
<td>PWM</td>
<td>Pulse Width Modulation</td>
</tr>
<tr>
<td>QUARC</td>
<td>Quanser Real-time Control</td>
</tr>
<tr>
<td>RAM</td>
<td>Random-Access Memory</td>
</tr>
<tr>
<td>RMSE</td>
<td>Root Mean Square Error</td>
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<tr>
<td>SMC</td>
<td>Sliding Mode Control</td>
</tr>
<tr>
<td>SSE</td>
<td>Steady-State Error</td>
</tr>
<tr>
<td>STR</td>
<td>Self-Tuning Regulator</td>
</tr>
<tr>
<td>TCP/IP</td>
<td>Transmission Control Protocol/Internet Protocol</td>
</tr>
<tr>
<td>VSC</td>
<td>Variable Structure Control</td>
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<tr>
<td>VSS</td>
<td>Variable Structure System</td>
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Chapter 1

Introduction

1.1 Motivation

The term pneumatic originates from the Greek word pneumatikós meaning “breath” or “wind”. In modern-day, pneumatics is referred as a branch of technology that utilizes compressed air as the power transferring medium to perform mechanical work for solving engineering problems [1]. The history of pneumatics can be traced as far back as 2000 years ago [1]. Ctesibius, a Greek inventor in ancient Greece who earned the title “father of pneumatics”, described the use of pneumatic cylinders to accelerate an arrow of a catapult. A temple doors which automatically opened and closed by means of hot air was developed by a Greek mathematician known as Heron of Alexandria. In the early 1600s, a German scientist named Otto von Guericke invented a vacuum pump that can be used to draw air or gas out of any vessel it was attached to. He had demonstrated how the vacuum pump could be used to separate pairs of copper enclosures called hemispheres using pressurized air. Late 19th century commemorated the introduction of pneumatics in industrial sectors on large scale when a center compressor station was installed in Paris to supply the city with compressed air. By 1950s, the use of pneumatics in the automation applications especially in production and assembly lines was commenced.

Since then, the field of pneumatics has experienced tremendous advancement. Today, pneumatics occupies many different areas of applications including packaging, food pro-
cessing, pharmaceutical, electronics industries, robotics [2] and biomedical engineering [3-5]. One of the most popular applications of pneumatics is in materials transportation over a relatively short distance. It was reported that the pneumatic systems are conceivably the most economical solution for transferring masses of up to 20kg with required power of up to 3kW over distances of up to one meter as compared to its counterpart power systems, that is, the hydraulics and electric motors [1]. The reasons for the extensive use of pneumatics are that it offers high power to weight ratio actuators, high-speed output response, relatively low-cost and less-maintenance components. Besides, the flexibility in routing the flow of compressed air, safe and environmental friendly fluid medium are other significant factors contributing to the utilization of this technology in those applications.

Despite the aforementioned advantages, servo control of pneumatic actuators is challenging. This is due to the inherent problems associated with the natural characteristics of air such as high compressibility and nonlinear flow phenomenon when passing through the pneumatic system components. In addition, friction presents in the pneumatic actuators and uncertainties in the parameters and model characterizing the dynamics of pneumatic systems further contribute to the existing problems. These drawbacks cause larger steady-state and tracking errors, stick-slip motion and limit cycles, which degrade the control performance. As a consequence, the practical implementation of pneumatics is still largely around pick-and-place positioning, which requires only two end positions of an actuator stroke [6]. However, as the deployments of pneumatic actuators in robotic manipulators and in mechatronics applications increase over the years, the demand to have accurate position tracking ability and arbitrary positioning between the two end positions of actuator stroke is becoming greater than ever. To achieve a good positioning performance in such a demanded application, it is necessary to implement a controller that is capable of dealing with the problems associated with pneumatic systems. This research, thus, focuses on the design and experimental evaluation of a controller that consists of control elements that capable of dealing with the issues of nonlinearities, parametric/model uncertainties and friction for servo-pneumatic positioning applications.
1.2 Research Objectives

Objectives of this research are:

(i) To design a controller that is capable of dealing the problems associated with pneumatic systems, that is, nonlinearities, parametric/model uncertainties and friction, for servo-pneumatic positioning applications.

(ii) To experimentally investigate the effectiveness of the proposed controller in accomplishing the positioning tasks, that is, tracking and regulating, under various operating conditions using commonly used actuator-valve setups.

In order to achieve the above objectives, four research questions are addressed:

(i) Which type of controller is more appropriate for a pneumatic system; linear or nonlinear?

(ii) How the proposed controller does perform when the pneumatic actuator is subject to a varying external load operating in two different modes of motion assisting and resisting?

(iii) Can performance of the proposed controller be maintained if the pneumatic system utilized a different combination of actuator-valve setups?

(iv) How effective is the proposed controller in comparison to the other well-established control schemes?

1.3 Scopes of Research

The research focuses on the design, implementation and experimental evaluation of a controller applicable only to the servo-pneumatic positioning applications of both tracking and regulating tasks. The controller is designed using a systematic design methodology adopted from the literature. The proposed controller is equipped with control elements that are capable of directly dealing with the issues related to pneumatic systems, that is,
nonlinearities, model/parametric uncertainties, and friction in the actuators, and thus suitable for servo-pneumatic positioning applications.

An experimental test rig is developed and used as a performance evaluation platform. The test rig consists of a different combination of actuator-valve setups commonly found in the literature. The utilized actuators are a single-rod, double-acting pneumatic cylinder and antagonistically paired pneumatic artificial muscles (PAMs). The valves include a 5/3-way proportional directional control valve, two 3/2-way and four 2/2-way Pulse Width Modulation (PWM)-controlled valves. The PWM-controlled valves are employed to investigate the possibility of cost reduction. It has been reported that, the application of PWM-controlled valves contributes to cost reduction of up to 60% as compared to using proportional valves [7, 8]. The test rig is also equipped with a mechanism that enables the actuators to be exposed to a varying external load operating in two different modes of motion resisting and assisting. A pneumatic simulator is also developed to assist the performance evaluation process. Using the simulator, simulation of various positioning tasks using the proposed controller on setups similar to the experimental test rig can be conducted. In this research, the simulator is mainly used to verify the proposed controller and determine appropriate controller gains for producing acceptable performance.

The effectiveness of the proposed controller in positioning of a mass attached to the actuators is evaluated by measuring its tracking and regulating performances. The performances are characterized using two performance indices, that is, the Root Mean Square Error (RMSE) and Steady-state Error (SSE), respectively. Seven different criteria are considered during performance investigation: (1) performance under nominal operating condition\(^1\) with the mass flow rates of compressed air regulated by a 5/3-way proportional directional control valve; (2) effectiveness of the friction compensator; (3) influence of tracking frequency; (4) impact of varying external load operating in both motion assisting and resisting modes; (5) robustness to parametric uncertainty; (6) improvement

---

\(^1\) The nominal operating condition refers to the condition where the controller is tuned to operate best when the actuators are subject to an external load operating in both motion assisting and resisting modes and the mass flow rates of compressed air regulated by a 5/3-way proportional directional control valve. Note that, the external load is originated from a mass attached to the actuators.
over other well-established control schemes, that is, Proportional+Derivative+Integral (PID) and Sliding Mode Control (SMC), and (7) performance compromise when using PWM-controlled valves.

1.4 Significance of Research

The greater demand in an accurate arbitrary servo positioning, that is, tracking and regulating between the two end positions of the pneumatic actuators justifies the need for more effective controllers. This research, thus, provides an alternative controller that can be used for such servo-pneumatic positioning applications. The outcome of this research also helps in uncovering the critical issues associated with the pneumatic systems. By understanding these issues, challenges that one has to face in an attempt to utilize the technology can thus be appreciated. The experimental test rig developed for this research is also unique. Unlike other research studies, the test rig has been designed in such a way that a varying external load originating from a mass attached to the actuators can either assist or resist the motion. This circumstance is often encountered in practical applications such as in robot manipulators. Therefore, realizing such an operating condition is desirable. The test rig is also equipped with a different combination of actuator-valve setups. The performance evaluation and comparison of the proposed controller, on a setup similar to what is found in the literature can thus be emulated. Besides, the findings from this research work are also reliable as the results are equitably compared on the same experimental setup. The pneumatic simulator, also developed in this research, contributes greatly to the pneumatic research community. Researchers from other universities can use the simulator to evaluate the performance of their controller on the simulated setups similar to what are available in the test rig. The graphical user interface (GUI) equipped with the simulator provides a user-friendly environment and thus eliminates the effort needed to understand the computer programs written in the background. This will speed up the evaluation process of controllers’ performance.
1.5 Thesis Organization

The thesis consists of eight chapters. The rest of the chapters are organized as follow:

In Chapter 2, a review of literature related to servo-pneumatic positioning systems is presented. In this chapter, architecture of the considered pneumatic systems, the operating principle and system components are described. Besides, problems associated with the pneumatic systems are also highlighted. In addition, previously developed control strategies relevant to servo-pneumatic positioning applications are overviewed.

In Chapter 3, description of the experimental test rig, developed for performance evaluation of the proposed controller is given. The experimental test rig offers a different combination of actuator-valve setups and capable of exposing the actuators with a varying external load operating in both motion assisting and resisting modes. Mathematical modeling of the considered setup is also provided in this chapter.

Chapter 4 introduces a pneumatic simulator that is developed specifically for this research. The simulator is equipped with a graphical user interface (GUI) that provides a user-friendly environment and capable of simulating all the setups available in the experimental test rig.

In Chapter 5, derivation and stability proof of the proposed controller is documented in detail. Besides, descriptions of other well-established controllers, that is, standard PID and classical SMC, involved in performance comparison with the proposed controller are briefly given.

In Chapters 6 and 7, performances of the proposed controller are evaluated. Chapter 6 focuses on the servo positioning of the pneumatic cylinder while Chapter 7 is dedicated to servo positioning of antagonistically paired pneumatic artificial muscles. The effectiveness of the proposed controller in performing tracking and regulating tasks under various operating conditions using both proportional and PWM-controlled valves is analyzed. In addition, performance of the proposed controller is also compared with the PID and SMC schemes.

Chapter 8 presents the concluding remarks. The main contributions of this research are summarized and suggestions for future study are proposed.
Chapter 2

Overview of Servo-pneumatic Positioning Systems

The goal of this chapter is to describe a typical servo-pneumatic system for positioning applications, understand the challenges in utilizing it and study the relevant control strategies proposed in the literature. This chapter is divided into four sections. Section 2.1 describes architecture of the considered servo-pneumatic positioning system. Components that making up the system including their operating principle are also explained. Section 2.2 identifies the challenges associated with the servo-pneumatic positioning systems. Relevant control strategies proposed in the literature are presented in Section 2.3. The advantages and disadvantages of these control strategies are also discussed. Summary of the chapter is provided in Section 2.4.

2.1 Architecture and Operating Principle

Figure 2.1 shows block diagram of a servo-pneumatic positioning system considered in this research. The system is made up of the controller, control valve(s), actuator and sensors. The control valves utilized in this research are a 5/3-way proportional directional, two 3/2-way, and four 2/2-way Pulse Width Modulation (PWM)-controlled valves.
Figure 2.1: Block diagram of servo-pneumatic positioning system.

Figure 2.2 shows schematic diagram of the utilized 5/3-way proportional direction valve. The valve comprises of a spring, a solenoid (armature and coil) and a valve spool. The spring holds the armature in its initial neutral position. When an electric current is applied to the coil, magnetic flux is induced and force is generated. This applied force moves the armature and since it is coupled to the valve spool, it also displaces the valve spool from its initial neutral position. As the valve spool moves, the spring acts as feedback by exerting a restoring force upon the armature. When the restoring force becomes equivalent to the force exerted by the armature, the valve spool comes to a rest in which its position is proportional to the magnitude and corresponding polarity of the applied current in the coil.

Figure 2.2: Schematic of solenoid-driven, spool-type, 5/3-way proportional directional control valve.
As the position of valve spool is proportional to the applied current, the proportional directional control valve inherits unlimited number of stable states of flow paths between the start (minimum) and the end (maximum) positions. Likewise, architecture of the PWM-controlled valves are similar to the proportional valve. However, the valves can only have one of the two stable states of flow paths which are no flow or maximum flow. This is due to the nature of applied PWM signal which is either on or off.

In this work, a single-rod, double-acting pneumatic cylinder and two pneumatic artificial muscles arranged in an antagonistic configuration are used. The pneumatic cylinder is employed to achieve linear positioning while the muscles are for rotational positioning applications. Figure 2.3 shows the picture of a typical single-rod, double-acting pneumatic cylinder. The cylinder consists of moveable elements such as a piston and piston rod. The piston divides the cylinder barrel into two chambers while the piston rod serves as a link between the actuator and external load. To move the piston, one side of the cylinder chamber is pressured with compressed air through a port and the other is depressurized. The seals are used to prevent the compressed air from leaking out of the cylinder while the cushions are used to reduce impact of collision between the piston and cylinder ends.

![Typical single-rod, double-acting pneumatic cylinder.](image)

Figure 2.3: Typical single-rod, double-acting pneumatic cylinder.

Figure 2.4 shows the picture of a typical pneumatic artificial muscle. The muscle is composed of an air-tight inner rubber tube placed in a flexible hollow braided mesh construction called membrane. It is also equipped with an appropriate metal end fitting for external attachments and compressed air pressurization. Pressurizing the muscle
2.1 Architecture and Operating Principle

inflates the inner rubber tube, causing its diameter to increase and its axial length to shorten. At the same time, a corresponding contracting force is generated. When depressurized, the muscle returns passively to its initial condition. Therefore, like a human muscle, the pneumatic muscle is a unidirectional actuator and ideal for applications related to bio-inspired robotics [2]. In addition, the pneumatic muscles also offer many good characteristics such as natural compliance, mimicking behavior of human muscles, high power per volume ratio, compactness and high speed action. Developed by J.L. McKibben in 1950’s, pneumatic muscles were used the first time in an artificial limb for polio patients [9]. This application however, required a relatively heavy CO₂ tank that made it impractical. In 1980s, pneumatic muscle regained its popularity when the Bridgestone’s engineers re-developed and introduced it to a robotic application that mimics natural skeletal-muscle of human [10]. Today, there are various versions of pneumatic muscles available in the market.

Figure 2.4: Typical pneumatic artificial muscle.

With reference to Figure 2.5, the positioning of the pneumatic cylinder can be achieved as follows. The input command subject to the control valve displaces the valve spool to the right or left and causes high pressure supply port of the compressed air to be connected to one of the pneumatic chambers while the exhaust port to the other. As a result, a pressure difference between the cylinder chambers is created and the cylinder’s piston is then moved to the left or right, accordingly.
Figure 2.5: Positioning of pneumatic cylinder: (a) extending, (b) retracting.

With reference to Figure 2.6, the positioning of the pulley connected to the antagonistic pneumatic artificial muscles can also be achieved in a similar way. By manipulating the unbalance contracting forces created from the pressurized and depressurized muscles, the positioning of the pulley in both directions, that is, counter clockwise and clockwise can be achieved.
Figure 2.6: Positioning of antagonistic pneumatic artificial muscles: (a) counter clockwise, (b) clockwise.

2.2 Challenges

Pneumatic systems offer many advantages such as good power to weight ratio actuation, high-speed output response, relatively low-cost and less-maintenance components, flexibility in routing the flow of compressed air, safe and environmental friendly fluid medium. However, servo control of pneumatic actuators is difficult because of several reasons. First, the pneumatic systems are highly nonlinear. This is due to property of the utilized fluid medium, that is, compressed air. Air is compressible and exhibits highly nonlinear phenomenon when passing through the valve’s orifice areas. Figure 2.7 shows an
example of the theoretical calculations and experimental results of mass flow rate of the compressed air when flows through a control valve at 72.5 Psi supply pressure. As can be seen from Figure 2.7, the mass flow rate of the compressed air is nonlinear. A good mathematical model that can adequately describe behavior of the compressed air is thus needed [1].

![Graph of theoretical and experimental mass flow rates](image)

**Figure 2.7:** Theoretical and experimental mass flow rates at 72.5 Psi (gauge) supply pressure: (a) supply pressure to control valve, (b) flow versus pressure ratio for spool displacement, $x_v = 0.23\text{mm}$ (reproduced from [11]).
The second problem associated with controlling the pneumatic systems is the uncertainties in parameters and/or model characterizing the dynamics of pneumatic systems. Both types of uncertainty if not properly accounted, can impose strong adverse effects on the control system. Parametric uncertainty enters into the mathematical model from various sources. It may come from the model parameters that are the inputs to the mathematical model but their exact values are unknown or impossible to be acquired in a real-life application. It may also arise because of inability to infer exact values of the parameters. In addition, parametric uncertainty may also come from the variability of input variables of the model. The effect of parametric uncertainty can be exemplified by taking variation in temperature during operation of pneumatic systems. Air is composed of approximately 78% nitrogen, 21% oxygen and 1% of some fourteen other gases by volume. Although the composition remains substantially the same between sea level and at altitude of up to 20 km, its density varies as pressure and temperature changes. When the temperature changes, value of the mean velocity of gas molecules also change. Consequently, the rate of collision that is responsible for the pressure exerted by air is modified. Hence, as temperature influences the pressure and air density, consistency in system's performance is affected. Besides, parametric uncertainty also affects the accuracy of the pneumatic model. For example, an accurate model of pressures inside the cylinder chambers is difficult to achieve as air temperature varies considerably during the charging and discharging processes of pneumatic cylinder [12]. Both processes were found to be neither isothermal nor adiabatic but somewhere in between. Similarly, as the sliding piston seals soften at a higher temperature [1], effect of friction in the pneumatic actuators is also changing and become less predictable. Measuring the temperature variations during operations by means of temperature sensors is possible, but will increase the cost of pneumatic systems and therefore is undesirable. Model uncertainty (also known as structural uncertainty) causes inadequacy, bias and discrepancy in the employed mathematical model. It originates from the lack of knowledge of the underlying true physics or from the purposeful of a simplified representation of the system's dynamics. This type of uncertainty depends upon how close a mathematical model is to the true system in a real-life application, respecting the fact that mathematical models are always approximations to real systems. An
example, would be a model describing the process of a free falling object neglecting dynamics of air resistance. The model itself is inaccurate and contains unmodeled dynamics, since it does not account for air resistance which always exists. In this case, even if all other parameters are known in the model, a discrepancy still lies between the model and true physics. Implementing highly accurate nonlinear models may be possible; but it is undesirable since the complexity as well as number of the immeasurable parameters will be tremendously increased [13]. To deal with this issue, a controller that is robust and able to maintain performance of the system in the presence of parametric and model uncertainties is necessary.

The third issue related to pneumatic systems is friction. Friction is an impediment to servo control and if it is not properly dealt with, will substantially degrade performance of the pneumatic systems. Hysteresis, instability, slow response, large tracking and steady-state errors, and limit cycles are among the adverse effects created by friction. In pneumatic cylinder, friction occurs as a result of relative motion between the contacting surfaces of cylinder wall and piston-rod's seals. Similarly, in pneumatic artificial muscles, a relative motion between the air tight inner rubber tube and flexible hollow braided mesh (membrane) construction causes friction (also called thread-to-thread friction [14]). In order to effectively deal with the adverse effect of friction, it is important to first understand its phenomenon.

Friction is created as a result of relative motion between the surface contacts. The phenomenon can be explained using different friction models as proposed in the literature [15-17]. In this thesis, the Strubeck friction model shown in Figure 2.8 is used, as it can sufficiently describe the friction phenomenon. With reference to Figure 2.8, friction is said to be in a static friction regime before the motion commences. In this regime, friction is referred as static friction and is determined by a very small displacement between the contacting surfaces. The relative motion between the contacting surfaces in this regime is considered to be zero. As an actuating force becomes greater than the static friction, a relative motion commences and friction enters into the dynamic regime. In this regime, friction is referred as dynamic friction (also known as Coulomb friction) and is a function of relative velocity of the contracting surfaces. Note that, the Strubeck friction model as-
sumes that engineering surfaces are lubricated, which is true for most engineering applications. Hence, unlike friction models of dry and unlubricated contacting surfaces, which has discontinuous [16] transition, the transition from static into dynamic friction regime in Strubeck model takes place in a continuous form with its magnitude continuously decreasing while the relative velocity of contacting surfaces continuously increasing [15]. This phenomenon is known as Strubeck effect and the resulting continuous transition curve in referred as Strubeck curve. As a result of changes in friction magnitude, a stick-slip motion is created. In pneumatic systems, stick-slip motion commonly occurs at the beginning of actuators’ motion and when the actuators move at very low speed. This reduces the ability of controllers to accomplish an accurate position control of pneumatic actuators as well as preventing a consistent and repeatable performance. For further physical understanding of friction behavior, refer to Appendix A1.

Conventionally, the effect of friction can be reduced by increasing the stiffness of the control mechanisms [15, 16]. However, due to the compliance nature of the pneumatically operated system, an alternative approach based on friction compensation techniques is always preferred [15, 18-20]. These techniques which are based on the knowledge of suitable friction model, predict the real friction presents in the actuators and then supplies the controller with the corresponding control effort for an appropriate control action.

![Figure 2.8: Strubeck friction model.](image-url)
2.3 Control Strategies

The control of pneumatic systems can be achieved using two major approaches. The first approach is to use a linear control method augmented by additional schemes such as adaptive [21-24], fuzzy logic [25-27], gain scheduling [28-30] and neural network [31-33]. These schemes help linear controllers deal with nonlinearities by modulating the controller parameters. The effectiveness of these schemes, however, is restricted by the slow updating of controller parameters in comparison to the relatively fast nonlinear dynamics of the systems [34]. As a result, only limited compensation can be achieved and performance of the systems may show significant degradation or instability as the systems rely on a linearized model which is inaccurate.

The second approach is to implement a nonlinear controller. There are three main advantages of using the nonlinear over the linear controller: (1) consistent performance upon larger operation range can be achieved due to the employment of a more accurate nonlinear model, (2) applicable to non-linearizable systems with discontinuous behavior and (3) provide high tolerance to uncertainties as these problems are intentionally accounted and incorporated in the controller [35]. Derivation of the nonlinear controllers can be accomplished through a traditional Lyapunov design method. In the Lyapunov design method, a scalar positive energy-like function, $V(x)$, of the system states, $x$, called control Lyapunov function (CLF) is first formulated. An appropriate control law that satisfies the Lyapunov stability criteria to make the CLF decreases along the system trajectories is then chosen. Figure 2.9 illustrates the CLF, $V(x)$, of two state variables $x_1$ and $x_2$ plotted in a 3-dimensional space. The CLF, $V(x)$, typically corresponds to a surface looking like an upward cup with the lowest point of the cup located at the origin. As the states, $x$, moves towards the equilibrium point at the origin, the CLF contour curves also move to the corresponding lower values of $V(x)$. This indicates that the energy is dissipated from the system and stability condition will be eventually achieved.
2.3 Control Strategies

Figure 2.9: Illustration of Lyapunov stability criteria satisfied Control Lyapunov Function (CLF) (reproduced from [35]).

The drawback of the traditional Lyapunov design method is that, it is difficult to find the suitable control law that satisfies Lyapunov stability criteria of the system as this method solely relies on one's intuition and experiences. Therefore, this method is only suitable for systems with the order of one or two (i.e., the control input can be separated from the output of interest by one or two integrations). For systems with higher order, the design obstacle can be avoided via order reduction [36]. The traditional Lyapunov design method can then be applied on the cascading lower-order subsystems. Such a control approach is called cascade control and had been introduced for servo control of pneumatic actuator by Guenther et al. [18]. Figure 2.10 shows a pneumatic system described as two interconnected subsystems, that is, pneumatic and mechanical. The mechanical subsystem is assumed to be driven by a force generated from the pneumatic subsystem.

Figure 2.10: Pneumatic system in cascading representation (reproduced from [18]).
2.3 Control Strategies

Alternative to the cascade control method is to use the well-established nonlinear controller design procedure called backstepping. Introduced by Kristic et al. [37], the backstepping procedure enables application of the traditional Lyapunov design method to be extended to a nonlinear system with higher-order. The procedure provides a recursive design methodology for the construction of both controllers and associated CLFs in a systematic manner [38]. The procedure begins by first defining a scalar equation of the output of interest which is separated from the control input by a number of integration. A suitable virtual control law is then selected to stabilize an appropriate CLF established from the corresponding output of interest. The procedure is repeated for the subsequent system states until the final control law is determined. Note that, the Lyapunov-based controllers synthesized using this procedure are globally stable and capable of asymptotically tracking of a reference signal [39]. This procedure has been used in [40-42] to construct a nonlinear controller for a servo control of the pneumatic actuators.

In general, the performance of nonlinear controllers is superior to the linear ones [43]. However, the effectiveness of the controllers heavily relies upon the accuracy of the employed nonlinear model. Implementing highly accurate nonlinear models even though may be possible, it is undesirable since the complexity as well as number of the unmeasurable parameters will be tremendously increased [13]. Consequently, simpler nonlinear models of pneumatic systems such as those applied in [44-46] are often preferred. To account for the imprecise model and parametric uncertainty, robust control schemes such as sliding mode controls [47-49] and nonlinear adaptive controls [50-52] can be applied or incorporated into the control schemes.

The concept of sliding mode control (SMC) originates from the variable structure control (VSC) theory of variable structure systems (VSSs). It was first introduced by Utkin in 1977 [53]. The basic idea behind SMC scheme is that the controller is allowed to deliberately change its structure, that is, to switch at any instant from one to another set of possible functions of the states in order to achieve robust performance with respect to bounded modelling imprecision. Using this approach, high performance control systems that are reliable can be implemented at low cost [54] and stability of the control systems can be preserved. Figure 2.11 shows block diagram of a control system utilizing the SMC
scheme. The SMC scheme is composed of a sliding surface, \( s \), and two control components: equivalent and robust. The sliding surface represents the dynamics of the control systems in which the actuator is supposed to track the desired position trajectory. The equivalent control component is similar to a feedback linearizing or inverse control law. The robust control component is responsible for dealing with parametric and model uncertainties. To ensure the system’s trajectory is always on the sliding surface, the applied robust control component has to satisfy the sliding condition.

![Diagram](image)

**Figure 2.11: Sliding mode control system.**

The sliding condition can be interpreted as a condition where the squared “distance” to the surface, as measured by \( s^2 \), decreases along all system trajectories as illustrated in Figure 2.12. The robust component forces the trajectories outside the sliding surface towards the surface and once on it, the system trajectories will remain there. In short, satisfying the sliding condition makes the surface an invariant set and thus guarantees convergence of the system trajectory to the sliding surface. When the system trajectories are on the sliding surface, \( s \), the systems are said to be on sliding regime or sliding mode. The incorporation of robust control component into SMC controller, however, causes a switching phenomenon known as chattering. Figure 2.13 illustrates the chattering phenomenon which is due to the introduction of a discontinuous term across the sliding surface in the SMC control law. In practice, chattering is undesirable as it involves high control activity and may excite the neglected high frequency dynamics of the system model [35]. Chattering can be reduced if not completely eliminated at the expense of less tracking precision, by adjusting the boundary layer’s thickness of the switching surface.
Therefore, when using the SMC controller, an optimal trade-off between the control bandwidth and tracking precision has to be determined. The implementation of SMC controller for servo-pneumatic positioning applications has been documented in [55-57].

![Figure 2.12: Illustration of sliding condition (reproduced from [35]).](image)

![Figure 2.13: Illustration of chattering phenomenon caused by control switching (reproduced from [35]).](image)

The adaptive control on the other hand, achieves robustness by estimating the uncertain plant parameters or the corresponding controller parameters from the measured system signals using an on-line parameter estimator. The estimated parameters are then employed in the control input computation. There are two main approaches for constructing an adaptive controller based on how the on-line parameter estimator (also known as adap-
tive law) changes the control gains in response to the changes in the plant and disturbance dynamics. These approaches are: (1) model-reference adaptive control (MRAC) and (2) self-tuning regulator (STR). The MRAC method utilizes a direct control approach where the plant model is parameterized in terms of the controller gains. Using this method, the parameterized model is estimated directly without intermediate calculations of the plant parameter estimates. Adjustment of parameters continues until the tracking errors converge to zero. The MRAC can be schematically represented by Figure 2.14. The control system consists of four elements: a plant with unknown parameters, a reference model for specifying the desired output of the control system, a control law with adjustable parameters, and an adaptation mechanism for updating the adjustable parameters. The STR on the other hand, is realized using indirect control approach. As illustrated in Figure 2.15, the plant parameters are estimated on-line and used to calculate the controller gains. Unlike the MRAC, parameter estimation in the case of STR can be simply understood as the process of finding a set of parameters that fits the available input-output data from a plant [35]. Example of research works that employed adaptive control approach for servo pneumatic system can be found in [58-60].

![Figure 2.14: Model-reference adaptive control system.](image-url)
2.3 Control Strategies

![Control System Diagram](image)

Figure 2.15: Self-tuning adaptive control system.

The problem of friction in pneumatic actuators can be dealt by incorporating an appropriate compensation scheme into the control schemes. In many friction compensation techniques, a dynamic friction model is often utilized in comparison to a static friction model. This is because, the static friction models although provides good representations for static friction, Coulomb friction, and viscous friction, are lacking of the dynamic friction behaviors such as Stribeck effect, static friction variation, hysteresis, and friction lag. Therefore, to achieve relatively accurate friction compensation and better control performances, the dynamic friction models are always preferable. One of the most commonly applied friction compensation scheme for servo control of pneumatic actuators is the LuGre-based friction compensator [18, 19, 61]. This friction compensation scheme is popular for control applications due to the utilization of a type of dynamic friction model called LuGre friction that is capable of: (1) capturing many friction behaviors such as Stribeck effect, varying breakaway force and frictional lag, (2) adequately representing the real friction behavior that occurs between the lubricated contacting surfaces of pneumatic actuators’ components, (i.e., the transition from static to kinetic friction is modelled as a continuous process), and (3) providing a dynamic behavior of the friction internal state at low-speed motion which is essential for construction of the adaptive laws to compensate the adverse effect of the stick-slip motion. The LuGre friction model is an extension of the Dahl model and was developed for the purpose of simulating control systems with friction. In LuGre friction model, friction on contacting surfaces of two rigid bodies
is visualized as elastic bristles [15, 17]. The bristles are assumed to deflect like springs when a tangential force is applied as depicted in Figure 2.16.

![Diagram of bristles on contacting surface]

Figure 2.16: Bristles on contacting surface.

As the deflection is becoming sufficiently large, the bristles begin to slip. The average deflection force of these elastic bristles which is considered as an internal state, \( z \), is used to represent friction. The dynamics of the internal state, \( z \), is given as follows:

\[
\dot{z} = v_r - \sigma_0 \frac{|v_r|}{g(v_r)} z
\]  

(2.1)

where \( v_r \) and \( \sigma_0 \) are the relative velocity of sliding surface and stiffness of the bristles, respectively. Function \( g(v_r) \) which models the Striebeck effect can be expressed as follows:

\[
g(v_r) = F_c + (F_s - F_c)e\left(-\frac{v_r}{v_s}\right)
\]  

(2.2)

In equation (2.2), \( F_s \) and \( F_c \) denote the static and Coulomb friction, respectively. \( v_s \) is the Striebeck velocity, beyond which the average bristles deflection become sufficiently large and breakaway occurs. As the breakaway takes place, friction suddenly drops and stick-slip motion occurs. The stick-slip motion eventually disappears at the higher relative velocity, \( v_r \), as the Coulomb friction, \( F_c \), starts to dominate the Striebeck effect. The total friction force for the LuGre friction model including viscous friction, \( \sigma_2 \), is given as follows:
References


References


