Component-based Mixed Reality Environment for the Control and Design of Servo-pneumatic System

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Declaration

No part of the material described in this thesis has been submitted for the award of any other degree or qualification in this or any other university or college of advanced education.

Ashraf Saleem

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Synopsis

Considerable research efforts have been spent over the last two decades on improving the design, control, and modelling of pneumatic servo drive systems including the development of dedicated controllers and control valves. However, the commercial updates in employing pneumatic servos are still largely limited to laboratory research usage and the initiatives in developing seem to have lost their momentums. Although this situation has some to do with the rapid development and availability of cost effective electric servo technologies, one reason is considered to be a lack of design and simulation tools for employing pneumatic servo drives. This research has therefore been conducted to address these concerns, and to demonstrate how appropriate tools and environments can be developed and used to aid in the design, control and commissioning of pneumatic servo drives. Because of the inherent high nonlinearities associated with pneumatic systems, it would be highly desirable if the simulation environment could be run in time domain so that it can be ***mixed*** with the real system. This would make the simulation more accurate and reliable especially when dealing with such nonlinear systems. Unfortunately, the tools that are available in the market such as Propneu (Festo, 2005) and Hypneu (Bardyne, 2006) are dedicated for pneumatic circuit design only.

This research is aimed at developing a mixed reality environment for the control and design of servo-pneumatic systems. Working with a mixed reality environment would include both the capability to model the system entirely as a simulation, the so-called "off-line", as well as being able to use real components running against simulations of others "on-line", or in a Mixed Reality (MR) manner. Component-based paradigm has been adopted, and hence the entire pneumatic system is viewed as a series of component modules with standardised linking variables. The mathematical model of each individual component has been implemented in simulation software which produces time domain responses in order to allow for mixing the simulation with the real system.

The main outcome of this research can be seen as a successful development and demonstration of the Component-based Mixed Reality Environment (CMRE), which would facilitate the control and design of servo-pneumatic systems. On the one hand, the CMRE facilitates the identification of some nonlinear parameters such as frictional

parameters. These parameters could cause great difficulties in servo-pneumatic modelling and control. Accurate friction parameters would give the ability to attain an accurate model, and therefore provide more flexibility in applying different control and tuning strategies on the real system. On the other hand, the CMRE facilitates the design process by enabling the designer to evaluate the servo-pneumatic system off-line prior to building the system. This would reduce the design time, increase the reliability of the design, and minimize the design cost.

The concept of the CMRE was validated by tests carried out on laboratory-based prototype servo-drive. Close agreement between the experimental and simulated responses was obtained showing that the models have represented the real system adequately. Case studies were then conducted to demonstrate the validity of the proposed methodology and environment. In these case studies, PIDVF controller and cascade control structure were successfully implemented, synthesised, and tuned. The results revealed that the CMRE is an easy, accurate and robust way of implementing different control and tuning strategies on servo-pneumatic systems. Furthermore, the research has shown how the CMRE can lead to significant improvements in certain life cycle phases of the system, e.g. commissioning, maintenance, etc.

This research has contributed to knowledge in the following:

(1) Adopting the mixed reality concept and the component-based approach in order to create a CMRE in facilitating the control and design of servo-pneumatic systems.
(2) A method to identify the friction parameters of a single-axis pneumatic machine,
(3) Encapsulate existing control methods within the CMRE to be applied on the real system.
(4) A scheme for controller tuning, in which the controller is tuned off-line and then applied on the real system, and hence avoided on-line tuning which can be troublesome and time consuming.

It is anticipated that the concept of the CMRE can be extended to include multi-axes servo-pneumatic system, servo-hydraulic, and servo-electric drives. Therefore, conceptual model structures have been introduced in this research which can be considered as the foundation for creating similar environments for those systems.

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Nomenclature

a, b	Subscripts for inlet and outlet chambers, respectively
A	Ram area
P_e	Exhaust pressure
C_d	Discharge coefficient
P_d	Down-stream pressure
l	Half the stroke length
V	Volume in cylinder chamber
c_f	Viscous damping coefficient
F_c	Coulomb friction
F_{static}	Static friction or stiction
$\dot{m}$	Mass flow rate
$\dot{m}_L$	Leakage mass flow rate
M	Payload
P_{atm}	Atmosphere pressure
P_r	Downstream to upstream pressure ratio
P_s	Supply pressure
P_u	Up stream pressure
P_d	Down stream pressure
r	Specific heat constant
R	Gas constant
T	Air temperature, K
T_s	Supply temperature
T_u	Up stream pressure
Y, $\dot{Y}$, $\ddot{Y}$	Load position, velocity, and acceleration
P_{drop}	Pressure drop
λ_L	Material friction factor
ρ	Air density
L_p	Pipe's length
v	Average air velocity
D	Pipe's inner diameter
K_p	Proportional gain
K_i	Integral gain
K_d	Derivative gain
K_v	Velocity feedback gain
K_f	Velocity feedforward gain
K_L	Leakage coefficient of mass flow
C_v	Specific heat capacity of air at constant volume
C_p	Specific heat capacity of air at constant pressure
ρ	Density of air
V_{da}	Residual volume in cylinder chamber A
V_{db}	Residual volume in cylinder chamber B
$\dot{V}$	Rate of change of volume
w	Port width
X	Spool displacement of valve
av, bv	Subscripts for valve port to cylinder Chamber A and B, respectively.

Chapter 1
Introduction

1.1 Background

Substantial research efforts have been spent over the last two decades on improving the modelling, control, and design of pneumatic systems, which resulted in a potential growth for their market. This increasing acceptance for pneumatic technology is due to their economy and simplicity compared to other technologies of equivalent performance. Moreover, pneumatic systems are robust, simple, and cost-effective. There is a continuous and increasing flow of new pneumatic products with more features, lower prices, and far less maintenance than their competing products (Stands and Steve, 2004).

However, servo-pneumatic application fields are still more limited than their electric and hydraulic counterparts. One of the restricting factors preventing wider applications of pneumatic systems arises from the lack of tools and simulation environments that facilitate the control, design, and the utilization of these systems in more industry sectors (Houck, 2004).

This implies that there is a need to develop the necessary tools and the suitable working environment in order to facilitate the control and design of servo-pneumatic systems. This would enable developments in pneumatic technology to go forward in an efficient manner taking advantage from the advancements of the PC-technology, field-bus systems, microprocessors, and data acquisition technologies.

1.2 Motivations

Pneumatic systems exhibit highly nonlinear dynamic behaviours, and hence it is a challenge to appreciate the effects of these nonlinear phenomena on the overall system. These nonlinear properties are due to air compressibility, pressure drop, and static and dynamic frictions, which introduce major modelling and control problems relating to time delay, low efficiency and low natural damping (Li, et al., 2004, Lin and Chen, 2003,

Yang, et al., 2004). Therefore, the available simulation tools in the market, such as Propneu (Festo, 2005) and Hypneu (Bardyne, 2006), are dedicated only for pneumatic circuit design. This impedes the design, implementation of different control methodologies, and the controller optimisation for pneumatic systems and made it more complex than their electric and hydraulic counterparts.

To cope with these complexities, it would be more advantageous if the simulation environment could be run in time domain so that it can be ***mixed*** with the real system. This would make the simulation more accurate and reliable especially when dealing with nonlinear systems.

Therefore, this research strives to develop a mixed reality environment for the control and design of servo-pneumatic systems. In order to provide more flexibility and reliability, component-based paradigm has been chosen as the general framework for the design and implementation of the proposed environment.

This environment should help the designers and the end users to cope with the complex phenomena associated with servo-pneumatic systems without involving with the sophisticated mathematical analysis. This can be attained by facilitating the identification of some nonlinear parameters and the evaluation of different control methodologies and apply them on the real system.

The proposed environment should also provide the capability of predicting the dynamic behaviour of servo-pneumatic systems and increase their usability which will enhance their market potential considerably. Moreover, this environment would improve the system's life cycle by facilitating the design and control of the system.

In few words, further investigation is required on (1) adopting the component-based approach and mixed reality concept to create a working environment for the design and control of servo-pneumatic systems, (2) adopting this environment with real systems, (3)

utilizing this environment to improve the system model accuracy, and (4) demonstrating how this environment would improve the life cycle of the system.

1.3 Aim and Objectives

1.3.1 Aim

The aim of this research is **to develop and implement a Component-based Mixed Reality Environment (CMRE) that facilitates the control and design of servo-pneumatic systems, and to explore its role in improving their life cycle**.

The focal research questions that arise are:

(1) How effective is the mixed reality concept and the component-based approach in the development of the CMRE, and what are the advantages of adopting such approaches?

(2) How can CMRE improve the design and control of servo-pneumatic system, and what are the effects on the system's life cycle?

(3) How can the user cope with the nonlinearity aspects associated with pneumatic systems using the CMRE?

(4) How can the CMRE be utilised to facilitate the implementation of different control schemes, commissioning, and machine service support for servo-pneumatics?

1.3.2 Objectives

The objectives of this research are:

(1) Understand the dynamic behaviour and identify the nonlinear characteristics of servo-pneumatic system.

(2) Formalise a framework to decompose the system into components and determine the platform which will be employed to develop the CMRE.

(3) Establish a mathematical model for the system's components.

(4) Develop and implement the CMRE on the real system.

(5) Demonstrate and validate the developed CMRE through a series of experiments, and evaluate the main concepts through conducting case studies.

(6) Demonstrate its potential in the control and design of servo-pneumatic systems.

(7) Explore how the CMRE would improve the life cycle phases of the system.

(8) Determine conceptual model structures for multi axis pneumatic system, servo-hydraulic systems, and servo-electric systems in order to widen the scope of the proposed environment to include these systems.

1.4 Scope of the Research

The research scope has been delimited to the following:

(1) The research has centred on developing the CMRE to cope with the difficulties associated with the control and design of servo-pneumatic systems.

(2) Developing the CMRE, which involves modelling and programming approaches, requires in-depth knowledge and understanding of the servo-pneumatic systems dynamic behaviour.

(3) Demonstration of how the system's life cycle can be improved using the CMRE. The life cycle study will focus on six phases; (a) system requirements analysis phase, (b) system design phase, (c) software components integration and testing phase, (d) control system evaluation and implementation phase, (e) mechanical components production and integration phase, and (f) machine service support phase.

(4) A validation study will be conducted to validate the proposed environment. This will be established on a laboratory-based setup. The set-up is a single axis drive system based on pneumatic actuation. Analysis and evaluation of the experimental results will be provided.

(5) Conceptual model structures for multi-axis pneumatic system, servo-hydraulic systems, and servo-electric systems will be established for potential future work. These models will provide a foundation for extending the proposed environment to include those systems.

1.5 Research Approach

The adopted research approach is the design and creation strategy which involves analysing, designing and developing a computer-based product. This sounds more like a project strategy rather than a research strategy. However, this research project will demonstrate not just technical skills but also academic quality such as analysis, explanation, argument, justification and critical evaluation. They must also contribute to knowledge in some way. This depends on role that the product/system plays (Briony, 2006). Typically design and creation is a problem-solving approach consists of the following iterative main steps/activities: (1) identification of research questions (via intensive literature reviews), (2) concept development, (3) concept implementation, (4) validation and evaluation, and (5) documentation/reporting findings and identification of directions for future research and development.

The research study was initiated with a two-stage literature review consisting of: technical and theoretical reviews. The technical review was conducted for the recognition of the recent technologies in the development of servo-pneumatic system components (mainly commercial components), which are important for the author to be familiar with since the research has adopted component-based approach. It focused also on the applications spectrum of the pneumatic systems in different industry sectors. This should help the author to identify the potentials of these systems as well as the requirements to achieve wider applications range.

The second stage of the literature review – theoretical review, was conducted for the identification of specific research problems. It also served the purpose of identifying potential solutions for the identified problems. In this part of the review, the focus was on the control methodologies, modelling techniques, and simulation tools for servo-pneumatic systems. Bear in mind that, this step was conducted continuously and in parallel with the other steps (from step 2 to 5), throughout the period of this research. This was necessary to ensure that potential developments in related areas could be continuously fed back to the other activities throughout the research period.

The second step – concept development, a conceptual framework were formulated for breaking the system down into components. These components were employed to develop a model structure for servo-pneumatic system with standard interfaces. Mathematical model for each component was established to create virtual system components.

In the third and fourth steps – implementation, testing and evaluation, the formulated model structures were implemented, which involved the development, experimentation and evaluation of prototypes for demonstrating and validating the developed environment. Due to various resource constraints, it has only been practical (within the three-year time scale of this research study) for the testing and evaluation process to be based only on a laboratory test bed. In this stage, the potential of the proposed environment was demonstrated by conducting different real scenarios. These scenarios included (1) identification of the system's frictional parameters, (2) implementation of the different control schemes, and (3) controller tuning. In the last step, findings were documented and reported. In addition, a number of potential extensions for this study have also been identified for future research and development.

The design and creation research strategy, which is adopted in this research project, has several advantages and potential weaknesses. Following points summarise these advantages and weaknesses.

Advantages:

- There is something tangible to show the efforts rather than just abstract theories or other knowledge.
- It appeals to people who enjoy technical and creative development work.
- Because the technology is advancing rapidly, there is plenty of scope for proposing and developing new computer-based systems/products and hence making a contribution to knowledge.

Potential weaknesses:

- It is a challenge to justify why the work is not just a normal design and creation.
- It is risky if the researcher does not have the necessary technical skills. Enthusiasm is not enough.
- The success of the system/product may depend on the researcher being present. Once they disappear the system may not work so effectively.
- Rapid advances in technology can invalidate the research results before they have been tried out in a real life context.

1.6 Thesis Outline

This thesis is organised into 10 chapters. The following paragraphs present the outlines of the remaining chapters of this thesis.

Chapter 2 presents an overview of state-of-the-art in servo-pneumatic systems' components development technology. It also investigates the applications spectrum of pneumatic systems, and hence it demonstrates what distinguish pneumatic systems from their electric and hydraulic counterparts as well as some requirements to attain wider range of applications.

Chapter 3 presents an overview on the control methodologies, modelling techniques, and simulation tools that have been employed for servo-pneumatic systems. Critiques in relation to tuning, control, modelling, and simulation approaches/techniques, and improvement opportunities, derived from the review are presented in the discussion sections within the chapter.

Chapter 4 presents a detailed description of component-based framework for the system decomposition. Based on this framework, servo-pneumatic system has been decomposed into the key components and the system model structure has been formulated. This

followed by details on the test bed that will be employed for the demonstration of the main concepts of the proposed environment.

Chapter 5 establishes a mathematical model of the servo-pneumatic system components described in chapter 4. This mathematical model is established based on previously published work.

Chapter 6 presents the design and development of the proposed CMRE based on MATLAB/Simulink. It also introduces a method for frictional parameters identification using the CMRE.

Chapter 7 the CMRE was validated in chapter 7. The validation was conducted on two different systems; simple open loop pneumatic system and servo-pneumatic system. A case study has been also carried out to facilitate the validation of the proposed methodology and the developed environment.

Chapter 8 demonstrates how the CMRE would improve the system life cycle.

Chapter 9 outlines some recommendations for future work. Conceptual model structures have been introduced in order to include multi-axes pneumatic system, servo-hydraulic systems, and servo-electric systems.

Chapter 10 highlights the thesis findings, contributions, and conclusions.

Chapter 2

State-of-the-art in Servo-pneumatic Drive Systems

2.1 Introduction

This chapter presents an overview of the state-of-the-art in servo-pneumatic technology. The review focuses on the recent developments of servo-pneumatic components, including actuators, valves, and sensors and monitoring devices. Servo controllers are also explored. A brief description of the applications spectrum of servo-pneumatic systems in different industry sectors is also presented. The objective of this chapter is to recognise the potentials that distinguish the pneumatic systems from their hydraulic and electric counterparts, and to extract a set of requirements that should be considered in order to widen the applications rang of pneumatic systems in the future.

2.2 Recent Technologies in Servo-pneumatic System Components

2.2.1 Background

In the early stage of introducing servo-pneumatic systems to the industry, some hydraulic components were directly used in their design due to the similarities between pneumatic components and their hydraulic counterparts. However, hydraulic components are more expensive because they have been developed for high pressure and high forces. The essential difference arises from the nature of the working media of each drive type. Moreover, some measures, which have been developed for hydraulic component design, do not work for pneumatic (Zhang, 1999).

Some of pneumatic components suppliers such as Festo, Hoerbiger-origa, Bosch-rexroth, Martonair, SMC, ...etc, in conjunction with some research efforts, have provided an advanced technology in pneumatic components development which increased their acceptance in the market, and thus employed in a variety of applications. Since this research is adopting component-based approach to develop a mixed reality environment, it is important to recognise these advancements and technologies, and to appreciate the performance and the operation characteristics of the new generation of

pneumatic components. This will also serve the recognition of the potentials those distinct pneumatic systems from their equivalents. The following sections will review briefly the recent developments in pneumatic components technology which include actuators, valves, and sensors and monitoring devices.

2.2.2 Recent Actuators Technology

Pneumatic actuators are classified in terms of their motion characteristics. In the case of limited motion within two physical extremes, such as cylinders, air muscles and vane-type, they are known as ***reciprocating actuators***. Actuators with continuous motion, where exchanges of working media occur between the driving and exhaust chambers, are regarded as ***continuous actuators*** (e.g. air motors). Within the context of this research, reciprocated actuators will only be considered.

Reciprocated actuators or cylinders can be divided into two categories: (1) rodded cylinders, and (2) rodless cylinders. Rodded cylinders (conventional cylinders) are classified as either single-acting or double-acting cylinders. Single-acting cylinders produce driving force in one direction only because it has one compressed air inlet. The return movement of the piston is effected by a built-in spring or by the application of an external force. Double-acting cylinders are more often used in servo-pneumatic systems. These cylinders provide driving forces on both sides of the stroke and therefore bi-directional motion can be achieved. Rodless cylinders have two distinct advantages: saving space and effective ram area utilisation. However, the basic design of rodless cylinders normally requires external guides if they are subject to side loading. Friction can be high in rodless cylinders due to the more seal arrangements that are often required because of the cylinder construction. The operational life is relatively less than for rodded cylinders.

Recently, new types of pneumatic cylinders have been introduced to the market to meet special requirements. For example in the case where normal cylinders cannot deliver the required power, or the available installation space is limited, ***high-power cylinders*** are used which provide up to four times the power for the same cross-section (Festo, 2005).

For applications that demand high force and high side load capacity, ***guided cylinders*** can be employed (Festo, 2005; Origa, 2005; Bosch, 2005). One of the most recent cylinder types is ***multi-position cylinder***, which consists of several cylinders of different lengths connected end to end and allows up to 5 positions to be approached (Festo, 2005).

Valves may also be manufactured as integral part of the actuator, which can make the system more compact and reduce the effect of pressure loss and time lags in air pipes. Origa offers 3/2 way valves integrated into the cylinder's end caps which can be used as a compact and complete solution. They also introduced the ***twin rode cylinder*** (see Figure 2.1), which provides high load, non-rotational guidance, and more compactness.

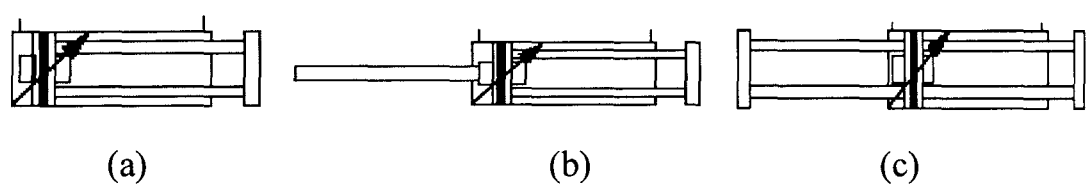

Figure 2.1 (a) Double acting, double rod, (b) Double acting, 3 rods, (c) Double acting, 4 rods.

2.2.3 Recent Valves Technology

Generally, there are two types of control valves: pressure control valves and flow control valves. This review will be devoted to flow control valves since this type of control valves are primarily used to achieve servo motion control of pneumatic actuators. There are several methods for operating the valves. The valve can be solenoid-driven, torque-motor driven, or piezo-actuation. From the control perspective, the valve can be either servo controlled valves, proportional valves, modulated on/off valves (PWM mode), or servo/proportional valves. More detailed study is presented in (Zhang, 1999).

The performance of a pneumatic valve can be determined by measuring a number of parameters which describe its characteristics most adequately.

(1) Flow capacity

The ability of the valve to deliver an amount of air flow at a given time is referred to as flow capacity, which can be represented by either mass flow rate or volumetric

flow rate. The flow rate is a main measure used to quantify the flow capacity of a valve.

Recent valves, which are available commercially, have maximum flow capacities of 2000 litre/min, and 1500 litre/min which manufactured by Festo and Origa respectively (Festo, 2005; Origa, 2005).

(2) Frequency response/bandwidth

One of the important features that distinct pneumatic actuator from the hydraulic one is the rapid and frequent movement of fluid into and out of a pneumatic actuator (Sandoval and Latino, 1997). Basically, the valve must be designed for rapid response to ensure adequate system bandwidth and to follow highly dynamic actuating signal. The bandwidth of a valve must be high enough to get a fast response, which is normally determined by measuring the valve response to a sine wave input. The corresponding -3dB frequency is defined as the valve bandwidth.

Some specifically designed pneumatic servo valves have been made with a great improvement to the frequency. For example, one of the latest proportional valves produced by Festo (MPYE series) can achieve a critical frequency of 125 Hz, which corresponds to the 3dB frequency at the maximum movement stroke of the piston spool (Festo 2005). Even more, Origa introduced a "Piezo chip" technology which offers a new method of piloting a standard proportional valve. The piezo chip is extremely fast which enable a switching speed up to 2 ms (equivalent to 500 Hz) (Origa, 2005). This frequency is enough to obtain sufficiently accurate positioning results.

(3) Crossover characteristics

Crossover characteristics are the measure of the dimensions, clearances and tolerances of the finely machined edges of the spool and the mating sleeve ports. The less precise they are, the greater the dead zone, the slower the response, the less the accuracy and the linearity. The valve configuration and actuation, the spool lap

condition, and the precision and repeatability of manufacturing process influence the linearity of the valve flow.

(4) Hysteresis, repeatability and threshold

Which are the measures of how the actual flow of the valve follows the theoretical flow when mapped against input curves, and how small is the signal that will cause a measurable change in the valve output. Electric offsets can be programmed to make compensation to the input versus output response, so the valve spool shifts quickly across the dead zone region near the null point. Commercially, Festo offers a proportional valve with maximum hysteresis of 0.4% of the maximum stroke of the piston spool.

2.2.4 Recent Sensors and Monitoring Devices Technology

Starting with positioning, there are different technologies for position sensing ranging from encoders (optical or mechanical) and LVDTs (Linear Variable Differential Transformer) to proximity sensors. The typical sensors for measuring position in servo-pneumatic systems are absolute encoders. Encoders are divided into two types: relative position encoders (incremental encoders) and absolute position encoders. Using absolute encoders eliminate the need for "homing" the axis, since exact position information is available immediately upon power-up.

LVDTs have the advantages of high resolution, long mechanical life, law mass, small in size, and robust, but they only have limited working range. Therefore, they are rarely to be used in servo-pneumatic systems.

For home position and position limits, proximity sensors are typical. Proximity sensors are either optical or inductive based. Inductive proximity sensors are more popular for servo-pneumatic systems due to their long service life, high reliability, no switching error, high switching frequency (up to 3000 Hz), ease commissioning, and vibration resistant (Festo, 2005). The proximity sensor generates an electrical signal when a

metallic object approaches the active surface and is situated within the specified switching distance.

Recently, Festo released the latest position sensor which based on proximity measuring principle. It can provide a continuous monitoring for the cylinder piston position within a range of 50 mm with reproducibility of 0.1 mm. This sensor produces an output analogue signal proportional to the piston position with sampling rate of 1 sample every 3 ms, and it can be connected directly to analogue inputs for 0-10 V (Festo, 2005).

Velocity and acceleration transducers such as tachometer are also incorporated into closed loop system for velocity feedback. It would not therefore be acceptable to include too many transducers and networks to generate all the theoretical required derivatives. With advanced technology in position transducers and the utilization of the advantages of electrical signal circuits and digital signal processing, it is preferable to use the derived velocity and acceleration by the differentiation of the position signal in velocity and acceleration feedback, which results in a less expensive and less complex control system (Weston, et al., 1984). The other advantage of the acceleration feedback is the equivalent effect as the pressure feedback on the system's performance due to the relationship between the acceleration and the pressure difference (Wang, et al., 1999).

2.3 Recent Technology in Servo Controllers

A servo controller is the brain of the servomechanism and has direct influence on its performance. It acts as a dynamic servo-loop controller that continuously monitors the actual axis position and compares this to the desired position. Any differences will result an error signal that is sent to the servo-valve, which will correct the position of the actuator accordingly. This is known as close-loop controller, which can provide automatic compensation for changes in command friction, temperature, transient, loading, and leakage.

The advent of microprocessor technology facilitates the implementation of more complex control algorithms. Moreover, different designs including the choice of working

conditions to compensate the frictions can be achieved. For systems requiring more than one axis of motion, a multi-axis controller provides the capability to coordinate individual axis controllers. This type of controllers also greatly simplifies the wiring among the servo actuators and the host controller.

Nowadays, pneumatic servo-controllers available in the market offer a good solution for most of the industry applications. For example, the SPC-200 servo-controller produced by Festo is a universal axis controller for pneumatic positioning axes. It includes position control for up to 4 axes. Self-tuning control technology is embedded within this controller. Depending upon the utilised drive, travel speeds of up to 3 m/s, acceleration values of up to 30 m/s^2 and accuracies ranging from ± 0.2 mm to ± 0.8 mm can be achieved. The user is able to freely create up to 100 programs in the "start-stop" mode, which includes more than 30 commands for complex automation tasks (Festo, 2005).

2.4 Application Spectrum

This section provides some application examples for pneumatic systems. The objective of this study is to recognise the potentials that distinct pneumatic systems from their hydraulic and electric counterparts. This should also help in identifying some requirements that should help to increase the acceptance of pneumatic systems in the market.

(1) Automobile Industry

Pneumatic servo-systems is involved in the automobile industry's production processes, providing important functions such as the secure and cost-effective handling of sheet metal in press shops, the clean control and process air for paint shops, and the smooth operation of assembly lines (Festo, 2005). Obviously, pneumatic solutions were used for all kinds of demanding tasks in the automobile industry.

One example on automobile industry is automatic seat testing. Some of car manufacturer needed to automate their seat testing procedure to achieve a reliable solution that was more cost effective than electrical or hydraulic alternatives. Any seat model can be

loaded onto the test plate which travels along the X and Y axis powered by two rodless cylinders. A third actuator is built into the vertical gantry facing the test plate; this positions the pneumatically operated test arm. The whole system, which runs from 10,000 to 15,000 cycles, is controlled by a single PLC allowing the operator to select any number of movements within the testing process.

The rodless cylinders provide a compact and integrally strong solution integrated into the test rig itself, this achieves a smaller footprint and reduces engineering costs. Although highly flexible, machine operation is straightforward and maintenance is low, especially compared to other forms of automation

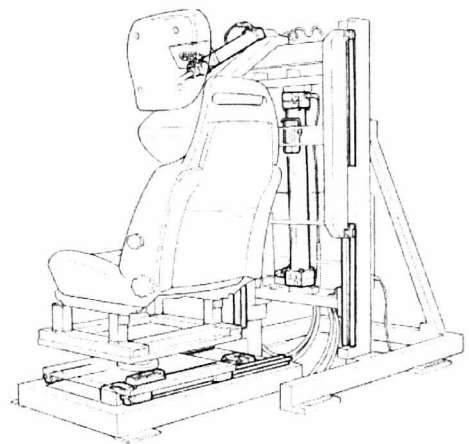

Figure 2.2 Automatic seat tester (Origa, 2005)

(2) Packaging Industry

A number of manufacturers require a flexible and reliable method of applying labels to the top, sides and around the corners of packages. Design criteria demand a rugged construction, compactness, low maintenance, high accuracy, simplicity and high speed operation.

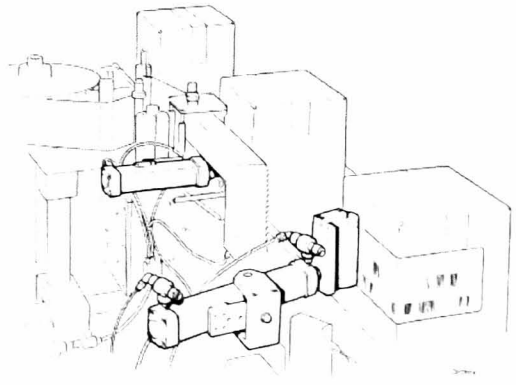

Figure 2.3 Labelling system (Origa, 2005)

The corner wrap system has become a key selling feature, providing a fast, simple and effective way of automatically labelling two adjacent panels of a box or package. The system will consistently print and apply 14" long labels at the rate of 20 products a minute, with assured wrinkle-free operation.

(3) Food Industry

The production of food is subject to strict hygiene regulations. Machines must be as easy as possible to clean on a daily basis in order to meet these regulations. Due to their high hygiene standard, resistance to aggressive environmental conditions and the frequent use of strong cleaning agents, servo-pneumatic drives have been employed in the packaging, labelling, filling, milk processing and meat processing industries.

(4) Process Industry

Pneumatic systems are used to fulfil a wide variety of functions in different process industry applications. In the chemical, petrochemical and gas industry, a high safety standard requirements have to be met in this industry, since the flow media used is often aggressive, which can lead to explosion in the event of sparking. In such cases, pneumatic systems offer solutions which can be specially adapted to individual applications. Other examples of process industry where pneumatic drives proved to be optimal solution are: paper industry, bulk goods industry, and water treatment technology.

(5) Other industries

Pneumatics has been utilized in many more industry sectors which will be not detailed here. These sectors include: welding technology, shoe industry, printing industry, clean room technology, medical technology, rail and vehicle technology, wood working industry, sheet metal working, and handling and assembly systems. For more details refer to (Festo, 2005; Origa, 2005; Bosch, 2005).

2.5 Pneumatic Systems characteristics

Based on the investigation of the recent development in the pneumatic components and their application spectrum, this section illustrates common characteristics properties that distinguish servo-pneumatic drives from their electric or hydraulic counterparts.

2.5.1 System Modularity

Modular devices are machines, assemblies or components that accomplish an overall function through the combination of distinct building blocks or modules. From this perspective, servo-pneumatic system design can be regarded as *modular design* which is based on the principle of creating components and systems by aggregating more primitive, previously tested, and possibly standardised building blocks (Isaacs, 1980). The approach can lead to extremely large savings in cost and time when building systems. These modular systems usually exhibit two distinct characteristics, namely:

(1) Standardised components and interfaces: This can much improve the cost effectiveness of manufacturing process and the quality of the product realised through the process. The inherent ability to directly actuate using pneumatic drives, and the ease with which compressed air can be supplied are the two primary factors which lead to a natural modularity of servo-pneumatic and of the components from which they are constructed (Pu, et al., 2000).

(2) Flexibility in connecting and disconnecting components: This allows pneumatic actuators and devices to be used in a distributed manner, without the need to duplicate energy sources (Pu, et al., 2000). In contrast to the distribution of energy with hydraulic drives which is more complex and requires local supply, thereby much increasing the cost and size of such drives.

This can allow servo-pneumatic systems to be tailored to specific application requirements, thereby providing opportunities for optimising performance and cost factor.

2.5.2 Speed Performance

In many application areas, pneumatic drives can perform similarly to their electric and hydraulic counterparts. Indeed a capability to facilitate the high speed operation of machinery is often claimed by manufacturers as being a distinctive feature of servo-pneumatic (Sands, 1995). Very commonly the operational speed of a machine is quoted by in terms of a *cycle time*, where the cycle time will depend upon the complexity and content of the various movements involved with a single cycle of the machine.

The high speed capabilities achieved in pneumatic systems does not involve cost penalties or impose extra constraints on the manufacturing precision of the actuator. The fastest pneumatic cylinder can now attain speed up to 30 m/s while linear electric drives can achieve a maximum speed of 10 m/s (Origa, 2005).

Nevertheless, any momentous specification of a drive system's operating speed must be evaluated together with the magnitude of associated load. Experimental evaluations manufacturers show that a load with 75 kg can be positioned within a 1 meter working stroke at 5 m/s (Origa, 2005).

2.5.3 Position Accuracy

Servo-pneumatic systems are capable of positioning accuracy on the order of 0.2 mm for a 0.2 m stroke with a typical accuracy and repeatability of ±0.1% of stroke. On the other hand, linear electric drives can achieve an accuracy of 0.06 mm for a 0.2 m stroke with a high repeatability of 0.005 mm (Festo, 2005). However, the level of precision of pneumatic system positioning is sufficient for an estimated 80% of typical industrial positioning requirements (according to Festo study). Therefore, servo-pneumatic systems are more to be considered as the lower-cost alternative to their electric and hydraulic counterparts.

2.5.4 Hygiene

The fact that working fluid can be returned to the atmosphere without contaminating the environment pays an important part in reducing the cost of manufacturing and using pneumatic actuators and valves. The resulting simplification in the construction of the components much simplified the manufacturing operation required to produce them. Moreover, during the operation in the field, there is no need for a return tank, as the case in the hydraulic systems, making the distribution of air supplies a relatively straight forward matter. Secondly, leakage is not a problem in pneumatic systems, in contrast to that of hydraulic, except in the sense that it can increase running cost. Therefore, pneumatics can be employed in many application areas, especially those applications which demand a high hygiene such as food industry. Whereas the alternative use of hydraulic or electric drives would contaminate the environment either through sparks, where electric systems are employed, or leakage in the case of hydraulic systems.

2.5.5 Cost

The inherent simplicity and modularity of pneumatic drives leads to very significant opportunities to reduce their capital cost and therefore the capital of building machines from them. This particularly for a certain power range, this corresponds to wide applications in different industry segments. Although linear motor technology has been emerged to eliminate the transmission element and fulfill the requirements of wide range of applications, "*the cost of a linear motor system is still higher than that of an equivalent pneumatic or conventional electric servo solution*" (Festo, 2005).

2.5.6 Energy Efficiency

Pneumatic systems are poor in terms of energy efficiency. The energy analysis of a compressed air system shows that only 10% of the input power supplied to the compressor is delivered as compressed air to the system [Zahng, 1999]. Inherently, the compression process rejects about 80% of the input power in the form of a heat. On the other hand, the design, analysis, and use of pneumatic drives ignore the occurrence of the chocked flow which characterises compressed air systems and affects the energy efficiency (Al-Dakkan, et al., 2003).

2.6 Discussion and Summary

To achieve high system performance in terms of position accuracy, speed capacity and an excellent cost/benefit ratio, high-quality pneumatic components are needed. Moreover, the implementation technology must be well proven and the choice of a given drive type is often strongly related to its perceived user friendliness, i.e. how easily the system can be used, controlled, supported and maintained.

This chapter reviews the recent developments in pneumatic components technology, as well as recent developments in servo controllers. This review was based on the commercial products available in the market. We can conclude that the operation and the advanced technology achieved in pneumatic components and devices made them fairly simple and robust. However, the associated control problems are more complicated to tackle.

A brief description of some servo-pneumatic applications in different industry segments is presented. It can be concluded that there are several distinct advantageous features that distinguish pneumatic systems from their electric or hydraulic counterparts, which resulted in a wide spread of pneumatic systems in the market. However, the tools capabilities for a digitally controlled servo-pneumatic system need to be directly comparable with its electric and hydraulic equivalents. In electric servo drives, for instance, "*the response of the servo positioning and motion control solution can be implemented optimally with little or no knowledge of control system stability theory. One can simply follow a recipe provided with the motion control canned software package, and get almost optimum performance from the drive*" (Houck, 2004).

Therefore, tools with such capabilities are required to make servo-pneumatic systems more user-friendly so that a long process of familiarity for the builder and the end-user is not essential. Furthermore, these capabilities should help in system commissioning which may cause poor performance of servo-pneumatic control system if not performed appropriately.

This lack of such tools and capabilities is due to the extraordinary nonlinearity associated with pneumatic systems, which make it hard to the end user to comprehend the system behaviour, and therefore it requires an in-depth knowledge from the user to be able to deal with such systems.

Chapter 3

Review of Control Methodologies, Modelling Approaches, and Simulation Tools for Servo-pneumatic Systems

3.1 Introduction

The main objective of this chapter is to provide the author with a comprehensive understanding of the control methodologies, modelling approaches, and simulation tools in the area of servo-pneumatic systems. This will assist in identifying specific research problems. It also serves the purpose of identifying potential solutions for these problems. The review focuses on the following three main areas: (1) control and tuning methodologies, presented in section 3.2, (2) modelling approaches, presented in section 3.3, (3) simulation tools and techniques, presented in section 3.4. Finally, section 3.5 summarises the findings of the literature survey and concludes the motivations for undertaking this research study.

3.2 Control and Tuning Methodologies

This section reviews some control and tuning methodologies which have already been used with some degree of success in controlling servo-pneumatic drive.

3.2.1 Background

Shearer (1956) is the first who attempted to analyse pneumatic control systems. He studied an open loop system consisting of a pneumatic servomotor driving an inertia load acted upon by an external force. Shearer advanced his study further to include closed loop pneumatic system with position feedback. However, the system was designed for military purposes, whereby the hot, high pressure exhaust gas was used as the working media. For safety reasons, low supply pressure is mandatory in most of the industry applications.

Burrows and Webb (1967) simulated an on-off pneumatic servo-system using on-off valves rather than servo-valve which was more expensive and less availability during that time. Their control system was based on the model formulas that developed by Shearer which are still valid for low pressure. Burrows and Webb (1969-1970) extended their previous work by investigating a low-pressure on-off servo-pneumatic system with a linear inertia load, in the presence of ideal coulomb friction, and with position, velocity, and pressure feedback.

Shearer, Burrows, and Webb described load motion with the piston operation about an operating point which was the mid-stroke position. It was then proved in (Burrows, 1969) that there is an inherent damping dependency on the position, and the system is inherently less stable when operating about the mid-stoke position compared with any other positions.

There was no substantial advancement made available until 1980s when Mannetje (1981) introduced a high gain differential pressure feedback loop which alters the valve characteristics to a constant pressure source independent of the amount of flow. The method showed significant improvement of the system bandwidth. Weston, et al., (1984) considered microprocessor-based low cost servo-pneumatic drives. Based on a linearised model, a three loop controller with position, velocity, and acceleration feedback was implemented. It achieved a substantial improvement in both the static and dynamic performance of the drive when positioning loads in a "point-to-point" mode with a brake subsystem.

Moore, et al. (1985) proposed a general-purpose gain-scheduling control scheme, which can be used to tackle the problem of position dependency of the optimised solution which employed to the control of non-linear pneumatic system. A three-loop controller (position, velocity, and acceleration feedback with velocity and acceleration derived from position signal) was developed by Backe (1986; 1993). The system stiffness and damping were improved and it was realised that the position performance is strongly affected by the quality of the servo valve, its time response, and the friction behaviour. A three

control phases (full opening, full reverse, and original real time control) scheme based on "front-end" control was proposed by Pu and Weston (1988; 1989) to optimise the position time.

Klein and Backe (1993) introduced a state loop fuzzy controller for servo-pneumatic systems. The fuzzy logic is especially suitable in the case of a high complexity of the mathematical description of the system but depth knowledge is required to form the rules. Consequently, a thorough understanding of the servo-pneumatic behaviour is essential for those who adopting fuzzy logic control approach. Fuzzy controller for servo-pneumatic was extended by (Choi, at el., 1995; Shih and Ma, 1998; Situm and Novakovic, 2003; Dindorf and Takosoglu, 2005). Other researchers developed different controller schemes based on multi-layer neural network (MNN) (Gross and Ruttan, 1997; Song, et al., 1997).

In parallel with that, some efforts have been spent on applying online controller adjustment mechanism, referred to as adaptive control. This approach has been implemented on servo-pneumatic positioning control by many researchers (Wikander, 1988; Tanaka, et al., 1998; Li, et al., 2003). Sliding mode control has been investigated for servo-pneumatic system which is claimed to attain good accuracy and sufficient stiffness (Tang and walker, 1995; Pandian, et al., 1997; Jalal, et al., 2004). The following subsections explore the most popular control methodologies that have been applied with some success on servo-pneumatic systems, and state their strengths and drawbacks.

3.2.2 PID-based Control

The Proportional-Integral-Derivative (PID) controller are well established, both in practical and theoretical terms, and provide a foundation for the use of other control approaches. The controller looks at the current value of the error, the integral of the error over a recent time interval, and the current derivative of the error signal to determine how much the correction value as well as how long it should last for. The controller stops working as soon as the process variable has matched the demand value for a certain time. On the other hand, if the current error is large or changing rapidly or has been accumulated for a long time, the controller will attempt to produce a large output.

One of the control structures that based on the PID controller is a cascade control structure, which is commonly used for motor drives because of its flexibility. It consists of two distinct control loops; the inner speed loop is followed by the position loop. If position needs to be controlled accurately, the outer position loop is superimposed on the speed loop. Bandwidth (speed of response) required to be increased towards the inner loop. Unfortunately this structure has received a little consideration for servo-pneumatic systems (Wakui, et al., 2003; Hildebrandt, et al., 2003; Guenther and Perondi, 2004).

Although it is easy to implement any PID-based control scheme on the servo-pneumatic system, the tuning procedure of the gain parameters K_v, K_i, and K_f is complex due to the extraordinary high non-linearities associated with servo-pneumatic system. The system can be tuned to work adequately with one operating point but fail with other operating conditions. For example, the critical proportional gain values of a closed-loop pneumatic vary with the load position of the drive. Therefore, correct identification of the tuning procedures will be vital to ensure a reliable and robust operation of servo-pneumatic drives.

Many methods for tuning PID controllers have been proposed but every method has some limitations. Cominos and Munro (2002) presented a survey on different tuning methods and techniques and their limitations. In the context of pneumatic drives, the most popular methods for PID tuning are gain-scheduling, fuzzy logic, neural network, and self-tuning.

3.2.2.1 Gain-scheduling

One method for tuning the controller is "gain-scheduling". This method aims to overcome the nonlinear behaviour by "scheduling" the associated control gains in response to variations in the system conditions or status. A study of a gain-scheduling method for controlling the motion of pneumatic actuators was introduced by Pu, et al., (1993). Basically, to compensate for the nonlinearity arise, for instance, from the valve, a look-up table can be built to characterise the actual performance characteristics of the control valve. As a result, the table can be used to associate the control gain constants, typically, the proportional gain term can be "scheduled" accordingly.

Another approach for "gain-scheduling" is to use a "gain table", where table entries are made automatically during system operation. Such a system could establish (or learn) appropriate controller parameters for different operating and initial conditions. Therefore gain-scheduling provides a means of compensation for known and repeatable nonlinearities. However, one major drawback of the method is that the system design is time-consuming if the learning capabilities are not included with the control system (Wang, et al., 1999). Another drawback is that the method is extremely difficult to be employed if the dynamics of the system are not known sufficiently either through the use of learning or priori modelling. More intensive research and critique for this method can be found in (Rugh and Shamma, 2000).

3.2.2.2 Fuzzy Logic

The tuning of the controller gains tends to be of a "fuzzy" nature. Therefore, some researchers utilised fuzzy logic for PID tuning (de Bruijn and van Wal, 1993; Visioli, 2001; Situm, et al., 2004). With fuzzy logic, new forms of tuning techniques can be realized without the need for mathematical identification. To describe the control logic, the knowledge of human experts in the form of a rule base is used instead of mathematical equations based on linear analysis.

This approach does not require significant potential power; therefore it can be easily embedded within a digital controller. However, the process of fuzzy logic tuning is related largely on a trial-and-error basis. It can be difficult to predict the learning steps and the length of time required. It may also be difficult to determine the maximum allowable gain values and they may have to be chosen through trial-and-error experimental means, which involve a tedious process.

3.2.2.3 Neural Network

Tuning approaches based on neural networks negate the need for detailed knowledge of the system being controlled. The network can be used to emulate the servo-pneumatic system after being trained and validated. As soon as the designed motion profile is fully defined, a feature extraction module can be used to analyse and extract the response

features (e.g. overshoot, damping ratio, and steady state error). If the chosen criteria are not met, these features will then be fed to the rule-based parameter adjustment mechanism (PAM). New values can then be assigned to the associated control parameters by the PAM, based on the characteristic values of the response features, the pre-defined rules and the specified performance requirements. Iteration continues until the specified performance achieved. Fujiwara and Matsukuma (Fujiwara, et al., 1995; Matsukuma, et al., 1997) present neural network based tuning PID controller.

Tuning using neural networks has a number of advantages over conventional, manually based, tuning practice. The method requires only a representative input-output data set, and therefore no detailed models need to be derived or used. Furthermore, tuning is achieved off-line, thereby reducing potential for accidental damage to the drive system during the commissioning phase of the system. However, this method suffer from two main disadvantages; firstly, the quality of the emulated results, i.e. how closely and with what level of robustness it can mimic the actual response characteristics, and secondly, the length of time involved before a converged result can be obtained.

3.2.2.4 Self-tuning

Self-tuning is also referred to as automatic tuning. The term automatic tuning covers a variety of concepts such as adaptive tuning, self tuning, tune on demand, and pre-tuning. Hardie (1988) in his article clarified some of the terminology related to this issue. However, self-tuning is not just a matter of automating the tuning process which typically carried out manually. With self tuning more mathematical mechanisms are employed which may incorporate some methods which are difficult to realise by manual operation. Two approaches currently adopted for process control which may be widely applicable for tuning pneumatic drives are (1) the pattern-recognition method (Kraus and Myron, 1984; Swiniarski, 1990; Xing, et al., 2001), and (2) the relay auto-tuner (Hang and Sin, 1991; 1992; Liu, et al., 2004).

With the pattern recognition approach, the response to step changes or disturbances is observed, and the controller parameters are adjusted based on the response pattern. The

procedure imitates the procedure used by an expert. It is necessary for reasonable controller setting to be known and selected prior to the self-tuning process. On the other hand, the relay auto-tuner is based on the belief that knowledge of the ultimate frequency (i.e. frequency where the phase lag of the open loop system is 180 degrees) is the critical information for tuning the controller. Many works are dedicated to the PID self-tuning for pneumatic systems (Shih and Huang, 1992; Shih, et al., 1994; Fok and Ong, 1999).

3.2.3 Feed-forward Control

Feed-forward control is to feed the command input signal or a disturbance signal forward in some form and to combine it with signals produced by a feedback loop or loops. The objective is to modify the feedback information in such a way that error resulting from the change in the command input or the disturbance can be compensated for. The addition of feed-forward disturbance control does not normally affect the stability of the system or the nature of its response to the changes in the command.

The main advantage of the feed-forward is that a low gain proportional controller can be used. The use of a high gain proportional control loop tends to increase the closed-loop stiffness of the system but deteriorates its stability. The feed-forward controller is designed so that it aims to eliminate system error. This arrangement is desirable when the system stability is crucial and only a small following error is allowed. For pneumatic drives, Pu et al. (1992) have shown that a velocity feed-forward term can be effective in reducing speed following errors. Because of all these advantages, this control methodology has been applied widely in servo-pneumatic drives (Balasubramanian and Rattan, 2003; Wang, et al., 1999; Varseveld, et al., 1997; Pu, et al., 1992).

3.2.4 Adaptive Control

Typically adaptive control systems consist of two loops; an ordinary feedback loop and a second loop which adjusts parameters of the feedback loop. The parameter adjustment loop comprises two main modular elements. One module performs operations to estimate the parameters of the model governing the plant process. Based on this knowledge, the second module computes the value of control parameters for the regulator. Both

parameter estimation and control adaptation can be done in many different ways. Recursive least-squares estimation is one of the most commonly used estimation technique.

For pneumatic drives, appropriate plant parameters can be the supply pressure, the moving mass, the ram area, actuator stroke length, etc.... For a given application system, the ram area and actuator stroke length will be fixed. The plant operator may be required to input information concerning other parameters (such as the moving mass) to the controller prior activating the system. Other parameters, such as upstream inlet pressure may vary while the system is running and their value measured and reported to the control adaptation module.

As noted, the adaptive control can be viewed as an online controller adjustment or adaptation mechanism. Many examples of conventional adaptive control application in servo-pneumatic can be found in (Wikander, 1988; Bobrow and Jabbari, 1991; Baoren, et al., 1997). A conventional direct and indirect self-tuning, and model-reference adaptive system (MRAS) were utilized in these works. Tanaka and Yamada (Tanaka, et al., 1998; Yamada, et al., 2000) adopted neural network in the conventional MRAS demanding satisfactory control performance. The controller is used to perform model matching and the neural network is employed to compensate for nonlinearities. In later work, Li and Tanaka (Li, et al., 2003) combined internal model control (IMC) with neural network.

Despite the fact that adaptive control improves the dynamic performance of the system and reduces the influence of the nonlinearities associated with servo-pneumatic systems, it is aimed at or based on online evaluation of the process variation and updating the controller. Online evaluation and tuning may cause accidental damage to the drive and results in poor safety.

3.2.5 Neural Networks and Fuzzy Logic Control

Considerable research efforts have been spent on developing controllers based on fuzzy logic (Matsui, et al., 1990; Shih and Hwang, 1997; Wang, et al, 1996; Dindorf and

Takosoglu, 2005) and neural networks (Fujiwara, at el., 1997; Song, et al., 1997; Chen, et al., 2004, Ahn and Yokota, 2005) for servo-pneumatic systems.

Fuzzy logic can be regarded as a set of heuristic decision rules which based on intuition and experience. It relies on integration the understanding and experience of skilled operatives by expressing it using a natural description language. Therefore it employs the system descriptive model rather than complex numerical equation (Lee, 1990). The structure of the fuzzy controller largely depends on the input and output classifications. Some of the common used in pneumatic systems and well understood models are (1) Fuzzy PI controllers (Li, et al, 1999), (2) Hybrid Fuzzy Controller (Parnichkun and Ngae, 2001), and (3) Fuzzy adaptive Controller (Gao, et al, 2005). The advantages of the fuzzy controller are firstly, a thorough understanding of the system or an accurate model is not required unlike other optimal and adaptive control strategies, and secondly the simplicity of design and implementation, and thus significantly reduces the time required to develop the entire system (Ferraresi, et al., 1994). Experience shows that the success of a fuzzy control depends on the level of knowledge concerning the positioning system's physical behaviour.

On the other hand, neural networks are used nowadays to tackle some of the pneumatic control problems because of their inherent parallelism and the ability to learn any kind of nonlinearity associated with the system. Generally, neural networks based control system performs a specific form of adaptive control. It is represented as a nonlinear map between the inputs and outputs which represent the dynamic behaviour of the system. This map forms a network that can be trained to implement any kind of control strategy. The network is able to change the parameters of the plant. It also can be trained on-line to adapt to the parameters changing in the plant. In the literature on neural networks architectures for implementing a large number of control structures have been proposed and used for servo-pneumatic systems: (1) Model reference Control (Sakamoto, et al., 2002), (2) Internal Model Control (IMC) (Li, et al., 2003), and (3) Predictive Control (Sorensen, et al., 1999). This controller offers many advantages over others: (1)

Parallelism, (2) Capability of non-linear mapping, and (3) Training is possible for various operating conditions, therefore it is adaptable to different operating conditions.

Despite all the advantageous features that are offered from the intelligent controllers, they are still suffering from many drawbacks which made them not as popular as conventional PID controllers in the market. Fuzzy with fixed structures fail to stabilise the plant under wide variations of the operating condition (Chen, et al., 1993). Moreover they lack the parallelism of Neural Controllers. On the other hand the neural networks are more adaptive by manipulating their weights according to the operating condition. However, their performance may effected by the presence of noise and other uncertainties. Furthermore, in some circumstances neural controller structures require the model of the plant, which is tedious to get in case of plants whose model is uncertain (Dash, et al., 1997). It may also be difficult to design controllers based solely on the relationships between the input-output magnitudes, without a thorough understanding of the system behaviour (the operative experience) (Ibanez and Ketata, 1994)

3.2.6 Control Systems with Friction Compensation

Friction in pneumatic systems is highly nonlinear and may result in steady state errors, limit cycles, and poor performance. Therefore it is important for control engineer to consider the friction in the design of the control system. Following are some methods for friction compensation.

- *Dither signal*: One of the simple ways to compensate for friction is to use a high frequency signal that added to the control signal which known as dither signal. This signal introduces an extra force that makes the system move before the stiction level is reached. The influence of this signal is like removing the stiction force from the system.

  In some situations, the friction compensator should be added to the current loop of the control system. This would be difficult to be implemented with conventional systems because it is not easy to modify current loop. In such cases the proposed CMRE

provides an advantageous feature since the whole control system is implemented with computer control which facilitates the adoption of the friction compensator in any loop of the control system.

- *Model based scheme:* this scheme can be applied if a good friction model is available. In this scheme the friction force is estimated using a model, and a signal that compensates the estimated friction force is added to the control signal.

- *Feed-forward compensation:* the friction, for tracking tasks, can be predicted and partially compensated for. The advantage of this method is the elimination of the lag and the noise effects of the velocity prediction. It is only suitable for tracking since the desired speed profile is known in advance.

- *LuGre model:* in this method the system can be decomposed into a standard feedback configuration with a linear block and a nonlinear block. Passivity theory can be then used to derive conditions on the controller that guarantees the stability of the closed loop system.

- *Bliman and Sorin LSI model:* the main difference between this method and the LuGre model is that the control design is realized directly in the spatial coordinates instead of the time domain. This is particularly suitable for systems with dominant elasto-plastic effects.

For more details on friction models and compensation you can refer to (Olsson, 1998).

3.2.7 Discussion

Even though various control methodologies have been employed in servo-pneumatic systems to some degree of success, further research and investigation is still required for them to be available commercially. To attain that, a reliable tools and working environments should be employed in order to facilitate the application of different control methods on servo-pneumatic systems.

The review states that all control methodologies have their own strengths and weaknesses. Nevertheless, PID-based controllers can be considered the simplest methodology among them and the most commercially available. Practically the most challenging issue associated with PID-based controllers is the tuning procedure, which can be extremely difficult due to the high nonlinear characteristics of pneumatic drives.

Generally, to facilitate the application of different control schemes and the tuning procedure of the controller in servo-pneumatic systems, three main issues should be considered:

(1) The modelling technique should be efficient to attain a good model. Conventionally, the control performance of a traditional controller fully depends on the accuracy of a known system dynamic model. The complex servo-pneumatic drive positioning process has non-linear and time varying behaviours; thus it is crucial to choose an effective modelling technique to achieve an accurate model (Shih and Tseng, 1995).

(2) The simulation tools and environments that will be employed to smooth the progress of controller design and optimisation. Therefore, a powerful simulation tools and environments are required, which capable to simulate different system complexity and has a certain degree of flexibility to adopt different controller schemes on the real system.

(3) System parameter identification, classic controls can meet the requirements for stability, accuracy and rapid response; providing that there is an optimal match between the real values of the system's physical parameters (e.g. frictional parameters) and the values used for system design, and there is no external interference (change in load).

The following sections review the existing modelling techniques and simulation tools that have been employed to model and simulate servo-pneumatic systems. The aim of this review is to focus on the crucial aspects that should be considered in modelling and simulating servo-pneumatic systems, which should help to select the most useful

modelling technique and simulation platform in order to design and implement the proposed environment.

3.3 Modelling Approaches

The main gain of system modelling is to solve practical problems by creating mathematical models or equations, these equations can be utilised to analyse the system performance. The results of the analysis will assist to show how well the system has met the design requirements and how best to modify and optimise the design as necessary. Contrasting to hydraulic or electric systems, pneumatic systems generally exhibit highly nonlinear characteristics due to the compressibility of air, the complicity of friction presence, and the pressure drop (Moore, et al., 1993). To overcome the nonlinearity of the model's governing equations, it has become a common practice to replace a non-linear model of pneumatic system by a linearised version which provides a reasonable description of the system dynamics around an operating point (Uebing 1997). However, this approach cannot predict the global behaviour of the positioning system which considered as important as a particular solution around the operating point (Mo, 1989).

The following sections review and discuss the techniques normally employed in the modelling and analysis of pneumatic servomechanism.

3.3.1 Black-box Modelling

Black-box technique or "*empirical curve fitting*" is employed when only the input/output behaviour of the system, and not the physical meaning of the model parameters, is of interest. The basic concept of the method is to deal with the physical system as a "black-box" with the input and output variables defined from the parameters which can be measured in a series of experiments on the system.

As stated before, pneumatic systems reveal a nonlinear behaviour if they considered over a wide operating range. Therefore, it is more effective to employ nonlinear modelling techniques to describe nonlinear behaviours. Nonlinear modelling techniques are essentially the regression methodology applied to the input-output data sets which is

regarded as nonlinear black-box structure. In this context, this structure is based either on neural networks or fuzzy logic. This approach has been employed by many researchers (Lai and Lin, 1993; Kroll, 1996; Kleutges, et al., 1998; Semerci, et al., 2005; Ke, et al., 2005).

The disadvantage of this method is the need to acquire a sufficient set of experimental test data of the system which may not be exist at the time of the design. Furthermore, this approach can not be adopted as a general analysis or configuration for modular servo-pneumatic system since the model is created for particular actuator with certain dimensions. Therefore, a new data collection and training procedure should be conducted if any modification on the system is applied. Finally, problems are still present particularly in the computational difficulties of the methods and validity test.

3.3.2 Grey-box Modelling

This modelling approach is utilised when some physical insights are available, but several parameters need to be determined from test data. Grey-box modelling can be thought of as a combination of a "white" model and a "black" model. The "white" model should be established by explicitly utilising prior knowledge and physical insights such as that on the clear nonlinear structure derived from physical laws. On the other hand, the "black" model should be used to approximate the neglected or un-measurable nonlinearities. In such a grey-box, the system will not rely entirely on artificial structures such as fuzzy logic and neural networks as seen in black-box model approximators.

State-space modelling is considered as a typical example on grey-box modelling. Basically, the model consists of a set of ordinary differential equations (ODEs) which are usually derived from the physical operating mechanism of the system. These ODEs consist of state variables and state coefficients. In the case of servo-pneumatic systems the piston position and velocity, and the pressure in both chambers are the state variables, and the friction coefficients and flow rate in the valve and the chambers are the state coefficients.

If the state variables are known at some time, then any plant output can be computed, at all future times, as a function of the state variables and the present and future values of the inputs.

In the context of servo-pneumatic drives, Scavarda, et al. (1987) proposed two state space models of nonlinear servo-pneumatic drives: one is valid around the equilibrium position rather than only at the centre position and the other describes the behaviour of the system a bout constant speed steady state, based on some previous work (Shearer, 1956; Burrows, 1969).

Many researchers employed state-space modelling technique to create an optimal model for pneumatic system (Acarman, et al., 2001; Wang, et al., 1998; Sorli, 1994; Marchant, et al., 1989). However, this approach suffers from many drawbacks. Firstly, this approach is lack of a well-established rule for determining the state variables (Stoica and Jansson, 2000). Secondly, estimating the state variables demands high computational process and high storage requirement, which have prevented, its real-time application (Gambler, 2004).

3.3.3 Linearisation of Non-linear Model

Linearisation techniques are commonly used in nonlinear system analyses. Most of them are based on Taylor series expansion to formulate a systematic procedure for obtaining a linear model for a nonlinear system. The operating curves of the physical system can be used to formulate the relation of the input and output variables when it is not available in mathematical form. The system with nonlinear relationships is linearised to obtain a set of linear algebraic equations which can be solved readily.

Linearisation process for the nonlinear model of pneumatic system has been employed by many researchers (Ke, et al., 2005; Semerci, et al., 2005; Hamiti, et al., 1996; Liu and Bobrow, 1988; Burrows, 1969; Shearer, 1956). They claim that linearisation analysis is valid for dynamic characteristics analysis and effective for control system stability investigation around the operating point. However, the major disadvantage for

linearisation is its reference to the linearisation point, so that when the operating point gets further away from the linearisation point, the linear model's behaviour will increasingly deviate from the behaviour of the real system which will invalidate the linearisation approach.

3.3.4 Component-based Modelling

Component-based approach has been applied firstly in the software industry over the last decade (Pour, 1998, Meijler and Nierstrasz, 1997). The main concept of component-based approach is that the common system *components* can be created once, and then reuse it as many times as it is required. A component is an entity or subsystem which designed to perform a certain functions and tasks and can be used as a building block in the design of a larger system. Hardware and software components are sharing common characteristics: (1) component is visible in some forms, (2) component can individually and completely perform one or more tasks, (3) component can set-up some relationship with other components in connections, and (4) a component can be a part of whole system or part of another component.

Mo (1989) has initially introduced the adoption of the component-based approach as a modelling technique for pneumatic servo-drive, which provides flexibility of modelling a wide variety of components. Pu et al. (2000) has conducted further study in advancing the capabilities of servo-pneumatic system with the component-based approach. Basically, this modelling approach starts by the process of identifying the components within the physical system, and then determining the linkage nature between the components when several of them are connected together. Components model are created, evaluated and validated separately. This will lead to a reduction in model complexity, an increase in their robustness, and a faster model formulation.

All components should include parameters providing adaptability to different design. They can be stored in a library as validated ready-to-use models, which enable reusing

(Kamigaki and Nakamura, 1996). The complete model is obtained by successively aggregating components until having a model sufficiently representing the whole system.

3.3.5 Discussion

Among the various modelling techniques presented above, the component-based modelling approach is particularly relevant to the development of the new generation of software/simulation tools. Component-based techniques, in general, have been used extensively in systems development due to their great potential for reducing time and cost, improving system maintainability and flexibility, and enhancing system quality (Pour, 1998). Therefore, adopting this technique to model and simulate servo-pneumatic drive will provide the following potentials:

(1) Component-based model is valid over the whole working range of the servo-pneumatic drive. This is because of including the nonlinear phenomena of servo-pneumatic drive without the need for linearization.

(2) Provide flexibility of configuration choice and direct interpretation because the system is broken down to subsystems or entities.

(3) Changes on the real system components can be reflected easily on the model.

(4) The capability of modelling a wide variety of machines and handling different types of pneumatic drives.

(5) Facilitate the modelling of different complexity in order to allow for variation in future.

3.4 Simulation Tools and Techniques

As explained previously, the system model can be derived analytically as a complete mathematical model, empirically as a physical or test based model or semi empirically where the mathematical model is supplemented by test data. In any of these cases, the model is a simplification of the reality that can be used to approximately describe the real system characteristics. However, the model becomes useless unless it can be converted into a form that is suitable for computer manipulating. With servo-pneumatic systems, which are highly nonlinear, a simulation tool with the capability for predicting the system behaviour is definitely needed. It will be more advantageous if the model can be

simulated with respect to time so that it can be easily attached to the real system to form a mixed reality environment.

Most simulation studies for servo-pneumatic systems are implemented using a simulation package, for example, Propneu (Festo, 2005), Hypneu (Bardyne, 2006), and Automation studio (Famic, 2005). The advantages of such packages are reduced programming requirements and natural framework for simulation modelling. **However, these tools are designed for pneumatic circuit design rather than servo-pneumatic systems design and simulation.** On the other hand, a simulation model can be built using general purpose simulation platforms, such as MATLAB and MATRIX-X, which are familiar to the analyst and less expensive.

As discussed in the previous section, component-based modelling will be adopted in this research study, therefore a component-based simulation environment is required. One example of component-based simulation packages is ControlShell (Schneider, et al., 1995) which is dedicated solely for control system design. It based on building data-flow systems from small, reusable components, each of which implements a specific functionality within a sampled data environment via methods that run at well-defined times. Components read input signals, generate output signals, and use reference signals. A library of pre-defined components is provided, ranging from hardware device drivers and controllers to trajectory generators and sophisticated motion planning modules. New or custom components can be easily added to the system via a graphical data interchange editor and C++ code generator.

In this project, MATLAB/Simulink has been selected as a platform to develop this tool because of the following reasons: (1) the availability of templates/building blocks, (2) it provides interactive entry and editing of parameters, (3) its ability to graphically display the simulated data as well as tabulate them for fine inspection, (4) its ability to visually program a dynamic system and look at results, (5) its flexibility where the user can create different reconfigurable and reusable components, and (6) the ability to be attached to the real system through a data acquisition card.

3.5 Discussion and Summary

This chapter has presented an overview of current developments in the area of servo-pneumatic control, modelling, and simulation. It has identified various issues and opportunities with regard to the implementation of different control methodologies under a simulation environment and based on a specific modelling technique. In general, the identified issues can be summarized into the following:

(1) Limitations of existing control methodologies that have been applied in pneumatic systems whether they are conventional or intelligent approaches.
(2) Lack of an optimal strategy for controller tuning and optimization.
(3) Limitations of existing simulation tools for pneumatic systems that are developed for commercial purposes and dedicated for circuit design only.
(4) Lack of working environment which allows the implementation of different control methodologies for servo-pneumatic systems and facilitate the procedure of tuning and optimisation.

On the other hand, various developments, which have the potential for facilitating the realisation of a more advanced control and tuning strategy, modelling, and simulation techniques can be summarized into the following:

(1) Advances in microprocessor and information technologies.
(2) Emergence of component-based concept in the field of software design and development.
(3) Advances in some general purpose simulation tools, such as MATLAB, that allow for attaching the simulation environment to the real system.
(4) Advances in sensor and data acquisition technologies.
(5) Advances in pneumatic components and technologies.

The review has pointed out to the cascade control strategy that has been employed in servo-electric systems and has received low attention for servo-pneumatic systems. This extended study is aimed at further exploring the prospective of component-based approach for creating an integration platform or mixed reality environment for improving the design and control of servo-pneumatic systems. It is expected that such environment

would allow various existing control methodologies/strategies to be implemented more easily, sufficiently, and cost-effectively.

Chapter 4

Component-based Framework for System Decomposition

4.1 Introduction

This chapter presents a framework for pneumatic system decomposition using component-based approach. Section 4.2 explains the adopted framework and demonstrates the concept through a practical example. It also presents how the adopted framework will be employed in the context of this research. Section 4.3 presents a brief description of the test bed, at the Mechatronics Research Centre, De Montfort University, UK, where the experiments for validation were conducted. Finally, section 4.4 summarise the main concepts introduced in this chapter.

4.2 Component-based Framework

4.2.1 The Concept of "Components" in this Study

A component is an entity which designed to perform a certain functions or tasks and which can be used to compose more complex entities for a given purpose. Components, virtual or hardware, share some common characteristics:

a. ***Completeness:*** the completeness of a component means that: (a) each component can individually and completely do one or more tasks (the cylinder component produce force, sensor components provides certain information about the system; (b) the status of a component itself and its output events or functions must be clear while any input event or function of the component is active at any time. The completeness of a component establishes a limit for the minimum components into which the system can be decomposed. *In other words, the completeness of a component will be a useful as how to decompose a machine into components.*

b. ***Connectivity:*** Each component can set up some relationship with other components in connection; (the valve component provides an air passage to the cylinder component),

c. ***Encapsulation:*** is one of the basic characteristics of components. A component can encapsulate two or more basic components. This characteristic of a component provides some basic idea on how to compose basic components into a sophisticated component. It is also helpful to decompose the machine into composite components then into basic components,

d. ***Reusability:*** a component can be used as many times as required to design a system. Furthermore, components' designer does not need to design all parts of a component each time. When a practical component is available, it could be reused as a part of another component. That will simplify the component design, save time and costs of designing and building new components.

The image of a component is a virtual component which implements a real component and is used in the virtual environment of the servo-pneumatic systems design. The image of the component has a rectangular form and associated I/O pins. The I/O pins represent the virtual connections which are event driven to this component or other components. An event is a stimulus from one component to another for calculating and passing values. In other words, the I/O pins are used to set up a logic connection with other components. Each pin has a pin type name which stands for the event type name.

The image of the component can be thought of as a black box from the perspective of a system designer. The functions of a component are embedded within the black box. The duty of the system designer is to set the required parameters of each component, to know what functions or events will be done by a component, and the connections between the I/O pins of each component. The designer does not need to know how the functions or events will be achieved within the component, and therefore designing a servo-pneumatic system is similar to designing an electronic schematic.

A component can be a basic component or a compound component which comprises of many components. Practically, the machine system can be decomposed into some compound components or many basic components. A component can be reused or duplicated in the same workspace to form a system. For example, two axes servo-pneumatic systems have two valves and two actuators and different loads. In this case the valves and the actuators are copies of the valve and actuator components.

4.2.2 The Approach

The adopted framework is a hierarchical structure with three different layers to target different categories of users (Figure 4.1). These layers are (1) device component layer, (2) composite components layer, and (3) modular machine layer. This component-based framework should lead to a great potential for: (I) significantly reducing the development cost and time, (II) improving system maintainability and flexibility by allowing new pneumatic components to replace old ones, (III) enhancing system quality by allowing pneumatic components to be developed by those who are specialized in the mechanical design engineering, and the pneumatic systems to be built by mechanical, control, and application engineers who are specialized in system developing. This framework can be interpreted by the Automated Manufacturing Research Facility (AMRF) model (Albus, et al., 1984).

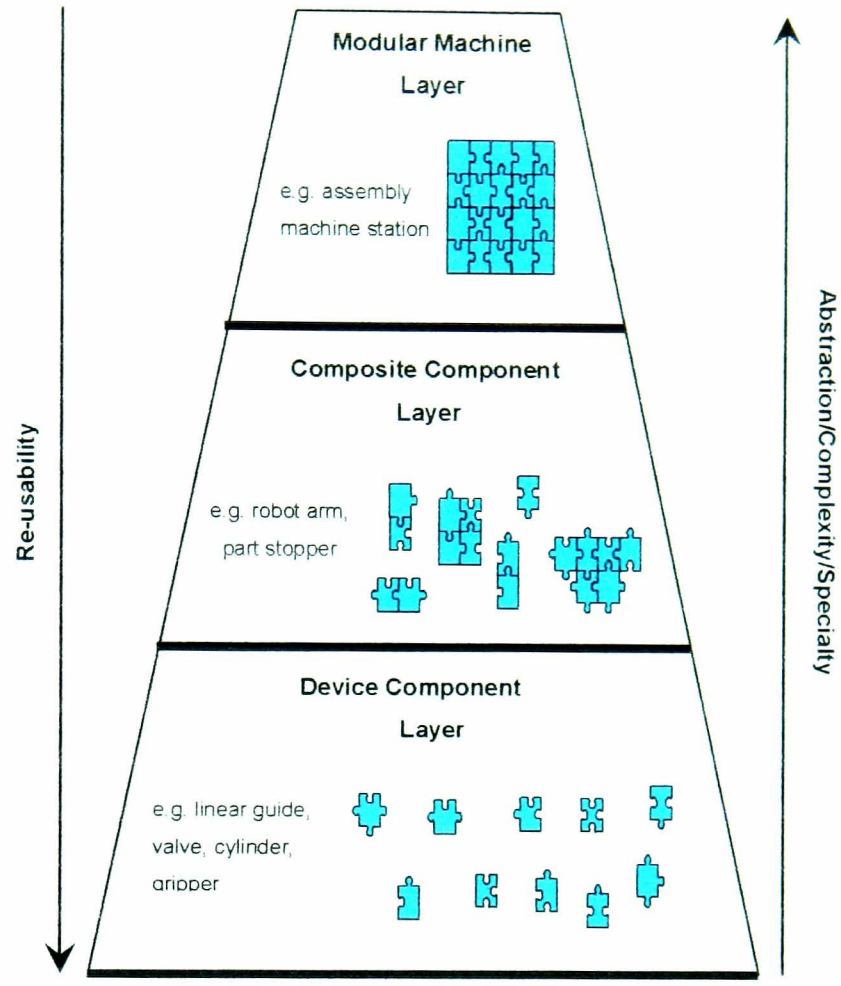

Figure 4.1 Conceptual framework for the pneumatic system decomposition

4.2.2.1 Device Components Layer

The device components layer provides the basic building blocks (components) to design and evaluate the target machine. Component models should be created and evaluated separately. These components are formulated in such a way that they can be replaced by alternative components that have the same interface specifications.

The system segregation allows a reduction in model complexity and an increase in their robustness, which leads to faster model formulation. Ideally, the component manufacturers supply the component models because they are likely to have insight knowledge about their developed components. Other sources to obtain the component models could be either from component designers (modellers), i.e. derivation of a mathematical model, or experimentally (e.g. empirical models). The level of model complexity for individual component is dependent upon the accuracy requirements,

modelling efficiency, and time constraints. The beauty of this flexibility allows the system model benefit from the subsequent improvement on individual component without re-design the system model itself.

4.2.2.2 Composite Components Layer

Composite components layer (or system layer) builds upon the component layer. A complete system is built using the components from the previous layer. In terms of complexity, it is more complicated than the components layer, but it is not complex enough to achieve a complete task in the same way that the modular machine layer does. This layer is important to facilitate component development and reuse by grouping a number of device components to provide a higher-level functionality.

4.2.2.3 Modular Machine Layer

This layer corresponds to the machine level, which executes the detailed operation sequence in order to complete a certain job; assembly machine station is a typical example. The following section presents an example of a modular machine that comprises of composite components.

4.2.3 An Example: Assembly Machine Station

Assembly machine station is one of the practical examples that can be used to interpert the introduced framework. Figure 4.2 shows the machine decomposition to different levels of complexity referring to the adopted framework. The advantages of the adoption of this framework are (1) the ability to reconfigure the system for different application and (2) the ability to accomplish rapid changeover from the assembly of one product to the assembly of a different product, the so-called *Agility* (Quinn, et al., 1997).

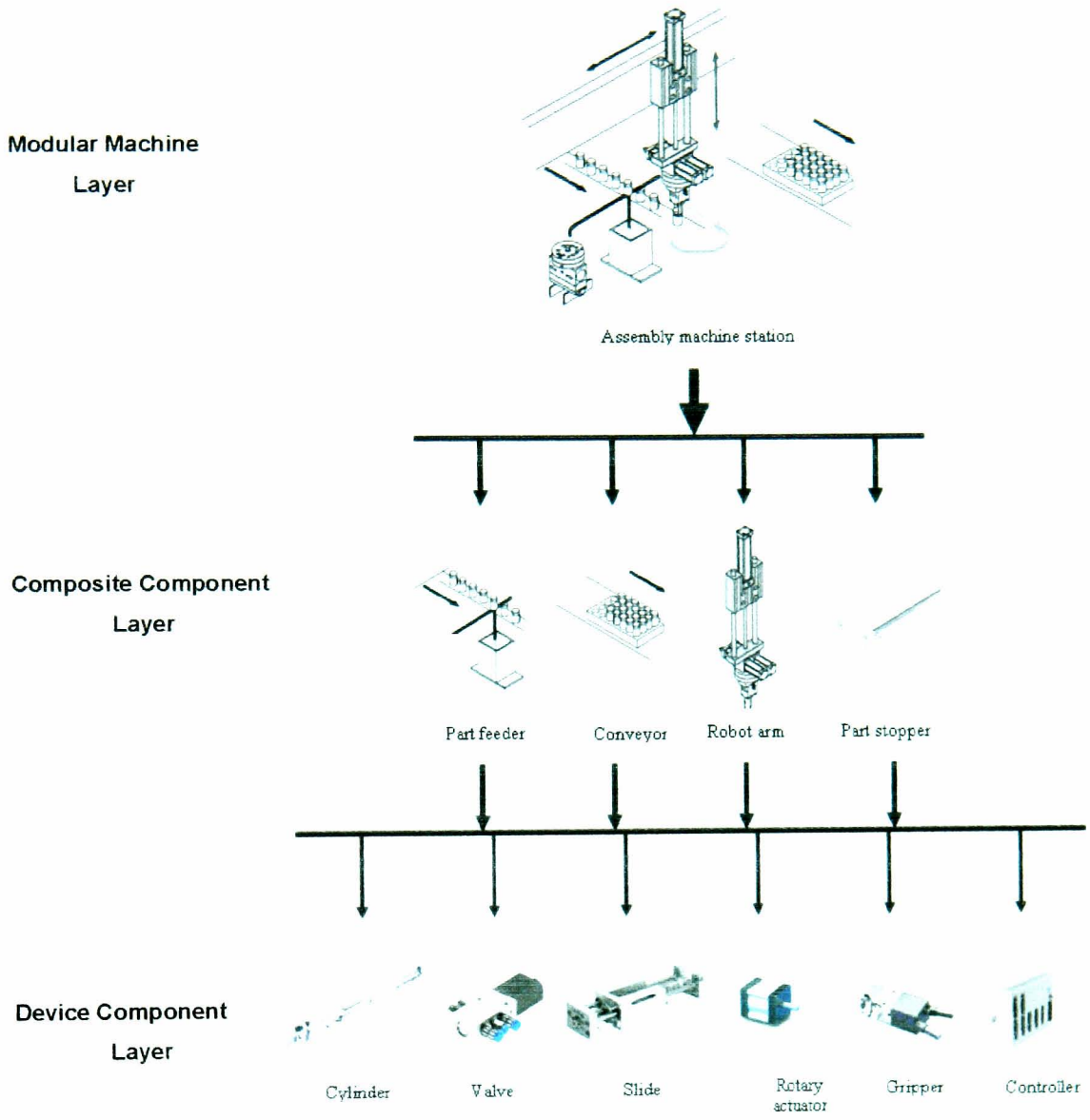

Figure 4.2 Assembly machine station decomposition in different levels of complexity

The key components of the system are:

(1) ***Robot arm:*** the robot arm utilizes re-configurable modules, which include two slide modules, one rotary module, and one gripper. The slides are attached to a metal plate and are positioned in perpendicular to each other. The rotary module is attached at the end of one slide and the gripper at the other. The rotary module allows the gripper to rotate. The gripper has two ports that allow air to travel from one end to the other, thereby providing open and closed states.

(2) ***Part stoppers:*** consist of a part stopper and a sensor. One set is positioned directly in front of each robot arm to stop the pallets as they move along the conveyor, so

that the robot arm can precisely place the parts on the pallets. The sensor detects the presence of a pallet, while the part stopper stops and starts the movement of the pallet depending on the availability of downstream robot arm. The part stopper consists of a solenoid and a pneumatic cylinder. The solenoid directs airflow, thereby causing the cylinder to extend or retract.

(3) ***Part feeder and conveyer:*** Two part feeders provide the parts for the robot arms to assemble into the work piece on top of the pallet. One part feeder is located next to each robot arm. A pneumatic air cylinder with a short stroke pushes parts forward along a guide chute. A conveyor transfers the pallet from station to station.

4.2.4 Decomposition of Servo-pneumatic Systems

In the context of this research, servo-pneumatic system will be considered since it comprises of the most commonly used pneumatic components. Referring to the adopted framework, the system can be decomposed into the following key components; a pneumatic actuator, controller, proportional or on-off valve, tubes, feedback devices, and load as indicated in Figure 4.3.

(1) The valve component, which provides a guide passage for the compressed air between the supply, the actuator, and the exhaust.

(2) The actuator and load component, which consists of two sub-components; the actuator which produces mechanical motion as a result of differential air pressure between the chambers, and the load which implement the force that opposes the actuator. This force is resultant from different natures, e.g. weight and friction forces. Combining the actuator and the load in one component simplifies the system models building since the user will not have to involve with adding any logical components between the actuator and the load. For example, if the actuator is separated from the load, then the user has to add more than one differentiator in between.

(3) The controller component, which can be treated as a black box which transforms the position information from the load to a command signal to the valve. This

black box could represent any control methodology (such as PID, fuzzy, neural, sliding mode controllers) which suits a particular application.

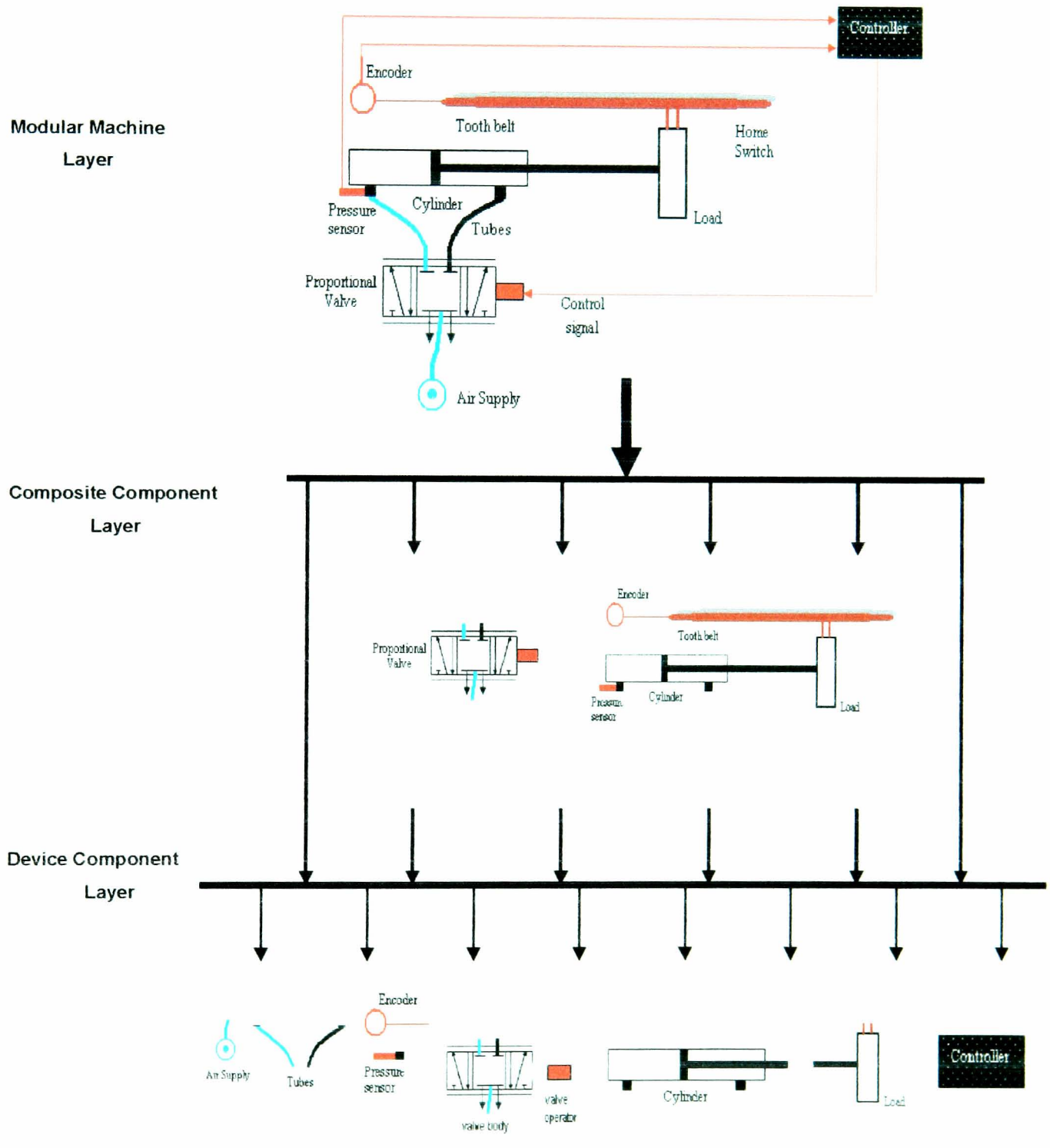

Figure 4.3 Key components in servo- pneumatic system after decomposition process

(4) The tubes, which passes the air from the air supply to the valve, from the valve to the actuator, and vice versa. The pneumatic conditions are affected by parameters which are characteristics of the tubing such as friction, upstream and downstream pressures, and length of the tube. Therefore this phenomenon cannot be neglected in pneumatic modelling.

It can be noticed that some components have been moved from the machine modular layer straight to the device component layer (such as the controller and the pipe components) skipping the composite component layer. This can be explained as these components are inherently primitive and they can not be decomposed into more primal components.

After decomposing the system into the basic components, the system model structure can be formulated. Mo (1989) has initially introduced the adoption of the component-based approach as a modelling technique for pneumatic servo-drive, which provides flexibility of modelling a wide variety of components. Pu (2000) has conducted further study in advancing the capabilities of servo-pneumatic system with the components-based approach. Figure 4.4 shows the system's components which connect to each other through a well defined interfaces. The connection signals of the components are derived from the inter-relationship of the successive components. These consist of:

(1) Pneumatic link between the valve and the actuator in the form of air mass flow rates.
(2) Pneumatic link between the actuator and the tubes, the tubes and the valve in a form of pressure and temperature.
(3) Mechanical link between actuator and load in terms of force and acceleration.

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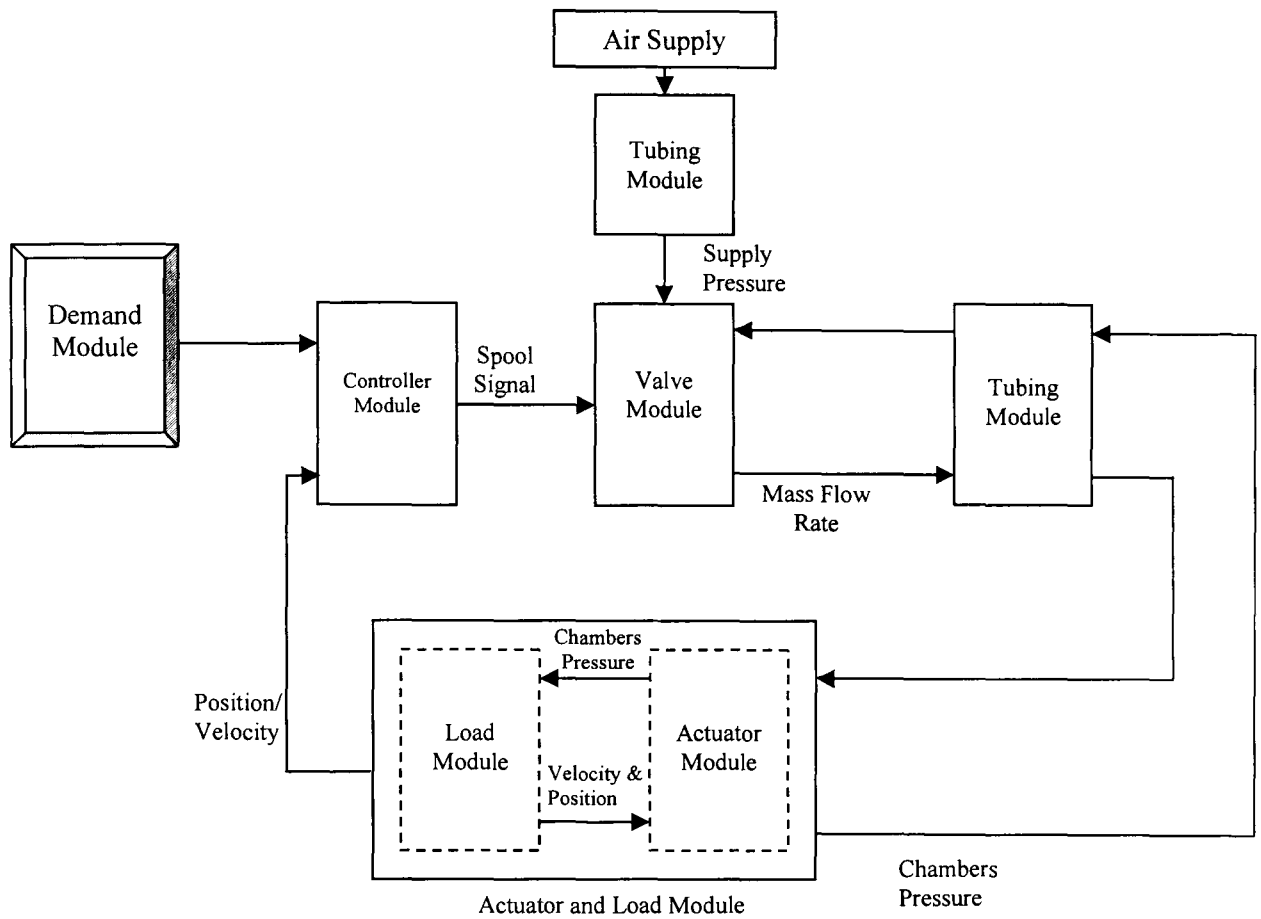


Figure 4.4 Servo-pneumatic component-based model structure with the interface variables

The level of complexity of the component model is highly dependent on the application. For applications that demanding accurate solutions, the analysis must take as many factors as possible which requires long time to achieve a solution, either by mathematical or computational methods. However, the component-based approach provides a good environment to include any model irrespective of its sophistication. Therefore, the complexity level of the component should not be affected by the level of complexity of the other components in the system model. In other words, the most critical components can be modelled as closely as possible to reality while the other components can be simplified to reduce the system sophistication and evaluation time. The following two tables summarise the function, inputs outputs, and subcomponents for the decomposed system.

Components	In real system	Virtual image (model)				
	Function description	Inputs	Outputs	Subcomponents		
				Name	Inputs	Outputs
Valve	Provides a guide passage for the compressed air between the supply, the actuator, and the exhaust.	P_a, P_b: Pressure in both chambers (from actuator-load component through tube component) P_s: Supply pressure (from compressor component through tube component). *Control*: control signal (from controller component)	$\dot{m}_a$, $\dot{m}_b$: mass flow rate in both chambers	Valve operator	**Control:** control signal (from control component)	A_{eff}: effective area.
				Mass flow function	P_a, P_b: Pressure in both chambers (from actuator-load component) P_s: Supply pressure (from compressor component).	$\dot{m}_a$, $\dot{m}_b$: mass flow rate in both chambers
Actuator - load	The actuator produces mechanical motion, and the load which implement the force that opposes the actuator. This force is resultant from different natures, e.g. weight and friction forces	$\dot{m}_a$, $\dot{m}_b$: mass flow rate – from valve component. P_s: Supply pressure – from compressor component through tube component.	P_a, P_b: Pressure in both chambers. *Acc*: Load acceleration. ***Spd***: Load speed. ***Disp***: Load displacement.	Actuator	$\dot{m}_a$, $\dot{m}_b$: mass flow rate – from valve component. P_s: Supply pressure – from compressor component.	P_a, P_b: Pressure in both chambers.
				Load	P_a, P_b: Pressure in both chambers.	*Acc*: Load acceleration. ***Spd***: Load speed. ***Disp***: Load displacement.

Controller	Transforms the position information from the load to a command signal to the valve.	***Error***: error signal ***Vel***: velocity feedback (from actuator-load component) ***Acc***: acceleration feedback (from actuator-load component)	Control signal	N/A
Tube	passes the air from the air supply to the required components	Pressure from air compressor and actuator-load components	Pressure after the drop	N/A

Table 4.1: Summary of the system's components after the decomposition process.

Components	*Parameters to be set by the user*
Valve	Port width in mm Max. spool displacement in mm Empirical coefficient K1 Empirical coefficient K2
Actuator-load	Bore Diameter in mm Stroke length in m Rod diameter in mm Internal leakage coefficient Initial piston position in m Initial pressure in the driving chamber in bar Initial pressure in the driven chamber in bar Load in Kg Static friction in N Coulomb friction in N Viscous damping coefficient
Controller	Proportional gain (K_p) Integral gain (K_i) Differential gain (K_d) Velocity feedback gain (K_v)
Tubes	Pipe's length in m Pipe's inner diameter in mm Friction coefficient Air velocity Temperature
Profile generator	Speed in m/s Acceleration in m/s^2 Deceleration in m/s^2 Time delay in s Profile length in s

Table 4.2 The parameters that have to be set by the user for each component

4.3 The Test Bed

The validation of the component-based model structure relies on the verification experiments on specific hardware. Variant hardware systems are required to cover as many aspects of the model as possible but unfortunately due to the time constraint, the experiments were conducted on typical single axis modules specially designed for

research purposes. Nevertheless, the test bed comprises mostly of standard parts (Festo parts) which will assist in developing the proposed environment in a way that make it feasible not only for research but also for industrial design.

In the single axis system, a conventional asymmetric pneumatic cylinder is used to power the actuation mechanism. A schematic diagram of the servo control system is shown in Figure 4.5.

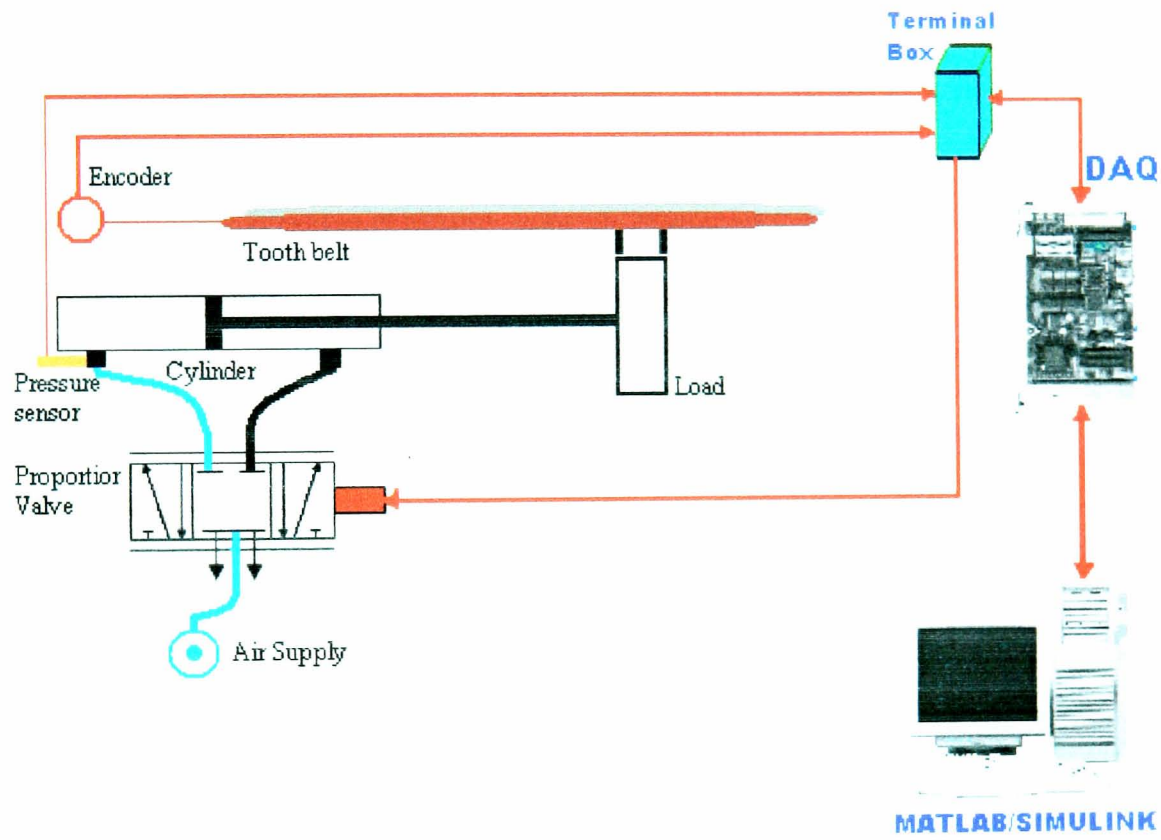


Figure 4.5 Schematic diagram of the test bed

The pressure difference between the two chambers produces the working force which moves the load along the tooth belt. An incremental encoder is fixed on the tooth belt which senses the position of the load/piston and feeds it back to the controller. Afterward, velocity and acceleration can be derived from the position signal. The pressure sensor feeds back the pressure in both chambers to the controller. Matlab/Simulink platform in conjunction with Data Acquisition Card (DAQ) is employed in the system to accomplish the signal processing and control algorithms implementation. The analogue output signal

of the controller is converted by a DC/DC converter from -10V ~ 10V to 0V ~ 10V which is used to control the valve spool movement of the proportional valve. The main parameters and characteristics of components are listed in Table 4.1:

Table 4.3 Components main parameters and characteristics

The Component	Parameters and characteristics
Proportional Valve	FESTO MPYE-5-1/4-010B:Nominal flow rate 1400 l/min (± 10%); Maximum leakage rate in new condition (6 bars, outlets closed) 251 l/min; Mid-position at 5 V
Cylinder	Asymmetric cylinder: Bore diameter D=30 mm; Rod diameter d=13 mm; stroke length L=1000 mm
Encoder	HEIDENHAIN ROD 426000b-2500 111.5 counts/mm
Pressure sensors	BT6010G4-FL: Operating pressure 10 bar (Maximum); response time (10% - 90%) in 1 ms; non-linearity and hysteresis ±0.2% FSO; repeatability ±0.1% FSO.
PCI-DAQ Card	NI-6024E; 16 channels of analogue input, two channels of analogue output, a 68-pin connector; 8 lines of digital I/O.
Supply Pressure	0 ~ 7 bars
Pipe Diameter	7 mm

4.3.1 Pneumatic Actuator

The actuator employed in the test bed is an asymmetric double acting cylinder (see table 1 for specifications). The double acting cylinder has a power stroke in two directions. The piston is moved both positive and negative by compressed air. The advantage of double-acting cylinder over single-acting cylinder is that the ability of the cylinder to carry out work in both directions of motion.

4.3.2 Proportional Valve

FESTO MPYE-5 series proportional valve has been used in the experiments, in which an analogue electrical input signal is converted into appropriate opening cross-sections at the outputs. At 5V, i.e. half the nominal voltage, the pneumatic actuator assumed to be on it's mid-position, where all control edges are closed, so that no air apart from minimal leakage passes through the valve. At 0 and 10V, the valve assumed to be on one of its end positions, with maximum opening cross-section area. The flow characteristic of the valve is schematically shown in Figure 4.6.

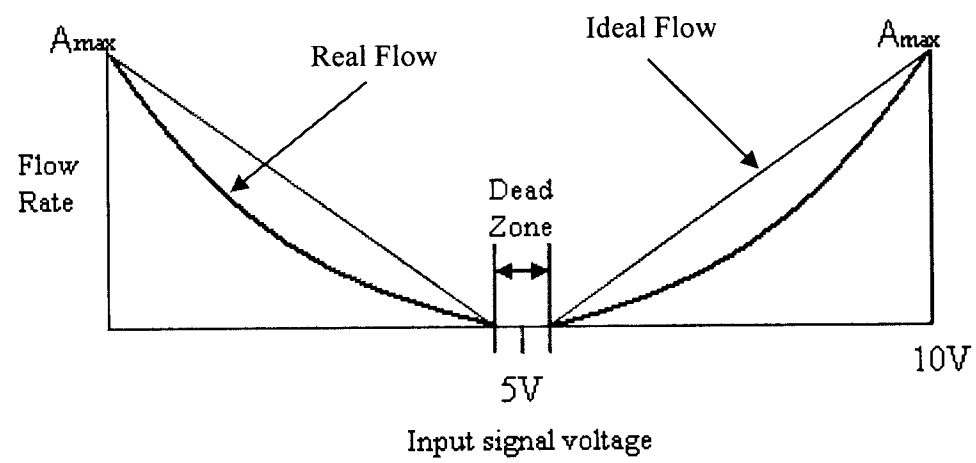


Figure 4.6 Proportional valve's flow characteristics

4.3.3 Data Acquisition System

NI-6024E data acquisition card has the task of converting analogue input signals to digital form as well as converting the digital output signal to analogue form. It features a sampling rate of 200 KS/s which is enough to perform the necessary data acquisition in real-time. The 6024E features 16 channels of analogue input (eight of them differential) with 12-bit resolution, two channels of analogue output with 12-bit resolution, a 68-pin connector, eight lines of digital I/O, and two 24-bit counter/timers.

4.3.4 Pressure Sensor

Pressure sensors were calibrated to guarantee their measurement accuracy. In the calibration process, pressure was chosen to change within the whole working pressure

range. Supply pressure was kept constant at every pressure calibration point for a long time such that constant cylinder chamber pressure could be established. Every pressure sensor calibrated was connected to the cylinder chamber to transform the pressure electrical signal. The transformed pressure signals are converted to voltage, which is fed to the DAQ. Each volt is equivalent to 1 bar in our case.

4.3.5 Software Platform

MATLAB/Simulink was chosen as the software platform. Simulink is a platform for modelling, simulating, and analyzing dynamic systems. It enables to build graphical block diagrams, simulate dynamic systems, evaluate system performance, and refine designs. Simulink integrates seamlessly with MATLAB, providing the user with immediate access to an extensive range of analysis and design tools. It also provides an interactive graphical environment and a customizable set of block libraries.

There are several reasons behind choosing MATLAB/Simulink as a platform to implement and develop the proposed environment. The reasons are summarized as following (Mathworks, 2005):

(1) Simulink provides extensive and expandable libraries of predefined blocks
(2) Interactive graphical editor for assembling and managing intuitive block diagrams
(3) Full access to MATLAB for analyzing and visualizing data , developing graphical user interfaces, and creating model data and parameters
(4) With Simulink, models can be built by dragging and dropping blocks from the library browser onto the graphical editor and connecting them with lines that establish mathematical relationships between the blocks, which facilitate the procedure of creating models of different complexity.
(5) Simulink enable the user to organize the model into clear, manageable levels of hierarchy by using subsystems. Subsystems encapsulate a group of blocks and signals in a single block.
(6) Simulink gives the ability to segment the model into design components and to model, simulate, and verify each component independently. Components can be saved as separate models or as subsystems in a library. They are compatible with

any registered Source Control Provider application on Windows platforms. The user can reuse the design components on multiple projects.

(7) Enable the user to visualize the system by viewing signals with the displays and scopes provided in Simulink.

Models built in Simulink can be configured and made ready for code generation. Using Real-Time Workshop, code can be generated from the model for real-time applications.

4.4 Summary

The chapter presented the component-based framework for pneumatic system decomposition. The component-based framework was adopted in order to facilitate the system modelling and the implementation stages of the proposed environment development process. It holds the property that the components are actual insight of the physical counterpart. Replacement of any component in the model can be associated with corresponding change in the real pneumatic system, which would improve the system maintainability and flexibility.

Component-based framework nature allows a vast number of pneumatic devises to be implemented as well as the ability to choose the suitable components which should enhance the system quality. It would also reduce the model complexity and increase their robustness, which leads to faster model formulation.

The chapter also presents the test bed that provides the test environment which will be employed to validate the proposed environment developed in this research study. Since the components employed in the test bed are products available commercially, the developed environment is anticipated to be applicable in industrial design. The test bed is a single axis pneumatic drive system which comprises of an asymmetric cylinder, proportional valve, DAQ, pressure sensors, and position encoder.

Chapter 5

Component-based Mathematical Modelling

5.1 Introduction

In chapter 4, a servo-pneumatic model structure has been formulated based on the component-based framework. The mathematical model for each component within the structure is established in this chapter. This chapter has been organised as following: Section 5.2 presents the mathematical model of pneumatic cylinder. Sections 5.3, 5.4, and 5.5 present the mathematical model for the load module, valve module, and tubes module respectively. Summary and discussion are presented in section 5.6. The mathematical model presented in this chapter is based on the derivation introduced in (Pu, 1988; Mo, 1989).

5.2 Pneumatic Cylinder Module

The governing equations of the pneumatic cylinder dynamic behaviour rely entirely on the study of charging and discharging processes of air to the controlled volume in the cylind+er chambers. This type of thermodynamic analysis has been initially applied to servo-pneumatic system by Shearer (1956). The traditional approach on the analysis is based on linearisation which makes the analysis valid only for small perturbations about an operating point. Linearisation is not acceptable in this research since it requires that the model is valid over the stroke length which allows the simulation to be mixed with the real system under the same environment.

The following analysis refers to the double acting asymmetric cylinder which can be generalized to include symmetric cylinder (by assigning the bore diameter to zero). Figure 5.1 illustrates diagrammatically the relationship of the cylinder's chambers and the inlet connections.

The assumptions that considered in this model analysis are:

(1) Supply pressure P_s and supply temperature T_s are constants

(2) Heat transfer between working gas and its environment is negligible. Temperature of the gas is at all times equal to supply temperature, T_s. that means the inherent effect of temperature variation on system dynamics is considered to be negligible.

(3) The working fluid is assumed to be an ideal gas so that the general gas laws can be applied (for example, $m = \frac{PV}{RT}$).

(4) The state condition of air, such as pressure and temperature, in the bounded volume are homogeneous. For instant, the pressure at the inlet is the same as that near the piston in the same cylinder chamber.

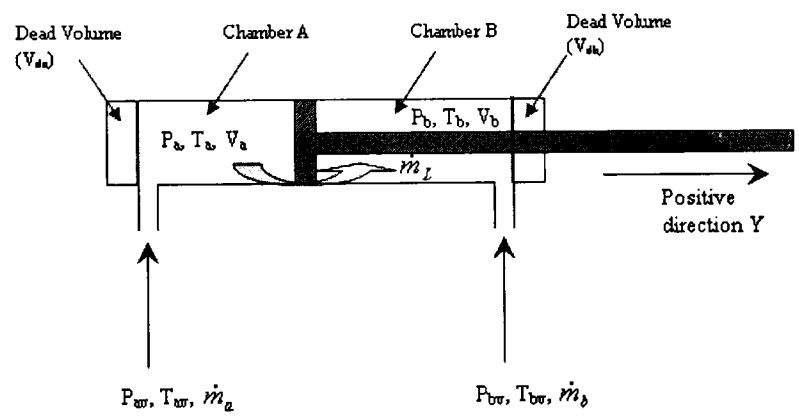


Figure 5.1 Schematic diagram of cylinder's chambers and inlet connections

Assuming that chamber A is the driving chamber (i.e. P_a>P_b), then the control volumes in the cylinder chamber can be expressed as follows:

$$\dot{m}_a C_p T_{av} - \dot{m}_L C_p T_a - P_a \frac{dV_a}{dt} = \frac{d(C_v \rho_a V_a T_a)}{dt} \tag{5.2.1}$$

The first term indicates in flow of energy from supply. The second term denotes leakage energy from chamber A to chamber B. The work is represented by the third term and the right hand side represents the internal energy.

The ***internal leakage*** represents a small amount of air flow from high pressure chamber to the low pressure chamber of the cylinder through any gap between the piston seal and cylinder. Generally the leakage is ignored due to the complexity that it brings to the

model. In this research, the component-based approach considers the internal leakage for more accurate and reliable model. Laminar flow model has been considered for the internal leakage. The equation of the leakage flow from chamber A (in the case of chamber A is the driving chamber) can be expressed as

$$\begin{aligned} \dot{m}_{La} &= K_L(P_a - P_b) \\ &= \dot{m}_L \end{aligned} \tag{5.2.2}$$

where K_L is the coefficient of mass flow leakage which represents a lumped constant for value of viscosity, leakage gap size, and passage length.

Similarly if chamber B is the driving chamber, then

$$\begin{aligned} \dot{m}_{Lb} &= K_L(P_b - P_a) \\ &= -\dot{m}_L \end{aligned} \tag{5.2.3}$$

Back to the main equation (5.2.1), if the air flows out of the cylinder chamber, the energy source of the first term at left hand side is changed. The equation becomes

$$\dot{m}_a C_p T_a - \dot{m}_L C_p T_a - P_a \frac{dV_a}{dt} = \frac{d(C_v \rho_a V_a T_a)}{dt} \tag{5.2.4}$$

However, based on assumption (2), $T_a = T_{av} = T_s$, and $T_b = T_{bv} = T_s$, equations 5.2.1 and 5.2.4 become the same and can be expressed as follows:

$$\dot{m}_a C_p T_s - \dot{m}_L C_p T_s - P_a \frac{dV_a}{dt} = \frac{d(C_v \rho_a V_a T_s)}{dt} \tag{5.2.5}$$

Assuming an ideal gas ($P = R\rho T$ and $P/R = \rho T$), then the right hand side can be simplified as

$$C_v \rho_a V_a T_s = \frac{C_v}{R} P_a V_a \tag{5.2.6}$$

For ideal gas:

$$r = \frac{C_p}{C_v}, \text{ and } \frac{C_p}{r} = \frac{R}{r-1} \tag{5.2.7}$$

Therefore equation 5.2.6 becomes

$$\frac{C_v}{R} P_a V_a = \frac{P_a V_a}{r-1} \tag{5.2.8}$$

Equation 5.2.5 becomes

$$\dot{m}_a C_p T_s - \dot{m}_L C_p T_s - P_a \frac{dV_a}{dt} = \frac{d}{dt}\left(\frac{P_a V_a}{r-1}\right) \tag{5.2.9}$$

Manipulating and rearranging terms, the equation becomes

$$\dot{m}_a C_p (r-1) T_s - \dot{m}_L C_p (r-1) T_s - (r-1) P_a \dot{V}_a = \dot{P}_a V_a + P_a \dot{V}_a$$

$$\dot{m}_a r R T_s - \dot{m}_L r R T_s - r P_a \dot{V}_a = \dot{P}_a V_a \tag{5.2.10}$$

Similarly, if chamber B is the driving chamber (i.e. $P_b > P_a$), then

$$\dot{m}_b r R T_s + \dot{m}_L r R T_s - r P_b \dot{V}_b = \dot{P}_b V_b \tag{5.2.11}$$

There are two unknowns (P_a, P_b) in these equations (5.2.8, 5.2.9). Rearrange for P_a and P_b:

$$P_a = \int \frac{(\dot{m}_a + \dot{m}_L) rRT_s - rP_a\dot{V}_a}{V_a} \tag{5.2.12}$$

$$P_b = \int \frac{(\dot{m}_b - \dot{m}_L) rRT_s - rP_b\dot{V}_b}{V_b} \tag{5.2.13}$$

The volumes V_a and V_b are calculated from the cylinder geometry:

$$V_a = A_a Y + V_{da} \tag{5.2.14}$$

$$V_b = A_b (L - Y) + V_{db} \tag{5.2.15}$$

where V_{da} and V_{db} are small constants which is added because of the dead or residual volume (i.e. volume unaffected by piston movement) in the cylinder. The rate of change of volumes $\dot{V}_a$ and $\dot{V}_b$ are related to the velocity of the actuator $\dot{Y}$:

$$\dot{V}_a = A_a \dot{Y} \tag{5.2.16}$$

$$\dot{V}_a = -A_b \dot{Y} \tag{5.2.17}$$

Substituting equations 5.2.14 and 5.2.15 in 5.2.10 and 5.2.11 respectively results in

$$P_a = \int \frac{(\dot{m}_a + \dot{m}_L) rRT_a - rA_a P_a \dot{Y}}{A_a Y} \tag{5.2.18}$$

$$P_b = \int \frac{(\dot{m}_b - \dot{m}_L) rRT_b + rA_b P_b \dot{Y}}{A_b (L - Y)} \tag{5.2.19}$$

5.3 Load Module

The load module describes the behaviour of the drive mechanism under the influence of the actuator force. The total mass of the load is accelerated by the net force after taking out the net friction force and any opposing weight. The velocity and the position can be

determined by integrating the acceleration. The following equation describes the dynamic behaviour of the load by applying Newton's second law to the actuation system.

$$(A_a P_a - A_b P_b) - (A_a - A_b)P_{atm} - F_l - F_f sign(\dot{Y}) - c_f \dot{Y} = M\ddot{Y} \qquad (5.3.1)$$

The first and second terms represents the net force applied from the actuator. F_l represents the total load system weight. However for the analysis cases presented in this research, the cylinder is placed horizontally and hence F_l is zero.

$c_f\dot{Y}$ represents the viscous friction due to lubricating oil. The value of c_f is estimated based on the published viscosity data of the oil or using the proposed environment that developed in this research **(see Chapter 6 and 7)**.

For the pneumatic cylinder considered in this research, the friction model is simplified to obtain the more commonly used threshold model. A graphical description of the friction model used in this research is depicted in Figure 5.2. Equation (5.3.2) describes the adopted model.

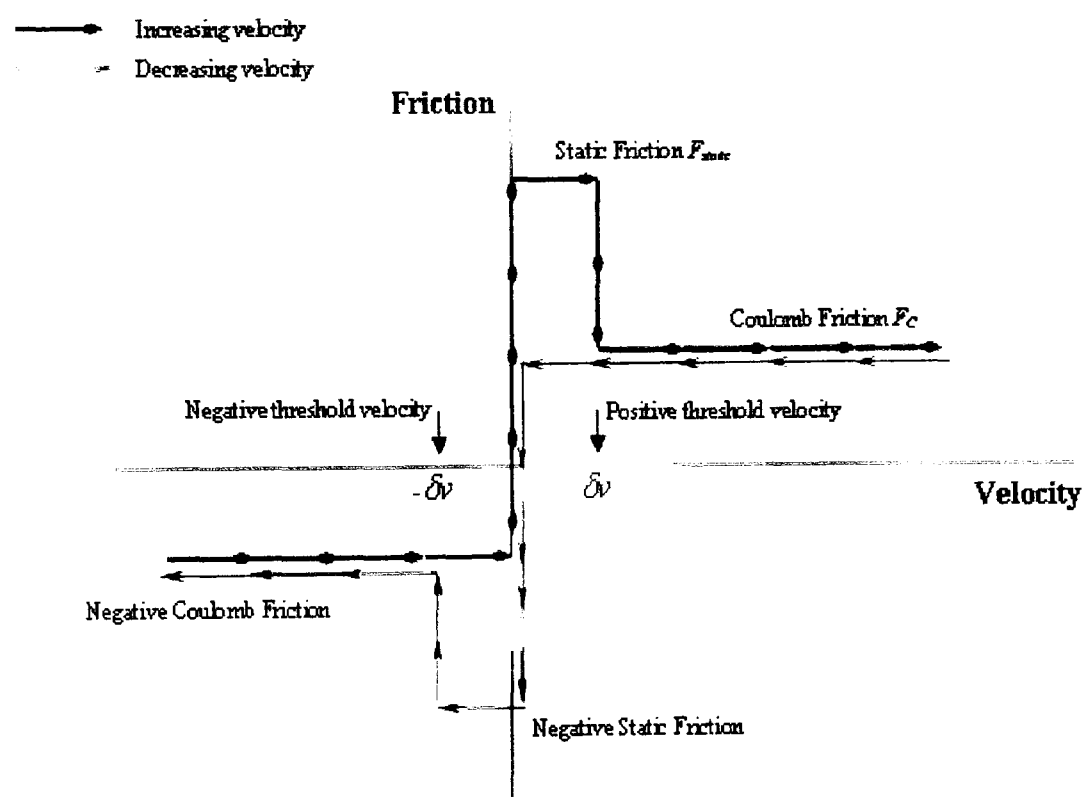


Figure 5.2 Static and dynamic friction model in increasing and decreasing velocity

$$F_f = \begin{cases} F_C & |\dot{Y}| > \delta v \\ F_{static} & |\dot{Y}| \le \delta v \end{cases} \tag{5.3.2}$$

However, it has been realized through the friction tests on the test bed that the frictional parameters during actuator extension (i.e. chamber A is the driving chamber) are larger than that of the retraction. This is mainly due to the design of the piston seal which adheres more tightly to the cylinder bore during extension. The phenomenon requires a modification of Coulomb's model to allow for a different set of friction parameters at positive (extend) and negative (retract) actuator velocity. ***Therefore, to avoid complexity in results analysis and to avoid duplication, all experiments have been performed in one direction (retraction).***

5.4 Valve Module

The valve consists of two parts. The valve body comprising of the spool and the sleeve, and an embedded circuit, known as valve operator, controls the position of the spool. The valve module provides the mass flow rates for the actuator module, and the valve operator receives signal command information from the controller and produces spool displacement accordingly. Figure 5.3 shows the flow paths in a 5-ports proportional valve.

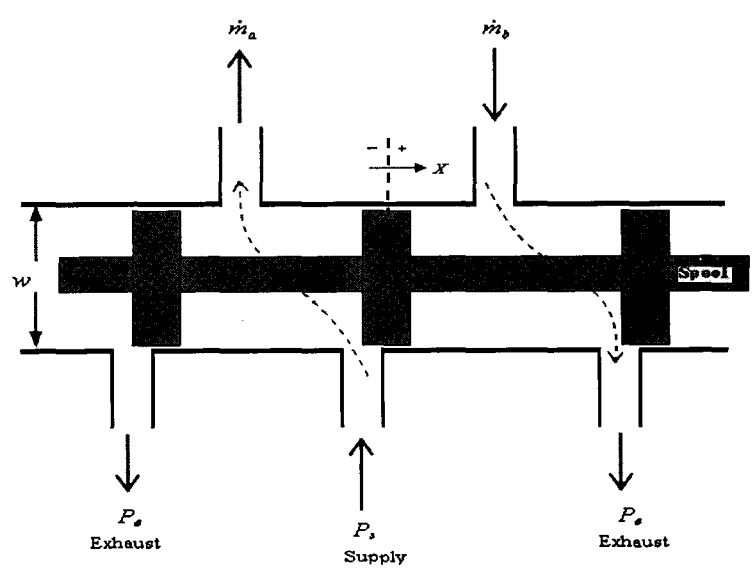


Figure 5.3 Flow path diagram for 5-ports proportional valve

The assumptions that considered in the valve model analysis are as following: (1) each port of the valve is assumed to act like a standard orifice, so that standard orifice theory can be applied, and (2) control valve constants C_d and C_o do vary with valve opening.

According to the standard orifice theory (Shearer, 1956; Blackburn, et al., 1960; Anderson, 1976), the mass flow rates across the two control ports of the control valve can be regarded as a function of the valve spool displacement and the chamber pressures, which can be expressed as

$$\dot{m}_a = C_d C_o w X f_{pa}(P_r) \tag{5.4.1}$$

$$\dot{m}_b = -C_d C_o w X f_{pb}(P_r) \tag{5.4.2}$$

For air $C_d = 0.8$.

C_o can be expressed as following:

$$C_o = \sqrt{\frac{r}{R(\frac{r+1}{2})^{\frac{r+1}{r-1}}}} \tag{5.4.3}$$

where r= 1.4, R=280, then C_o =0.04.

The following formulae are employed to calculate the mass flow rate function $f_{pa,pb}(P_r)$

$$f_{pa,pb}(P_r) = \frac{P_u}{\sqrt{T_u}} \tilde{f}(P_r) \tag{5.4.4}$$

where:

$$\tilde{f}(P_r) = \begin{cases} 1, & \frac{P_{atm}}{P_u} < P_r \le C_r \\ C_k[P_r^{2/r} - P_r^{(r+1)/r}]^{\frac{1}{2}} & C_r < P_r < 1 \end{cases} \tag{5.4.5}$$

where

$$C_k = \sqrt{\frac{2}{r-1}\left(\frac{r+1}{2}\right)^{\frac{r+1}{r-1}}} \tag{5.4.6}$$

for air $C_k = 3.864$

$$C_r = \left(\frac{2}{r+1}\right)^{\frac{r}{r-1}} \tag{5.4.7}$$

C_r is the critical value of the pressure ratio that determines the increase of the mass flow rate through the orifice (here C_r=0.526). The mass flow function for chamber A and chamber B can be expressed as following

$$f_{pa}(P_r) = \begin{cases} \frac{P_s}{\sqrt{T_s}}\tilde{f}(\frac{P_a}{P_s}) & when \quad chamberA \quad is \quad a \quad drive \quad chamber \\ \frac{P_a}{\sqrt{T_a}}\tilde{f}(\frac{P_e}{P_a}) & when \quad chamberA \quad is \quad a \quad exhaust \quad chamber \end{cases} \tag{5.4.8}$$

and

$$f_{pb}(P_r) = \begin{cases} \frac{P_s}{\sqrt{T_s}}\tilde{f}(\frac{P_b}{P_s}) & when \quad chamberB \quad is \quad a \quad drive \quad chamber \\ \frac{P_b}{\sqrt{T_b}}\tilde{f}(\frac{P_e}{P_b}) & when \quad chamberB \quad is \quad a \quad exhaust \quad chamber \end{cases} \tag{5.4.9}$$

It has been observed that if air is supplied at constant inlet conditions and the exhaust pressure is decreased gradually, the mass flow rate of air increases until the pressure ratio (P_u/P_d) reaches the critical value of 1.89. Thereafter, further decrease in the exhaust pressure has no effect on the flow, and the flow is said to be "chocked" (Figure 5.4).

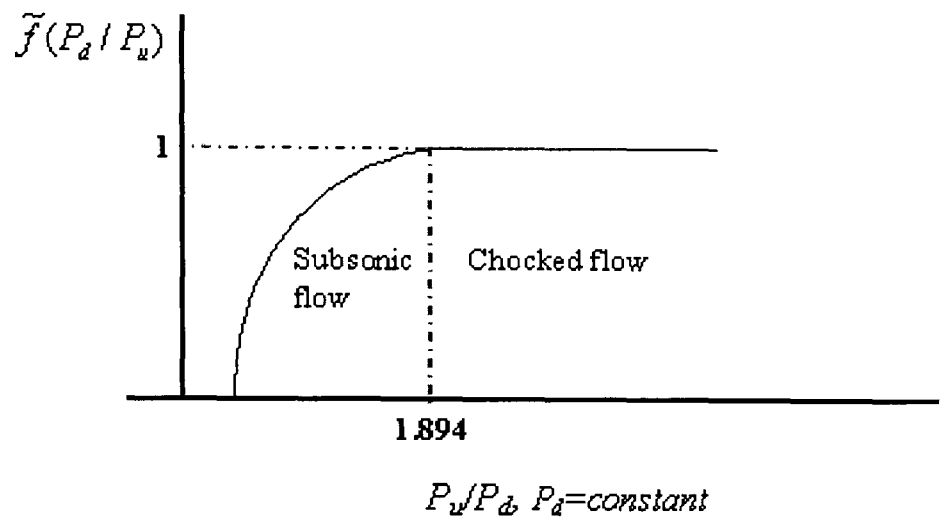


Figure 5.4 Mass flow function $\tilde{f}(P_d / P_u)$

The amount of fluid flowing through the spool valve depends on the flow passage area. The passage area is directly related to the control signal. It has been observed through the experiments that the relation between the control signal and the spool displacement is not linear (Figure 5.5). Therefore, an empirical model has been created for this relationship as shown in equation (5.4.10).

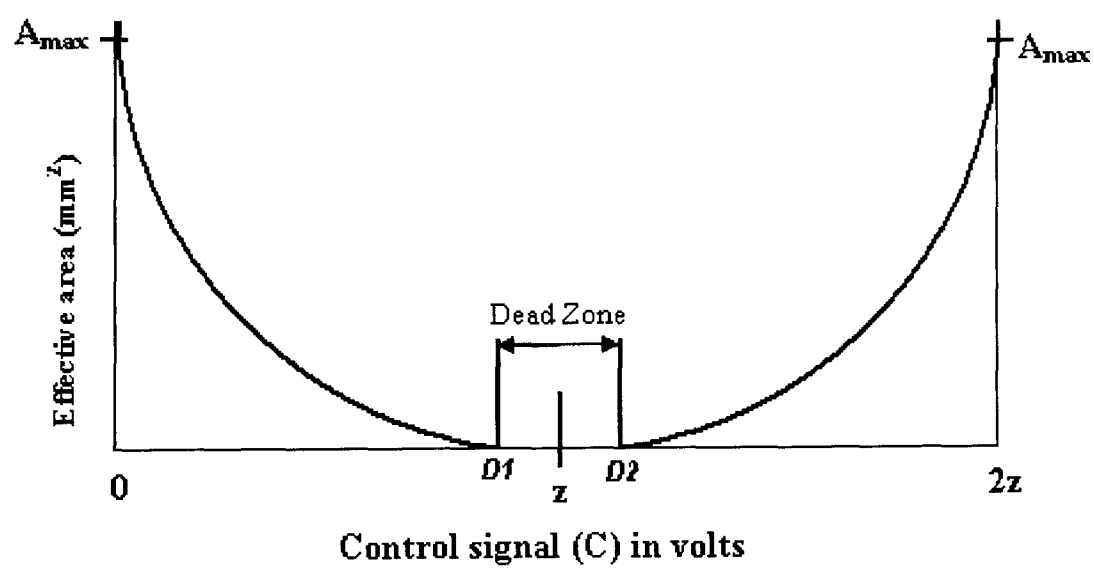


Figure 5.5 Empirical model for the effective area/control signal relationship.

$$A_{eff}(C) = \begin{cases} \dfrac{|C-z|^{K_1} - K_2}{z^{K_1} - K_2} \times A_{\max} & D1 \geq C \geq D2 \\ 0 & D1 \leq C \leq D2 \end{cases} \tag{5.4.10}$$

where K_1 and K_2 are coefficients to be determined experimentally. z is the voltage where the valve should be off. K_1 is related to the curve concavity and K_2 to the width of the dead zone. K_1 and K_2 are different from valve to another depending on the manufacturing materials and precision. According to the empirical equation the maximum effective area is achieved at 0 and max voltage. This model has been verified for the valve used in the test bed.

5.5 Tubes Module

Significant pressure drop can be appeared in the system's tubes, in which many factors would contribute the effect. The major one is the isentropic expansion of fluid in the tubes. The pressure drop is varying according to a span ratio L/D where L is the pipe length and D is the pipe inner diameter (Mo, 1989). The model, which is adopted to represent the pressure drop, is as follow:

$$P_{drop} = \frac{\lambda_L \rho L_p v^3}{2D} \tag{5.5.1}$$

λ_L is depending on the material of which the pipe is manufactured, and the air velocity can be measured through experiments.

5.6 Summary and Discussion

A mathematical model for an individual component within the pneumatic system model structure has been established in this chapter. The main components that invent the dynamic behaviour of the servo-pneumatic system are the actuator, the load, the valve, and the tubes. The actuator module is formulated by considering the air flow into the two chambers. The asymmetric cylinder is the general case from which the model equations

(5.2.18, 5.2.19) are derived. Symmetric cylinder is the special case by setting the bore diameter to zero (i.e. A_a=A_b).

The load module is created by applying Newton's second law to the actuation system. The nonlinearity associated with the friction behaviour acquired an empirical method to determine the frictional parameters which is introduced in **chapter 6**.

The valve module has been established based on the standard orifice theory which considers the mass flow rate across the two control ports of the control valve. For the flow passage area, an empirical equation is created to describe the relationship between the control signal and the flow passage area. This equation contains two parameters, each of which should be determined through experiments. These parameters vary from valve to another depending on the dead area of the spool and the linearity of its behaviour. Finally, a simplified model is established for the pressure drop in the tubes. These components' models will be implemented in software to develop the pneumatic simulation toolbox and the proposed environment.

Chapter 6
Component-based Mixed Reality Environment Development

6.1 Introduction

The tools that are currently available in the market for the design, simulation and control of pneumatic drive systems, are mainly developed for conventional pneumatic circuit designs (Famic, 2005; Festo, 2005). Perhaps largely due to the fact that pneumatic drives are still not widely utilised in industrial applications because of their high nonlinearity. This implies the need for simulation tools and environments for the design and control of pneumatic servos. It would be more advantageous if the simulation environment could be run in time domain so that it can be mixed with the real system. This would make the simulation more accurate and reliable especially when dealing with such nonlinear systems.

This chapter presents the development process of the Component-based Mixed Reality Environment (CMRE). Working with the CMRE would include both the capability to model the system entirely as a simulation, the so-called "off-line", as well as being able to use real components running against simulations of others "on-line", or in a Mixed Reality (MR) manner. Therefore this chapter presents two stages of developments: (1) the development of the Component-based Simulation Toolbox (CST) for off-line simulation, and (2) the development of the CMRE using the CST for on-line applications.

The design and development of the CST is introduced in section 6.2. Section 6.3 presents the CMRE and the proposed method for frictional parameters identification. Summary and discussion are presented in section 6.4.

6.2 Design and Development of the CST

The MATLAB/Simulink software was chosen to act as the base-platform to implement the proposed CST which adopts components-based paradigm. The CST is part of the proposed CMRE in which the individual components are implemented as 'blocks' in a components library within the Simulink. Defining a servo-pneumatic system becomes much like drawing a block diagram.

The pneumatic components are organised into six major component categories. Each component category consists of a collection of components bearing the same linkage specification in the component interfaces. A system would be formed by selecting individual components from different categories and connecting them through their external interfaces.

Thus the CST allows building different pneumatic systems by simply selecting and integrating a set of **off-the-shelf** components. Moreover, it allows the addition and extension of any model irrespective of its sophistication requirements. The advantage is that some components can be modelled as closely as possible to the physical component (e.g. the most critical components), while others can be simplified to increase the modelling efficiency. Furthermore, empirical component models can be mixed with theoretical component models to form an integrated system model.

Figure 6.1 illustrates the basic steps in the design of CST. These steps are only dedicated to pneumatic systems and it may or may not be applicable for different types of dynamic systems.

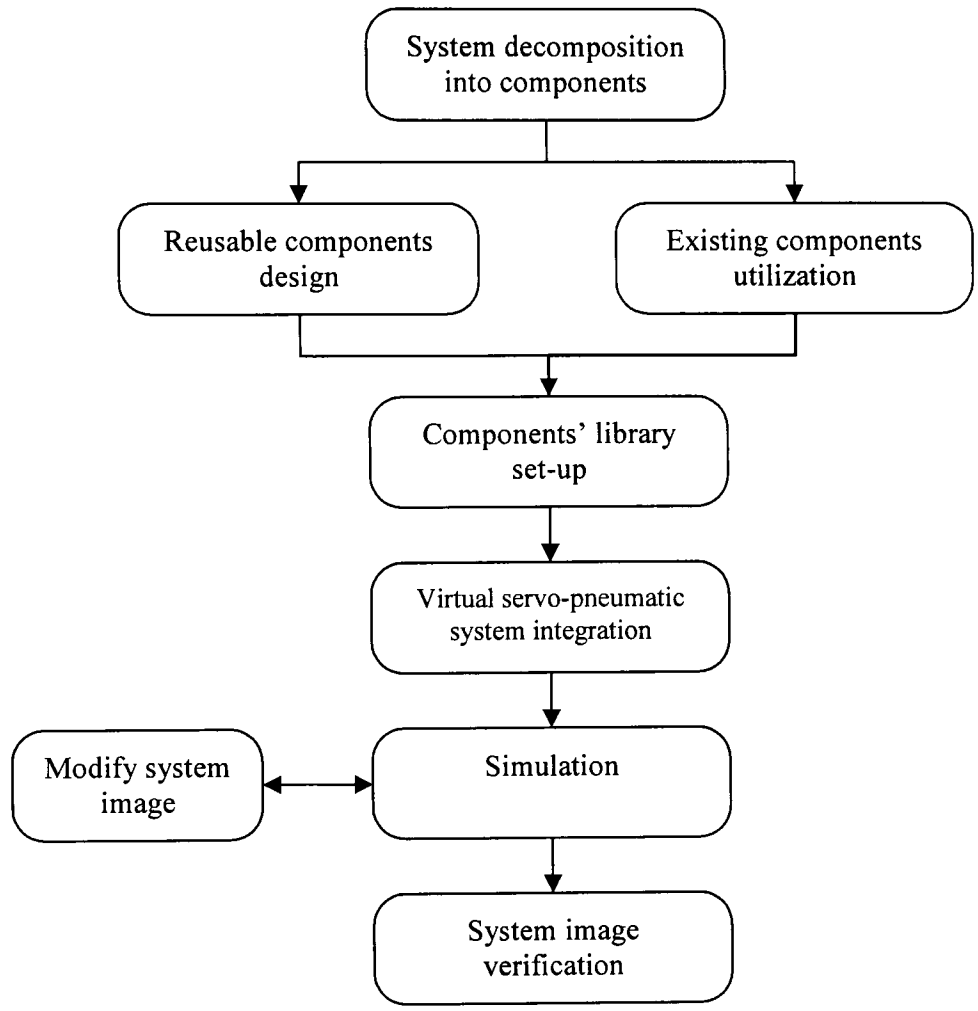


Figure 6.1 Basic steps for the design of the CST

The following sub-sections present only the first three stages of the design and development of the CRT, which are: (1) decomposing the system into components, (2) reusable components design, and (3) components library set-up. The virtual servo-pneumatic system integration, simulation, and system image verification will be presented in the validation and evaluation chapter.

6.2.1 Decomposing the System into Components

The philosophy of component-based approach demands the decomposition of the system into components. The smaller the components are decomposed, the more flexible is the resulting pneumatic system. This step is important to determine which components that have to be designed and build, and which components that can be used from the

platform's library. The system decomposition has been performed in chapter 4 by adopting the component-based framework.

6.2.2 Reusable Components Design

After decomposing the system into components, some components, which are not available in the platform's library, should be designed and built. In the context of servo-pneumatic system, the virtual components that had to be designed are pneumatic actuator, load, pneumatic valve, controller, tubes, DC-DC converter and profile generator. Some sinks, sources and scopes have been used from the Simulink library.

6.2.2.1 "Actuator-load" Component

The "actuator-load" component can be considered as a compound component comprises of two main sub-components; the actuator and the load. The inputs to the component are the mass flow rates ($\dot{m}_a, \dot{m}_b$) and the supply pressure (P_s). The outputs are the pressure in both chambers (P_a, P_b), the load acceleration, speed, and position.

6.2.2.1.1 Pneumatic Actuator

According to the mathematical model established in chapter 5 for the pneumatic actuator and based on the equations (5.2.18, 5.2.19), the actuator image can be devised as shown in Figure 6.2. The input variables to the actuator are identified as mass flow rates ($\dot{m}_a, \dot{m}_b$) which are fed-forward from the valve component, displacement Y, and velocity $\dot{Y}$ which are fed-back from the load sub-component, and the output is the pressure in both chambers (P_a, P_b). In case of symmetrical cylinder, the areas A_a and A_b are equal by setting the bore diameter to zero.

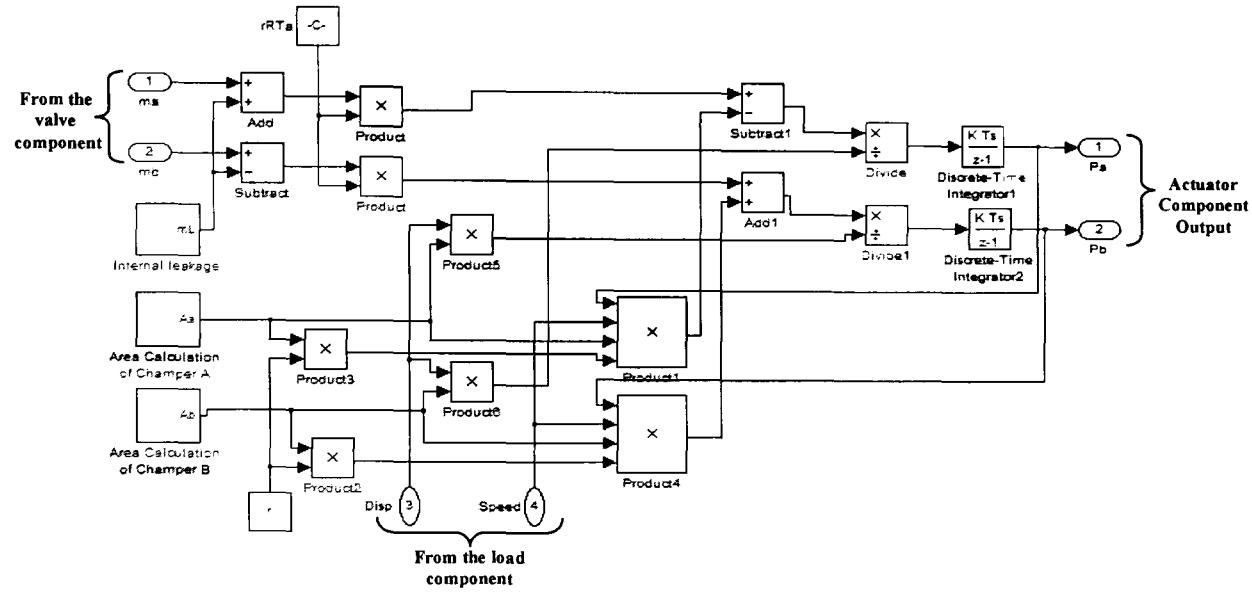


Figure 6.2 Actuator model block diagram

6.2.2.1.2 The Load

The load component has three inputs and three outputs according to the equation (5.3.1) which describes the dynamic behaviour. The inputs are the pressure in both chambers (P_a, P_b), which are fed-forward from the actuator sub-component. The outputs are the acceleration, velocity, and position of the load. Figure 6.3 shows the block diagram of the load image.

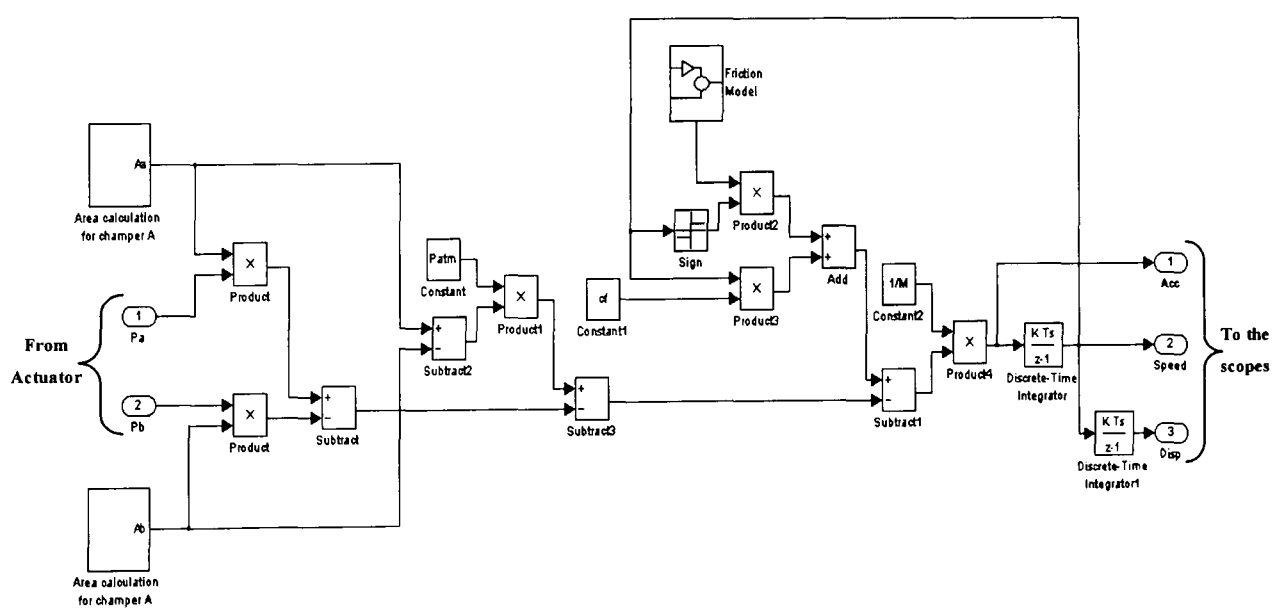


Figure 6.3 Load model block diagram

The "actuator-load" has some parameters which have to be set by the user according to the system design and the hardware specifications. For example, if the user double clicks on the "actuator-load" component, a user dialog window will appear as shown in Figure 6.4 (a). The user is required to set the cylinder's bore diameter, stroke length, rode diameter, and internal leakage coefficient according to the specifications of the cylinder used in the real system. The user also has to set the initial conditions and the friction parameters of the system. This flexibility allows for simulating a variety of cylinder types with different sizes and specifications without the need to change the whole system's model.

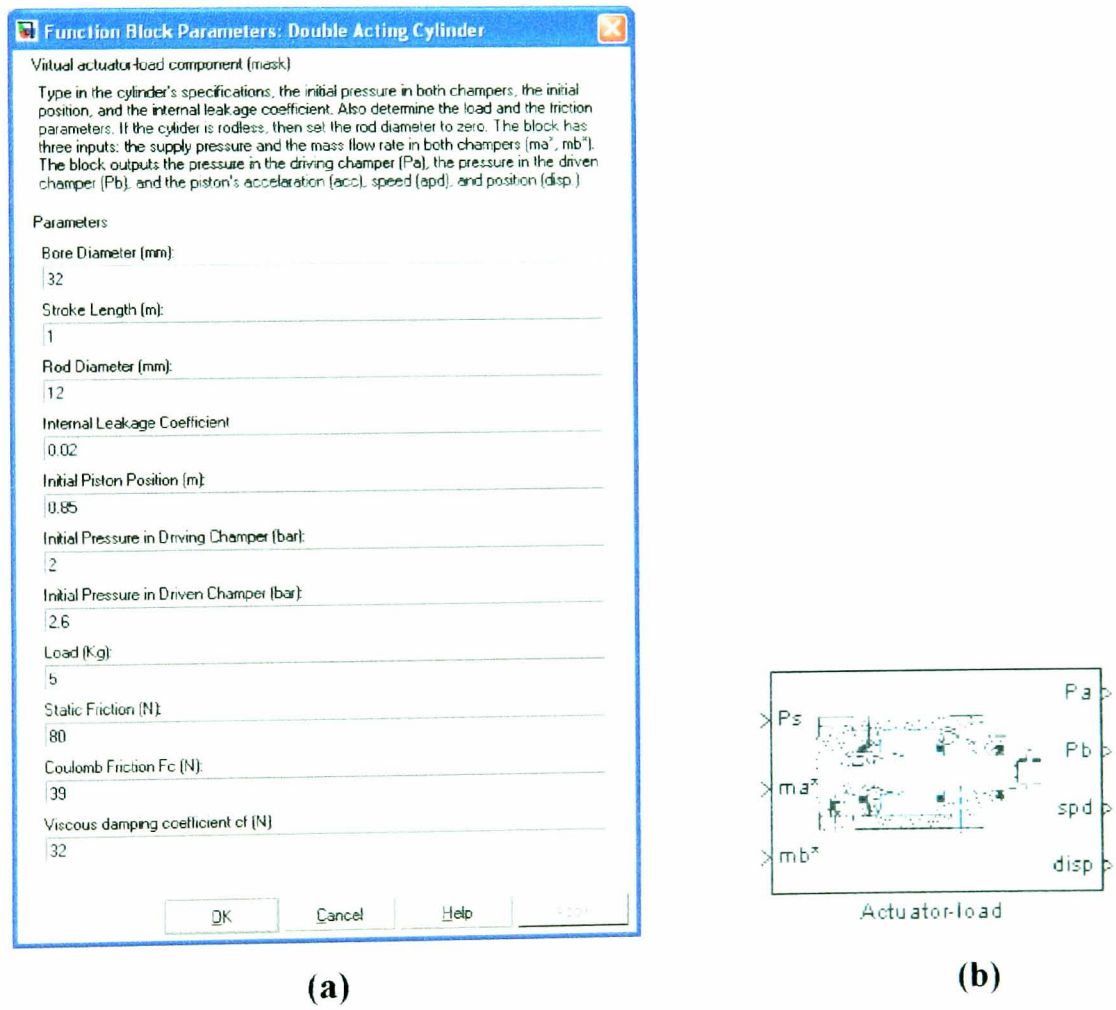


Figure 6.4 (a) User dialog for the virtual actuator-load component; (b) The component's I/O pins.

6.2.2.2 Pneumatic Valve Image

As described in chapter 5, the valve consists of two parts: the valve body comprising of the spool and the sleeve, and the valve operator, which controls the position of the spool. Therefore, the valve component image includes two sub-components: (1) the first one is designed to calculate the mass flow rate function and (2) the second implements the valve operator which involves with the control signal and the spool displacement.

6.2.2.2.1 Mass Flow Rate Sub-component

The mass flow rate sub-component is designed based on the equations (5.4.4, 5.4.5, 5.4.8, 5.4.9) that were derived in chapter 5. The sub-component consists of two subsystems; one of them is employed for the positive direction movement (extension) and the other for the negative direction (retraction). The inputs are the pressure in both chambers (P_a, P_b) provided from the actuator-load component, the supply pressure (P_s) provided from the air compressor component, and the effective area provided from the valve operator sub-component. The outputs are the mass flow rate in both chambers ($\dot{m}_a, \dot{m}_b$). Figure 6.5 depicts the block diagram of the mass flow function component in the positive direction.

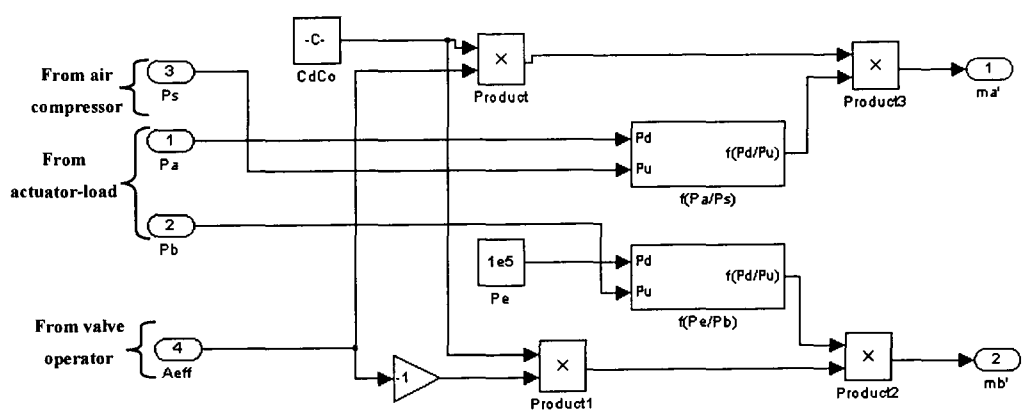
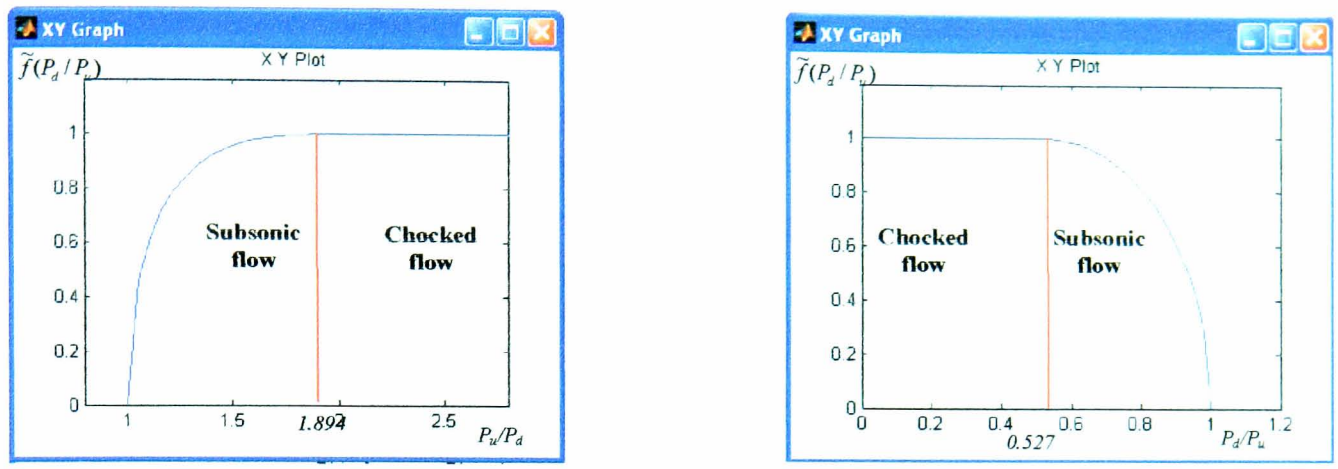


Figure 6.5 Mass flow function block diagram

To verify the designed sub-component, the "chock flow" was tested and the results were as following:

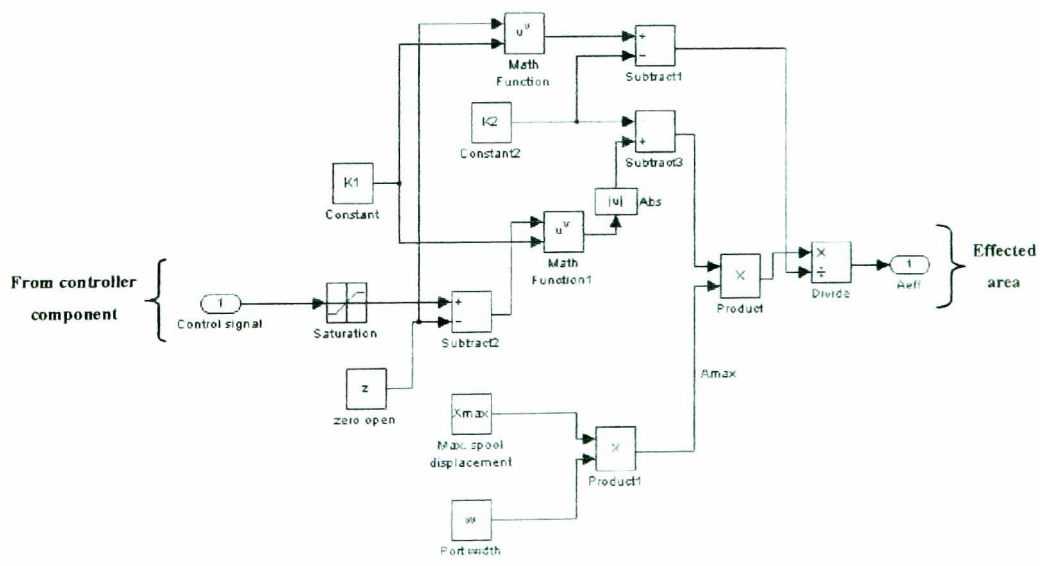


(a) **(b)**

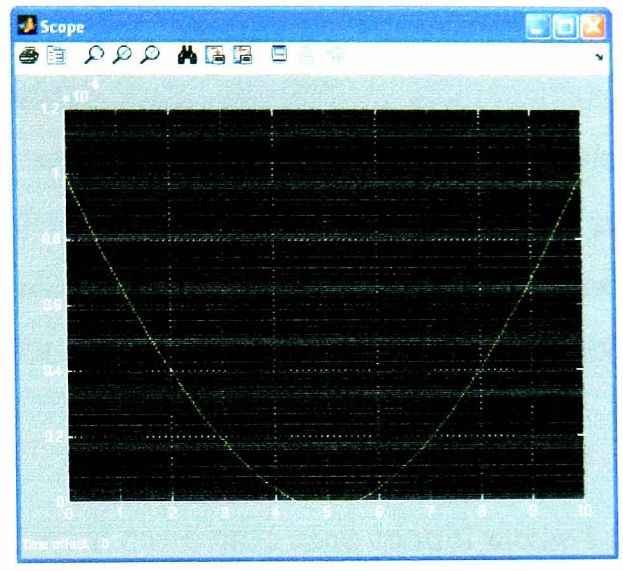
Figure 6.6 (a) Mass flow function with P_d is constant; (b) Mass flow function with P_u is constant

6.2.2.2.2 Valve Operator Sub-component

The main function of this sub-component is to convert the control signal provided from the controller into a spool displacement then calculates the effective area of the flow passage. The design of this sub-component is based on the empirical equation (5.4.10) which has been established in chapter 5. The user should determine the empirical coefficients K_1 and K_2 based on the valve that has been used in the real system. Figure 6.7 shows the block diagram of the component and the output over a range of 0-10V input voltage.



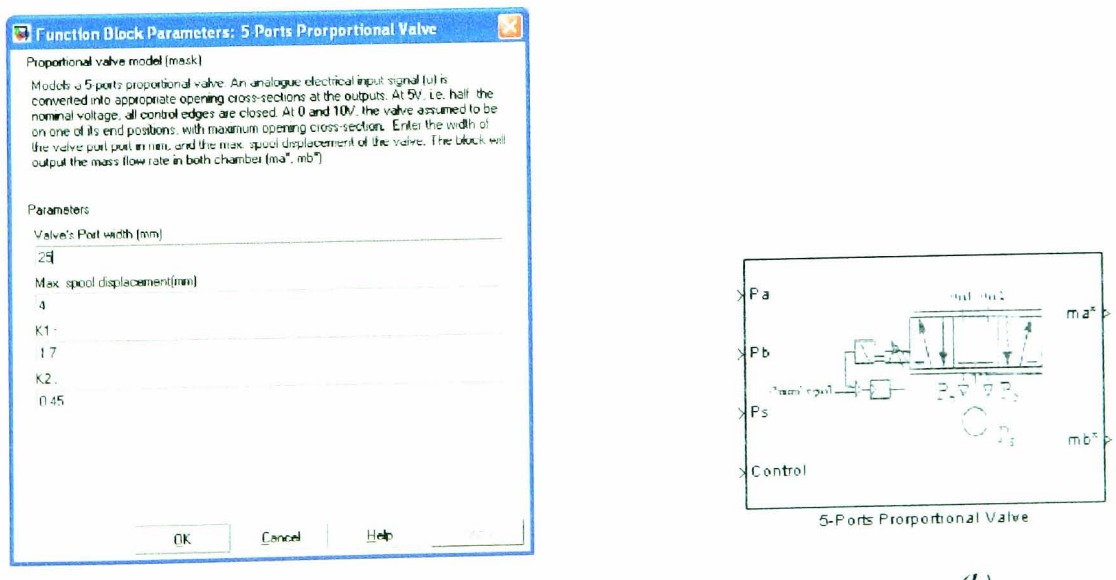
(a)



(b)

Figure 6.7 (a) Valve operator model block diagram; (b) The effective area over 0-10V input with K_1=1.73, K_2=0.45, max. spool displacement=4 mm, Port width=25 mm

The mass flow rate and the valve operator sub-components were compounded to form the Valve component. This will simplify the design of servo-pneumatic system especially in the case where more than one valve is needed. The Valve component has some parameters which have to be set by the user according to the specifications and dimensions of the real valve. These parameters are the maximum spool displacement, the port width, and the empirical parameters K_1 and K_2. Figure 6.8 illustrates the valve's virtual component and its user dialog window.



(a) *(b)*

Figure 6.8 (a) User dialog for the valve component; (b) The component's I/O pins.

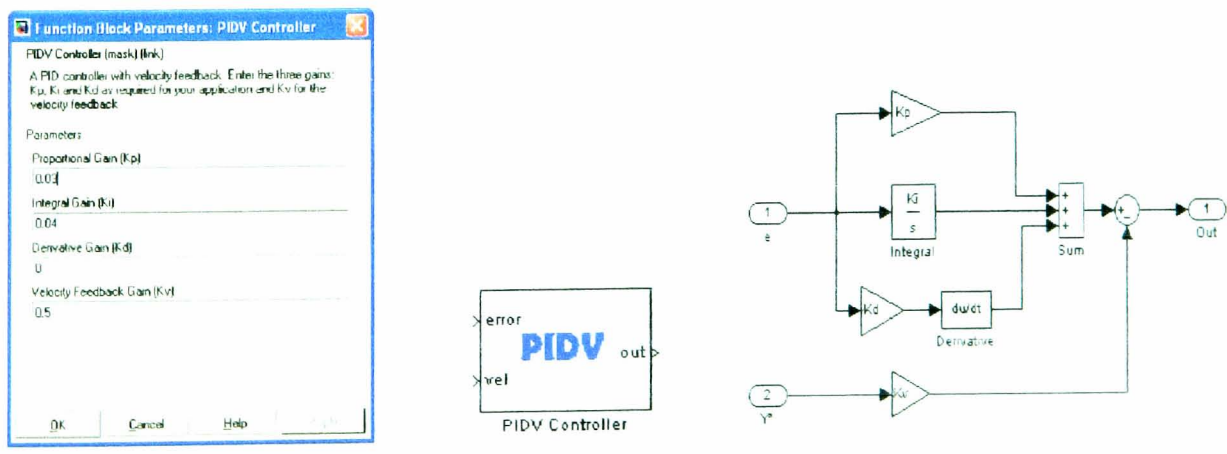
6.2.2.3 Controller Image

In the scope of this research study, the PID-based controller has been chosen as the controller module. However, all other control methodology explored in chapter 3, such as fuzzy controllers, neural network, and adaptive controllers can also be implemented within the simulation environment. This is one of the component-based approach advantages, where the user can implement several control methodologies according to the application.

PID-based control is the most commonly used control method for servo control. This is because it is simple and commercially available. The PID-based controller can be any combination of the *PIDV* controller e.g. *P, PI, PD, PID,* or *PIDV*. All gain values can be accessed by the user through the user dialog of the controller. PIDV control algorithm can be implemented in different forms. The most commonly used PIDV algorithm is regarded as ideal parallel PID, because it can be conveniently implemented in software and leads to separating out the effect of the individual control gains. The PIDV algorithm analyzed in this research is of the following form:

$$V_{out} = K_p e_i + \int e_i + K_d(e_i - e_{i-1}) - K_v \dot{Y} \qquad (6.1)$$

where i = instance of time interval. Figure 6.9 shows the block diagram and the user dialog of the virtual controller component.



(a) *(b)* *(c)*

Figure 6.9 (a) User dialog for the controller component; (b) The component's I/O pins; (c) The controller model block diagram.

6.2.2.4 Tubes Image

The tubes play an important role in the servo-pneumatic dynamic behaviour due to the pressure drop which is, in some cases, not negligible. Therefore it has been considered in the system model structure presented in chapter 4. The mathematical model of the tubes has also been established in chapter 5. According to equation 5.4.10 the block diagram and the user dialog of the virtual tubes component is depicted in Figure 6.10.

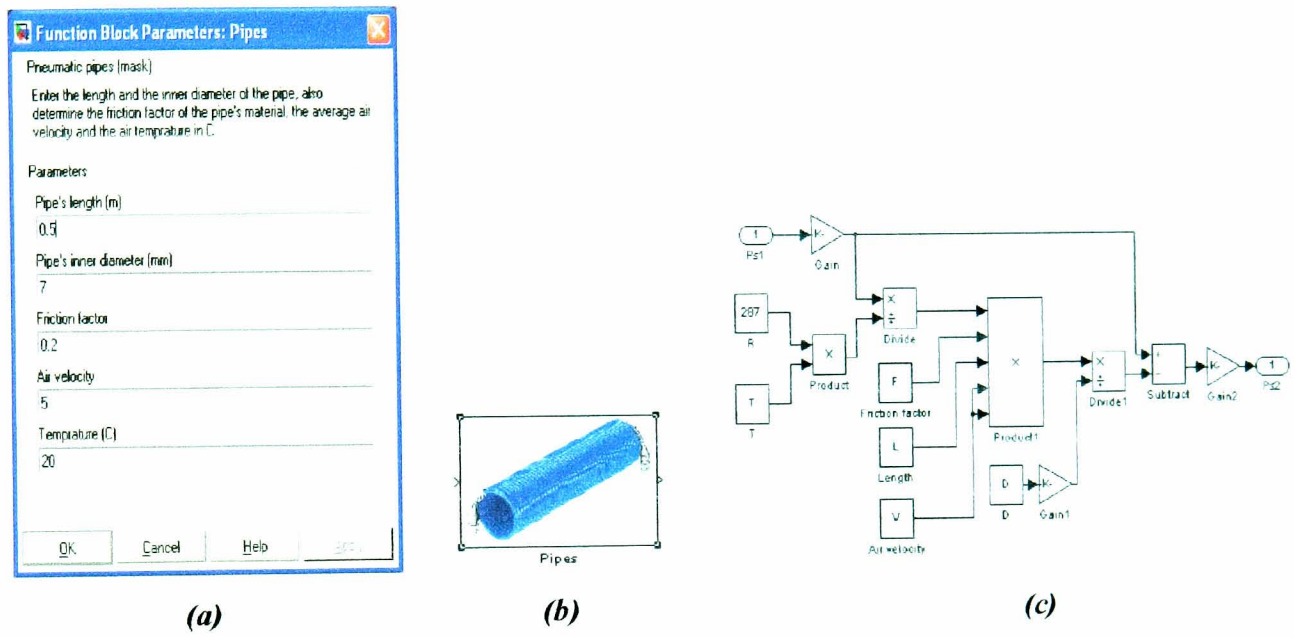


Figure 6.10 (a) User dialog for the tube component; (b) The component's I/O pins; (b) The tube model block diagram.

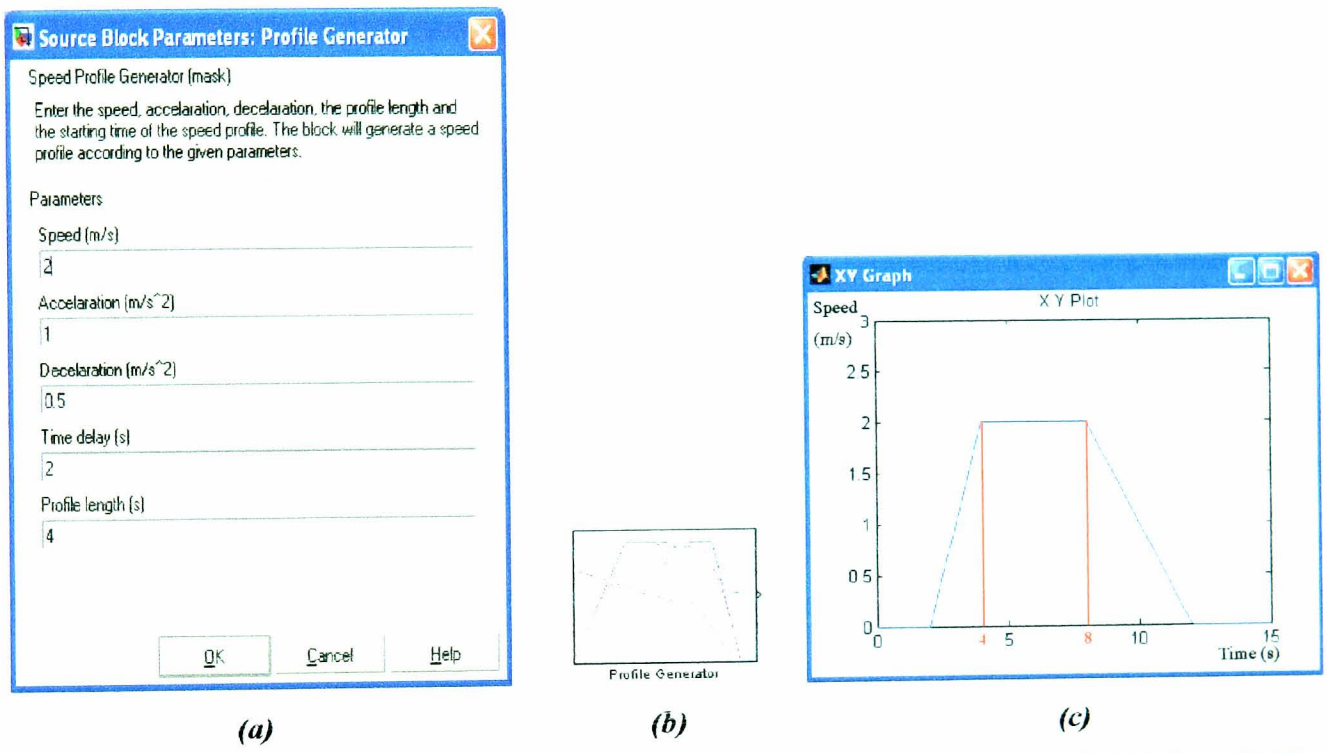
6.2.2.5 Miscellaneous Components

The servo-pneumatic system contains many components, other than the pneumatic components, which are necessary to complete the servo loop. Some of these components have been designed and implemented, others have been used from the existing components in the Simulink library. The following subsections explore the developed components which are profile generator and DC-DC converter.

6.3.2.5.1 Profile Generator

Many industrial applications involve profile following to accomplish some specific tasks. Servo-pneumatic drives have the potential to provide relatively low cost/high

performance actuation system. Therefore it is essential to have a profile generator component in order to facilitate the research on the profile following capability of servo pneumatic system. The profile generator component has been designed so that the user can set the speed, acceleration, deceleration, time delay and profile length. This flexibility will allow the user to generate different speed profiles according to the application in order to evaluate the profile following capability of the system. Figure 6.11 shows the user dialog of the component and an output example.

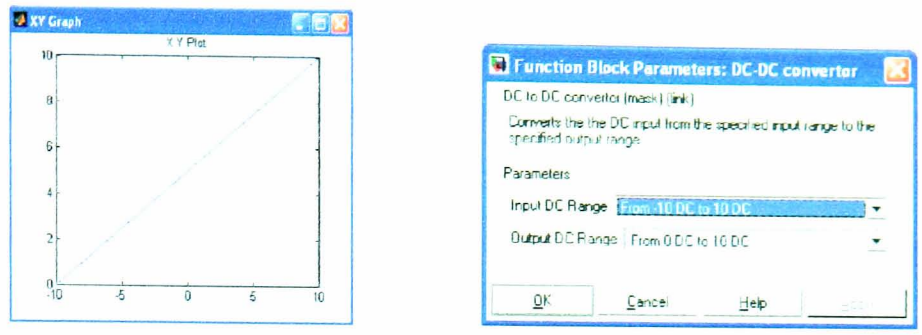


(a) *(b)* *(c)*

Figure 6.11 (a) User dialog for the profile generator component; (b) the component's I/O pins; (b) The speed profile output with speed = 2 m/s, acceleration = 1 m²/s, deceleration = 0.5 m²/s, time delay = 2 s, and profile length = 4 s

6.2.2.5.1 DC-DC Converter

The main task of the controller is to control the valve which controls the flow of the air into the cylinder, by which the load is controlled. However, the analogue output of the controller is a DC output ranged from -10V to 10V. This range of voltage must be converted to match the required input voltage for the valve which is ranged from 0V to 10V DC. A DC/DC converter was built to achieve that. The DC-DC component, its user dialog, and its characteristics graph are shown in Figure 6.12.

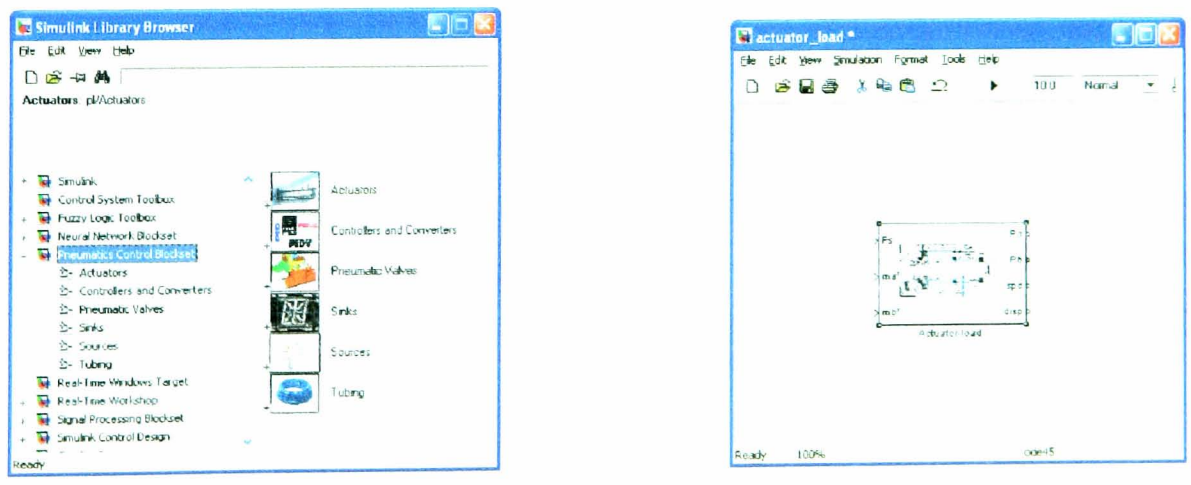


(a) *(b)*

Figure 6.12 (a) Virtual DC-DC converter output characteristics; (b) The component's user dialog.

6.2.3 Components Library Set-up

Once all main components are designed and ready for use, it is required to setup a components library in order to manage the procedure of building and designing different pneumatic systems. Figure 6.13 shows the components library named "*Pneumatics control blockset*". The pneumatic components are organised into six major categories; actuators, controllers and converters, pneumatic valves, sinks, sources and tubing. The component can be used by simply drag it from the component list and drop it at the workspace window. In the component image, there are a number of I/O pins each of which has a name. These names stands for the variable within the component function. For example, the "actuator-load" component has three inputs variables: **ma***, **mb***, and **Ps**, and five outputs: **Pa**, **Pb**, **Acc**, **spd**, and **disp**. The **ma*** stands for mass flow rate in chamber A, **Acc** stands for acceleration, **spd** stands for speed… and so forth. This library is, obviously, expandable and can include additional pneumatic components.



(a) *(b)*

Figure 6.13 (a) The pneumatic control blockset; (b) The simulation workspace.

6.3 Component-based Mixed Reality Environment (CMRE)

6.3.1 The Concept of Mixed Reality Environment

Generally, classical servo-pneumatic system comprises of controller, valve, actuator, and sensors. The controller generates an output according to the feedback signal from the sensors and sends it to the valve that manages the air flow to the actuator.

According to the above situation, the controller can be substituted by its model and simulated in real time. The simulated controller can be run in conjunction with real components under the same environment. This environment can be regarded as Component-based Mixed Reality Environment (CMRE). Figure 6.14 shows the concept of the proposed environment.

The CMRE is an environment whereby virtual controller component can be applied on real plant components. Working with CMRE should include "off-line" or "pure simulation" mode and on-line or Mixed Reality (MR) environment. Off-line simulation, on the one hand, will normally take place before moving onto MR simulation using the CST, where the system should be tested and the controller should be tuned or optimized. On the other hand, the CMRE will be used to apply the optimized controller on the real plant.

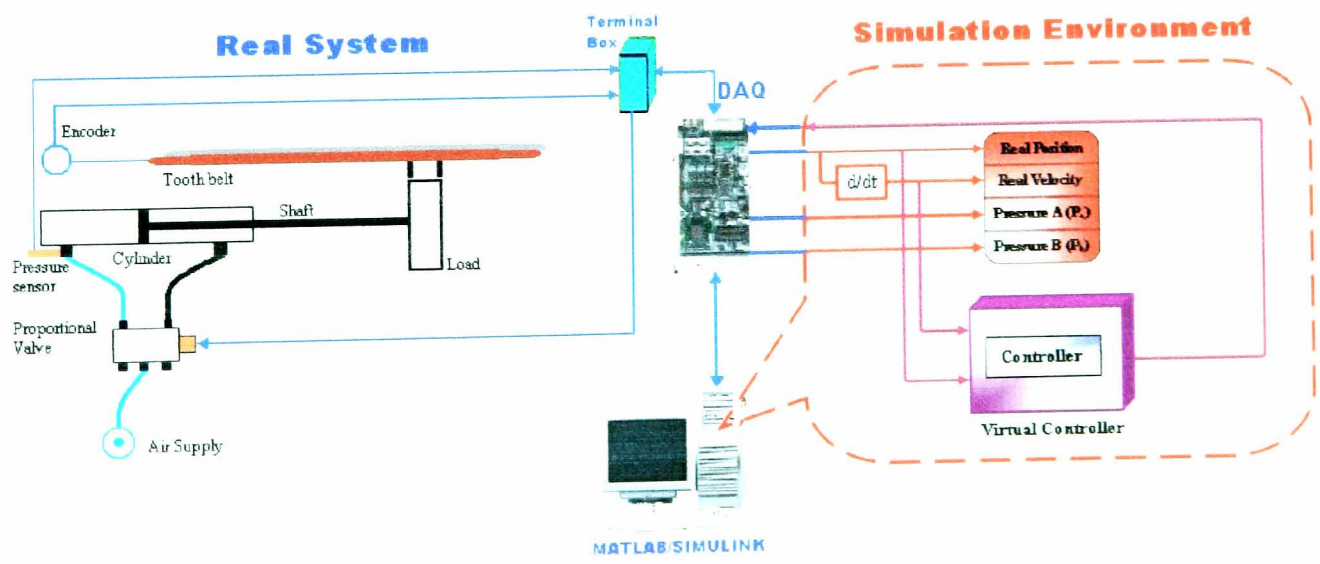


Figure 6.14 CMRE structure for control scheme implementation

This environment should allow the system to be controlled with different control schemes by simply replace the "Controller" component according to the application requirements. Furthermore the CMRE gives the capability to monitor the system's behaviour by observing the output signals such as pressure and position signals. These signals should facilitate the identification of the frictional parameters of the system which is normally difficult to be acquired once the system is assembled. The following sections explain a method for frictional parameters identification in more details.

6.3.2 Frictional Parameters Identification

This section outlines a method to identify the friction parameters for servo-pneumatic systems using the proposed CMRE. To acquire system friction parameters accurately could be extremely difficult once the servo-system has been assembled, which causes a great difficulty in servo-pneumatic system modelling and control. In this research project, the CMRE has been employed to determine these parameters. Therefore, an online estimation for the friction parameters can be conducted effectively and efficiently. This estimation can be performed manually or automatically using one of the traditional optimization method e.g. neural network. The advantages of the proposed environment and method are high accuracy in the estimated parameters, simplicity and less time consuming. An experimental case study has been conducted and the results are presented in chapter 7 to show the accuracy and effectiveness of the proposed environment and methodology.

A rich knowledge about structure and parameters of a servo-pneumatic system would facilitate to obtain an accurate model which will increase the success of the control strategy to be applied. However, when pneumatic drives are employed to achieve servo control, problems such as compressibility of the air, pressure loss, leakage and nonlinear frictional parameters considered to have significant effects on system modelling and control (Pu, et al., 1994). A method or technique is demanded to accurately determine the system's frictional parameters which is considered one of the crucial factors that influence the modelling and control of servo-pneumatic systems.

The conventional way is to conduct a series of experiments and calculate these parameters based on empirical data. This method is time consuming and poor in accuracy due to the nonlinearity nature of the pneumatic systems' friction. Daw (2002) employed genetic algorithm in order to estimate the dynamic friction parameters along the pneumatic cylinder. The evaluation function has been formed using the statistical expectation of mean-squared errors (MSE). Further study has been conducted by Wang, et al., (2004) to improve the convergence rate and the accuracy of the algorithm. Their work concentrated on measuring the friction parameters of the cylinder rather than the complete system.

In practice the dynamic behaviour of the load is influenced by the friction within the cylinder as well as other friction types such as air friction. In this research, an improved method, by which the engineer could work out the friction parameters accurately for the whole system considering all friction coefficients and types, online, and in relatively short time, has been proposed. The CMRE that introduced in this chapter has been devised to facilitate the realisation of the proposed method to identify the system frictional parameters.

6.3.2.1 CMRE for Frictional Parameters Identification

Figure 6.15 shows the proposed structure of the CMRE that is used for frictional parameters estimation of the system. The main concept of such structure is to use real signals acquired from the system's sensors to calculate the piston's speed and position. These calculations depend on three variables: (1) static friction F_{static}, (2) coulomb friction F_c, and (3) viscous damping coefficient c_f. Therefore, by comparing the position and the speed that resulted from these calculations and the real position and speed of the system on the same graph, the three variables can be identified.

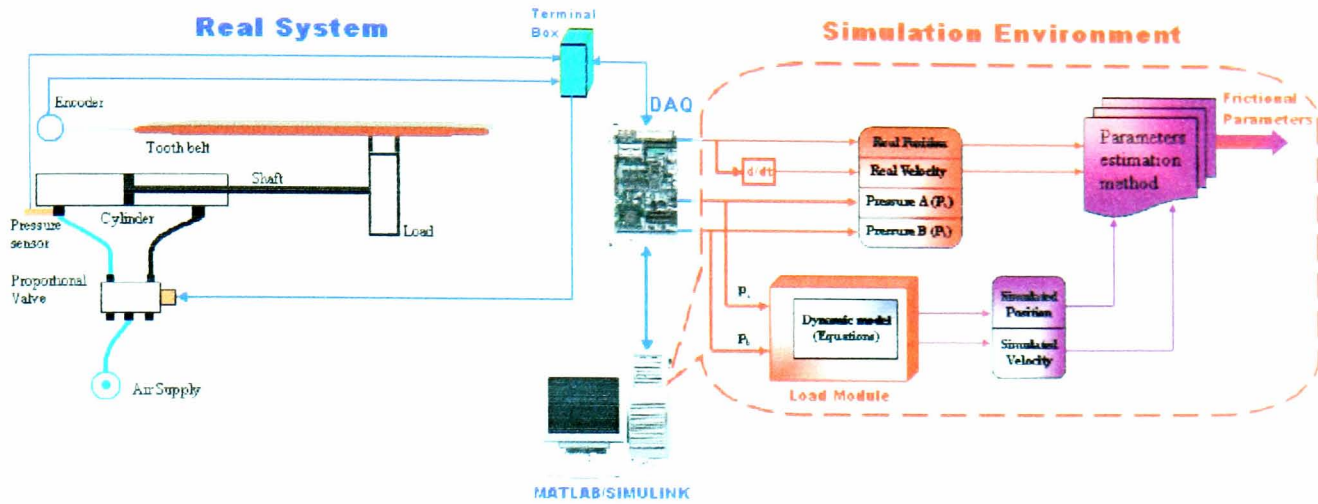


Figure 6.15 CMRE structure for frictional parameters identification

Based on the derivation in chapter 5/section 5.3, the following equations have been used to calculate the speed and position of the piston, which were implemented in the Load Module block:

$$\ddot{Y} = \frac{1}{M}((A_a P_a - A_b P_b) - (A_a - A_b)P_{atm} - F_f sign(\dot{Y}) - c_f \dot{Y}) \qquad (6.2)$$

$$\dot{Y} = \frac{d\ddot{Y}}{dt} \qquad (6.3)$$

$$Y = \frac{d\dot{Y}}{dt} \qquad (6.4)$$

The CMRE has been implemented using *"Real-time workshop"* provided by MATLAB. Figure 6.16 shows the CMRE structure under Simulink that used for the identification of the friction parameters.

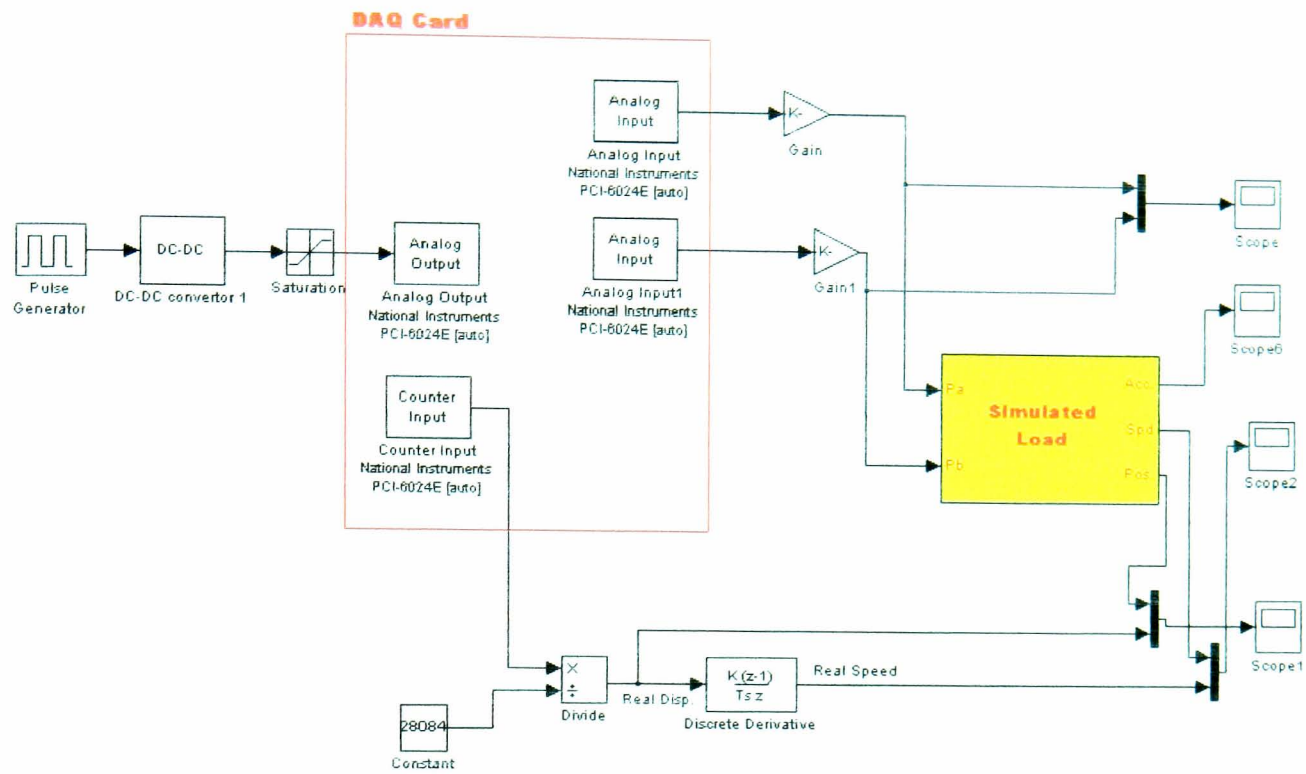


Figure 6.16 CMRE structure under Simulink for friction parameters identification

As shown in Figure 6.16 the inputs to the Load Model block are the readings from the pressure sensors for both chambers (P_a, P_b). To guarantee an accurate estimation of the friction parameters of the system, the initial conditions and components dimensions, which are required to calculate the acceleration in equation (6.2), should be set through the "User dialog" of the Load Module.

6.3.2.2 The Methodology

The frictional parameters would have significant influence on the behaviour of the simulated output as shown in the previous section in equation (6.2). Therefore, tuning these parameters and comparing the real position and speed with the simulated ones will help to establish them quickly and easily. This section presents the procedure to determine three frictional parameters, namely, static friction F_{static}, coulomb friction F_c, and viscous damping coefficient c_f.

Firstly, an initial value for the static friction (stiction) should be determined. This can be achieved by setting the other two variables (F_c and c_f) to zero and then increase the value

of the stiction until reaching minimum time delay between the real and the simulated position. Figure 6.17 shows the output for different stiction values.

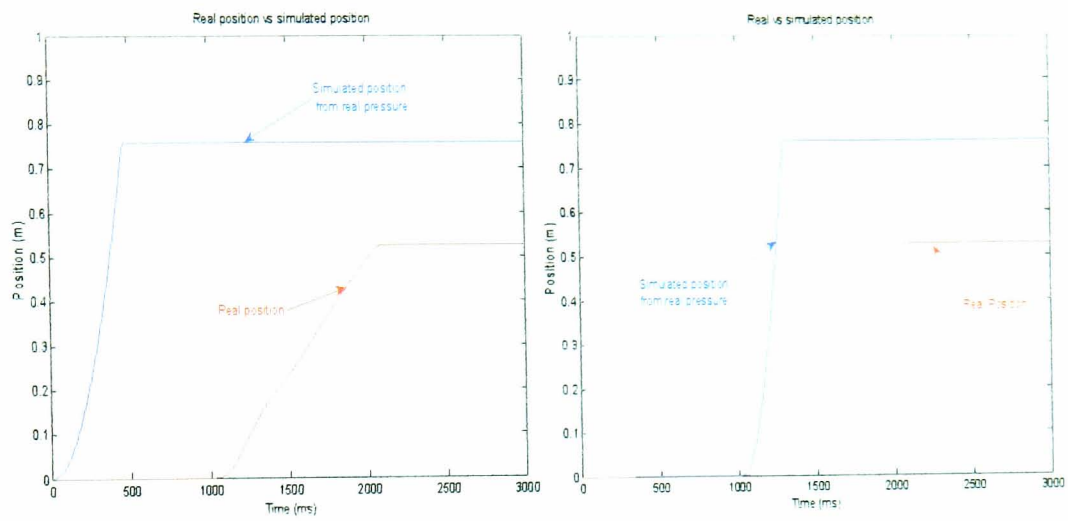


Figure 6.17 Simulated position vs. real position with (a) F_{static}=20, F_c=0, c_f=0, (b) F_{static}=60, F_c=0, c_f=0

The next step is to start increasing F_c and c_f with fixed steps. As shown in Figure 6.18, the simulated position becomes closer to the real position as the variables increased.

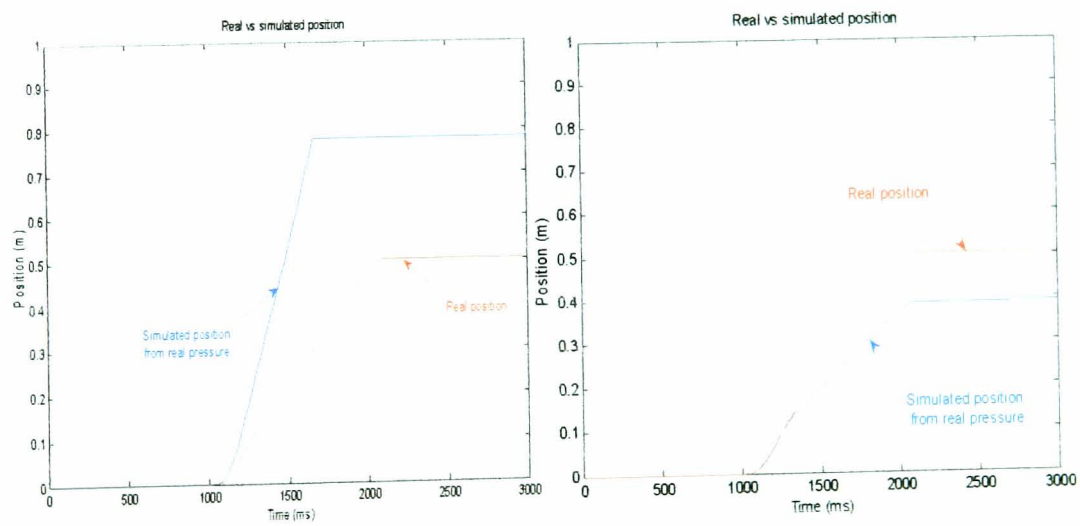


Figure 6.18 Simulated position vs. real position with (a) F_{static}=60, F_c=20, c_f=20, (b) F_{static}=60, F_c=40, c_f=40

Tuning the three parameters is needed to bring the simulated position closer to the real one (see Figure 6.19).

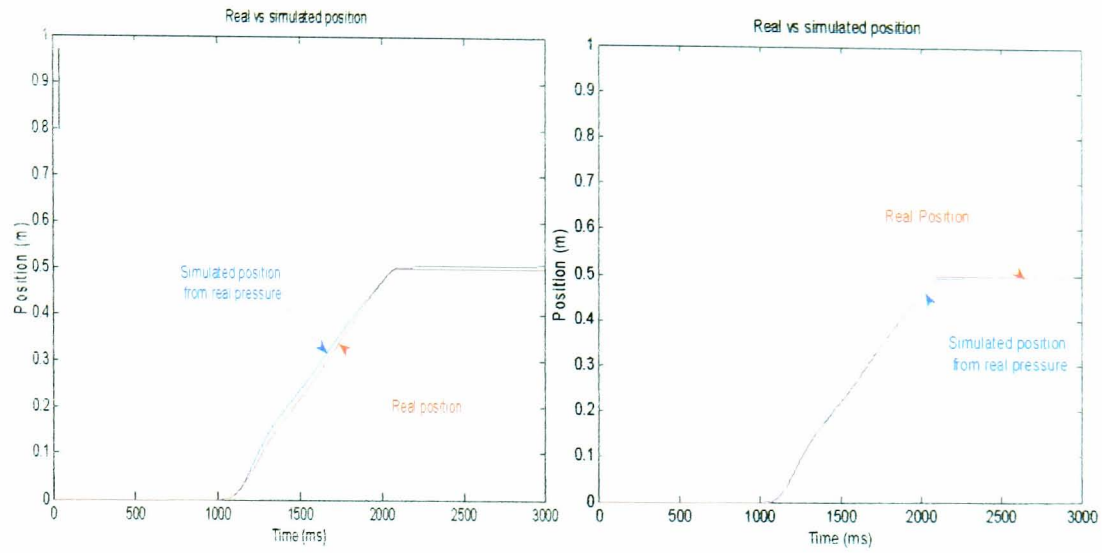
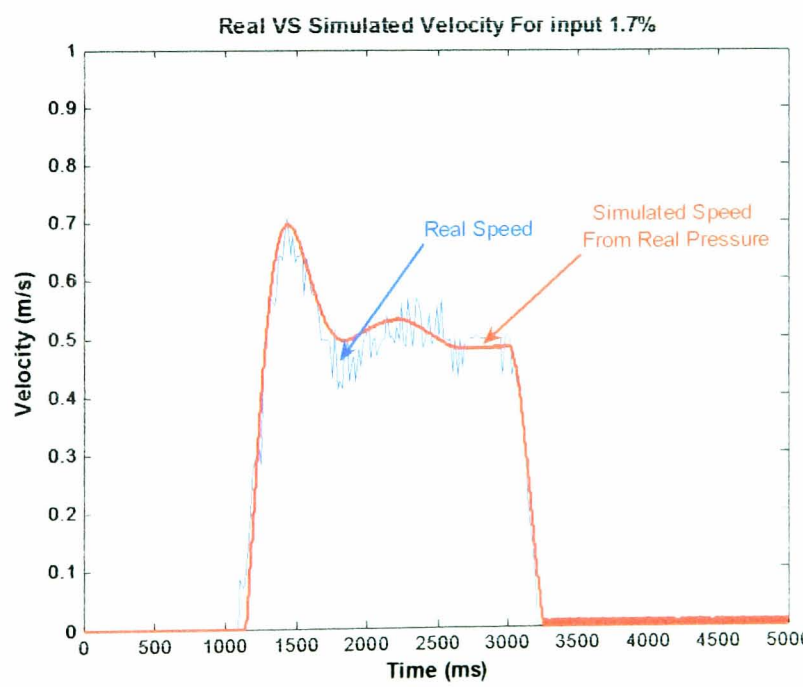
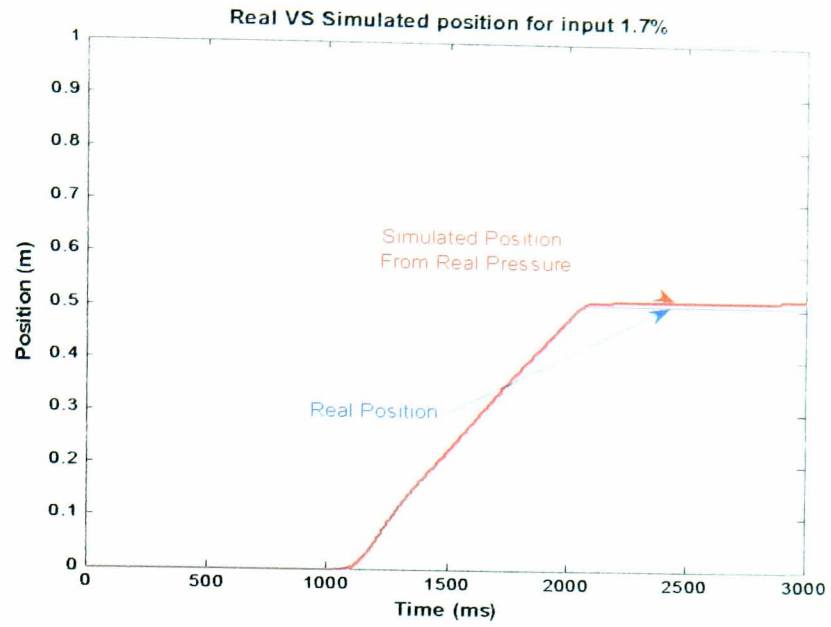


Figure 6.19 Simulated position vs. real position with (a) F_{static}=60, F_c=39, c_f=32, (b) F_{static}=80, F_c=39, c_f=32

Further fine tuning for the three parameters (e.g. finer increment step) should guarantee accurate frictional parameters. Figure 6.20 shows some experimental results of the speed and position which are obtained after the final frictional parameters are determined for different input.



(a)



(b)

Figure 6.20 Pulse response of the system at F_{static}=88, F_C=39, c_f= 31, input 17 %. (a) velocity, (b) position

Traditionally, estimating the frictional parameters would be achieved by conducting a large number of experiments and obtaining a large set of data, which is time consuming. Using the above method, a high degree of accuracy in the identification of these parameters can be achieved in a relatively short time.

6.4 Summary and discussion

After establishing the mathematical model of the servo-pneumatic system components, it is necessary to build the working simulation model of the system. The virtual components of the servo-pneumatic system were designed as independent modules with inter-module linking adapted from the linking variables as defined in the component-based model structure. The designing and developing of the CMRE comprises of two main steps (1) the development of the CST for off-line models, and (2) the development of the CMRE using the CST for on-line applications.

The CST was mainly developed for off-line system integration and testing and controller tuning. A components library is set-up and organised in six major categories; actuators, controllers and converters, pneumatic valves, sinks, sources and tubing. The component

can be used by simply drag it from the component list and then drop it at the workspace window. The user has to set the components parameters according to the specification of the real system's components.

The developed CST has then been employed for the development of the CMRE. The CMRE comprises of a real system working in conjunction with a simulation environment. The CMRE would assist in the implementation of different control schemes on the real system, which will be demonstrated in chapter 7. It also facilitates the identification of the system's frictional parameters. A method for frictional parameters identification using the CMRE has been introduced in this chapter.

The main difference between the CMRE and the hardware-in-the-loop (HIL) that the latter has been mainly employed to test real control systems against simulated plant before the production of the plant. Moreover, the CMRE provides more flexibility to apply different control strategies according to different applications on the same real plant.

Chapter 7
Validation and Evaluation

7.1 Introduction

In this chapter, two test cases are presented; open loop test and closed loop test. The test cases were conducted to demonstrate and explore the feasibility and validity of the CMRE. Section 7.2 presents the system's parameters identification. Section 7.3 presents details of the test cases and the validation study. A case study which shows the viability and efficiency of the developed environment is presented in section 7.4. Section 7.5 presents a summary and discussion of this chapter.

7.2 System's Parameters Identification

The model formulated in chapter 4 for the test bed described in the same chapter requires a number of parameters to be measured. These parameters were represented as numerical constants or coefficients of algebraic expressions in the virtual components' user dialog. These parameters are vital to the accurate interpretation of the simulation results. Apart from the dimensional parameters such as stroke length and diameter, which could be measured using venire or micrometer, separate measurements were performed in experiments utilising the proposed CMRE. Using the method for frictional parameters identification presented in section 6.4, the system's friction parameters has been determined.

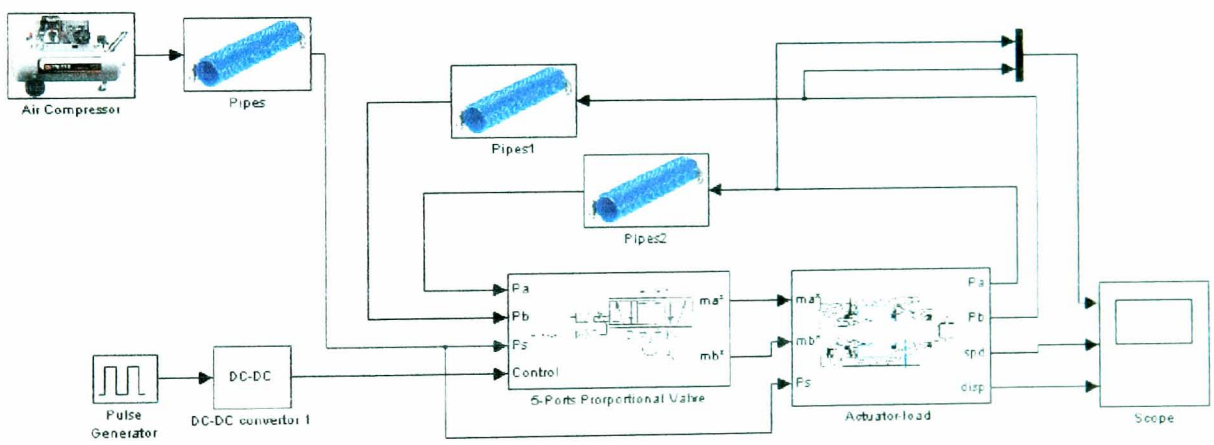
The parameters determined in the measurement tests were set through the user dialog of the corresponding virtual components. The initial conditions of the test bed were also measured and set into the appropriate virtual components. By inserting the same initial conditions and operating parameters into the simulation model, the results could be compared directly by magnitudes. Adding to visual comparison on the same graph, some statistical analysis is performed using some features of Simulink. This will provide additional indicators on the closeness of the virtual system to reality.

7.3 Validation through Simulation of Specific Models

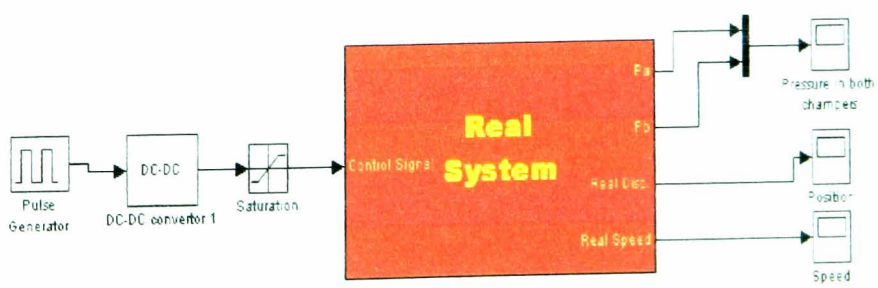
Two specific models of pneumatic system; open loop system and closed loop system, have been simulated to obtain the time response. In order to guarantee an exact comparison of the response, the initial parameters of the test such as starting position and initial pressures in both chambers, the dimensional parameters, and frictional parameters of the system were set in the virtual components.

7.3.1 Open Loop Servo-drive System

The typical open loop system (bang-bang system) consists of a pneumatic actuator, proportional or on-off valve, tubs and load. This open loop drive system was built in the virtual environment as shown in Figure 7.1(a). The "*real-time workshop*" has been employed to establish a connection between the real system and Simulink, as shown in Figure 7.1 (b). The objective is to secure a direct comparison of the time responses of the real system with the virtual system by merging the results on the same graph.



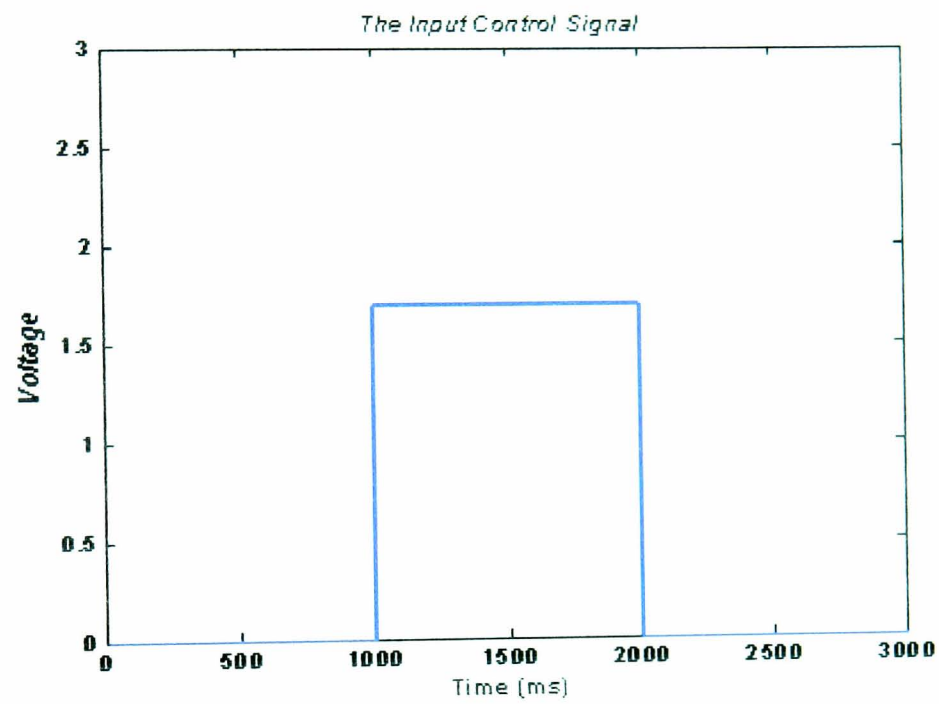
(a)



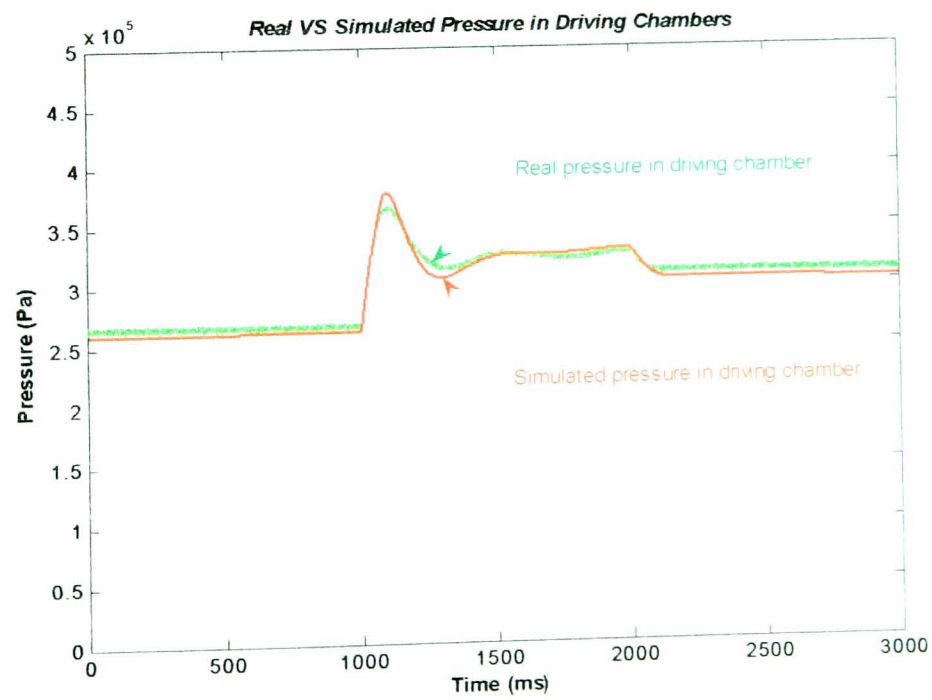
(b)

Figure 7.1 Servo-pneumatic system in the (a) Virtual environment, (b) CMRE

The open loop responses of the drive system are depicted in Figure 7.2. The Simulated and experimental position and speed curves agreed fairly well. Due to safety reasons, the input signal, or the spool command, has been chosen to be a pulse signal rather than step signal. The same timing and value of the spool command has been used for the virtual and real system.



(a)



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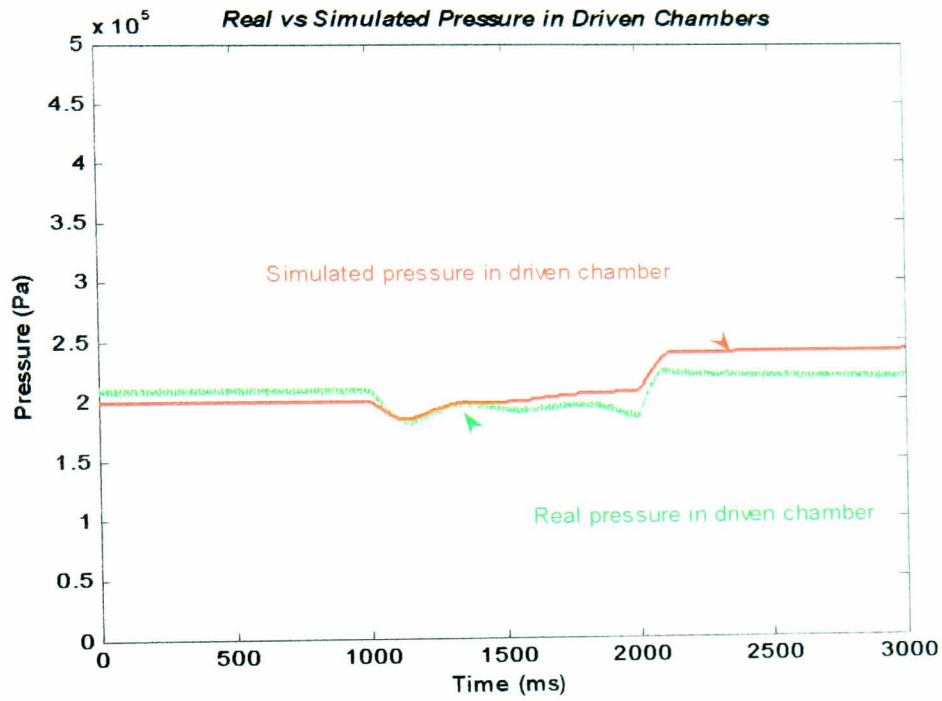
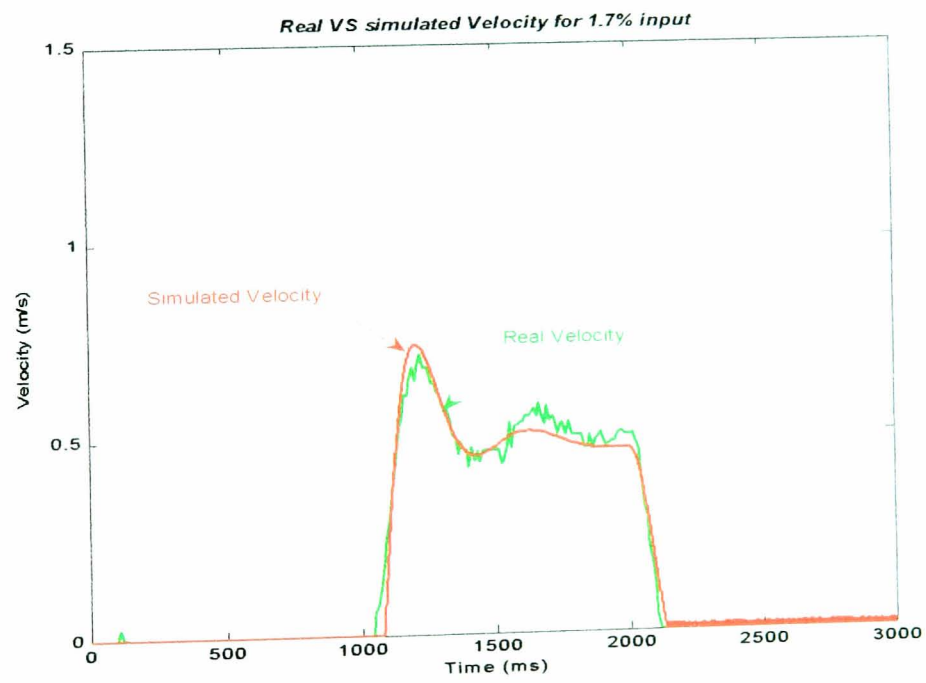
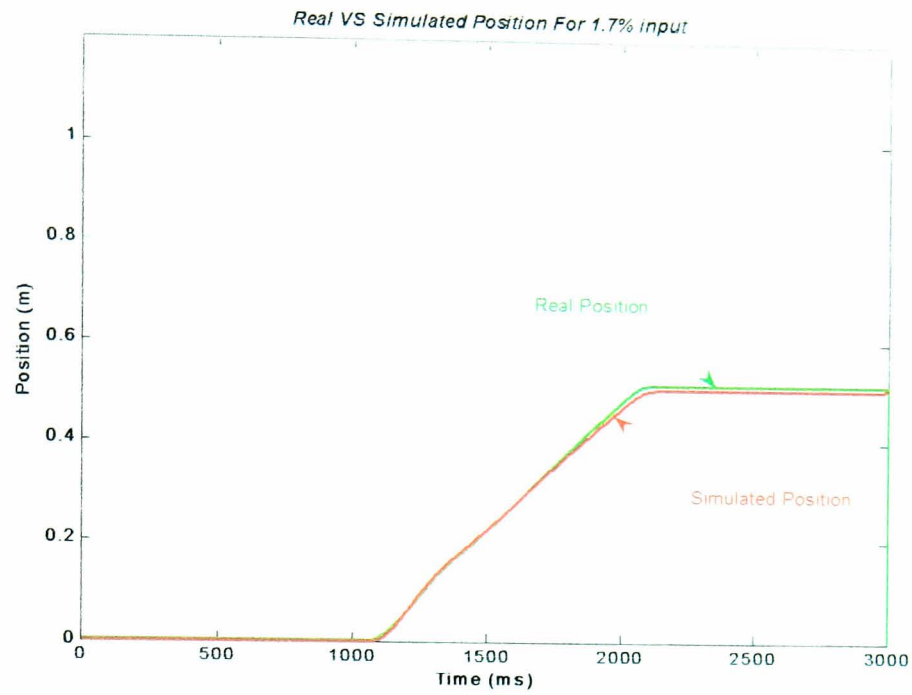

Real vs Simulated Pressure in Driven Chambers
x 10^5
Pressure (Pa)
Time (ms)
Simulated pressure in driven chamber
Real pressure in driven chamber
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
5
0
500
1000
1500
2000
2500
3000

(c)


Real VS simulated Velocity for 1.7% input
Velocity (m/s)
Time (ms)
Simulated Velocity
Real Velocity
0
0.5
1
1.5
0
500
1000
1500
2000
2500
3000

(d)



(e)

Figure 7.2 Simulation vs. experimental open loop system's responses

The pressure response did not agree satisfactorily in value especially in the driven chamber. However, agreement of the trend can be seen from both responses. The difference between the responses could be due to: (1) the internal leakage between the two cylinder chambers. The virtual actuator generating the curves in Figure 7.2 has a zero internal leakage coefficient. This is because of the instrumentation difficulties involved in the measurements of this coefficient. It is expected that if the internal leakage coefficient is provided, then a closer agreement can be achieved; (2) the simulation model assumes that the pressures in the control volumes of the actuator are homogeneous. The simulated values of the pressures are in fact the pressures inside the cylinder chambers whereas in reality the tubes connected to the inlet which is outside the cylinder causing non-homogeneity of the volume; (3) The fact that the tubing model has been established from steady state flow characteristics effectively implies that the mass flow rate flowing in is exactly the mass flow rate flowing out of the tubes which is not the case in the real system. More work is required on the propagation of the pressure wave in the tubes so that the proper tubing model can be established; (4) difficulties in predicting the performance of the valve operator or the spool mechanism; (5) the measurement points of

the pressure sensors are not the same as the simulated pressures which are instantaneous values in the course of integration.

By using an enhanced option in MATLAB, the simulated and experimental responses were compared statistically by computing the max, mean, median, and standard deviation values of the corresponding responses for different inputs. The results are shown in Table 7.1. The values do indicate the relative accuracy of the simulation output. However, the simulated pressure responses have some deviation from the experimental responses. This requires further investigation into the factors affecting the pressure levels in cylinder chamber.

Table 7.1 Statistics for the simulation and experimental open loop responses for different inputs.

Responses	Statistic variables	For Input 17%		For Input 18%		Unit
		Simulation	Experiments	Simulation	Experiments	
Pressure in driving chamber	Max.	3.77	3.677	3.798	3.697	bar
	Mean	2.976	3.007	2.998	3.031	
	Median	3.043	3.096	3.076	3.124	
	Std Dev.	2.929	2.662	3.055	2.746	
Pressure in driven chamber	Max.	2.412	2.28	2.554	2.324	bar
	Mean	2.126	2.07	2.166	2.109	
	Median	2	2.085	2	2.083	
	Std Dev.	1.894	1.176	2.608	1.842	
Speed	Max.	0.7358	0.7121	0.8145	0.8546	m/s
	Mean	0.168	0.1697	0.2029	0.2061	
	Median	0.0031	0	0.0015	0	
	Std Dev.	0.24	0.2422	0.2881	0.2967	
Position	Max.	0.509	0.5154	0.6087	0.6182	m
	Mean	0.2422	0.2457	0.2909	0.2982	
	Median	0.2234	0.2222	0.2703	0.2767	
	Std Dev.	0.2236	0.2275	0.2682	0.2737	

7.3.2 Servo-Pneumatic System with PIDVF Controller

After achieving satisfactory results from the open loop test, a servo-controlled system has been built and tested. As explained in the literature review, PIDVF controllers are widely used in industry due to their simplicity and efficiency. A block diagram of the complete servo controller loop is given in Figure 7.3. Every certain time the controller samples the

axis positions, calculates the axis demand on the bases of linear interpolation of the profiler output, compares desired and actual position and generates a new servo demand by running the five terms *PIDVF* servo loop. Where the controller looks at the current value of error, the integral of the error over a recent time interval and the current derivative of the error signal to determine not only how much of correction to apply, but also for how long.

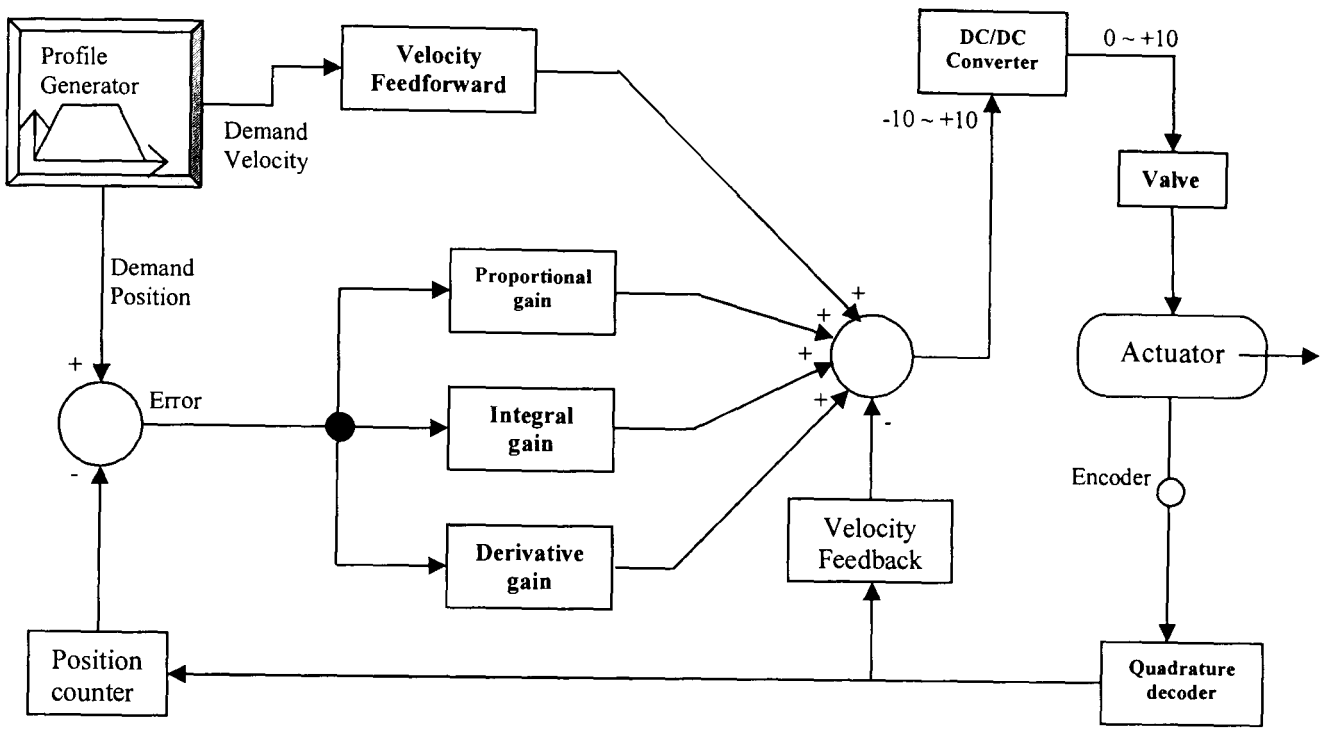


Figure 7.3 PID servo controller block diagram

Suppose that we have a small error between the demand and actual position, a proportional controller will simply multiply the error by the proportional gain K_p and apply the resultant to the actuator. However, if the gain is too high this will cause overshoot, which will cause the actuator vibrating back and forth around the desired position. This implies that as the gain is increased, the system will become unstable. The velocity feedback K_v is used to reduce the rapid movement of the actuator which reduce the instability in the servo loop algorithm. Therefore it is called a damping term. Too low velocity feedback and too high proportional gain will cause overshoot or instability in the actuation system. Whereas the term K_f (velocity feed forward) is used to reduce the following error and response time which is essential for pneumatic actuation systems.

With Proportional and Derivative actions it is possible for the actuator to exhibit a small positional error because the driving force derived from small positional error may not be large enough to drive the system and overcome the static friction. This error can be overcome by involving integral servo loop action (K_i) which sums the error over time. Although the integral servo loop action is useful to eliminate steady state positional errors, it will reduce the system dynamic response. Therefore high K_i values can cause instability during moves. Closed loop gain values tuning is crucial to achieve best performance for the actuation system. This issue will be discussed in more detail later in chapter 8.

The main objective is to synthesis the PIDVF controller in the virtual environment using the developed CST, as shown in Figure 7.4, and to tune the controller gain parameters. Once satisfactory responses attained, the optimised virtual controller will be applied on the real system through the CMRE (see Figure 7.5). Basically, the system under the CMRE consists of hardware and software components. The hardware components (cylinder, valve, and encoder) are located in the red block named "Real System". The software components are the PIDV, profile generator, DC-DC converter, integrator, amplifiers, and scopes. The gains values have been obtained from the virtual or off-line tuning

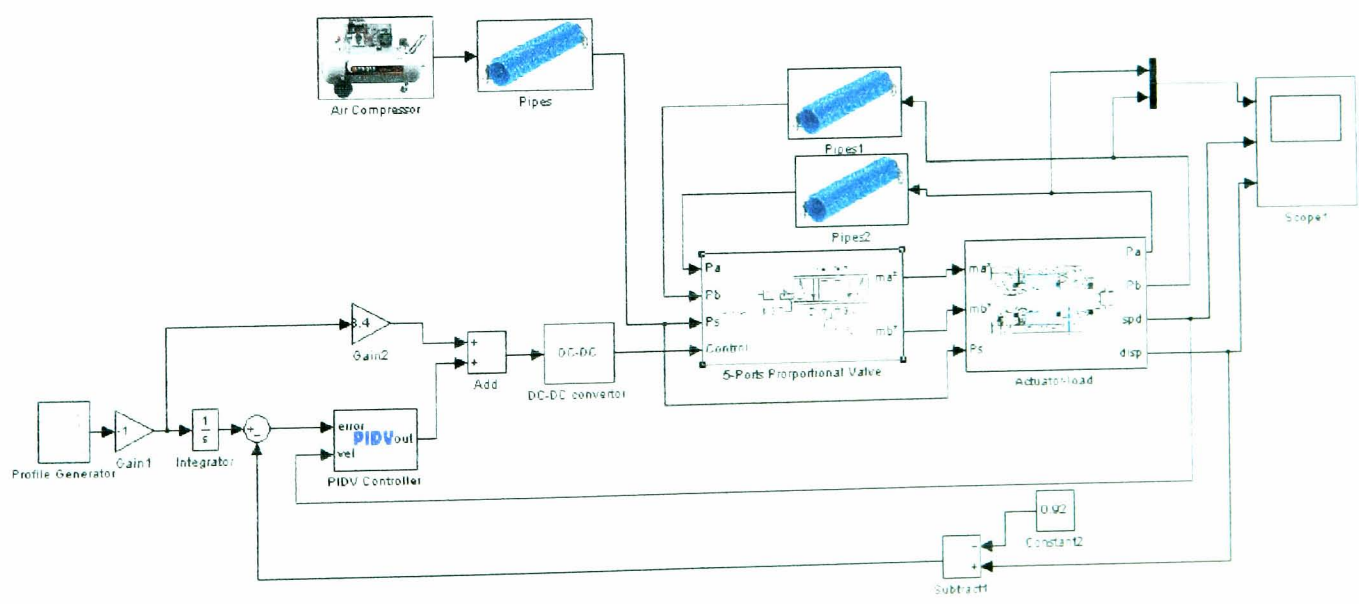


Figure 7.4 Servo-pneumatic system setup under the simulation environment using the CST

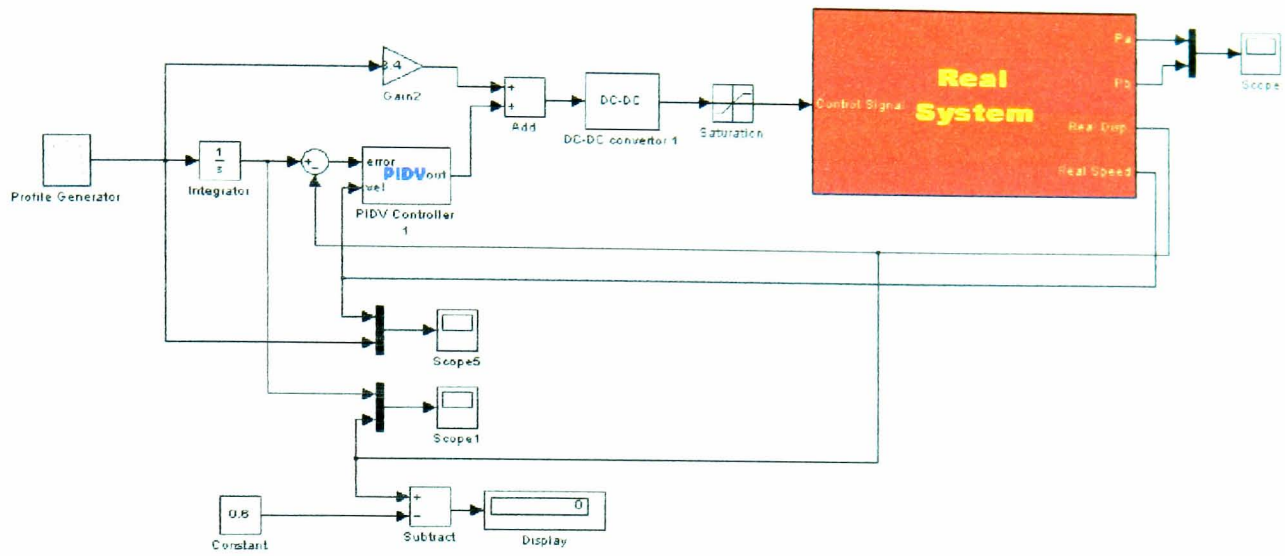
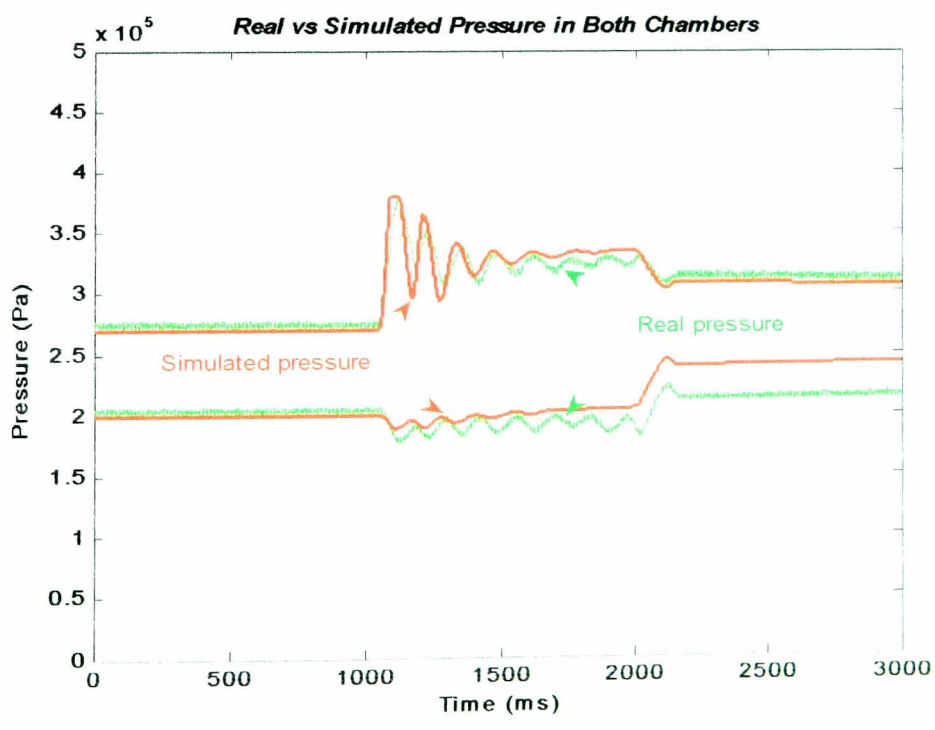


Figure 7.5 Servo-pneumatic system using software and hardware components through the CMRE

As shown in Figure 7.6, the responses from the virtual system are reasonably agreed with the experimental results. A close agreement was obtained from the speed response with slight variation in the amplitude of vibration. The pressure responses, however, deviated reasonably in magnitude, although there are signs for the simulated response to agree to the trend of the real pressure variation. This deviation is due to the reasons that mentioned previously.



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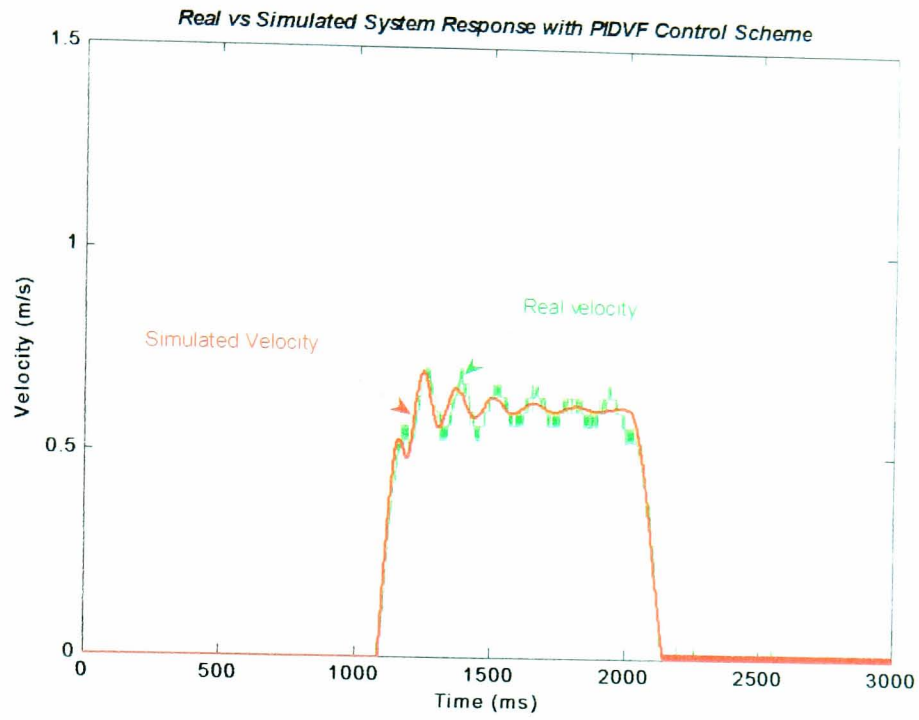

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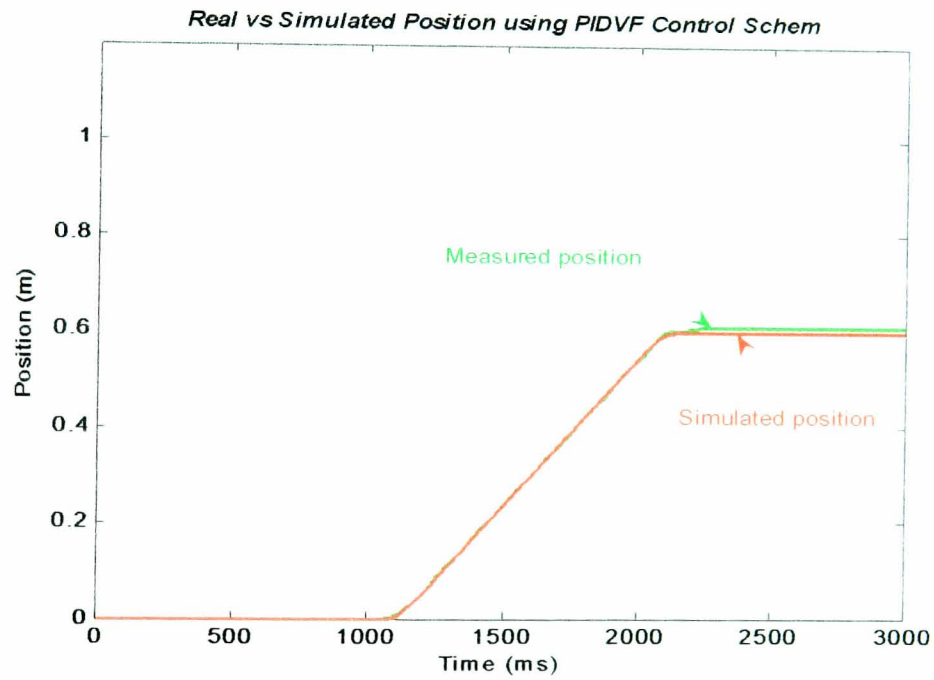

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Figure 7.6 Simulated vs. experimental responses for servo-pneumatic system

Statistically, the max, mean, median, and standard deviation values have been computed for the corresponding responses of the real and virtual systems. The results are shown in Table 7.2. The values indicate the high accuracy of the simulation output. However, the simulated pressure responses have some deviation from the experimental responses.

Table 7.2 statistics for the simulation and experimental responses of the servo-pneumatic system

Responses	Statistic variables	Simulation	Experiments	Unit
Pressure in driving chamber	Max.	3.799	3.799	bar
	Mean	3.02	3.038	
	Median	3.068	3.115	
	Std Dev.	0.2714	0.2381	
Pressure in driven chamber	Max.	2.446	2.256	bar
	Mean	2.126	2.024	
	Median	2	2.036	
	Std Dev.	0.1922	0.1018	
Speed	Max.	0.7018	0.7121	m/s
	Mean	0.1996	0.1965	
	Median	0.003338	0	
	Std Dev.	0.2799	0.2786	
Position	Max.	0.5996	0.6099	m
	Mean	0.2775	0.2808	
	Median	0.2324	0.2334	
	Std Dev.	0.2639	0.2674	

Generally, the results indicate the high efficiency, reliability, and accuracy of the developed CST and CMRE. Taking the advantage of this environment should facilitate the implementation of other control methodologies and should provide an efficient tool for controller tuning and system optimisation. A case study has been performed to demonstrate the capabilities of the developed environment.

7.4 Case Study: Cascade Control Structure

A case study has been conducted to facilitate the study validity of the proposed environment. In this case study, a cascade control structure was chosen to be implemented and synthesised.

The cascade control structure is commonly used for motor drives because of its flexibility. It consists of two distinct control loops; the inner speed loop is followed by the position loop. If position needs to be controlled accurately, the outer position loop is superimposed on the speed loop. Bandwidth (speed of response) required to be increased towards the inner loop. Figure 7.7 shows the cascade control structure that employed for servo-pneumatic system.

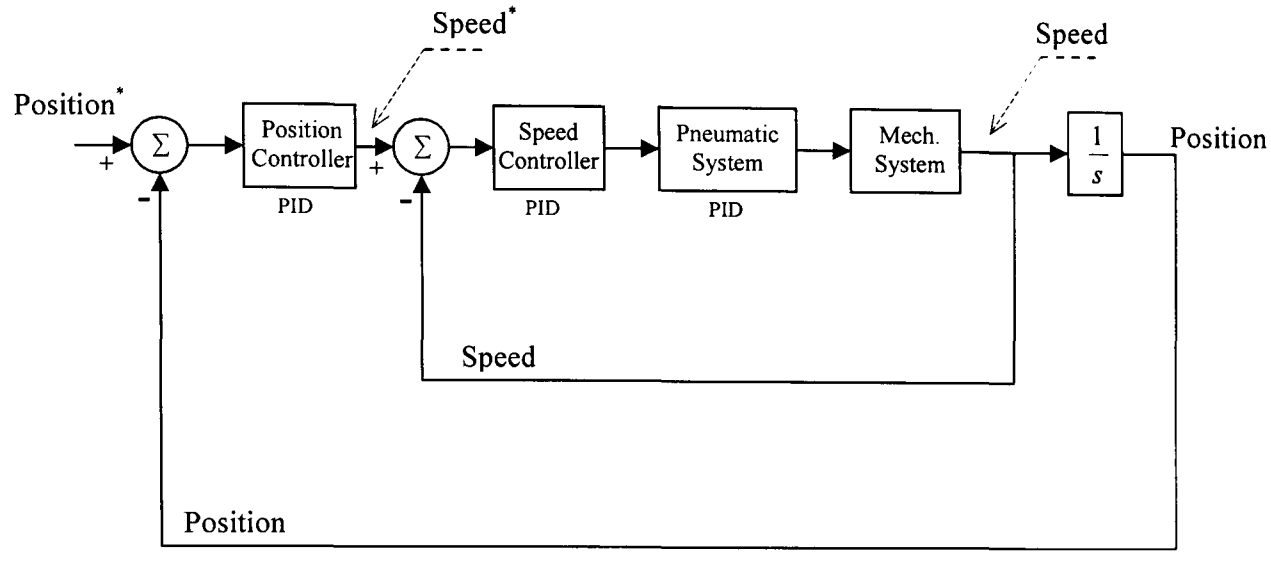


Figure 7.7 Cascade structure for a pneumatic drive

This structure provides the advantages of both velocity and position controls, since it allows following a desired velocity profile in the velocity loop, while stopping with high position accuracy. Cascaded control is responding to changes more slowly than a control system in which all of the system variables are processed and acted upon simultaneously. The feed-forward operation can minimize the disadvantage of the slow dynamic response of cascade control. Therefore, the demand speed and position values are fed forward, as shown in Figure 7.8.

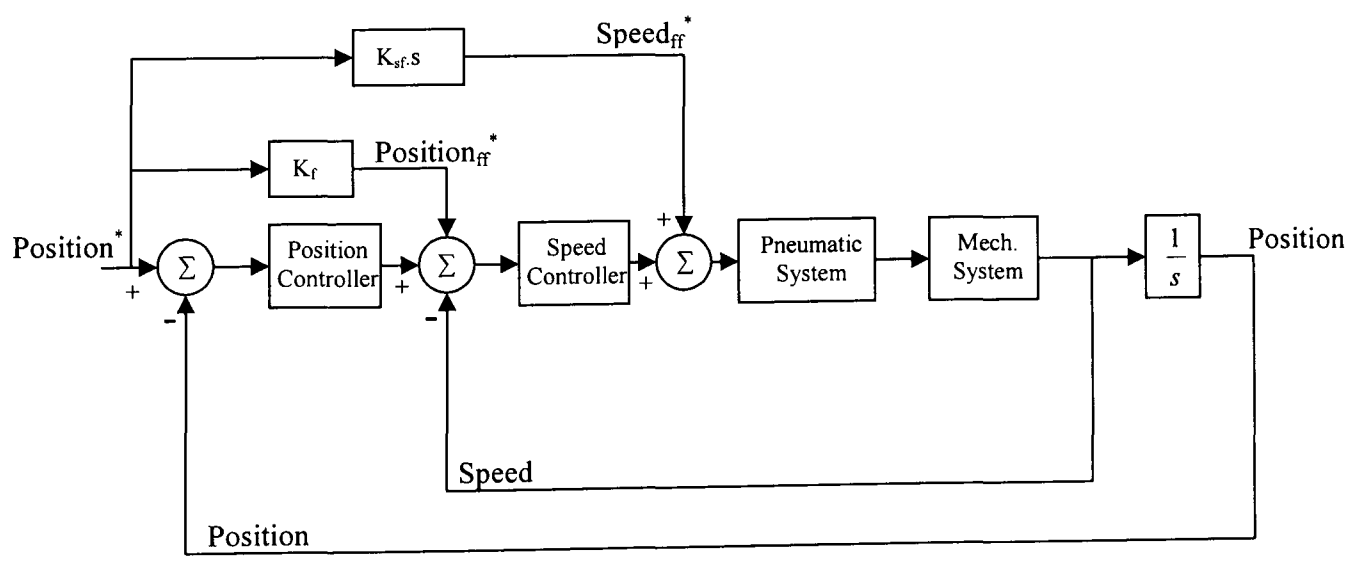


Figure 7.8 Cascade structure with feed-forward for DC motors

7.4.1 Cascade Control Implementation

7.4.1.1 Virtual Implementation

The cascade control structure has been implemented off-line using the CST developed in this research study. Figure 7.9 shows the virtual pneumatic system with the cascade control structure. The control structure consists of two feedback loops; the inner loop is applied to control the speed which allows the system to follow the demand speed profile. PID controller with velocity feedback and feed-forward has been employed within this loop. The outer loop is employed to control the position of the load and assure high position accuracy.

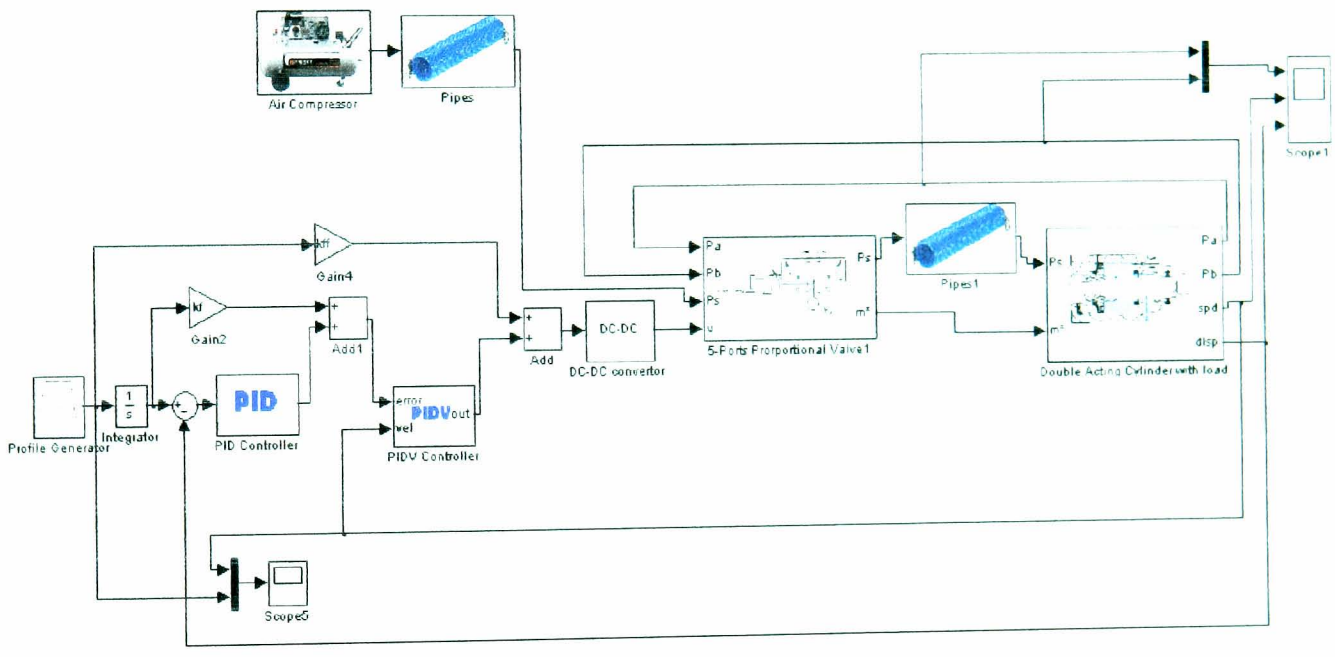


Figure 7.9 Virtual servo-pneumatic system setup with cascade control

After realizing the cascade structure, both controllers have been tuned to achieve minimum following error and maximum position accuracy. Figure 7.10 shows the dynamic response behaviour of the virtual system with cascade control.

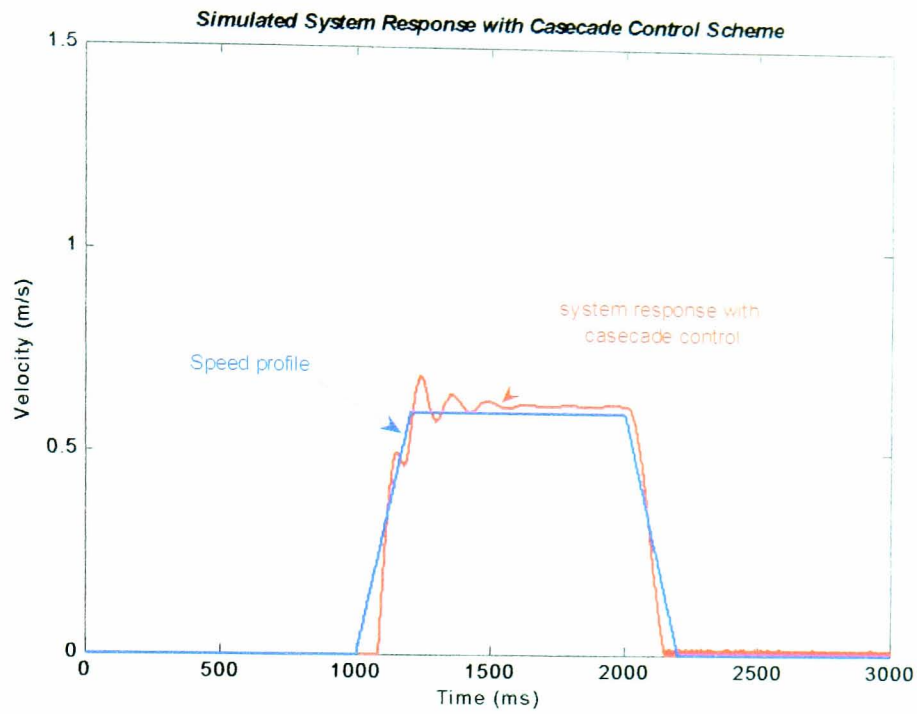


Figure 7.10 Simulation output of the servo-pneumatic system with cascade control

7.4.1.2 Implementation on Real System

The CMRE has been exploited to implement the cascade control structure on the real system. The flexibility and efficiency of this environment allowed for the implementation of this structure easily, efficiently, cost effectively and in a short time. In Figure 7.11, the system setup is shown.

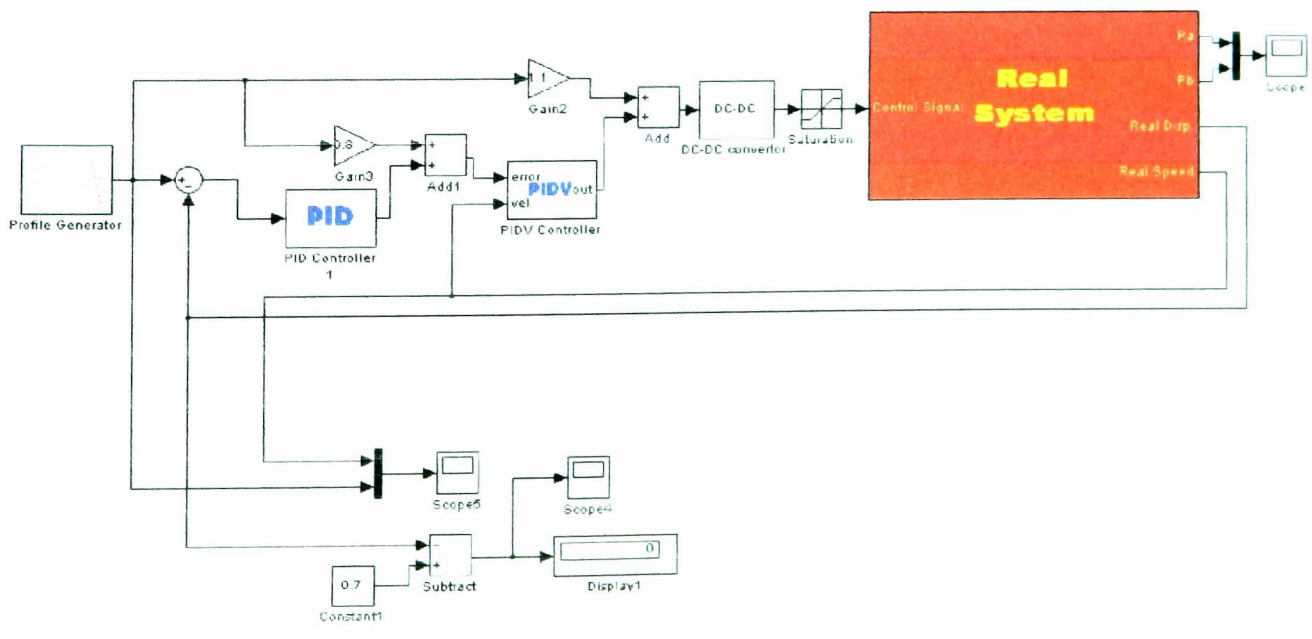
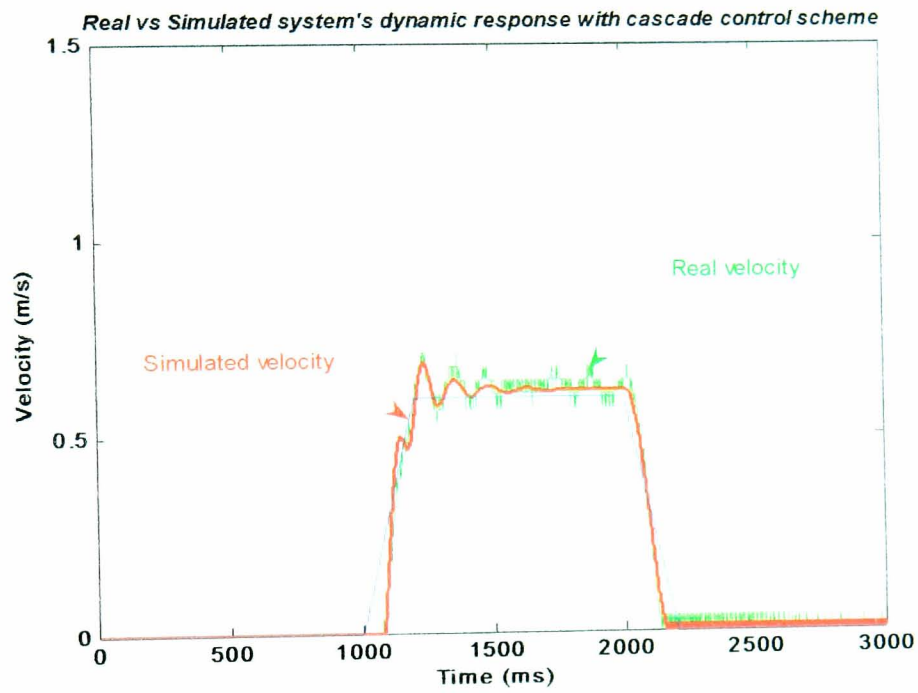


Figure 7.11 Real system with cascade control structure using the CMRE

The dynamic behaviour of the real system was fairly close to the one achieved in the simulation environment as depicted in Figure 7.12. This implies that the accuracy of the system model is high, and the developed CMRE is appropriate to apply different control schemes despite their complexity.



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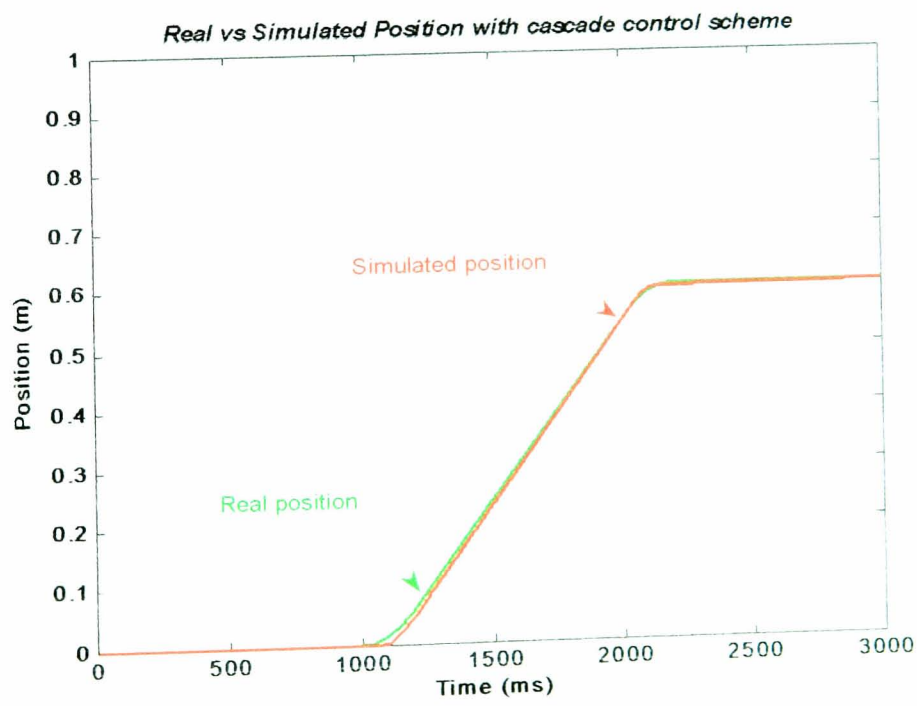

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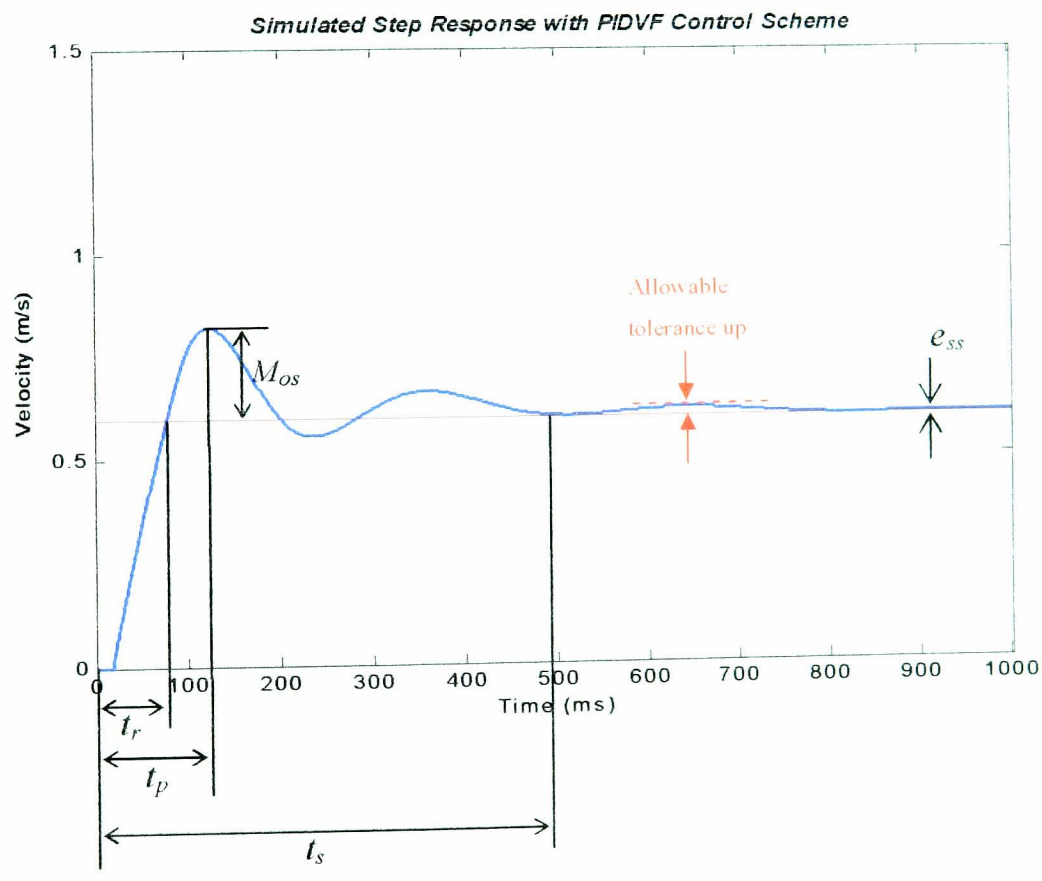
Figure 7.12 Simulation vs. experimental system's dynamic behaviour with cascade control

7.4.2 Cascade vs. PIDVF Control

Comparing the cascade control with the conventional PIDVF control can be carried out by using one of the test signals such as step-function input. The step response after optimising both control methodologies to the best output is shown in Figure 7.13. The step response test was performed using the simulation only since it is unsafe to run the real system with step input. However, the simulation was already approved to be accurate enough for test and evaluation.

A number of commonly used performance criteria which characterise the control system in the time domain were used to evaluate the performance of both control methodologies. Table 7.3 illustrates these parameters for the both responses.



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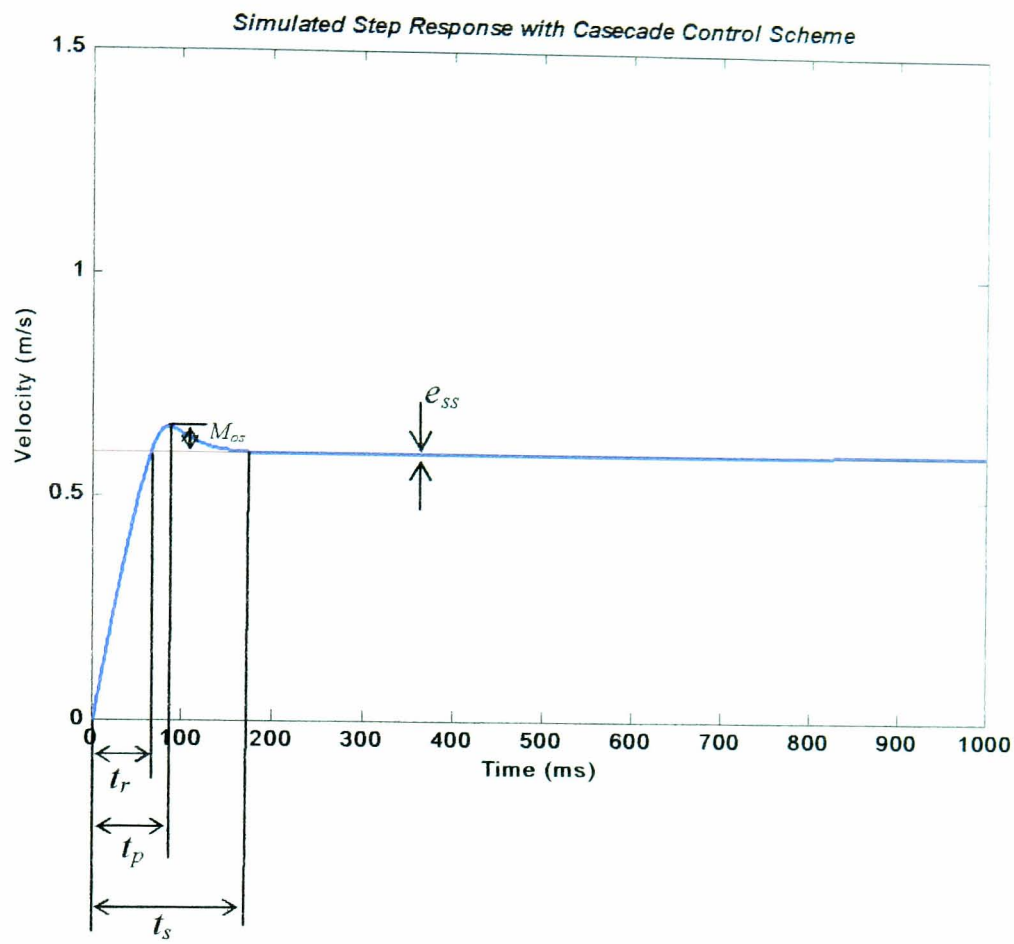


(b)

Figure 7.13 Servo-pneumatic system's step response with (a) PIDVF controller; (b) Cascade control structure

From Table 7.3, it can be observed that the system with cascade control structure is more stable and efficient than the one with PIDVF. The maximum overshoot percentage of the system employed cascade control is lower than the one used PIDVF, which indicates that the latter is less stable. Moreover, the system settled down faster in the cascade control case. The steady state error and oscillation are less in the cascade control case. This indicates that the cascade control structure has improved the system dynamic behaviour.

Table 7.3 performance criteria for the systems employing cascade control and PIDVF controller

Performance criteria	Cascade control response	PIDVF response
Maximum overshoot (M_{os})	0.0549 (9.15%)	0.2219 (36.9%)
Rise time (t_r)	0.064 s	0.076 s
Settling time (t_s)	0.173 s	0.506 s
Peak time, (t_p)	0.102 s	0.094 s
Steady-state error (e_{ss})	0.001	0.002
Number of oscillation to settling time	1	2

7.5 Summary and Discussion

This chapter aimed at validating and evaluating the proposed environment by conducting a series of laboratory-based experiments and carrying out a case study. Before performing the test, the system's parameters have been identified using the CMRE and used in the simulation. This allowed a direct visual comparison between the responses from the real and virtual system.

Two modes of operation were examined; open loop and closed loop tests. The open loop test was needed to determine the validity of the pneumatic system model. After achieving satisfactory results from the open loop test, a PIDVF controller has been adopted. The closed loop test was aimed at simulating the performance of the servo-pneumatic system test bed at normal working conditions.

Close agreement between the experimental and simulated position and velocity responses was obtained in most cases showing that the simulation models have represented the real system adequately, and the friction parameters have been identified accurately. The results were verified visually and statistically. The deviations in the pressure responses were attributed to the fact that the internal leakage coefficient was ignored in the simulation due to some measurement difficulties. The assumption that the pressure in the

control volumes is homogeneous and the difficulties in predicting the performance of the spool mechanism were also contributed to the difference.

A case study was carried out as the simulation environment was approved to be sufficient and reliable for control system application and implementation. Cascade control structure was chosen to be implemented and synthesised. The controller was first synthesised and optimised off-line using the CST, then it was applied to the real system using the CMRE. The results from both the simulation and the real system agreed fairly well, which endorse that ***the CMRE is an easy, accurate and robust way to implement different control strategies and methodologies on real servo-pneumatic system.***

From the control perspective, the cascade control structure improved the system dynamic response compared with the PIDVF controller in terms of the maximum overshoot and settling time. This has been proved by considering the step response for both methods and comparing a number of commonly used performance criteria which characterise the control system in the time domain.

In summary, the developed CST and the proposed CMRE were verified satisfactorily. More refinement can be made with additional tests on the parameters of the components models, specifically the valve operator and the internal leakage parameters. Also some extra research is required on the tubing model and its characteristics.

Chapter 8

Servo-pneumatic Life Cycle

8.1 Introduction

Part of this research project is dedicated to investigate the influence of the proposed CMRE and the developed toolbox on the servo-pneumatic system life cycle. The developed tool and environment, which facilitated the control and commissioning of servo-pneumatic system is anticipated to improve the design process of the system. Improving the design process and control implementation would improve the system's life cycle.

8.2 Servo-pneumatic Life Cycle Strategy

Professional machine developers and the customers they serve share a common goal of building pneumatic systems that effectively support industry objectives. In order to ensure that cost-effective and quality systems are developed which address different industry sectors needs; developers employ some kind of strategies and tools to direct the system's life cycle. In this chapter, a strategy for pneumatic systems life cycle is introduced based on the developed tools and environment. Based on the general machine life cycle (Jlab, 2005), the strategy can be split into six phases:

- System requirements analysis phase.
- System design phase.
- Software components integration and testing phase.
- Mechanical components production and integration phase.
- Control system evaluation and implementation phase.
- Machine service support phase.

As shown in the Figure 8.1, the proposed strategy of the pneumatic system life cycle can be developed using a systematic, requirements-driven process. This process starts with the system requirements analysis phase, in which the fundamental end-user requirements are stated. Then all important phenomena and/or constraints that potentially combine

through the pneumatic system to limit machine performance are enumerated and their impact assessed. In the design phase the designer generates potential system concepts and different conceptual designs, which must be individually reconciled with the fundamental requirements and constraints, and collectively evaluated to select the optimum solution. Iteration may be required at this stage as much as additional physical constraints or end-user requirements may be discovered. Finally, after attaining an optimized concept, a detailed mechanical design can be generated and the pneumatic components specifications can be released.

By the end of the design phase, the released pneumatic components' specifications can be used to create virtual pneumatic components (see chapter 6). These components can be then aggregated to form a virtual system using the developed CST. Afterwards simulation can be conducted to validate the components specification and the system design. After achieving satisfactory results from the simulation of the dynamic behaviour of the system, the mechanical components can be produced and the system can be integrated.

The control system design can be then performed. Open loop control simulation should be carried out after the identification of the system friction parameters using the CMRE. Once the satisfactory results achieved from the open loop control test, a control scheme should be evaluated and adopted off-line and on-line using the CMRE.

Finally, the machine service support phase begins after the production of the mechanical components and the implementation of the control system; this process involves commissioning, testing, evaluating and optimization of the pneumatic machine system on-site using the CMRE.

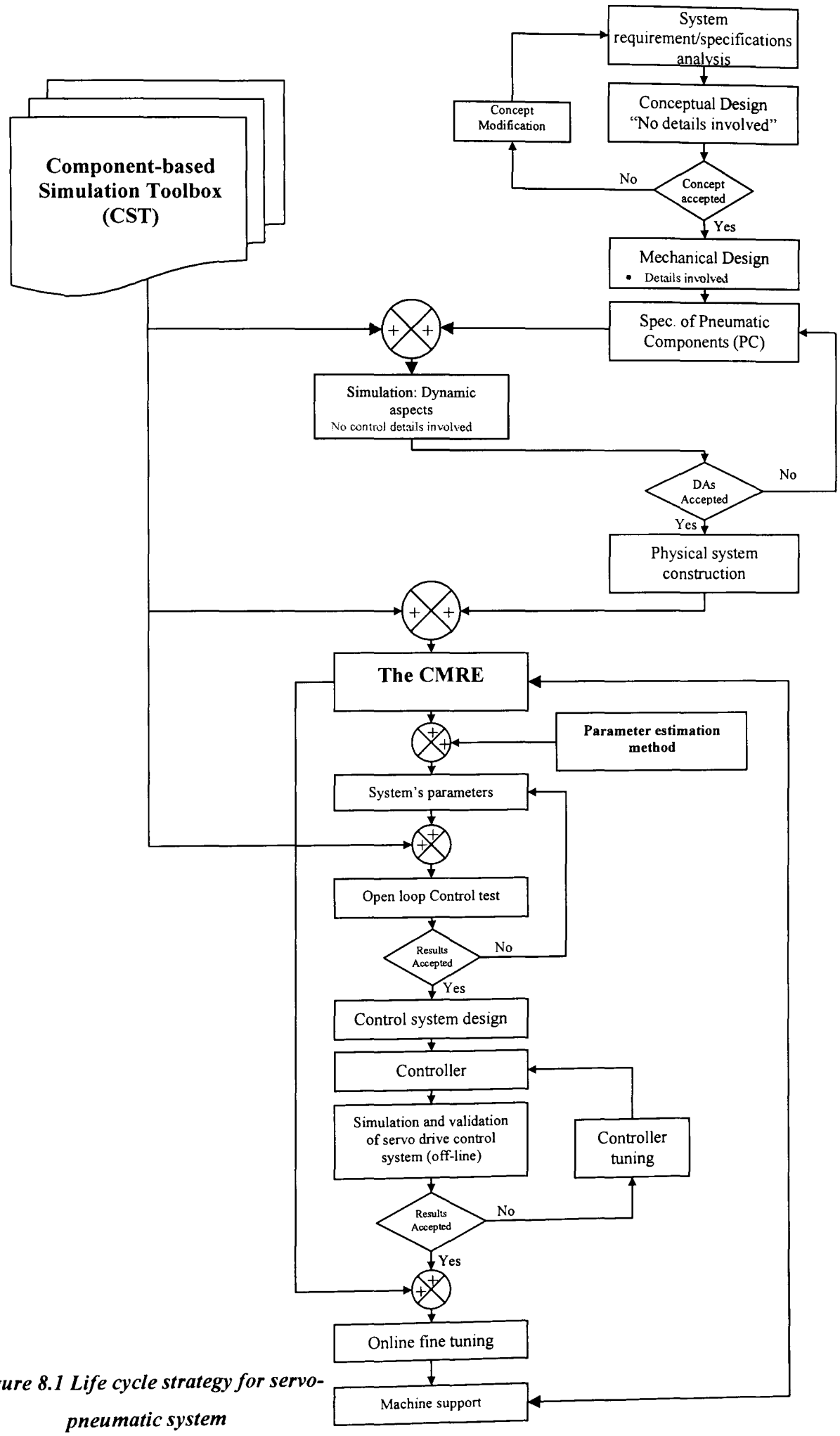


Figure 8.1 Life cycle strategy for servo-pneumatic system

8.2.1 System Requirements Analysis Phase

The system requirements analysis phase includes analysis of user requirements and analysis of system requirements, constraints, and phenomena.

8.2.1.1 Analysis of User Requirements

Typically, machine system design life cycle starts with the user requirement analysis stage in order to acquire the system's specifications and requirements. Understanding user requirements is critical to the success of machine design. It is now widely understood that successful systems and products begin with an understanding of the needs and requirements of the users. The benefits can include increased productivity, enhanced quality of work, reductions in support and training costs, and improved user satisfaction.

The basis for the application of user requirements is a simple process, as shown in Figure 8.2, around four elements: (1) information gathering, (2) user needs identification, (3) envisioning and evaluation, and (4) requirements specification (Maguire and Bevan, 2002).

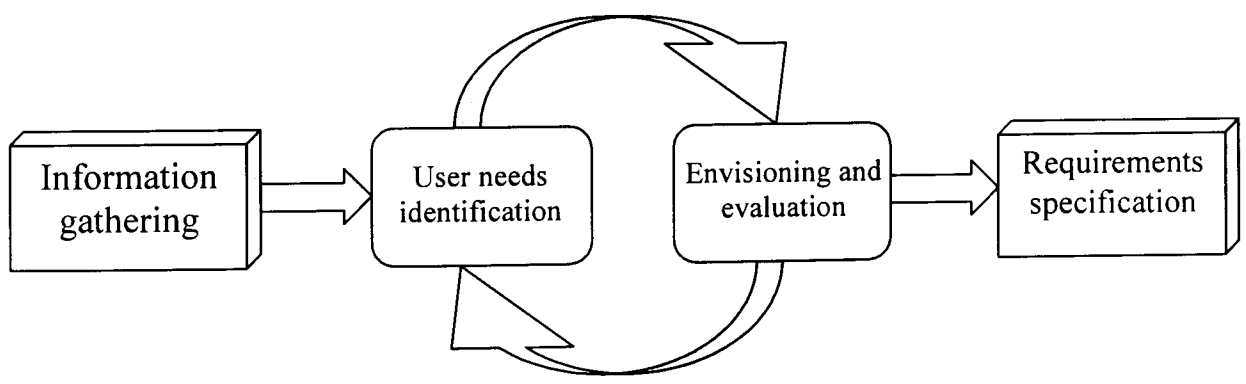


Figure 8.2 General processes for user requirements analysis (Maguire and Bevan, 2002).

The information gathering stage aims to gather background information about the users and the machine system that currently used in their industry. Once user data has been collected, user needs can start to be identified. A number of methods exist for identifying these needs such as user surveys, interviewing, scenarios and use cases, and evaluating an existing or competitor system (Maguire and Bevan, 2002).

After developing an initial set of user requirements, it is important to develop a model to illustrate them. User feedback can then be obtained on the model to validate and refine the user requirements. Finally, the requirements should be organized by documenting a clear statement of design goals, the requirements with an indication of their priority levels, measurable benchmarks against which the emerging design can be tested, evidence of acceptance of the requirements by the user, and acknowledgement of statutory or legislative requirements, e.g. for health and safety. It is also important to manage changing requirements as the system develops (Maguire and Bevan, 2002). There are techniques and methods to organize the final user requirements such as task/function mapping, requirements categorisation, prioritisation, and criteria setting.

8.2.1.2 Analysis of System Requirements, Constraints, and Phenomena

System requirements analysis can be a challenging phase, because all of the major customers and their interests are brought into the process of determining requirements. The quality of the final product is highly dependent on the effectiveness of the requirements identification process. Since the requirements form the basis for all future work on the project, from design and development to testing and commissioning. It is of the highest importance that the machine designers create a complete and accurate representation of all requirements that the system must accommodate. Accurately identified requirements result from effective communication and collaboration among engineers from different disciplines, and provide the best chance of creating a system that fully satisfies the needs of the customers.

For most of the applications in the industry sectors, servo-pneumatic system must meet at least two fundamental requirements. Firstly, it must provide a high performance and reliability, and secondly, it should be cost effective systems.

A high performance servo-pneumatic system should offer an improved performance in terms of speed and load capacity, but priced comparably with their electric and hydraulic equivalents. Some applications require high force output per unit weight as well as linear, fast and accurate response, low friction, and low mechanical impedance (Richer and

Hurmuzlu, 2000). Such systems should consist of very carefully matched actuators and valves which are manufactured in relatively high cost, offering reliable and predictable response characteristics (Backe, 1986). Also, a well structured control strategy is essential to guarantee optimized and robust operation from pneumatic servos. On the other hand, cost-effective systems, which provide low to medium level of control but still offer acceptable performance characteristics at significantly low price, are required in many applications.

An accurate and comprehensive description of the phenomena and constraints of the servo-pneumatic systems is excessively complex. However, it is crucial for the designer to have a fully appreciation of these phenomena and constrains. This can be attained by understanding the governing equations of servo-pneumatic systems which are highly nonlinear. The assumptions with the governing equations for servo-pneumatic system will be detailed in chapter 5.

8.2.2 System Design Phase

After analyzing the user requirements, setting-up the system requirements, and identifying the system constraints and phenomena, it is time to move forward into the design work. The first step is to create a conceptual design of the anticipating system and then to design the detailed mechanical system.

8.2.2.1 Conceptual Design

A conceptual design embodies the structure of the system, as perceived by the user. This structure is a formulation of the conceptual ideas to meet the requirement specifications which are the results from the previous phase. The input and goals of the conceptual design process is the requirement specifications to be met. *It is always more expensive to improve a bad conceptual design than a bad detailed design, therefore this process is crucial and essential* (Delta, 2005).

This process generates solutions without detailed design parameters, and acts as a blueprint for the subsequent design and implementation stages. New requirements may

arise within the process, which should be considered. After generating concepts and meeting all the requirements, the optimal concept and solution can be then selected, if accepted by the customer. Conceptual design mainly involves mechanical designers and the corresponding technical personnel. The 3D graphical virtual environments can be employed at this stage to facilitate the communication between different parties (e.g. designer, sales and end user) and convey the solution concepts to them without difficulty.

8.2.2.2 Detailed Mechanical Design

Traditionally, the machine system design involves the design of all mechanical aspects in full detail to meet the machine system requirement specifications. The design can be supported by CAD/CAE tools which are widely available in industry. During the mechanical design process, the control engineer may give advices related to control system issues such as positioning of sensors. The following items are the major outputs from the mechanical design process:

- CAD models for individual pneumatic components (including components' dimensions).
- Pneumatic system layout which also can be represented in the form of CAD model.
- Kinematics properties (e.g. maximum allowable velocity)

The outputs produced at this stage will serve as the essential inputs for building more concrete simulation model in the next phase.

8.2.3 Software Components Integration & Testing Phase

To achieve efficient machine design, engineers need to simulate the integrated mechanical design and the dynamic behaviour of the system in software before moving to the production and integration stage. This helps them make trade-offs in the early stages of design, visualize the system in action, and save money on creating prototypes after every design iteration.

- Therefore, the objectives of this phase are: (1) evaluate, compare, and verify the mechanical design, and (2) generate concrete control system requirements for the subsequent control system design process.

The developed CST can be used to perform the simulation of the dynamic behaviour of the system. Therefore, rather than building systems from scratch, designers could rapidly realise a servo-pneumatic system by means of assembling existing pneumatic components from the components library after setting the required parameters which are resulted from the previous stage.

Dynamic behaviour simulation provides conflict detection for verification, which helps a designer to find possible collisions due to either incorrect event logic (soft fault), or faulty mechanical design (hard fault). In addition, the conflict detection gives some essential information about the machine operation limit for the control system. Moreover, simulation helps to verify the performance of the design, so that simulation outputs can be compared to the desired output. This will help to find any discrepancy between the requirement specifications and the design.

Compared to the traditional design process, implementing and testing servo-pneumatic system off-line has big advantages in terms of reducing machine development time and cost-effective. Moreover, in the virtual world, enhanced graphical representation and visualisation facilitate the discovery of potential logic errors.

8.2.4 Mechanical Components Production and Integration Phase

Once the off-line simulation results fulfil the system and user requirements and the physical components are available, the virtual system can be implemented and aggregated into the real machine. After constructing the physical system, The CMRE can be employed to identify some of the nonlinear parameters such as the frictional parameters. These parameters should provide a high efficiency in the evaluation and optimization of the control system.

8.2.5 Control System Evaluation and Implementation Phase

Designers and engineers need to simulate not only the dynamics of the servo-pneumatic mechanism but also the control algorithms that act on the mechanism. The mechanical simulation involves parameters such as friction, forces, and gravity, while the control simulation involves control algorithms such as PID.

In this phase, a control architectural foundation should be set. It involves the important engineering activities that transform the control requirement specifications into control architecture. In detail, the tasks include system design, control logic programming, and simulation in virtual environment.

Tuning the controller is considered more complex in servo-pneumatic drives than their electric and hydraulic counterparts. Therefore, a well structured tuning strategy is essential in order to guarantee optimised and robust operation from servo-pneumatics. According to this strategy, the controller can be tuned off-line and then applied on-online through the CMRE. The advantage of using of the CMRE is reducing controller tuning time and the hazard of tuning online. Moreover, it allows for evaluating and applying different control schemes on the real system according to a specific application.

8.2.6 Machine Service Support Phase

8.2.6.1 Servo-pneumatic Commissioning

Servo-pneumatic system commissioning can be performed over two main stages; system synthesis and testing and balancing. The following subsections explore these stages and discuss how the developed simulation environment can improve the commissioning quality and reliability.

8.2.6.1.1 System Synthesis

System synthesis can be performed virtually, as described in chapter 8. After satisfactory outcomes from the virtual system, the system can be synthesised on the shop floor. The work required under this stage should include the following:

- *Set-up inspection:* the aim of the set-up inspection of all system is to verify that each piece of equipment is properly installed and prepared for start-up.
- First run inspection: all items outlined in the set-up inspection should be rechecked to insure proper operation.
- System manoeuvre inspection: in which the mechanical system should be observed under operating conditions for sufficient time to insure proper operation under varying conditions, such as day-night and heating-cooling states. This can be performed after making sure that the system is safe to manoeuvre by testing and verifying its operations under the simulation using the CST. Once confident of the system function is attained, the engineer can run the mechanical system and periodically check the main components.

The intent of this stage is to provide a proper installation, start-up, service and operation of the mechanical systems in preparation for system balancing. Replacement or adjustment of any component should be performed on the virtual system before the real system.

8.2.6.1.2 Testing and Balancing

Total system balance requires that all elements are not only individually correct, but also correct as a composite system. Therefore, participation of the virtual environment is required in the test and balance procedure.

The work required under this stage involves the mechanical and virtual systems:

(1) Tasks that should be performed for the mechanical system:

- Before any adjustments are made, the systems should be checked out for dirty filters, pipe leakage, filter leakage, damper leakage, equipment vibrations, correct damper operations, etc. All compressed air systems, major tubes sections, lubricator, etc., are to be adjusted to deliver the designed air quantities within certain tolerance. After balancing is completed the cylinder charging and discharging with the filters clean.

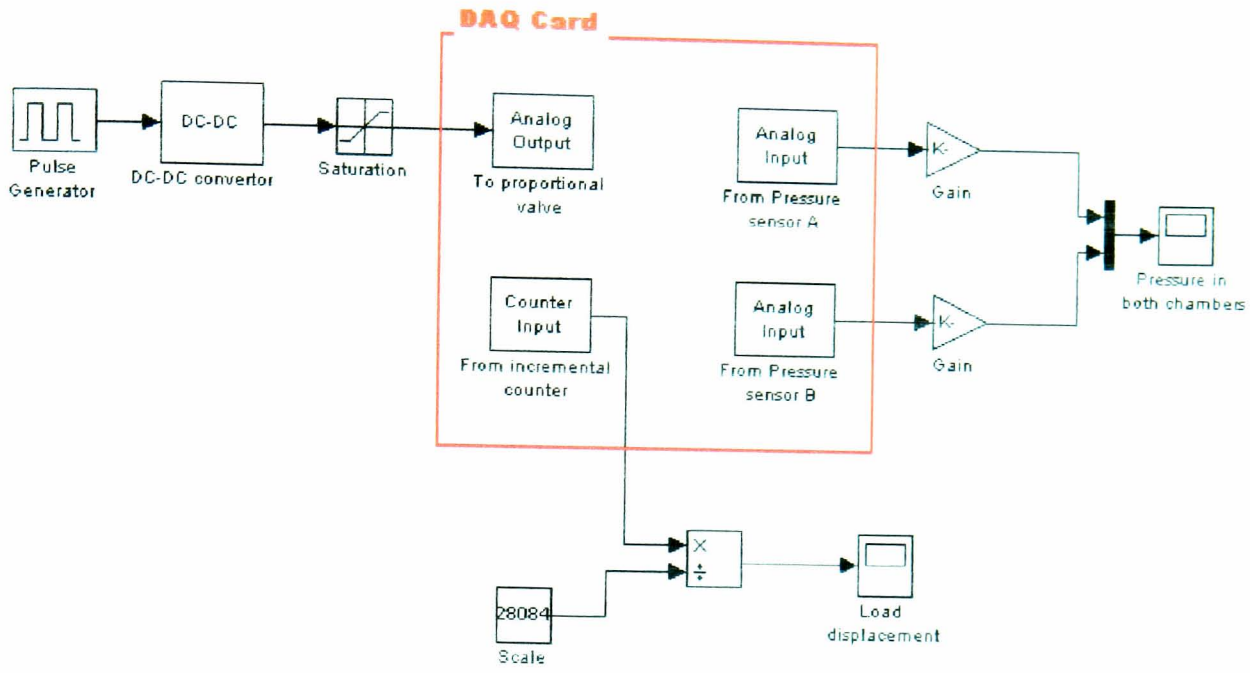
- Supply, exhaust and recirculation air systems should be adjusted for air quantities required. A proper relationship between supply and exhaust air should be established following the proportional balance procedures outlined by the independent firm for such work.

- Using system flow meters, pressure gauges, and/or contact pyrometer to adjust the quantity of the air handled by each filter and supplied to each regulator or valve to meet design requirements.

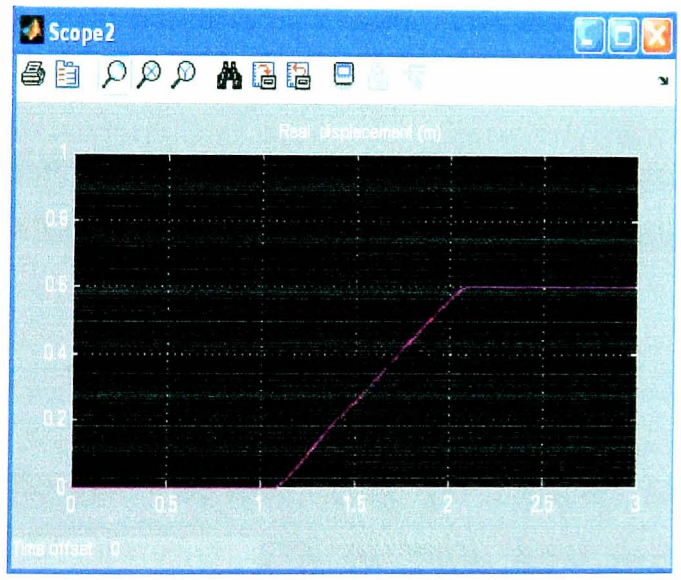
(2) The tasks that should be performed using the virtual systems and then apply them to the real system.

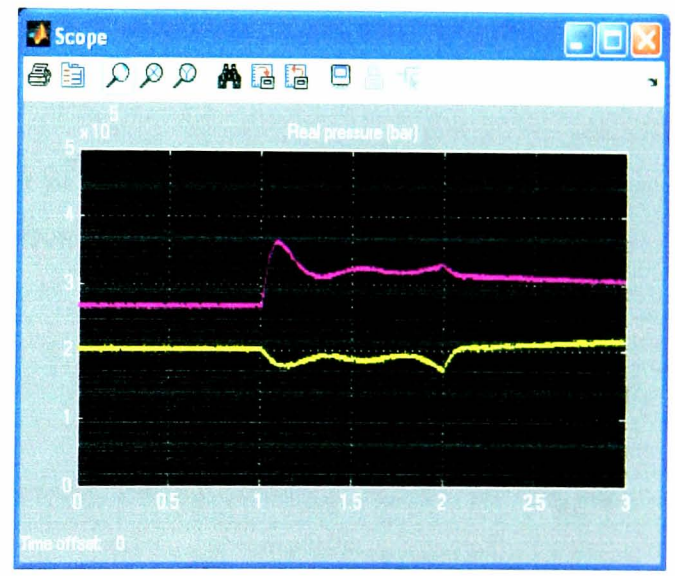
- The testing and balancing of some major components in operation under the virtual system and then perform the test on the real system through the CMRE.
- The testing and balancing specialist should go through the entire control system with the control engineer verifying proper operation of each and every device and the proper function of each system. This can be much easier by using the CMRE, where you can monitor the behaviour of the system with different control schemes and algorithms.

Moreover, data acquisition system should be checked and tested to ensure a reliable monitoring for the pressure in both chambers as well as the displacement of the load. Figure 8.1 shows the test panel that should be configured and an example of the pressures and load position signals that can be monitored using the CMRE.



(a)





(b) *(c)*

Figure 8.3 (a) Test panel for the data acquisition system, (b) Load displacement, (c) Pressure in both chambers

8.2.6.2 Tuning and Optimisation

Tuning work is usually performed manually by trying out different tuning parameter combinations on-line until a satisfactory or at least acceptable results are achieved. This

method is laborious, time consuming, unsafe and does not always give the best possible solution. The developed CST can be employed for off-line tuning. Off-line tuning can be performed using one of the tuning techniques that have been explored in chapter 3. After achieving satisfactory results, the optimised controller(s) can be applied to the real system through the proposed CMRE.

Using the above technique has the following potential advantages:

(1) Reducing the time of tuning since dealing with virtual system is much faster than real system.

(2) Increase the safety factor of the process, in some cases tuning can be dangerous on lives and equipments. For example, if the proportional gain has been set to a high value either by mistake or due to lack of experience, then system will behave aggressively which may harm anyone in the scene or cause a potential damage to the components themselves.

(3) The system becomes more user-friendly and thus it will reduce the need for specialists to perform system tuning and optimisation.

For simplicity and to demonstrate the potentials of the proposed environment, the tuning off-line has been performed based on Ziegler-Nicholas and Astrom (relay) methods.

1. Ziegler-Nicholas method

Hongtao Pan [Pan, 2001] proposed the following tuning strategy, based on Ziegler-Nicholas method, for pneumatic drives which has been used in the experiments to tune the gain factors and thus achieve a satisfactory dynamic response:

Step 1: The tuning is started by applying a small value to the proportional gain (K_p); i.e. K_p=0.01 can be applied at the starting stage, then increase K_p until the system oscillate. Decrease 15% of the total increased value of K_p.

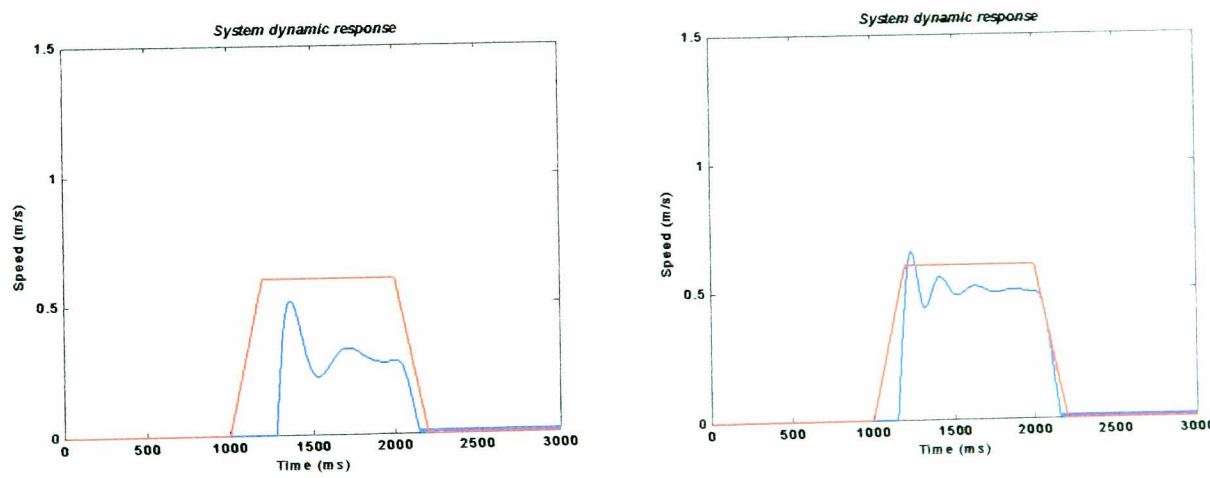
Step 2: Once the proportional gain (K_p) has been set, increase the velocity feedback gain (K_v) with a small value until the rigidity of the system achieved. Decrease 25% of total increased value of K_v and K_p.

Step 3: After increasing the velocity feedback gain (K_v), increase K_p until the system oscillate again. Then decrease 15% of the total increased value of K_p.

Step 4: Apply a small value of the derivative gain (K_d) and then increase it to enhance the stability of the system.

Step 5: Repeat steps 1 to 4 with adjusting each gain value carefully to achieve a better system's performance

Figure 8.2 shows the system dynamic response at different stages of the tuning process. The final values of the controller gains were K_p=0.04, K_i=1.1 , K_d=6, K_v=0.2 and K_f=3.4. The position error at the final setting is 0.2 mm.



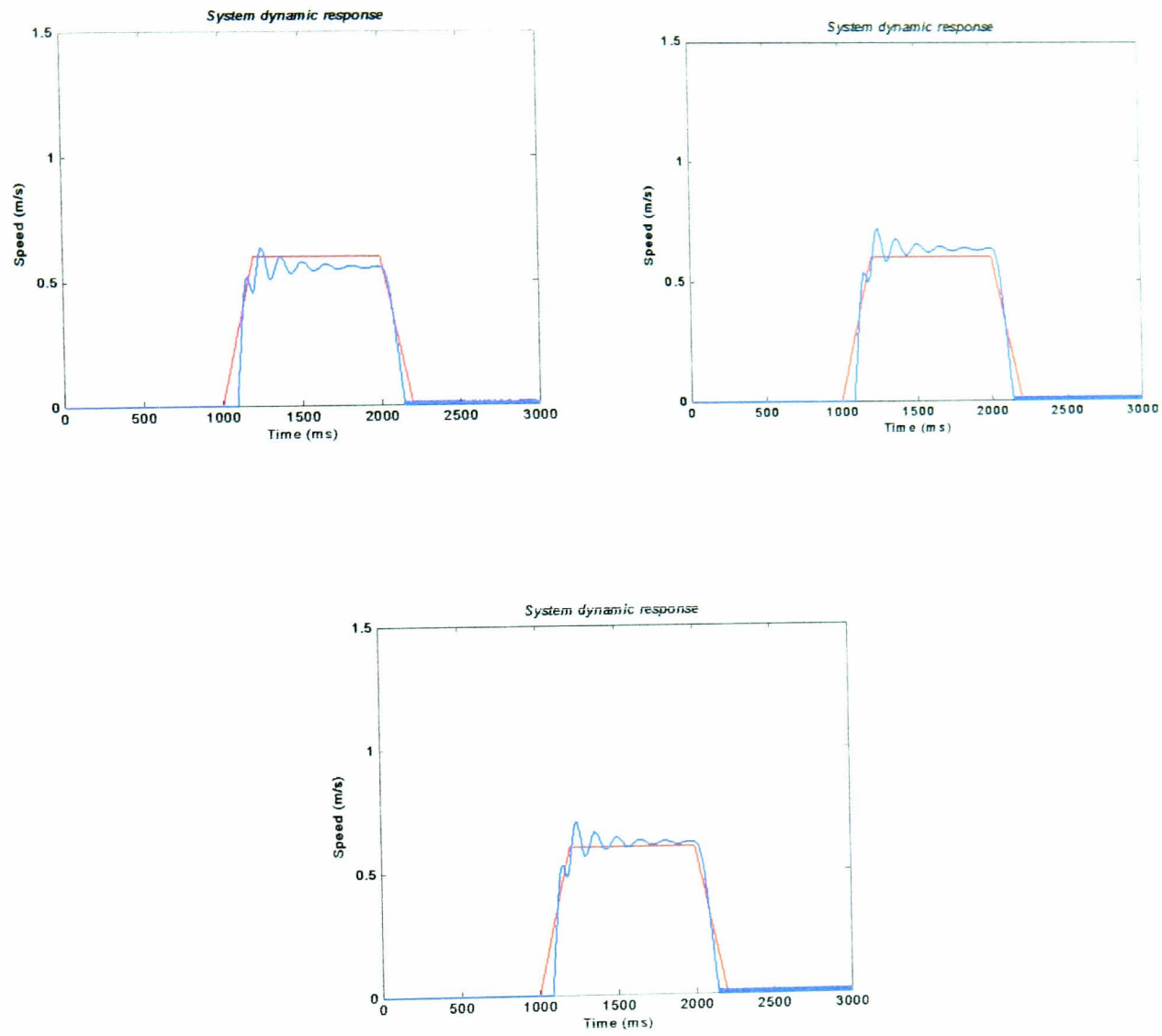
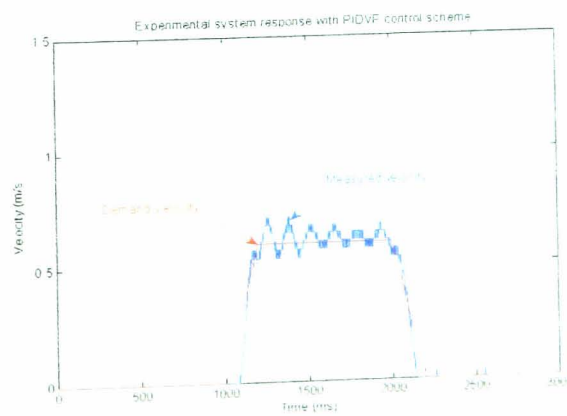
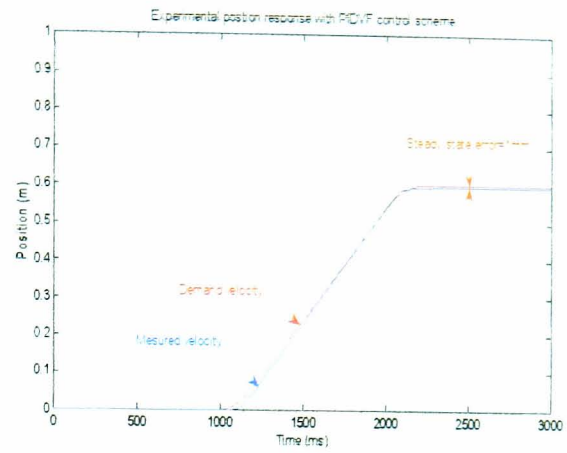


Figure 8.4 the system dynamic behaviour at different stages of the off-line tuning

The controller, with the optimised gain values, has been applied to the real system through the CMRE; the output of the real system is depicted in Figure 8.3.



(a)



(b)

Figure 8.5 (a) Experimental system dynamic response with the optimised PIDVF controller; (b) Experimental position response with steady state error=1 mm

2. *Åström (relay) method*

The concept behind this method is to replace the PID controller by an ON-OFF relay which will force the system to oscillate. The resultant signal is monitored at the error signal *e(t)* and processed to find the ultimate gain K_u and ultimate period, P_u. This data is then used in a rule base to compute PID parameters. Finally, these parameters can be used for the PID controller, and the relay can be swapped with the original controller [Wilkie, et al., 2002].

The procedures to perform this method of tuning is summarised as following:

Step 1: replace the PID controller within the closed loop system with an ON-OFF relay. Figure 8.6 shows the virtual servo-pneumatic system with the relay control.

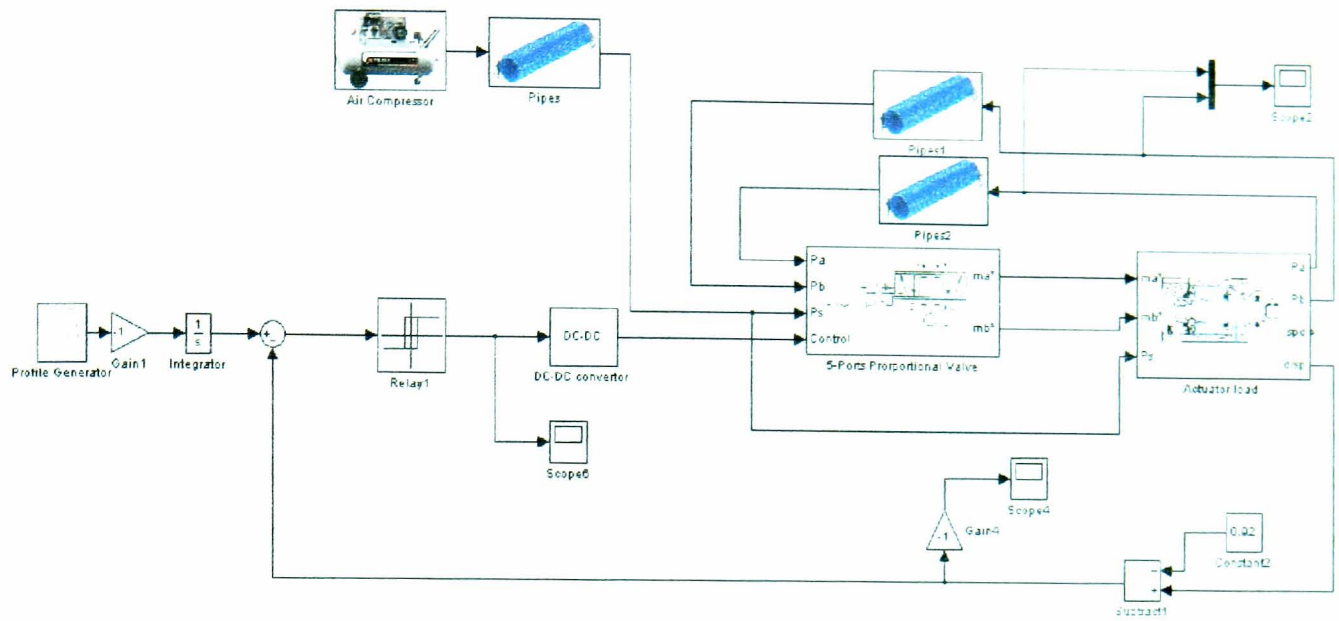


Figure 8.6 the relay experiment loop setup

Step 2: The relay height M was selected to 1.5 by trial and error which is sufficiently large to set up the limit cycle oscillation.

Step 3: Process the data from the output signal by using the steady state portion of the response for measuring the values of P_u and A_{osc} as shown in Figure 8.7.

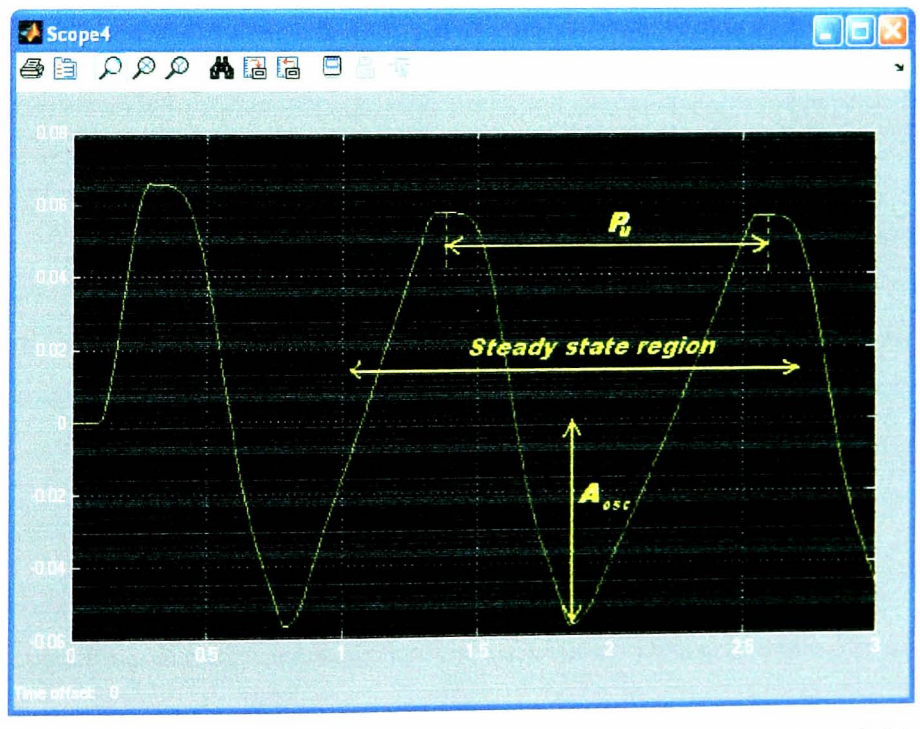


Figure 8.7 output response from relay experiment. P_u= 1.05, A_{osc}= 0.06

Step 4: After completing the data process, the following values are available:

1. the recorded value for $M = 1.5$,
2. the measurement for the ultimate period, $P_u = 1.05$,
3. the measurement for the oscillation amplitude, $A_{osc} = 0.06$,
4. the ultimate gain value, $K_u = \frac{4M}{\pi A_{osc}} = 31.847$.

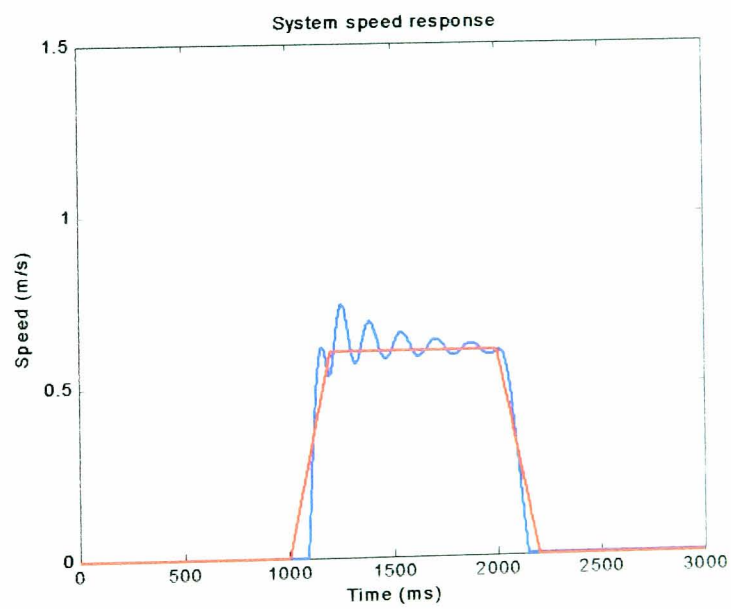
The values of the P_u and K_u can then be used in a rule base to compute appropriate PID controller parameters as following:

$K_p = 0.5\ K_u = 15.9$,

$\tau_i = 1.29\ P_u = 1.354$, $K_i = K_p / \tau_i = 11.756$,

$\tau_d = 0.3\ P_u = 0.315$, $K_d = K_p * \tau_d = 5.01$.

Step 5: Apply the result values of K_p, K_i, and K_d on the PID controller and assess the system's performance with these values. Figure 8.8 shows the speed and position of the load after implementing the new values of the controller gains.



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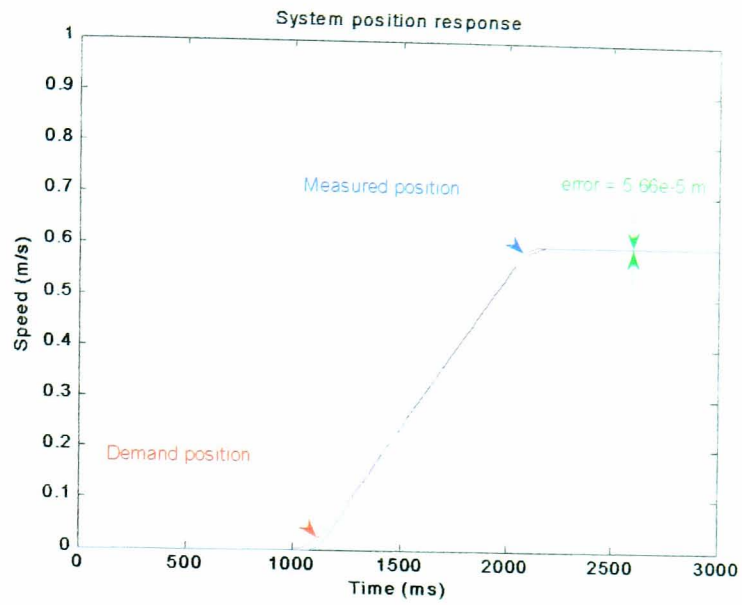


(b)

Figure 8.7 (a) System dynamic response using Åström method for controller tuning; (b) position response with steady state error = 5.66e-5 m

By comparing the speed responses from Figures 8.5 and 8.7, it is noticed that ***Åström method*** for controller tuning provides less error but more oscillation and less stability.

8.2.6.3 Machine Service Support

The flexibility of the CMRE allows the adoption of different machine service support strategies. These strategies can be embedded in the simulation tool as "*service support*" components. Using the CMRE, these "service support" components should facilitate machine builders/suppliers in providing cost-effective and efficient machine maintenance support for the equipment they supply. The main aims of the service support are: (1) to facilitate the maintenance of the equipment's *physical life*, and (2) to facilitate the continuous improvement of equipment during its *in-service life*. It should minimize the number of site visits and shorten the time-scale for completing an onsite maintenance service. The maintenance support functions should include the following: fault detection, diagnostics, prognostics, maintenance action decision support, re-configuration of the control system (both control parameters and control system software components) and onsite maintenance services.

The advantages of using the CMRE for machine service supports are: (1) it facilitates the continuous acquisition, visualization, analysis and re-use of equipment operational and maintenance-related data; (2) It allows experts to access a comprehensive set of data already available in tool's data base such as data that used/generated by the control system; (3) A continuous assessment strategy of the system performance can be adopted in this tool.

One more issue worth mentioning in this context is introducing a web-based remote maintenance support tool for servo-pneumatic systems. Recently Mathworks released a tool known as "MATLAB Web Server 1.2.4" that deploys MATLAB and Simulink applications over the Internet via HTML. This should allow for applying the remote maintenance systems, which are based on the integration of data acquisition, distributed intelligent/automated maintenance support, network/Internet and web-based visualization technologies.

Figure 2.3 depicts a typical (simplified) configuration of such support systems. The web-based visualization is typically a 2D graphics interface. It provides a remote user interface, which allows maintenance personnel/remote experts to access and visualize equipment operational data and analysis outputs of automated maintenance-support-modules. The automated maintenance-support-modules are mostly based on the combination of a number of software components performing various functions such as: digital signal processing, feature extraction, classification, diagnostics, prognostics etc.

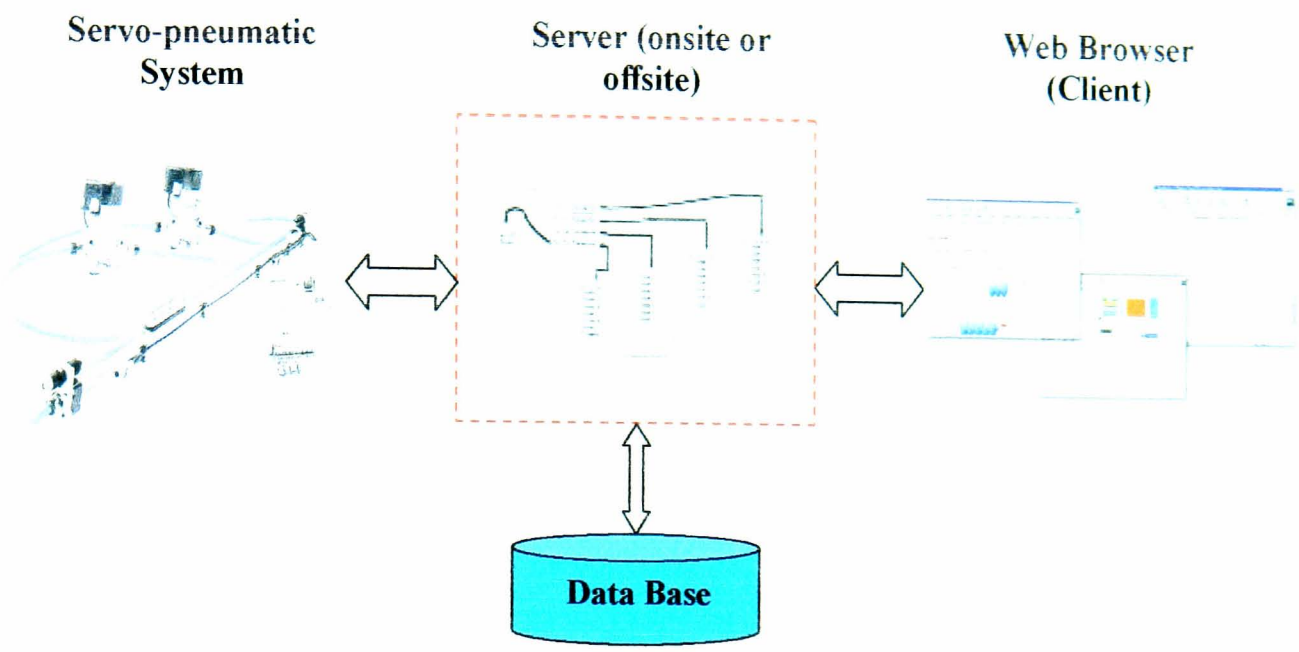


Figure 8.7 A typical configuration of web-based remote service support systems

8.3 Summary and Discussion

This chapter has demonstrated the capabilities of the CMRE in improving the system's life cycle. This was performed by presenting a detailed description of an improved life cycle strategy for servo-pneumatic systems. Six phases were investigated: (1) system requirements analysis phase, (2) system design phase, (3) software components integration and testing phase, (4) mechanical components production and integration phase, (5) control system evaluation and implementation phase, and (6) machine service support phase.

Through that investigation, it was shown how the developed tool is constructive in servo-pneumatic design, control, and service support. Furthermore, it can be observed how this tool supports the introduction of servo-pneumatic technology by assisting the designer, builder, and end-user in designing, implementing, synthesising, and optimising servo-pneumatic system in a user-friendly way.

It has been also demonstrated how the user can apply any tuning procedure the proposed environment with confidence, ease, and high efficiency to attain the demanded performance. A well tuned servo-pneumatic can then be just as easy as its electric

counterparts, thereby increasing the understanding and confidence of users which is important for the success of any technology.

From the machine maintenance and service perspective, an overview of various future developments with the aid of the developed tools. It has identified some issues and opportunities with regard to the implementation of a more cost-effective and comprehensive service support system.

Chapter 9

Recommendations for Future Work

9.1 Introduction

The outcomes of this research project reveal that the research is worthwhile to be pursued further in several directions. This chapter presents some recommendations for these directions and introduces some foundation concepts for extending the developed tool to include multi-axis pneumatic systems, servo-hydraulic and servo-electric systems. Section 9.2 presents the component-based model structure for 2-axis pick-and-place system. Sections 9.3 and 9.4 provide model structures to apply the concept of the CMRE on servo-hydraulic and electric systems. More directions for future research are given in section 9.5.

9.2 The CMRE for Multi-axis Pneumatic System

One of the recommendations for future work is to employ the CMRE for the design and control of multi-axis pneumatic systems. This section introduces a model structure for a high-speed pneumatically driven gantry type pick-and-place system, which exist in the pneumatic lab of the Mechatronics Research Centre at DeMontfort University. This machine is a typical example of a multi-axis pneumatic system. As shown in Figure 9.1 the machine has the capability to move in two-dimensional plane, and it can be described with an X-axis and Y-axis motions.

X-axis motion: X-axis provides the swing action. This is powered by a cylinder called X-axis cylinder (cylinder B) in this study. X-axis encoder measures the swing angle which caused by the rode movement of the X-axis cylinder. However, the X-axis is presented using the displacement of the cylinder.

Y-axis motion: Y-axis provides motion along the rod of cylinder A, which is called the Y-axis cylinder in this study. The resultant displacement is measured by an encoder. The motion generated by this system can be demonstrated in a simplified way in Figure 9.2.

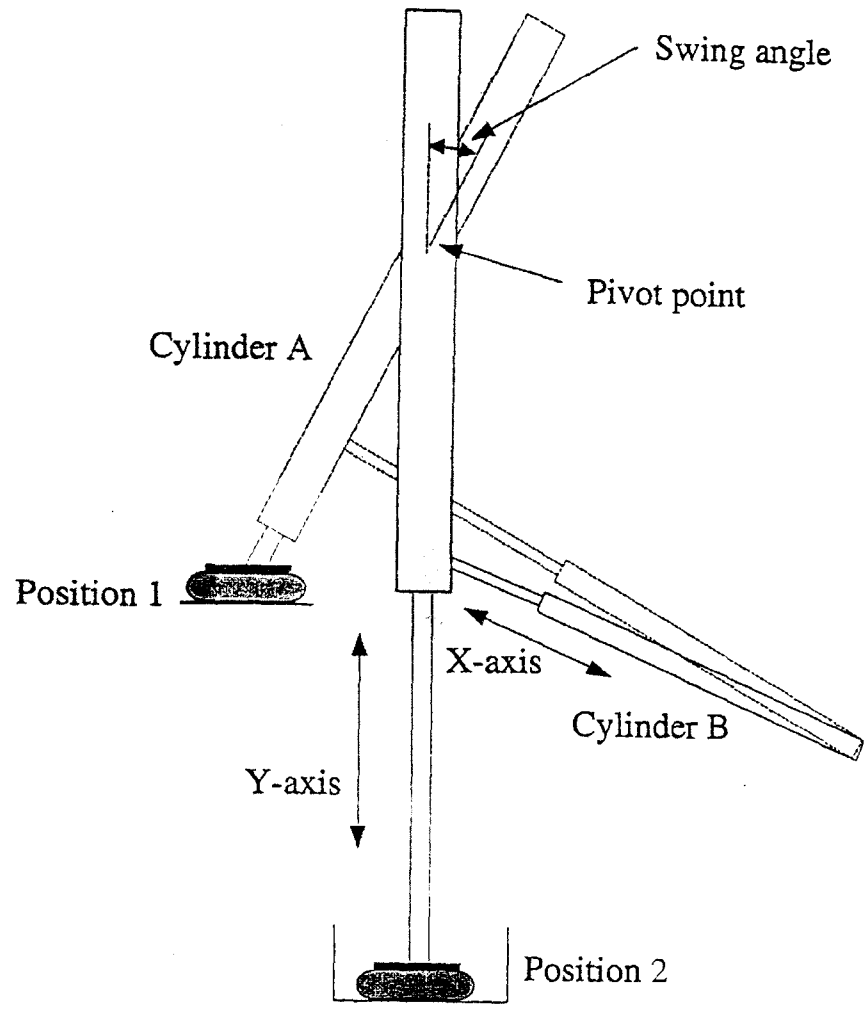
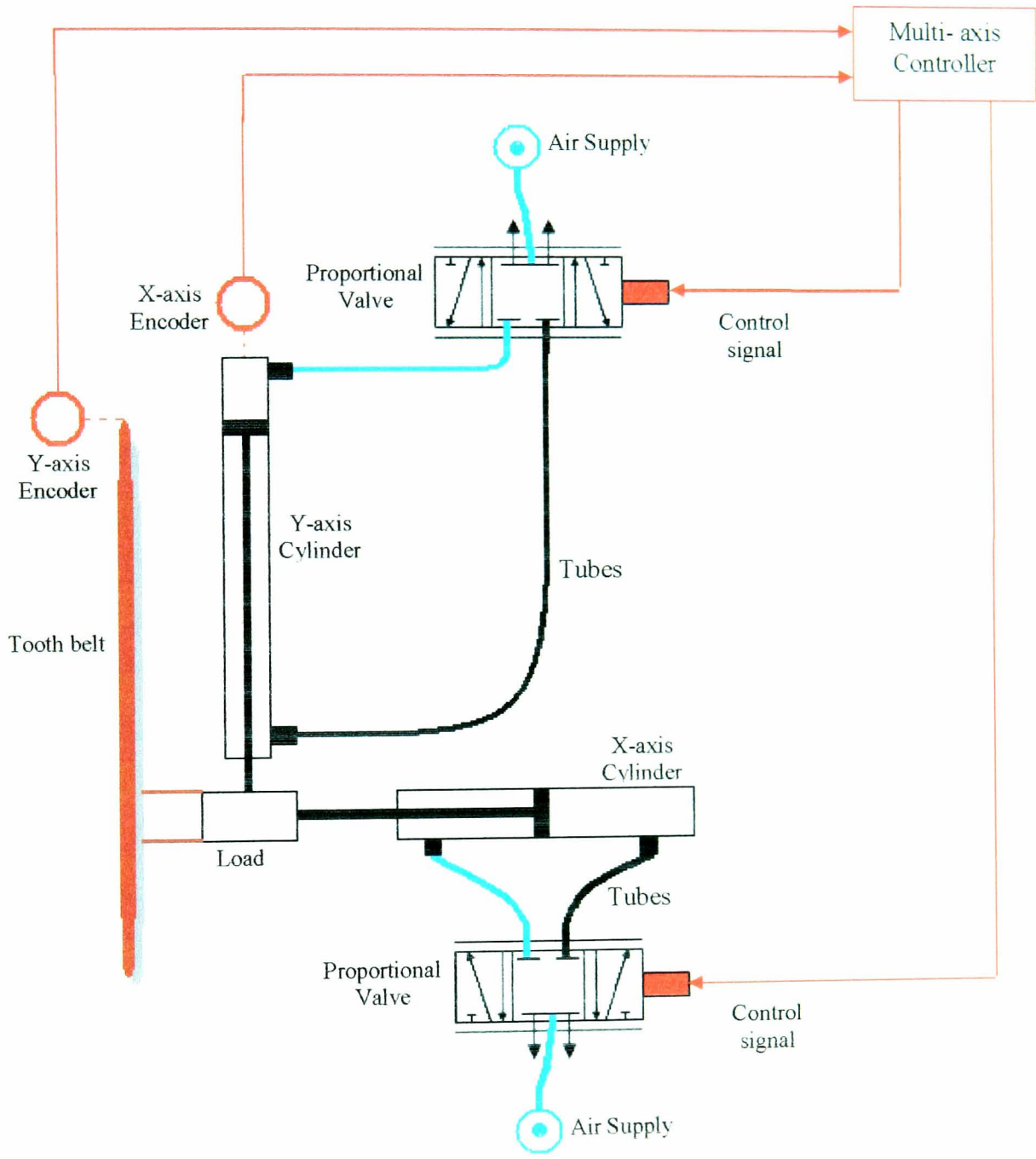


Figure 9.1 Schematic diagram for the pick-and-place system motion

Figure 9.2 (a) depicts the control loop functional diagram. Obviously there are two servo systems for both cylinders A and B respectively. A multi-axis motion controller accomplishes the signal processing and control algorithm implementations. The X-axis encoder measures the swing angle and the Y-axis encoder measures the rod displacement. Each of these encoders generates two quadrature pulse trains as the encoder's shaft rotated. The interface electronic circuit was designed to provide up/down counting of the pulses so that the displacement of the drive could be established.

The multi-axis controller generates two sets of analogue output signals which are converted by two DC/DC converters to control the proportional valves. Therefore the

mass flow of the compressed air across the proportional valve can be controlled by manipulating the valves spools.



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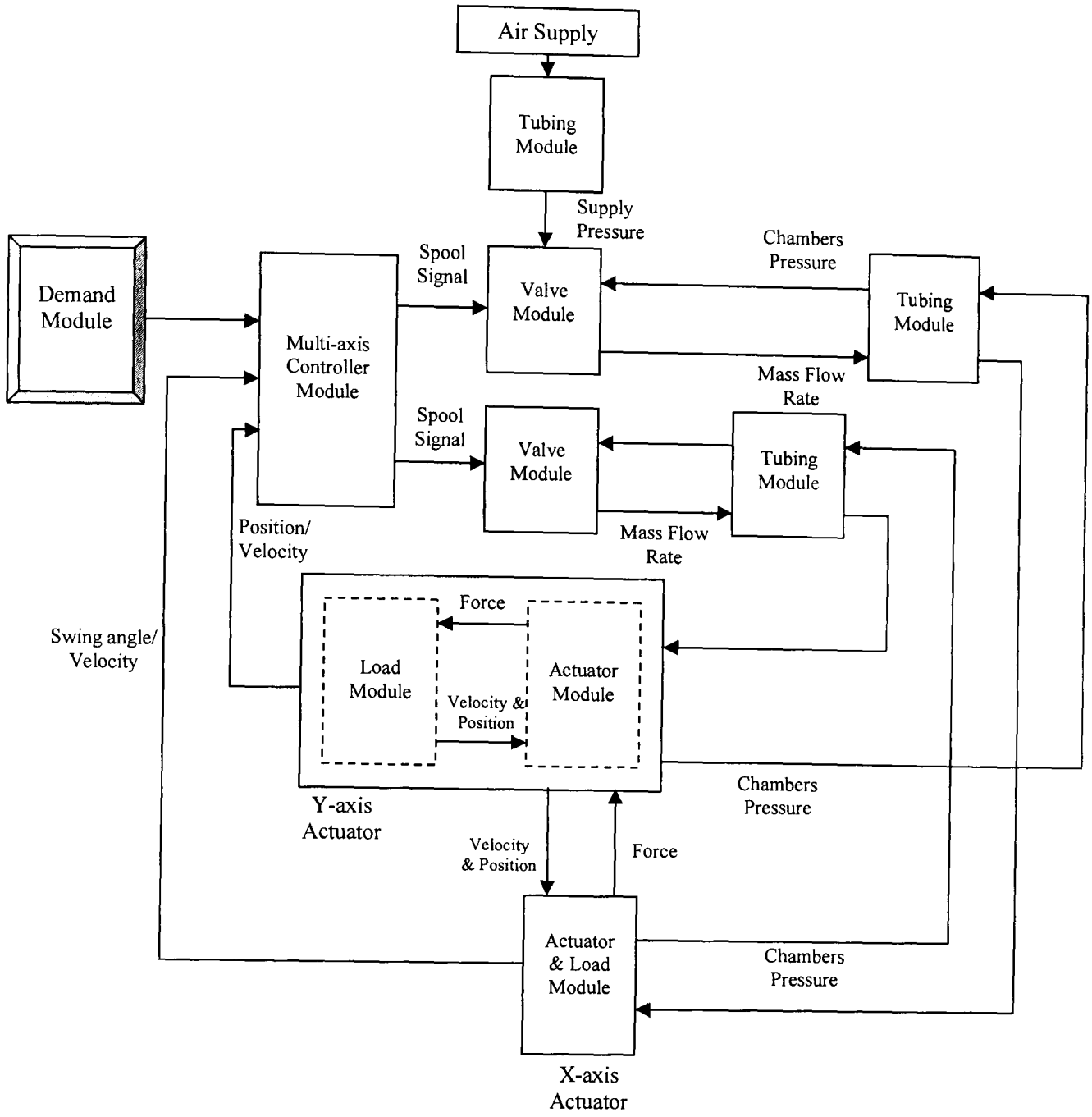

(b)

Figure 9.2 (a) Functional diagram of the pneumatic system's control loop, (b) The proposed component-based model structure for the two axis system

Figure 9.2 (b) illustrates the proposed component-based model structure for two axes servo-pneumatic system. The system has been broken down into five main components: (1) multi-axis controller component, which generates two analogue signals for both valves based on the feedback signals from the encoder, (2) valve component, two valves are required for both of the axes, (3) tubs component, which passes the air from the air supply to the valves, from the valves to the actuators, and vies versa, (4) actuator and

load component, this will mimic the Y-axis cylinder where the load is placed, and (5) actuator module, which implement the X-axis cylinder.

This model structure can be considered as a foundation concept for further developments. After this stage, it is required to establish a mathematical model for each component, and then implement it in a software package. Based on the process to create a mixed reality environment and on the components library, the CMRE for multi-axis pneumatic system can be developed.

9.3 The CMRE for Servo-hydraulic Systems

Similar to pneumatic system, the hydraulic system should be broken into a series of components which are connected to each other through well defined interfaces. This section aims at providing a model structure for servo-hydraulic system which should provide a foundation for further work in order to extend the CMRE concept for servo-hydraulic systems.

Servo-hydraulic system can be classified into three basic types: (1) valve-controlled systems, which consist of a valve controlling the flow from a hydraulic power supply to an actuation device regardless the valve type, (2) pump-controlled systems, where the hydraulic power control is performed by actuating a pump or a motor, and (3) load-sensing systems, where the flow is automatically adjusted to any load demand required by the actuators within the given pressure and flow boundaries. Nevertheless, the component-based model structure of the valve-controlled system has been considered in this chapter.

Typically, servo-hydraulic systems consist of a hydraulic actuator, controller, proportional or on-off valve, pump, reservoir, tubs and load as indicated in Figure 9.3.

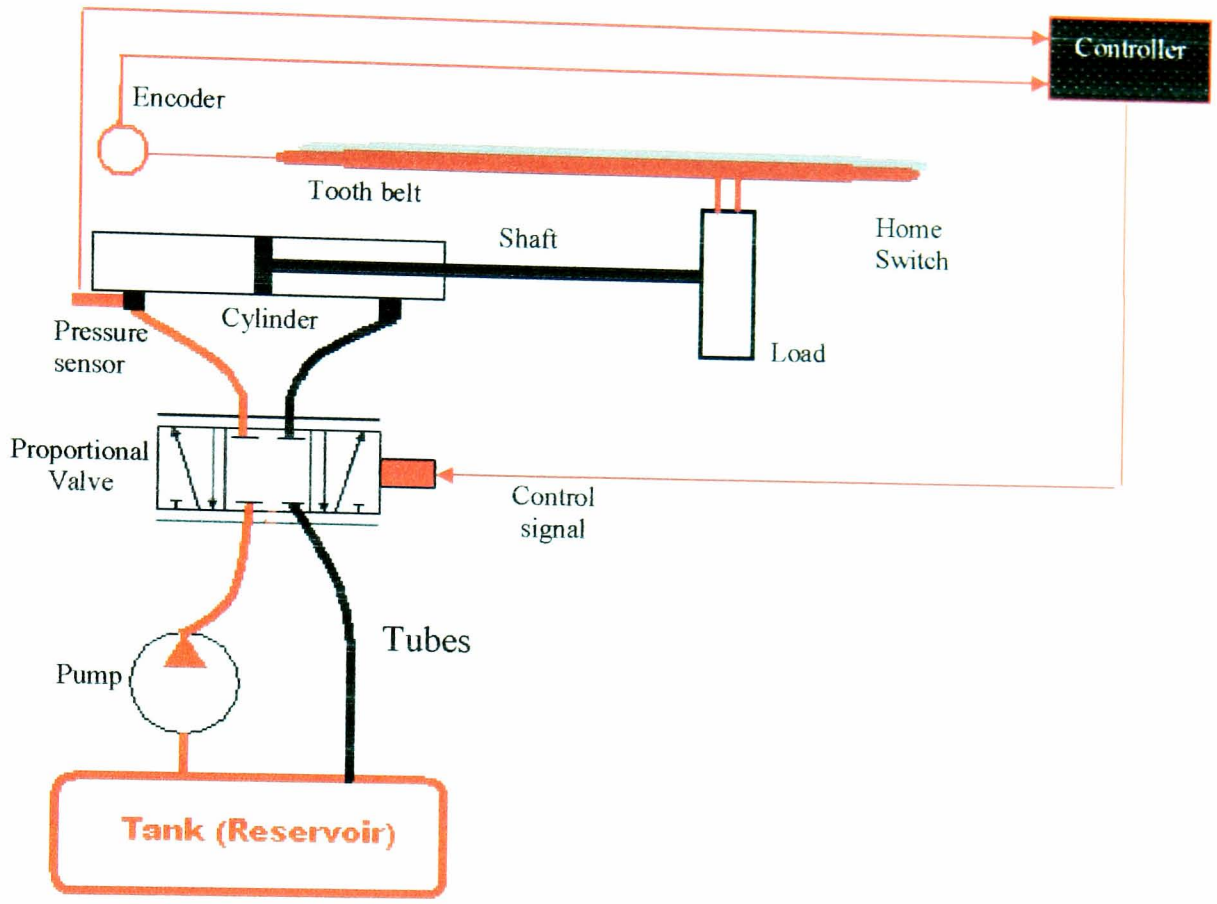

Figure 9.3 Key components in servo-hydraulic system

The model structure of the component-based open loop system structure can be formulated as shown in Figure 9.4. The system has been broken down into four main components:

(1) The valve component that provides the interface between the hydraulic pump and the hydraulic actuator which is either a rotary or linear actuator. The valve receives a control signal from the operator or the controller and adjusts the system output accordingly.

(2) The actuator and load component, similar to the one in pneumatic system, this component consists mainly of two components; the actuator that is responsible for delivering force and motion to the external load system. This force is produced as a result of differential pressure between the two chambers. The load opposes the actuator due to a force from different natures.

(3) The controller component which can be treated as a black box which transforms the position information from the load to a command signal to the valve. This

black box could represent any control methodology (such as PID, fuzzy, neural, sliding mode controller) which suits a particular application.

(4) The tubes between the pump and the valve, as well as the tank and the valve are to be considered especially when the distance between the main reservoir and the valve is long. Moreover, the tubes between the valve and the actuator should be considered, which is mainly important when the actuator has a long stroke.

(5) The hydraulic pump component, which can be regarded as the power source for hydraulic systems. The main task of the pump is to convert mechanical energy to hydraulic energy that may be controllably transmitted downstream to other output devices.

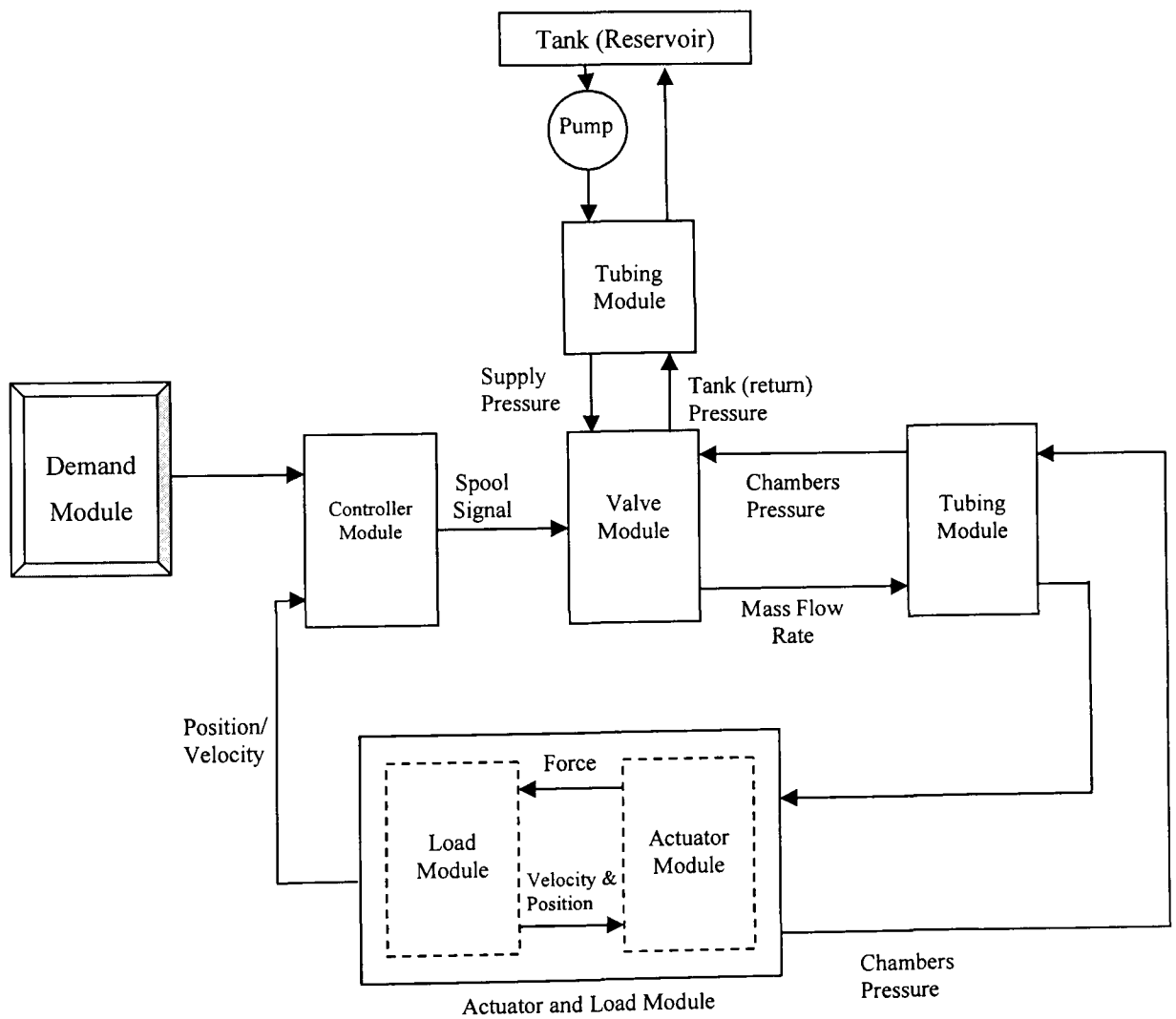

Figure 9.4 The proposed component-based model structure for servo-hydraulic systems.

The connection signals of the components are derived from the inter-relationship of the successive components. These consist of:

(1) Hydraulic link between the valve and the actuator in the form of oil volume flow rates.

(2) Hydraulic link between the pump and the tubes, the tubes and the valve in a form of pressure and temperature.

(3) Hydraulic link between the valve and the reservoir for the return pressure in a form of pressure and temperature.

(4) Hydraulic link between the actuator and the tubes, the tubes and the valve in a form of pressure and temperature.

(5) Mechanical link between actuator and load in terms of force and acceleration.

9.4 The CMRE for Servo-electric Systems

Figure 9.5 shows the key components in servo-electric system. In response to an input command, servo-electric drives efficiently control the speed and the position of the mechanical load, consequently eliminating the need for the valve. The controller, by comparing the input command for speed and position with the actual values measured through sensors, provides appropriate control signals to the power-processing unit (PPU) consisting of power semiconductor devices.

The PPU gets its power from the utility source with sinusoidal voltages of a fixed frequency and amplitude. In response to the control input, the PPU converts these fixed-form input voltages into an output of appropriate form (in frequency, amplitude, and the number of phases) that is optimally suited for operating motor. The input command to the servo-electric drive may come from a process computer, which considers the objectives of the overall process and issues a command to control the mechanical load.

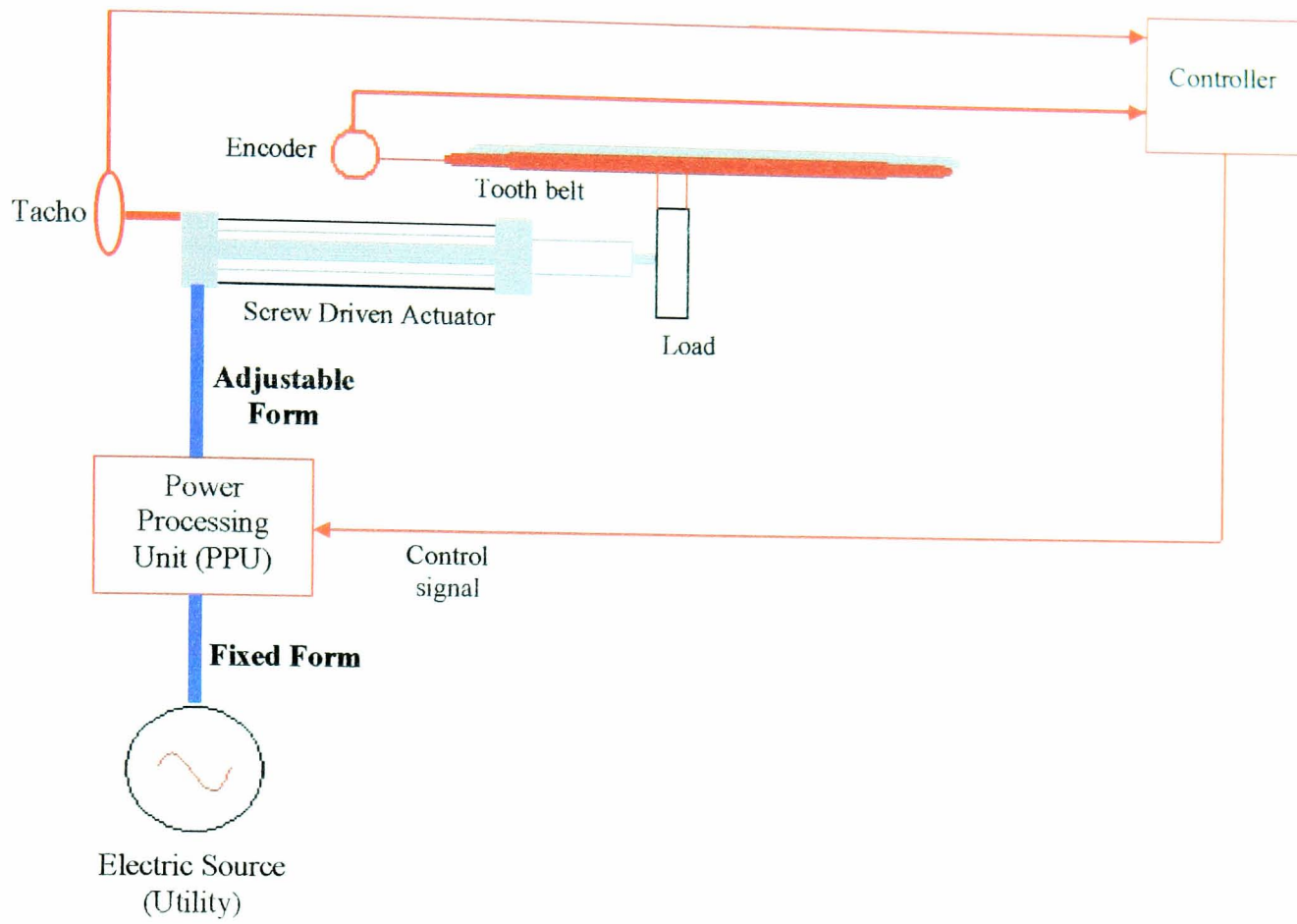

Figure 9.5 Key components in servo-electric system

The component-based structure is depicted in Figure 9.6. The structure is simpler than the one created for pneumatic and hydraulic system. It comprises of three main components: (1) the controller component, (2) power processing unit, and (3) actuator and load component. The connection signals of the components are derived from the inter-relationship of the successive components. These consist of:

(1) Electrical link between the PPU and the actuator components in a form of power signal with appropriate frequency, amplitude, and the number of phases according to the control input.

(2) Electrical link between the electrical source and the PPU component in a form of power signal with single-phase or three-phase sinusoidal voltages of a fixed frequency and constant amplitude.

(3) Mechanical link between actuator and load in terms of force and acceleration.

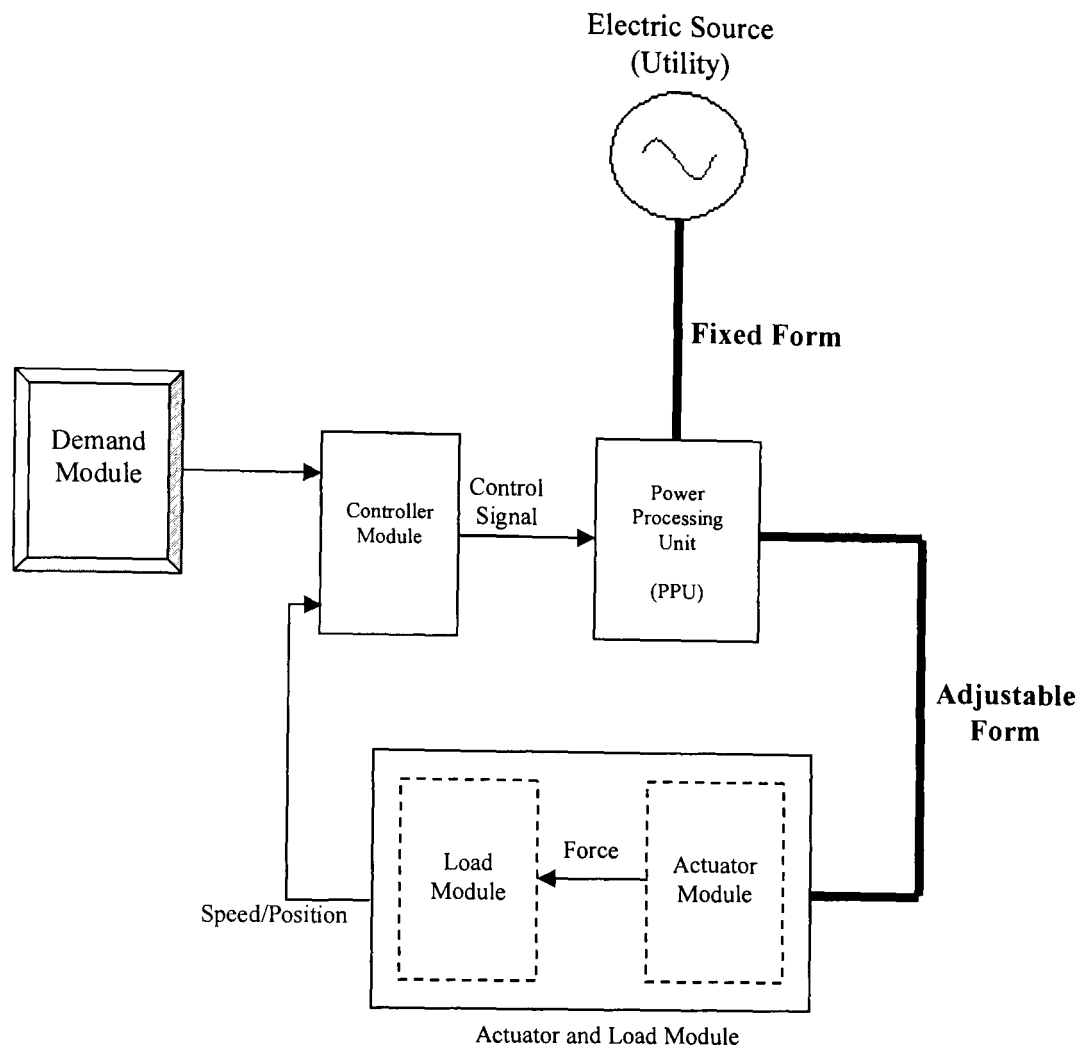

Figure 9.6 The proposed component-based model structure for servo-electric system

9.5 Further Areas for Future Research

More possible directions for further research would include component library expansion, further research on components modelling, system support service strategy, and different tuning strategies implementation.

- **Components library expansion**: the concept is to classify the components into groups. Each group should contain a wide variety of local components. For example, the actuators group may contain rotary components such as air motors and rotary chambers which are often used in conjunction with linear components to provide cylindrical working envelop. Moreover, valves can vary from the spool valve used in this research to poppet or flopper valves. Several model templates are required for developing the wide variety of valve configuration. The component library should

also include more controller components related to different control strategies and methodologies such as gain-scheduling, fuzzy logic, neural network, etc.

- **Further work on some components modelling:** further research is required on the following issues:
    - Tubes modelling, specifically on the propagation of the pressure wave in the tubes so that the proper tubing model can be established.
    - The internal leakage between the two cylinder chambers.
    - The valve operator or the spool mechanism
- **System service support strategy:** further research is required for developing a service support strategy by employing the developed tools to enhance the system maintenance and service. It is worthwhile to think of introducing a web-based remote maintenance support strategy/tool for servo-pneumatic systems. The web-based usually provides a remote user interface, which allows maintenance remote experts to access and visualize equipment operational data and analysis outputs of automated maintenance-support-modules. The automated maintenance-support-modules are mostly based on the combination of a number of software components performing various functions such as: digital signal processing, feature extraction, classification, diagnostics, prognostics etc.
- **Different tuning methodologies implementation:** tuning methods described in chapter 3, such as gain-scheduling, fuzzy logic, neural network, etc.., can be implemented using the developed simulation environment. It is advisable to spend more efforts on this aspect.

9.6 Summary

This chapter presented some foundation concepts in order to extend the developed CMRE to be applied in multi-axis servo-pneumatic system, servo-hydraulic, and servo-electric drives. The chapter introduced the proposed component-based structure for each of the three systems. These structures can be considered as the basis to create the environment.

Further work is required to establish a mathematical model for each and every component within the proposed structures. Then, by following the development process that has been

applied in this research for servo-pneumatic system, the simulation environment can be created for the hydraulic and electric servo drive. The chapter also presents some opportunities for future research.

Chapter 10
Conclusions

10.1 Research Findings

- The mixed reality concept and the component-based approach have been successfully adopted to create and implement the Component-based Mixed Reality Environment (CMRE) for the design and control of servo-pneumatic systems. A number of component modules were established under the criteria of the approaches. The components were defined clearly with the interfaces associated with the traceable physical parameters in the real system. Adopting these techniques has provided the following potentials to the CMRE:
  (1) Component-based model is valid over the whole working range of the servo-pneumatic drive. This is because of considering the nonlinearity phenomena of servo-pneumatic drive without the need for linearization; this was presented in chapter 5.
  (2) Provide flexibility of configuration choice and direct interpretation because the system is broken down to subsystems or entities. Refer to chapter 4 and 6.
  (3) Changes on the real system components can be reflected easily on the model. This was shown in chapter 6 and 7.
  (4) Facilitate the implementation of different control strategies/methodologies in order to allow for variation in future. This was proved by implementing two control strategies; PIDVF and cascade structure.
  (5) Provides the ability to interact with real system, which facilitates the identification of the system parameters and the implementation of controller tuning procedure. This was explained in chapter 6 and 7.

- It has been found that the CMRE can provide an effective environment for servo-pneumatic system control and design. On the one hand, it has the potential for allowing various control strategies and methodologies to be implemented/synthesised in a more integrated and coordinated manner. On the other hand, rather than building systems

from scratch, designers can rapidly realise a servo-pneumatic system off-line by means of assembling existing virtual pneumatic components from the components library after setting the required parameters. This can be performed using the enhanced graphical representation and visualisation provided by the software platform which facilitates the discovery of potential logic errors. Compared to the traditional design process, developing the servo-pneumatic system utilising the CMRE has big advantages in terms of:

(1) Reducing machine development time.

(2) Cost-effective.

(3) Reducing optimization time

(4) Reducing the hazard of optimization online.

As a result of improving the control and design of the system, the CMRE has improved the life cycle of the system. More specifically it has improved the system design phase, software components integration and testing phase, and machine service support phase. More details can be found in chapter 8.

- Decomposing the servo-pneumatic system based on the component-based framework introduced in Chapter 4 was successfully performed. However, some components have been moved from the Machine modular layer directly to the Device Component layer with skipping the Composite component layer. This means that servo-pneumatic systems can not be decomposed into composite components and then into basic components since it consists of basic and composite components in the machine level. Chapter 4 presented the system components after the decomposition process and the dependency relationship between these components.

- Coulomb and static frictional parameters are considered as one of the nonlinear aspects of the pneumatic systems. To acquire system friction parameters accurately could be extremely difficult once the system has been assembled, which causes a great difficulties in pneumatic systems modelling and control. In this research, it was proved that the CMRE assists in identifying these parameters. This resulted in attaining more

accurate model for the system, and therefore increasing the efficiency and reliability of the CMRE in evaluating and implementing different control schemes on the real system. A detailed demonstration has been presented in chapter 7 to show how the CMRE can be employed to apply different control schemes on the real system. This has been shown via the implementation of the PIDVF and cascade control schemes.

- From the control perspective, the tuning procedure of the controller is complex due to the extraordinary non-linearities associated with servo-pneumatic systems. However, it has been shown how the end-user can tune the system with confidence, ease, and high efficiency without the need for specialist. A well tuned servo-pneumatic become easier to achieve, which would increase the understanding and confidence of the users, and hence increase the success of the pneumatic technology in the market.

10.2 Research Contributions

- This research has contributed to knowledge in adopting the mixed reality concept and the component-based approach in order to create the CMRE that facilitates the control and design of servo-pneumatic systems, and hence improves their life cycle.
- A new method for system's friction parameters identification is proposed. The results revealed that a relatively high accuracy in friction parameters identification can be achieved using the proposed method in conjunction with the CMRE.
- Encapsulate existing control methods within the CMRE to be applied on the real system. This was demonstrated by implementing PIDVF and cascade control schemes on servo-pneumatic systems. The results have shown that the CMRE has facilitated the application of different control strategies with ease and high efficiency.
- A scheme to facilitate the controller tuning using the CMRE. It has been shown that from the end-user perspective, any tuning procedure can be applied using the developed environments with confidence, ease, and high efficiency to attain the demanded performance. This has been demonstrated by employing two different tuning strategies; Ziegler-Nicholas method and Åström (relay) method.

10.3 Conclusion

With the success of adopting the concept of the mixed reality environment and the component-based approach to create the CMRE, which improved the control and design of servo-pneumatic system, and the improvement achieved in the frictional parameters identification, it is reasonable to conclude that the aim and objectives set forth at the start of the project have been fulfilled and that the development of the software components has provided a valuable foundation for further refinement to become an optimum environment for servo-pneumatic system design and control. Laboratory experimentation has demonstrated the potential of the CMRE for facilitating and improving the control of servo-pneumatic system, and consequently improving its life cycle. The flexibility of the approaches utilised in this research allowed for providing a conceptual model structures to extend the idea of the CMRE to include multi-axis pneumatic systems, servo-hydraulic systems, and servo-electric systems

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Appendix I

Recent Pneumatic Components in the Market

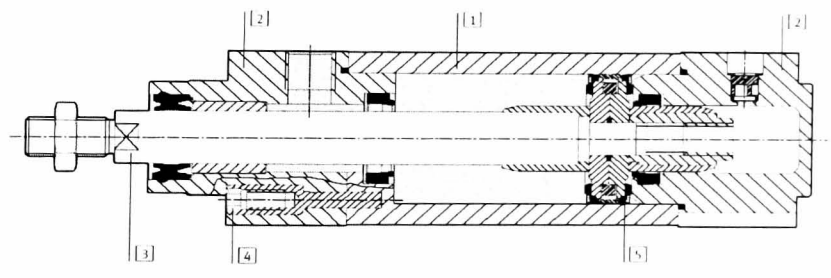
AI.1 Introduction

This appendix presents the latest pneumatic components technology that available in the market. The survey was conducted in Jan/2005. It focused on the products from the leading manufactures for pneumatic components; Festo, Origa, and Bosch. ***All the technical data, information, and supported pictures, have been obtained from their products catalogues***.

AI.2 Latest Pneumatic Components from Festo

AI.2.1 Standard cylinder (DNCB)

The latest member of the standard DNC cylinder range - the DNCB is recommended for simple tasks. The new system piston with its running characteristics is far more compact and absorbs far more energy. The space saved allows a greater bearing length. Naturally, all the existing accessories and mounting components can be used for this member of the DNC range too. Figure AI.1 shows a cross sectional view of this cylinder and Table AI.1 summarises some of the characteristics of DNCB cylinders.



Cylinder		
[1]	Cylinder barrel	Wrought aluminium alloy
[2]	Bearing/end caps	Wrought aluminium alloy
[3]	Piston rod	High alloy steel
[4]	Flange screws	Tempered steel
[5]	Dynamic seals	Polyurethane
-	Static seals	Nitrile rubber
-	Lubricant	Kluberplex BE 31-102
	Note on material	Free of copper and Teflon

Figure AI.1 cross sectional view of the DNCB cylinders

Table AI.1 the characteristics of DNCB cylinders

Criterion	Feature			
Model	DNCB-32-PPV-A	DNCB-40-PPV-A	DNCB-50-PPV-A	DNCB-63-PPV-A
Stroke	1-2,000 mm			
Piston Diameter	32 mm	40 mm	50 mm	63 mm
Cushioning	Pneumatic cushioning, adjustable at both ends (PPV)			
Assembly position	Any			
Design structure	Piston Piston rod Piston barrel			
Position detection	With proximity sensor			
Operating pressure	0.6 - 12 bar			
Mode of operation	Double-acting			
Operating medium	Dried compressed air, lubricated or unlubricated			
Ambient temperature	-20 – 80 °C			
Theoretical force at 6 bar, return stroke.	20 N	20 N	22 N	22 N
Theoretical force at 6 bar, advance stroke.	415 N	633 N	990 N	1,870 N
Moving mass with 0 mm stroke	483 N	754 N	1,178 N	1,870 N
Additional weight per 10 mm stroke	108 g	204 g	363 g	460 g
Basic weight for 0 mm stroke	460 g	760 g	1,225 g	1,800 g
Pneumatic connection	G1/8	G1/4	G1/4	G3/8
Material information for piston rod	High alloy steel			

AI.2.2 DNCI Standard Cylinders

- DNCI is the first piston-rod drive from Festo with an integrated displacement encoder. It contains sensor head in the bearing cap and a reference strip on the piston rod, which result a robust, simple and compact piston-rod drive with the necessary versatility for applications such as dosing/filling plants, welding devices, deflector actuators or pushers.
- The displacement encoder operates on the proximity principle and is thus free of wear. The fixed sensor head means no need for an external cable loop between the sensor and controller.

- This system eliminates any possibility of installation errors. The mounting accessories and guide units from Festo's modular DNC system can be used. The cylinders have the same dimensions as the DNC standard cylinders to ISO 6431, thus ensuring easy interchange ability.
- The DNCI allows free positioning, measuring and closed-loop-controlled tasks and the controlled acceleration and deceleration of large masses. The cylinders are operated in conjunction with SPC11 or SPC200 axis controllers. All the technical data are shown in Table AI.2.

Table AI.2 technical data for DNCI

Cylinder				
Sizes	32/40/50/63			
Stroke length	100, 160, 200, 250, 320, 400, 500 mm			
End-position cushioning	Elastic			
Sensor head cable outlet	At side of bearing cap			
Displacement encoder				
Repetition accuracy	< 0.1			
Electrical connection	M12			
Degree of protection	IP65			
Working temperature range	-20 – 80 °C			
Positioning data				
Diameters [mm]	32	40	50	63
Mounting position	Any			
Smallest load mass, horizontal	3	5	8	12
Largest load mass, horizontal	45	75	120	180
Smallest load mass, vertical	3	5	8	12
Largest load mass, vertical	15	25	40	60
Smallest positioning stroke	< 3% (relative to max. stroke, but not more than 20 mm)			
Repetition accuracy	< 0.4			
Stroke reverse	10	10	15	15
Min. travel speed	< 0.05 m/s			
Max. travel speed	> 1.5 m/s			

AI.2.3 Standard cylinders ADN/AEN

- Diameters ranging from 12 to 125 mm. Extensive range of variants can be configured to meet individual requirements.
- Comprehensive range of mounting accessories for just about every installation situation.
- High speeds and machine cycles leads to excellent running characteristics and outstanding cushioning characteristics.
- Up to 50 % less fitting space compared with standard cylinders ISO 6431
- Systematically thought out - 1 proximity sensor, can be used on three sides (type 8 slot) for all sizes and many other cylinder families

Table AI.3 ADN cylinders technical characteristics

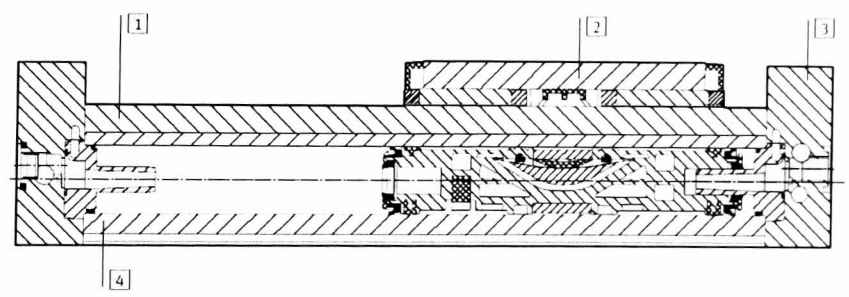
Criterion	Feature			
Model	ADN-12-	ADN-20-	ADN-25-	AND-40-
Stroke	1- 300 mm			
Piston Diameter	12 mm	20 mm	25 mm	40 mm
Cushioning	Pneumatic cushioning, adjustable at both ends (P)			
Assembly position	Any			
Design structure	S2: through piston rod K2: piston rod with extended external thread K5: special thread on piston rod K8: extended piston rod Single-ended piston rod			
Position detection	With proximity sensor			
Operating pressure	1 – 10 bar			
Mode of operation	Double-acting			
Ambient temperature	-20 – 80 °C			
Impact energy in end position	0.7 J	0.2 J	0.3 J	0.7 J
Theoretical force at 6 bar, return stroke.	51 N	141 N	247 N	686 N
Theoretical force at 6 bar, advance stroke.	51 - 68 N	141 - 188 N	247 - 686 N	686 – 754 N
Moving mass with 0 mm stroke	483 N	754 N	1,178 N	1,870 N
Additional weight per 10 mm stroke	108 g	204 g	363 g	460 g

Basic weight for 0 mm stroke	460 g	760 g	1,225 g	1,800 g
Pneumatic connection	M5	M5	M5	G1/8
Material information for piston rod	High alloy steel			

AI.2.4 Rodless Cylinders DGC

- DGC 8 - 40, the new rodless cylinders as basic drives, with plain-bearing or recirculating-ball guides, and with the option of shock absorbers and a 2nd slide. A common feature of all variants is better guidance quality and higher characteristic load values and torque capacity.
- Minimised installation space requirements and proximity sensors integrated into the cylinder profiles which provide faster installation.
- All accessible from the same side: End-position fine adjustment, sensor, mounting, air connections and pneumatic end position cushioning.
- There is a choice of foot mountings or profile mountings. These latter allow the cylinder to be mounted by means of a dovetail slot directly on the profile, which provide High strength and easy installation and replacement.
- Precise mounting interfaces with centring rings on the slide made of stainless steel and end caps with integrated shock-absorber brackets.

The cross sectional view of the DGC rodless cylinders is shown in Figure AI.2.



Cylinder		
1	Guide rail	High-alloy steel
2	Slide	High-alloy steel
3	End cap	Anodised aluminium
4	Cylinder barrel	Anodised aluminium
-	Piston seal	Polyurethane
-	Sealing band/cover strip	Polyurethane
-	Note on materials	Free of copper and Teflon

Figure AI.2 cross sectional view for Festo's rodless cylinders

AI.2.5 Linear Module HMP

HMP is a range of pneumatically driven linear axes for handling and assembly technology. By means of adapters, individual axes can be combined to form multi-axis devices. These can then be further completed with various profile components to form pick-and-place units. The HMP series forms the core of the modular system for handling and assembly technology (HAT).

HMP has been further expanded to include the following features:

- Longer strokes for diameters HMP-16 and HMP-20
- Sensor rail for all diameters and strokes
- Handling shock absorbers for all sizes
- Intermediate positions, option of up to 3 intermediate positions

Strengths:

- Exceptionally high guide precision
- Very low distortion with high torques
- Very short traversing times
- High repetition accuracy even at very high velocity
- Long strokes
- User-friendly installation technology
- Hi-tech design
- Infinitely adjustable stroke (between double shock absorber stroke and maximum stroke)
- Optimum cushioning characteristics thanks to the progressive action of the special handling shock absorbers
- Visual motion indication via LEDs
- Sensor rail with 5 SMx-8 sensors for axes configuration
- Mid position with up to 3 intermediate positions
- Integrated clamping unit for safety in case of emergency stop (optional)
- Choice of connection technology on mechanical interfaces

- Flexible cable guide system.

Table AI.5 the features of HMP module

Criterion	Feature			
Model	HMP-16-	HMP-20-	HMP-25-	HMP-32-
Mode of operation	Double-acting			
Operating mode of drive unit	Yoke			
Piston diameter	16 mm	20 mm	25 mm	32 mm
Ambient temperature	0 – 60 °C			
Position detection	With proximity sensor			
Operating pressure	4,000 – 8,000 bar			
Theoretical force at 6 bar, advance stroke	121 kN	188 kN	295 kN	483 kN
Theoretical force at 6 bar, return stroke	104 kN	158 kN	247 kN	415 kN
Operating medium	Filtered compressed air			
Corrosion resistance classification CRC	2			
Protection against torque/guide	Guide rod with yoke			

AI.2.6 Electronic End Position Controller

(Electronic end position controller Smart Soft Stop SPC11)

- Tuning for pneumatic drives that cuts travel times by up to 30 % and dramatically reduces vibration in the approach to the end position.
- Up to two freely selectable intermediate positions without fixed stop for ejector or rest positions. Replaces in-house constructions and expensive electromechanical drive solutions.
- Minimises maintenance costs due to the reduced vibration load.
- The Smart Soft Stop system enables the travel time from A to B to be reduced by up to 30 % and improves cushioning characteristics in the end position for linear and semi-rotary drives - thereby increasing the service life of the drives.
- Reduced travel times, fewer vibrations and longer system service life for optimum production and greater productivity, easy, quick installation and commissioning without costly in house constructions.

AI.2.7 Smart Positioning Controller (SPC 200)

(SPC200: Flexibility and maximised dynamism for servo-pneumatic positioning technology)

The Smart Positioning Controller SPC-200 is a universal axis controller for pneumatic positioning axes. It includes position and positioning control for up to 4 axes. It consists of a basic device with two different sizes, to which either 4 or 6 plug-in modules with various functions can be installed. A stepper motor controller is also possible.

The SPC200 is connected to the periphery of the respective pneumatic axis via a cable and the axis interface, and includes, amongst other components, displacement encoders and proportional valves for controlling the axis.

Some of the SPC200 Features are:

- High performance, self-tuning control technology for the pneumatic axis
- The controller is optimised for 2 axis systems with pneumatic drives.
- Expandable to 4 axes
- Supports series DGP linear drives with diameters from 25 to 63 mm
- Supports DNC/DNCM piston rod drives with diameters from 32 to 63 mm
- Supports DSMI rotary drives with 25 and 40 mm diameters
- Supports the new SEC-ST stepper motor controller with DGE drive range
- Depending upon the utilised drive, travel speeds of up to 3 m/s, acceleration values of up to 30 m/s² and accuracies ranging from ± 0.2 mm to ± 0.8 mm can be achieved.
- Incorporation of CP valve terminal products for actuating grippers, lift cylinders and semi-rotary drives.
- Modular design: analogue inputs, digital inputs and outputs, fieldbus interface for Profibus, DeviceNet or Interbus, stepper motor plug-in module
- Profibus with extended I/O mode allows for direct reading and writing of position data.
- The "set selection" mode is suitable for very simple positioning tasks. 32 motion sets can be controlled via I/O signals, or 1000 via fieldbus interface.

- The user is able to freely create up to 100 programs in the "start-stop" mode, which includes more than 30 commands for complex automation tasks.
- All SPC200 configuration data can be displayed and edited at the simple control panel.
- WinPISA assures easy commissioning, programming and diagnosis, and can be used with Windows 98, NT, 2000 and XP.

AI.2.8 High-Speed Pick & Place HSP

- 100 parts/min
- Independent adjustment of the Y and Z strokes in the pick or place position. The teach-in positions and speeds mean that the electrical variants can achieve greater functionality.
- The HSP permits the design of more compact machines without making any concessions to clarity. The simple project engineering tool also saves time and costs.
- The installation tubing holds all the tubes and cables and routes them to the rear. Matching grippers and rotary drives.

All the technical characteristics for HSP are shown in Table AI.6.

Table AI.6 HSP Technical Features

Criterion	Feature
Size	25
Y Stroke	130 – 170 mm
Z Stroke	50 – 70 mm
Z stroke, working stroke	25 mm
Cushioning	Soft characteristic Shock absorber on both ends (CC)
Assembly position	Guide rail Vertical
Mode of operation	Double-acting
Design structure	Cross guide Swivel model Positively-driven motion sequence

Position detection	With proximity sensor
Operating pressure	4 – 8 bar
Minimum cycle time	1 s
Repetition accuracy	± 0.02 mm
Operating medium	Compressed air, filtered, unlubricated Filtered, lubricated
Ambient temperature	0 – 60 °C
Max. torque Mx dynamic	3.2 Nm
Max. torque Mx static	15 Nm
Max. torque My dynamic	3.2 Nm
Max. torque My static	15 Nm
Max. torque Mz dynamic	3.2 Nm
Max. torque Mz static	15 Nm
Theoretical force at 6 bar	65 N
Pneumatic connection	M5

AI.2.9 Proportional Pressure Regulator VPPE

- Proportional pressure regulator with excellent characteristics for simple control tasks.
- Compact design, ready for use in a variety of different applications, particularly where an optimal cost-benefit ratio is a primary consideration.
- The VPPE is ideal for almost any sector or application where the price-conscious are looking for a solution.
- Can be connected in series with up to 8 valves.

Figure AI.3 shows the internal structure of the VPPE valve

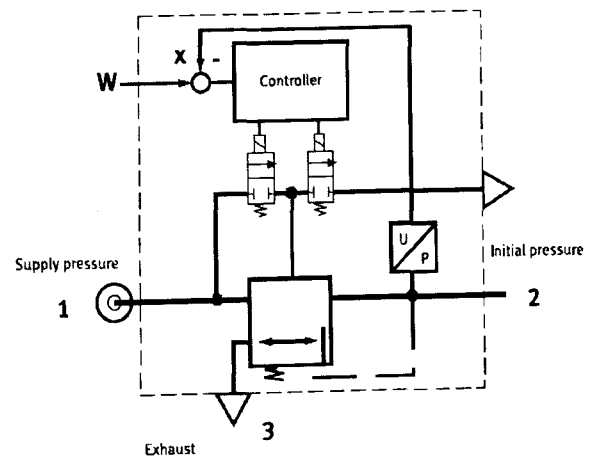


Figure AI.3 VPPE Valve Internal Structure

Table AI.7 technical data for the VPPE valve

Technical data	VPPE 1/8
Control signal	0 – 10 V
Ambient temperature	+10 °C – 50 °C
Operating medium	Filtered unlubricated air (40 µm)
Max. operating pressure (p1)	10 bar
Output pressure p2 min.	0.2 bar
Output pressure p2 max.	6 bar
Flow rate	600 ml/min with ref. to p2= 6 bar
Nominal value input	Analogue 0 – 10 V (differential input)
Accuracy	2 %
Nominal temperature	25 °C
Output connection	1/8" female thread
Input connection	1/8" female thread
Block design with no. of valves	1, 2, 4, 6, 8
Fitting position	Any
Response time	Max. 2 s on 95% nom. value step
Pressure retained in event of power failure	Yes, loss of pressure by leakage
Operating voltage	24 V DC ±10%
Nominal current	0.05 A
Max. current	0.1 A
Duty cycle	100% may be operated for several days
Degree of protection	IP65
Connection	4-pin M12 plug

AI.3 Pneumatic Components from Origa System Plus (OSP)

AI.3.1 Rodless Pneumatic Cylinders

The Origa rodless pneumatic cylinder OSP-P has many advantages, it has standard magnetic piston in which a non-contact position sensors may be fitted to three sides of the actuator. Furthermore it contains internal air channels in the profile section enable the connection of the air supply to one end only, whereas air supply to the other end is via an internal air passages. This is making it useful for situations where shortage of space, simplicity of installation or the nature of the process makes it desirable. It has also extremely low friction and high seal efficiency, due to the proven Origa stainless steal sealing system. The end-piston is adjustable as standard. Finally, it has high speed, fast

response positioning capabilities, with repeatability to 0.1 mm/m in servo-pneumatic positioning systems.

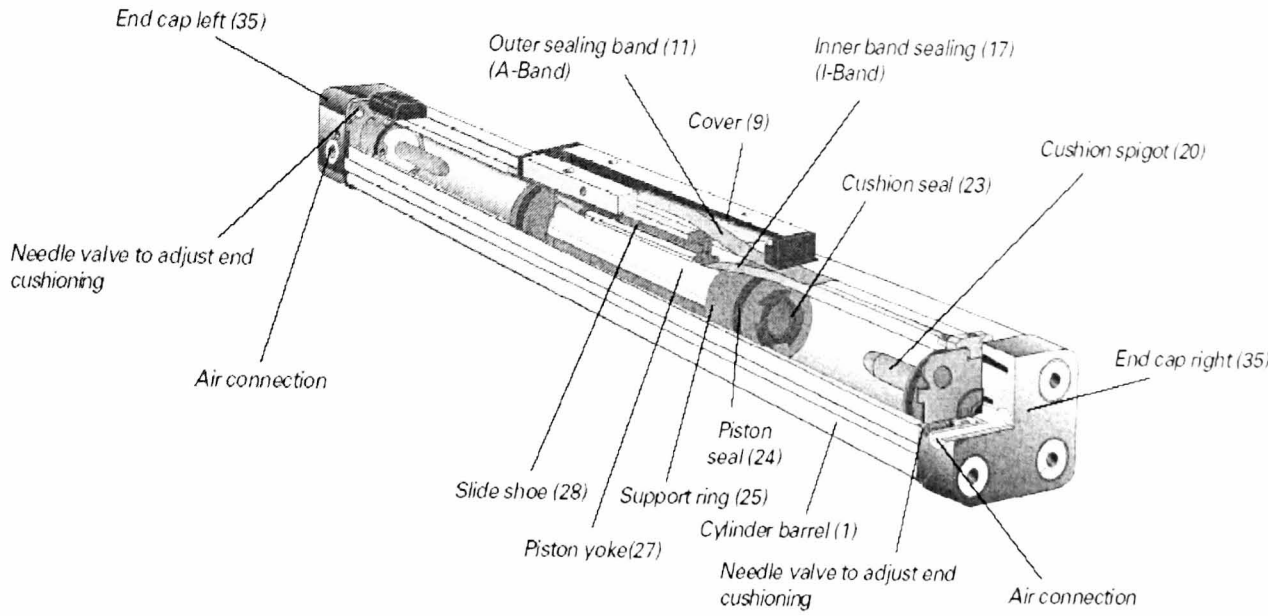


Figure AI.4 schematic diagram for an OSP-P rodless cylinder

The following table shows some technical characteristics of the OSP-P pneumatic cylinders.

Table AI.8 technical characteristics of the OSP-P series cylinder

Characteristics	**Description**
Type	Rodless pneumatic cylinder
Series	OSP-P
System	Double-acting, with cushioing, position sensing capability
Air connection	Threaded
Size	10, 16, 25, 32, 40, 50, 63, 80 mm
Stroke lengths	Free choice of stroke length up to 6000 mm, longer strokes on request
Installation	In any position
Speed	0.2 – 4 m/s, standard version 0.005 – 0.2 m/s, slow speed version max. 30 m/s high speed
Action force	62 – 3470 N depending on size at 8 bar

Positional accuracy	Standard control: up to ±3 mm Servo pneumatics: ± 0.1 mm/m stroke
Temperature range	-10 to +80 °C
Corrosion Resistance	Stainless steel screws and fittings are available option

AI.3.2 Integrated 3/2 Way Valves VOE

For optimal control of the OSP-P cylinder, 3/2 way valves integrated into the cylinder's end caps can be used as a compact and complete solution. They allow for easy positioning of the cylinder, smooth operation at the lowest speeds and fast response, making them ideally suited for direct control of production and automation.

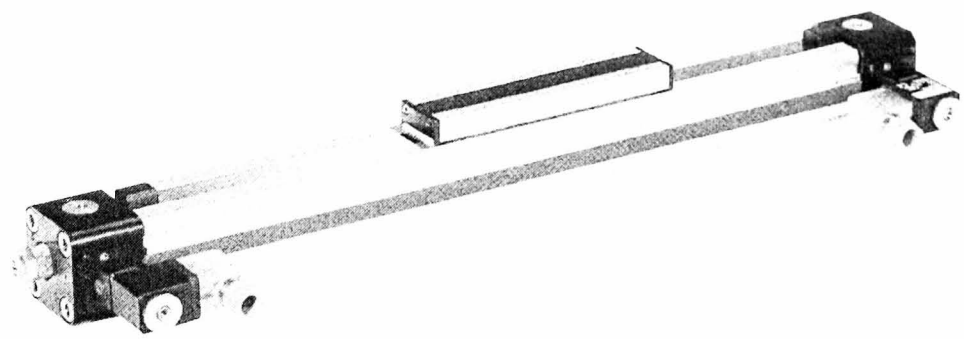
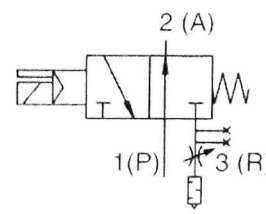
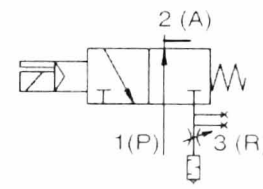


Figure AI.5 Integrated 3/2 Way Valves VOE Series

Table AI.9 characteristics of the Integrated 3/2 Way Valves VOE Series OSP-P25, P32, P40 and P50

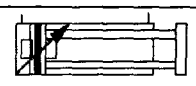
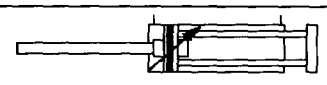
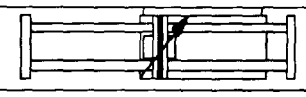
Characteristics	**3/2 Way Valves with spring returns**			
Pneumatic diagram				
Type	VOE-25	VOE-32	VOE-40	VOE-50
Actuation	Electrical			

Basic position	P → A open, R closed			
Type	Poppet valve, non overlapping			
Mounting	Integrated in end cap			
Installation	In any position			
Port size	G 1/8	G 1/4	G 3/8	G3/8
Temperature	-10 °C to +50 °C			
Operating pressure	2-8 bar			
Nominal Voltage	24 V DC / 230 V AC, 50 Hz			
Power consumption	2,5 W / 6 VA			
Duty cycle	100%			
Electrical Protection	IP 65 DIN 40050			

AI.3.3 AZV Twin Cylinder

Unlike conventional rod type cylinders, the AZV provides high load, non-rotational guidance, and is more compact and cost effective than alternative 'slide units'. Supplied with flexible clevis mounts and accessories, the AZV fully supports an applicator roller and withstands the high side loads necessary for the corner wrap operation. AZV twin rod cylinders supplied with Clevis mounts, proximity sensors and tooling plate.

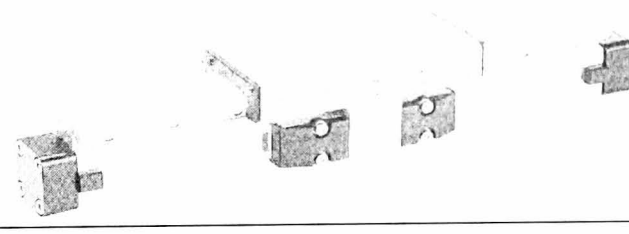
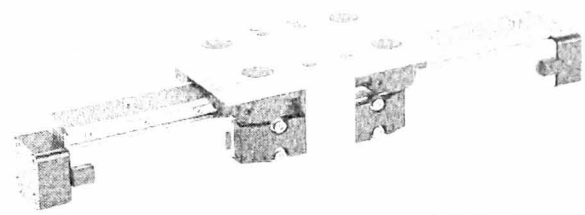
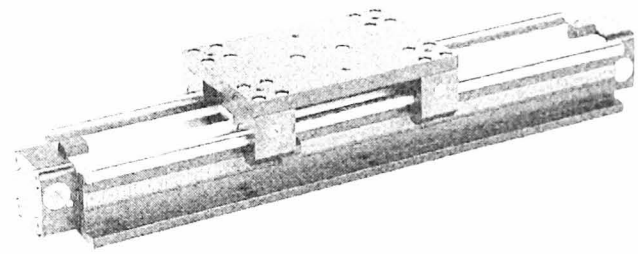
Table AI.10 technical features of the different AZV twin cylinders

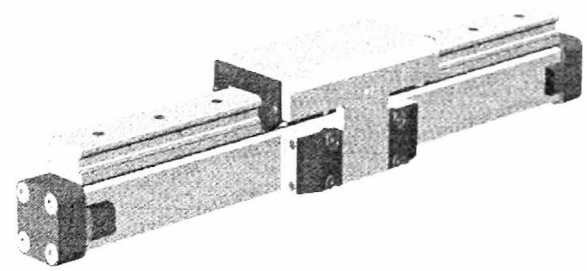
Features			
Type	Twin rod		
Series	AZV		
Configuration	AZV: Double Acting, Double Rod, Magnetic Piston, Cushions	AZV 3D: Double Acting, 3 Rods, Magnetic Piston, Cushions	AZV 4D: Double Acting, 4 Rods, Magnetic Piston, Cushions
Pneumatic Symbol			
Operating temperature	-20 °C – 80 °C		
Operating pressure	1 – 10 bar		
Normal operating pressure	6 bar		
Media	Filtered and regulated compressed air		
Installation	In any position		
Stroke length	Up to 20 inches		

AI.3.4 Linear Guides

The Origa system plus -OSP- provides a comprehensive range of linear guides for the pneumatic linear drives. It offers a pneumatic linear drives with different piston diameters: 16, 25, 32, 40, 50, 60 mm. There are many advantages for this product; they take high loads and forces, they provide a high precision with smooth operation and they can be retrofitted and installed in any position. The following table shows some different types of the OSP linear guides.

Table AI.11 different types of the OSP linear guides

Pneumatic Linear Drive – series OSP–P	
SLIDELINE The cost-effective plain bearing guide for medium loads. Brake optional	
POWERSLIDE The roller guide for heavy loads	
GUIDELINE The ball bushing guide for the heaviest loads and greatest accuracy	

PROLINE The compact aluminum roller guide for high loads and velocities. Optional with brakes.	

AI.3.5 Servo-Pneumatic Positioning Systems (ORIGA-Servotec)

The ORIGA- Servotec positioning system is an inexpensive alternative to electric positioning drives, and it is commercially available. The system consists of cylinder, measuring system, servo valves, and controller. The measuring system senses the exact position of the cylinder piston. This signal fed back to the position monitor of the controller, where the actual value is compared with the present value. The deviations found are eliminated by the extremely fast servo valves. With this constant electronic monitoring the cylinder piston is positioned within a very close rang of tolerance. The following table depicts the characteristics of this pneumatic servo-positioning system:

Table AI.12 the ORIGA-Servotec features

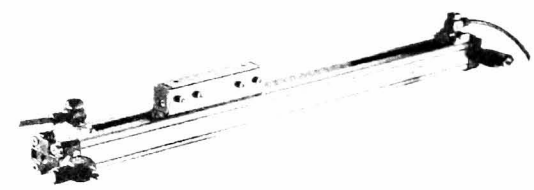
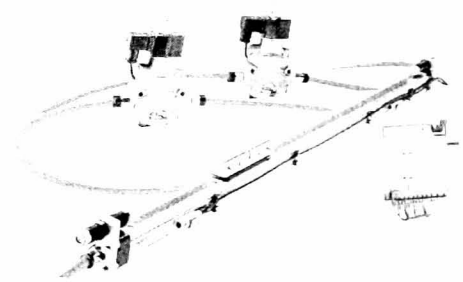
Characteristics	Description (Unit)
General Features	
Cylinder diameter	25, 32 (mm)
Positionable stroke length	0 – 3000 (mm)
Velocity	5 – 2500 (mm/s)
Reproducibility max.	± 0.1 (mm/m)
Mounting	Wall – or rail mounting (35 mm DIN-rail)
Temperature range	0 – 50 (°C)
Relative humidity	10 – 80 (%)
Pneumatic	
Operating pressure range	1 – 8 (bar)
Electric	

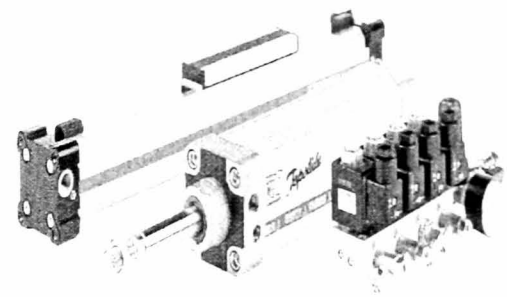
Voltage – controller	24 V (DC) (max. 50 W) 230 V (AC) (10 VA)
Voltage – control signal	Analogue: 0 – 10 / 0 – 20 mA (set value) Binary: PLC – level (24 V DC) for "enable" and "in position"
Power consumption – controller **- servo valves**	10 VA 2 x 25 W
Connection	Terminal screws
Electrical protection	42 (controller IP 20)

AI.3.6 Custom Solutions with Origa

The following table summarizes some of the custom solution for particular applications and end user requirements:

Table AI.13 examples of custom solutions

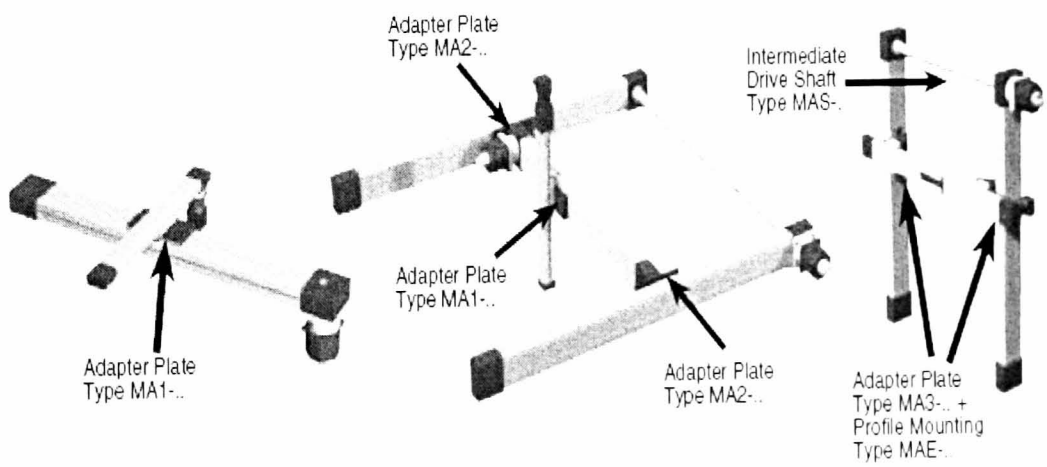
Category	Explanation	Picture
Clean Room Pneumatic Cylinders	Especially adapted for use in clean room environments, the Clean Room series incorporates wear resistant components and an in-built vacuum system which practically eliminates particle emissions from the actuator	
High Speed Pneumatic Cylinders	Developed specifically for high speed applications in excess of 10 m/s e.g. Flying knife installations. Solutions include an actuator and control package which provides for maximum air flow and the fastest possible response time.	

Door Drive Systems	Pneumatic door drive systems for commercial, industrial and transportation applications. Features may include • Extended end of stroke cushioning • Obstruction detection and auto-reverse capability. • Specialist control requirements • Electronic interface	

AI.3.7 Multi-Axis Connection System for Linear Drive Systems

Developed for the heavy duty belt drive series OSP-E..BHD, the system provides cross-connection with the same Series and also other linear drive series in the ORIGA SYSTEM PLUS range.

A wide range of adapter plates, profile mountings and intermediate drive shafts simplify engineering and installation. The connection system enables actuators to be mounted in carrier to carrier; carrier to profile; carrier to end cap mounting; and carrier to end cap configurations.



AI.4 Pneumatic Components by Bosch

AI.4.1 Rextreme Pneumatic Cylinders

Rextreme is an acid-proof pneumatic cylinder which meets all corrosion resistance levels. It has a hygienic design with smooth surfaces and positive sealing for easy cleaning. Furthermore it has been lubricated with edible grease allowing direct contact with food.

Table AI.14 features and technical information for the Rextreme cylinders

Features						
Type	Round cylinders Rextreme, series 299-21 in Acid-proof steel					
Configuration	Double acting for high temperature < 120° C, adjustable cushioning, 32-80 mm diameter					
Max. working pressure	10 bar					
Ambient pressure range	-25 °C to 120 °C					
Medium	Compressed air lubricated or non-lubricated					
Application areas	• Particularly suitable for the food industries with high demands of hygiene. • Pulp and paper industry • Chemical and petrochemical industries					
Technical information						
Piston diameter	mm	32	40	50	63	80
Theoretical piston force at 6 bar/push stroke	N	483	754	1178	1870	3016
Theoretical piston force at 6 bar/pull stroke	N	415	633	990	1682	2721
Cushioning length	mm	10.5	14	16	14.5	19.5
Cushioning energy	Nm	4.8	9	15	27	54
Cylinder weight	kg	0.6	1.2	1.7	3.0	4.0

AI.4.2 Mini ISO Cylinders

Made of stainless steel (304 grade) and plastic, this round cylinder is ideal for multiple uses. A wide range of sizes beginning from diameter 2.5 mm makes an optimal choice easy.

Table AI.15 features and technical information for the Mini ISO cylinders

Features				
Type		Piston cylinder		
Configuration		ISO mini cylinder, double acting with magnetic piston, 16 – 25 diameter with adjustable cushioning and push-in fitting		
Max. working pressure		10 bar		
Ambient pressure range		-25 °C to +75 °C		
Medium		Compressed air lubricated or non-lubricated		
Application areas		Food- and packaging- industries		
Technical information				
Piston diameter	mm	16	20	25
Theoretical piston force at 6 bar/push stroke	N	141	217	295
Theoretical piston force at 6 bar/pull stroke	N	124	186	247
Cushioning length	mm	10	13	16
Cushioning energy	Nm	0.6	1.5	2.3
Cylinder weight	g	70.1	122.2	168.1

AI.4.3 Mecstreme Cylinders

Stainless (304 grade) pneumatic cylinder that meets most levels of corrosion resistance. With its round profile it is easy to clean and can be used in most sensitive areas.

Table AI.16 features and technical information for the Mecstreme cylinders

Features							
Type		Round cylinders Mecstreme, series299 in stainless steal					
Configuration		Double-acting with magnetic piston and adjustable cushioning, 25 – 125 mm diameter					
Max. working pressure		10 bar					
Ambient pressure range		-25 °C to +70 °C					
Medium		Compressed air lubricated or non-lubricated					
Application areas		The cylinder is particularly suitable for food industries, pulp and paper, chemical and petrochemical industries					
Technical information							
Piston diameter	mm	25	32	40	50	63	80

Theoretical piston force at 6 bar/push stroke	N	295	483	754	1178	1870	3016
Theoretical piston force at 6 bar/pull stroke	N	227	415	633	990	1682	2721
Cushioning length	mm	11	11.5	14	17	16	19
Cushioning energy	Nm	1.0	4.8	9	15	27	54
Cylinder weight	kg	0.5	0.6	1.2	1.7	3.0	4.0

AI.4.4 KPZ Cylinders

This compact cylinder with anodised surface and screws of stainless steel will meets most levels of corrosion resistance. With its standardised dimensions (NFE 49 004) and the wide range of variants this cylinder is suitable for nearly all application.

Table AI.17 features and technical information for the KPZ cylinders

Features						
Type	Compact cylinder, series KPZ					
Configuration	16 – 100 mm diameter, double acting, piston rod with female thread					
Max. working pressure	10 bar					
Ambient pressure range	-20 °C to 80 °C					
Medium	Compressed air lubricated or non-lubricated					
Application areas	Compact exterior dimension Flushingly integrated cylinder switches Large fastening program					
Technical information						
Piston diameter	mm	16	20	25	32	40
Theoretical piston force at 6 bar/push stroke	N	106	164	259	422	665
Theoretical piston force at 6 bar/pull stroke	N	91	137	216	364	560
Cylinder weight	kg	0.07	0.098	0.143	0.223	0.333

AI.4.5 Guide Precision Cylinders (GPCs)

The Guide Precision Cylinder combines precise movement with high side load capacity. The GPC-TL (top loader) has an extra mounting surface on top for installation in food and packaging applications.

Table AI.17 features and technical information for the GPCs

Features				
Type	Guide cylinder, series GPC			
Configuration	Series GPC-TL, 12 – 20 mm diameter, double acting, magnetic piston			
Max. working pressure	1.3 – 8 bar			
Ambient pressure range	-10 °C to +70 °C			
Medium	Compressed air lubricated or non-lubricated			
Application areas	For precise movements with high side load capacity The guide cylinder can be used as: • Carrier of a second axis • Carrier of a grippers or suction cups in material handling applications • Carrier of tools like automatic screw drivers • Carrier of work-pieces			
Technical information				
Piston diameter	mm	12	16	20
Theoretical piston force at 6 bar/push stroke	N	67	120	180
Theoretical piston force at 6 bar/pull stroke	N	50	100	140
Max. velocity	m/s	0.5	0.5	0.5
Cushioning energy	Nm	0.1	0.11	0.15

AI.4.6 Rexmover Rodless Cylinders

RexMover is a rodless pneumatic cylinder with an integrated ball-bearing guide for heavy-duty applications. The RexMover offers space-saving design, highly flexible mounting options, virtually leak-free operation, and speeds up to 5 m/s.

Table AI.18 features and technical information for the Rexmover cylinders

Features						
Type		Rodless cylinders RexMover, series 277				
Configuration		Heavy duty, 16 – 63 mm bore with magnetic piston and adjustable cushioning				
Working pressure		4 – 8 bar				
Ambient pressure range		-10 °C to 60 °C				
Medium		Compressed air, non-lubricated				
Application areas		In compact installation where restricted external dimensions are to be found for controlled and long strokes. The heavy duty version will be used as a strong linear drive. Due to the strong ball bearing guide, heavy loads can be transported with high precision.				
Technical information						
Piston diameter	mm	16	25	32	40	50
Connection port		M7	G 1/8	G 1/4	G 1/4	G 3/8
Theoretical piston force at 6 bar	N	120	295	482	756	1176
Cushioning length	mm	20	20	20	20	20
Cushioning energy	Nm	1.5	4	7	10	15
Cylinder weight	kg	1.1	1.82	2.79	4.69	6.32

AI.4.7 ND7 Valves

The electropneumatic pressure control valve converts an electrical serial signal proportionally into a pneumatic pressure. It is highly dynamic, with a short response time and minimal hysteresis. Combines compact design and high flow rates with very accurate repeatability.

Table AI.18 features and technical information for the ND7 valves

Features	
Type	Poppet valve
Configuration	Electropneumatic pressure control valve, ND7, G 1/4, proportional solenoid, high performance
Supply pressure	12 bar
Output pressure	0 – 10 bar
Hysteresis	0.03 bar
Repeatability	0.01 bar

Nominal flow rate Qn at primary pressure = 7 bar and secondary pressure = 6 bar with pressure drop Δp = 0.2 bar	1300 Nl/min
Ambient temperature range	0 °C to +50 °C
Medium	Condensate-free and non-lubricated compressed air, filtered 50 µm
Weight	2.1 kg
Application areas	They are used where electrical control is required to act directly on a change of pressure or force
Technical information	
Supply voltage	24 V DC ± 20%
Permissible ripple	5%
Power consumption	1.4 W

AI.4.8 HF03 Valves

This type of valves is Ideal for high-flow applications where space is limited. Simple extension and valve replacement without the need for disassembling the unit provides easy handling. Due to the proven manifold design using aluminum end pieces and reliable electrical connections, the valve system is insensitive to vibration and temperature fluctuations.

Table AI.19 features and technical information for the HF03 valves

Features	
Type	Valve terminal system, series HF02
Configuration	Completely mounted valve terminal system
Working pressure, local pilot control	2.5 – 10 bar
Working pressure, external pilot control	Vacuum – 10 bar
Min. admissible control pressure	2.5 bar
Nominal flow rate Qn	700 Nl/min
Ambient temperature range	0 °C to +50 °C
Medium	5 µm filtered oil-free compressed air or 40 µm filtered and oiled compressed air
Application areas	Suitable for all applications Easy and fast mounting Low weight, short response time
Technical information	
Supply voltage	24 V DC ± 20%
Power consumption	0.34 W

Appendix II

Pneumatic Applications in Different Industry Segments

AII.1 Introduction

There are a wide range of applications in different industry sectors that utilise servo-pneumatic systems, such as automobile industry, food industry, process industry, and handling and assembly technology. This appendix aims at providing the reader with some details on these applications in order to demonstrate the wide spread of the pneumatic systems in the market. ***All the technical data, information, and supported pictures, have been obtained from the Festo, Origa, and Bosch websites.***

AII.2 Automobile Industry

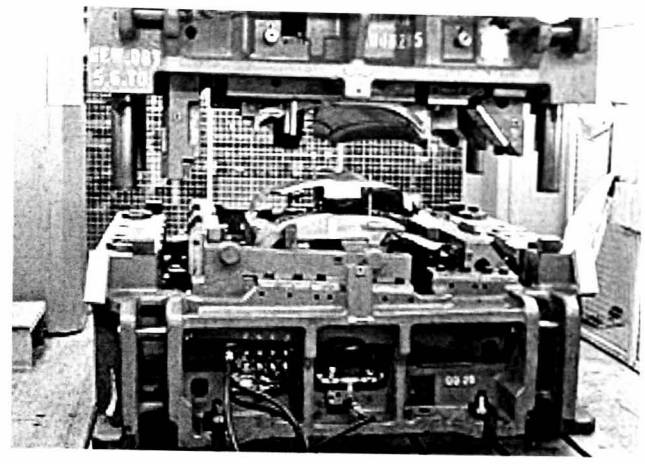
Pneumatic servo-systems is involved in the industry's production processes, providing important functions such as the secure and cost-effective handling of sheet metal in press shops, the clean control and process air for paint shops, and the smooth operation of assembly lines. Obviously, pneumatic solutions were used for all kinds of demanding tasks in the automobile industry.

Pneumatic components and systems can undertake major tasks such as

- handling and transporting sheet metal parts in the press shop,
- component clamping and fixing, positioning welding tongs for the unpainted body shell,
- nozzle control in the paint area,
- assembling or feeding units in final assembly

AII.2.1 Pneumatics in the Press Shop

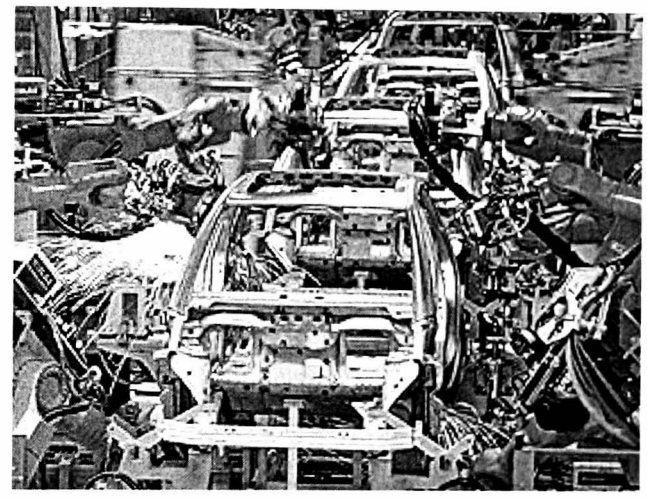
The pressing and shaping of bodywork components demand absolute accuracy. Pneumatic deliver the necessary precision, often under very cramped conditions, for all kinds of functions from work-piece handing through securing gripping, feeding and sorting to centring and holding.



One example is a special vacuum cylinder used to lift sheet metal work-pieces. This flat cylinder with a non-rotating piston rod saves valuable space and maintains the vacuum for only as long as necessary.

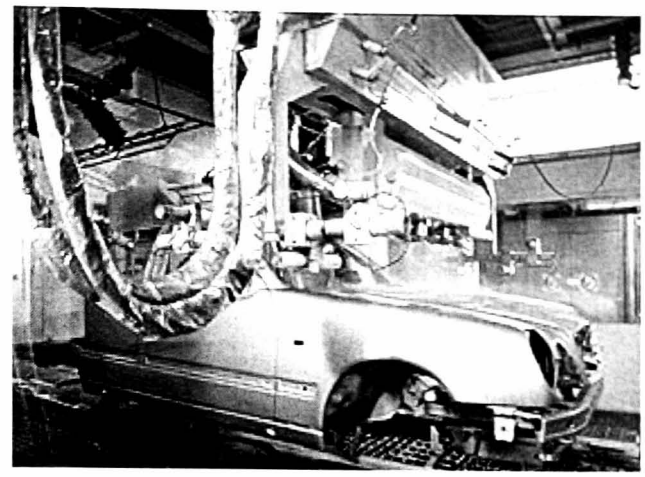
AII.2.2 Pneumatics in Bodywork Production

It is common for auto makers to produce several hundred bodywork variants of a single vehicle platform. This demands extremely flexible automation systems. Innovative pneumatics ensures that the right component is brought into the right position at the right time, whether in a welding jig, in welding tongs or in workpiece handling.



Pneumatic components are required not only to fix components into place but also to prevent these from being deformed. In daily operation, they are subjected to severe loads, both in terms of the forces present and the welding splashes falling onto their outer surfaces. As bodywork components are transported from one welding station to the next, pneumatics are used to hold and grip these. This calls for optimised individual solutions.

AII.2.3 Pneumatics in Paint Shops



Automobiles receive four or five coats of paint. The processes these days have been conceived with tomorrow in mind - for example, by specifying water-based paints and the lowest possible atmospheric pollution.

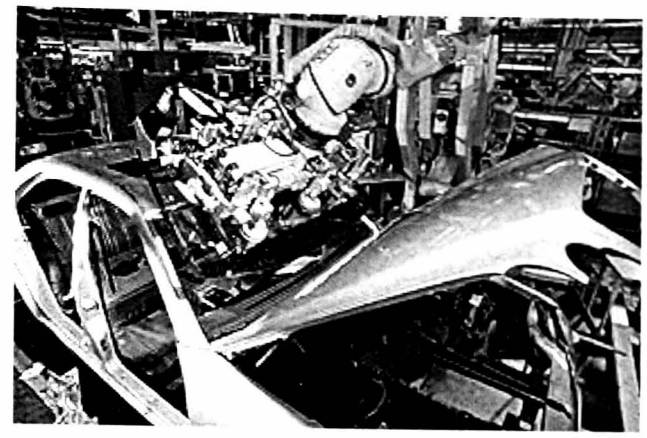
Pneumatic components match these demands and help deliver quality results in paint shops. Modern air preparation ensures there is no water condensation or other contamination in the form of dirt or oil.

Pneumatic service units ensure clean air and are available with standard or micro filters. Lockable on/off valves can be incorporated to provide total safety, particularly during maintenance work. This is how it works:

- **Filtration:** Solid matter may be present in compressed air in the form of dust, soot, abrasion particles and corrosion products. Some service units enhance system service life by filtering these substances out of the air.
- **Drying:** Dry clean air is essential for long service life of paint-shop equipment. Modern air preparation protects process air from water condensation, thus avoiding corrosion damage to compressed-air supply networks.
- **Removal of oil:** Even with compressors which operate oil-free, oil aerosols will contaminate the atmospheric intake air with traces of oil. This oil is not suitable for the lubrication of cylinders and may cause blockage of susceptible components. Filters within service units separate these oils out from the compressed air down to the µm range.
- **Freedom from PWIS (e.g. silicone):** These service units ensure through its choice of lubricants, seals and assembly methods that "paint-wetting impairment substances" (PWIS) such as silicone are not able to deposit themselves in or on its products. This eliminates the risk of craters in paint surfaces caused by these substances.

AII.2.4 Pneumatics for Assembly Lines and Test Rigs

Final assembly involves a wide variety of tasks: feeding work pieces, automatic assembly, holding or positioning. A smooth-running process is vital. Pneumatic components make this work easier by ensuring precise and efficient working operations.



Automobile components such as control pedals are subject to high loads. They are therefore required to undergo a comprehensive programme of long-term and endurance tests. The results of these must be 100% reliable. This is where pneumatic provides complete units for the fully-automatic control and evaluation of long-term tests.

In assembly operations, each device used must be matched to the relevant step in the production process. Each automobile manufacturer requires individual solutions in order to achieve this. Therefore high-flow-rate CPV valves can be utilised, it is used mainly on robot arms in order to optimise robot operation. Cylinders of special designs and with additional functions increase flexibility. For example, the double-acting DZF flat cylinders with protection against torsion and a wide range of mounting options, or the DFM twin-piston cylinders with integrated guides.

Balancers are used both in final assembly and other production operations, such as door fitting or cockpit assembly. The use of balancers reduces the amount of physical effort required, particularly when transporting heavy loads. Pneumatic components for balancers are commonly used by the manufacturers.

Advantages of using pneumatics in balancer systems:

- Low actuating force (necessary when handling different masses)
- Low investment costs
- Easy to operate

- Highly reliable
- Safe system in cases of compressed-air supply failure

The major components of a balancer system are as follows:

- Pneumatic cylinder
- Precision pressure regulator
- Piloted one-way flow control valve
- Air preparation system
- Fittings and tubing

AII.3 Areas of Activity in the Process Industry

Industrial applications are highly diverse. Pneumatic components are used to fulfil a wide variety of functions. Differentiation is made between industry sectors and process engineering criteria. The following are some application examples:

- **Chemical and petrochemical industry:** The chemical and petrochemical industry uses a large number of butterfly and ball valves which are as a rule either manually actuated or fitted with pneumatic actuators. Particularly stringent safety standard requirements are the norm in this industry, since the flow media used is often aggressive, which can lead to explosion in the event of sparking. In such cases, pneumatics offers solutions which can be specially adapted to individual applications:
    - Solenoid valves with explosion-proof coils mounted directly on the drive (Namur interface) in the explosion zone
    - Solenoid valves in an explosion proof cabinet with tubing leading to the drives
    - Solenoid valves in a control cabinet outside the explosion hazard area with tubing leading to the drives
    - Valve terminals outside the explosion hazard area with tubing leading to the drives

With the use of single-acting actuators, it is also possible to easily realise a safety setting on a valve actuator, for instance in the event of a power supply failure.

- **Gas industry:** Alternative energy supplies are becoming ever more significant in industry, where the use of natural gas as a powering fuel is increasingly gaining in importance. Natural gas-operated motor vehicles for instance represent an interesting alternative to crude oil.

  Today, the use of pneumatic actuators on high pressure valve actuators in natural gas fuel storage depots is already regarded as standard practice. These components are distinguished by their sturdy design and maintenance-free operation, as well as the use of compressed air as a drive medium that makes them suitable for use in potentially *explosive atmospheres*. The analogue control, necessary in part, is realised via electro-pneumatic and explosion protected positioners.

- **Food industry:** The food industry is divided into three zones: The food zone, the splash zone, and the non-food zone. If pneumatics are deployed in areas directly involved with the production of food, or entails coming into contact with food, easy to clean products and special lubricants have to be used.

  However, even the food industry has areas of application where standard pneumatics from process automation can be used, such as for instance in the following areas of application:

  - Water treatment (filtration)
  - Dosing
  - Mixing
  - Cleaning
  - Transport of cleaning chemicals
  - Filling
  - Waste water disposal

- **Paper industry**: Complex processes, aggressive environmental conditions, production in continuous operation – the demands in the paper industry are very stringent. Water, sludge, fibrous material, pulp and pollutants of various consistencies flowing through production stations, have to be separated, sorted or filtered.

  Pneumatic actuators offer an optimum solution in the form of drives for knife gate valves and butterfly valves. The robust and corrosion-resistant components fulfil all requirements pertaining to variable speeds, smooth surfaces, maintenance-free design, continuous operation and overload resistance. Pneumatics represents the driving force in the following areas of application:

  - Raw material handling
  - Sorting
  - Purification systems
  - Bleaching processes
  - Washing processes
  - Filter systems

- **Bulk goods industry:** Dusty, polluted and explosion hazardous environments such as those occurring in the handling of powders, granulates and flours are the predestined areas of application for sturdy pneumatic drives. These reliably control butterfly valves, knife gate valves and weirs of any size. Previously manually operated gate valve systems were automated via pneumatics with minimal spending. Areas of application:

  - Dosing
  - Silo emptying and filling
  - Granulate and powder transport
  - Bottom valves

- **Drinking water technology:** Water is the basis of all life. In many industrialised nations, an endless supply of water on tap is taken for granted. However, every

city and community requires the necessary equipment for drinking water treatment and purification in order to assure clean water for use in homes and businesses. Profitable and easy to use.

Pneumatic drives and other automation solutions greatly increase the efficiency of filtering plants in water treatment facilities. An additional advantage: The use of compressed air maximises operating reliability, and minimises maintenance and repair costs. Furthermore, compared with electrical actuators, pneumatics do not require any additional control and monitoring signals such as torque switching, over-temperature and clockwise-anticlockwise operation.

Pneumatic actuators are qualified for drinking water applications by means of overload protection and resistance to continuous loading, as well as zero service and maintenance requirements. Up to 1000 PLC inputs and outputs can thus be eliminated in a mid-sized water treatment plant - or even some valve terminal technology produced by Festo includes field-bus interface or integrated PLC systems.

AII.4 Food Industry

The production of foodstuffs is subject to strict hygiene regulations. In order to meet these, machines must be as easy as possible to clean on a daily basis. Pneumatic components by fulfil the following specific requirements of the food and packaging industry:

- High hygiene standard
- Resistance to aggressive environmental conditions and the frequent use of strong cleaning agents
- Products can be provided for all production zones; the food zone, splash zone and non-food zone

AII.4.1 Packaging

The packaging must meet a large number of different requirements. Packaging must at the same time provide a sales-promotion function and offer protection and information. Moreover, the packaging industry is required to be highly flexible and adapt quickly to changing consumer demand. Pneumatic products allow all kinds of solutions to be created:

- Compact space-saving design
- Exceptionally high efficiency
- Flexible applications
- Modular construction

AII.4.2 Pneumatics in Filling/Bottling Plants

The trade demands high quality, enhanced taste and longer product shelf life. In order to satisfy these demands, production conditions must be optimised or upgraded. Particular stress is placed on the hygienic design of filling/bottling plants.

There is no two filling/bottling plants are alike, even if the products themselves are broadly similar. A further factor is that each individual country has different requirements regarding machine operation and equipment. It is thus not possible to define a universal performance specification. One feature, however, is common to all filling/bottling plants: Flexibility. Packaging size and capacity will be different with almost all machines. Only through a high level of machine flexibility can the wishes of the various target groups be met.

Products must be packaged without any loss of quality or quantity. Total hygiene is vital. The objective is to optimise each individual step in the production process in order to secure long-term market positions. Pneumatics has proved its value with all sizes of machines, all performance classes and all levels of automation.

AII.4.3 Pneumatics in the Milk-Processing Industry

There is tough competition in the milk-processing industry. The demands placed on quality, product shelf life and product variety are high. At the same time, costs must be strictly controlled. Pneumatic systems operate in filling/bottling plants highly efficiently. For example, corrosion-resistant cylinder series and a wide range of matching accessories offer high resistance to chemicals and cleaning agents. Special application-optimised variants safeguard product quality, despite the presence of extreme forces and difficult environmental conditions.

The enormous performance density of pneumatic systems allows high speeds and high throughput volumes within plants. The components used fulfil the specific requirements of the industry, particularly with regard to hygiene.One example: The specially-coated cheese-press cylinder. With its large piston, it can easily take up the highly-viscous cheese whey and move this without difficulty. Its low-friction characteristics ensure that cheese pressing can be carried out with the necessary constant speed. Suitable wiper concepts provide guaranteed product quality and long component service life.

Corrosion-resistant pneumatic components are mandatory in cheese production, particularly with hard cheeses. The slow ripening process means that the cheeses are washed over with salt solutions over a period of several months. Only acid-resistant stainless steel can provide the necessary level of hygiene.
Customers expect a lot from their milk: It must be fresh, healthy and keep for as long as possible - and of course taste good. In order to meet all the requirements, production plants are equipped with complex control systems whose functions are closely matched one to another. Pneumatics supports these processes with a systematic range of products, from intelligent controllers to actuators.

The advantages which pneumatic components offer:

- Fast product changeover
- Ever-higher level of automation
- Long service life

- Fast but thorough cleaning of plants with CIP (Clean in Place) concepts integrated into processes
- High yield from raw materials
- Low energy consumption helps reduce costs

When the summer sun shines, thoughts turn to ice cream: Production facilities must be capable of reacting quickly. This demands maximum flexibility. Pneumatic components can continue to operate reliably even if the night is frosty, with temperatures below zero. This allows manufacturers to react promptly to changes in consumer demand.

AII.4.4 Pneumatics in Meat Processing

Sausages, both large and small, are popular once again, which means that plant and machine constructors in the meat-processing industry have considerable potential for growth. The level of automation in this industry at present is still quite low. Innovative solutions are needed, but customers demand absolutely top-quality products. Automation solutions must therefore be not only cost-effective but also deliver appropriate quality. One factor is of paramount importance: Hygiene. What the meat processing industry requires above all is easy-to-clean and corrosion-resistant pneumatic components.

Cutlets, fillets, steaks or sausages - the range of different meat products is very large. This demands a special range of machines for both small-scale butchers and industrial meat processing plants. No matter for what purpose a machine is used, carcass cutting, mincing or refinement of animal products, it is vital that hygiene regulations are fulfilled.

Day in, day out, machines and plant are disinfected with acids and highly-concentrated alkalis. The best solution here is the pneumatic stainless-steel component series (supplied by Festo). All the components of this series are optimised for maximum corrosion resistance and ease of cleaning.

AII.4.5 Pneumatics in the Pharmaceutical Industry

Maintaining an overview is the challenge in the pharmaceutical industry. Clean-room capability is the watchword here. The initial and operating costs of clean rooms are extremely high. It is thus vitally important to choose the right components from the very start. Pneumatics has a wide range of inexpensive standard components which are almost all clean-room-capable.

Tablets, pills, capsules, suppositories, creams and drops - all medication used by human beings is required to be manufactured in accordance with strict regulations. Every manufacturer must therefore establish and operate an efficient quality-assurance system. The "Clean Design" and "Hygienic Design" product series (produced by Festo) fulfil all the requirements of the pharmaceutical industry. An excellent ease of cleaning and minimal maintenance needs.

Clean-room capability is a common term used in the pharmaceutical and medical-technology industries and relates to the particle emissions of a piece of equipment. The cleaner an installation is, the less dirt generated during production operations.