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## **Advancements in the programmable motion control of pneumatic drives for robots and other flexible machines**

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ADVANCEMENTS IN THE PROGRAMMABLE MOTION CONTROL  
OF PNEUMATIC DRIVES FOR ROBOTS AND OTHER FLEXIBLE MACHINES

by

JUNSHENG PU

A Doctoral Thesis

Submitted in partial fulfilment of the  
requirement for the award of

Doctor of Philosophy

in the Department of the Loughborough University of Technology

Loughborough University

Department of Engineering Production

June 1988

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In Memory of Our Mother

## Declaration

No part of the work described in this Thesis has been submitted in support of an application for any other degree or qualification of this or any other University or Institution of learning.

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## SYNOPSIS

The overall objective of this research is to advance the programmable motion control of pneumatic drives. Enhanced performance characteristics and improved parameter tuning facilities have been evolved for a new generation of pneumatic servos. The evolution has been achieved with specific reference to the application of pneumatics in the motion control of modular robots and other modular machines.

The research has been largely experimentally-based with complementary new modelling studies providing a basis for suggesting and explaining the control strategies evolved. Novel realtime control algorithms have been implemented and their associated performance characteristics statistically analysed.

The algorithms have provided technological advance with respect to

- (i) minimising drift and hysteresis in the drive system through compensating and automating the system null conditions;
- (ii) optimising the positioning time through use of learning procedure;
- (iii) achieving velocity control through "null velocity" compensation and learning.

It is anticipated that the methodologies evolved will be incorporated within a range of microprocessor-based motion controllers through a UK manufacturing system supplier.



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## CHAPTER 1 INTRODUCTION

The objective of carrying out this research study is based primarily on the following understanding:

As in many other fields, with the advancement of microprocessor application technologies, traditional performance of servo pneumatic drives could be improved considerably without seriously impinging on traditional advantages like low cost, simple and reliable mechanical design. This has been shown to be the case with the initiatives aimed at designing "intelligent" pneumatic servo drives in the last decade. Existing commercial products released world-wide through these initiatives can be loosely classified into two extremes: one is with fast speed and high precision but utilising relative high cost and closely specified hardware components; the other is to use low-cost hardware components of various forms but offering poor control repeatability and tuning problems for users. So far, the applications of pneumatic servos have been mainly limited for the purpose of experiments and prototype investigations. Previous work indicates that there is great potential in this area. The objective of this work is the derivation and evaluation of new and improved control algorithms which would be eventually transferred into a general purpose pneumatic motion controller. These programmable pneumatic drives can be used to form multi-axis or modular industrial robots or other flexible machines. Thus, wider application areas can be found using pneumatics which have not yet been generally acknowledged by industry as servo drive system technology.

The excessive high nonlinearities introduced when using compressed air as the control medium, especially at low pressure, necessitates the use of largely experimentally-based approaches. Many other research studies have indicated that it is impossible to obtain detailed and exact analytical solutions to the governing differential equations for pneumatic drives. Simple but relatively comprehensive and useful modelling is highly desirable. The implementation of control theory in most present industrial robots has shown that it is wiser to use simple but correct control laws rather than advanced but complicated and incorrect ones. Furthermore, as yet there is no universal solution to the control of highly non-linear systems like the pneumatic one.

The following discussion provides the "backcloth" for the following research aims:

- (i) to make use of the power of modern control concepts.
- (ii) to derive simple but correct control methods so that they can be easily implemented and used.
- (iii) to take advantage of computer-based "intelligence" to achieve aims (i) and (ii). These analyses lead naturally to the adoption of learning and knowledge-base ideas which are an essential component of the control strategies evolved in this study.

The nature of such a research activity requires experimental evaluation tools in order to be able to draw valid conclusions. However,

experimental tests should not be blind, hence complementary mathematical studies are also required to derive adequate models for both designing and understanding of the underlying physical mechanisms involved. "Specific" control methods will be evolved for positioning or velocity control tasks. However, a "generic" control equation will be described which integrates the specific or individual control schemes evolved into a single control methodology suitable for general purpose pneumatic motion control. The architectural framework of such a controller will also be outlined along with a consideration of factors such as parameters tuning and control algorithm robustness.

## CHAPTER 2 REVIEW OF LITERATURE

### 2.1. Introduction

This chapter reviews research work in the field of motion control of pneumatic drives. A global view of manufacturing automation will be presented which highlights the potential application areas for pneumatic drives and summarises the functionality provided by the present generation of pneumatic servos. The control and modelling techniques previously applied will be reviewed along with a categorisation of the problems encountered when utilising servo drives in industrial positioning and speed control applications.

### 2.2. Universal and Modular Drives for Manufacturing Automation

Manufacturing machines exist between two extreme categories: special-purpose dedicated machines and universal (programmable) machines (which includes semi-dedicated machines). The use of special purpose dedicated machines is particularly appropriate in a large batch or flow manufacturing environment [1]. However, frequent process and/or product changes, can demand the use of universal machines such as industrial robots: these machines commonly incorporating some form of computer controller [2]. However, the conventional robot is overly sophisticated and consequently unnecessarily costly for a significant number of tasks [3]. Thus robots can demonstrate the following limitations, viz [4]:

- (1) limited dynamic and kinematic properties, e.g. reach, velocity, acceleration, load bearing, repeatability and accuracy restraints;

- (ii) mechanical flexibility is provided but mechanical optimisation is seldom achieved;
- (iii) configuration flexibility is limited;
- (iv) operational flexibility is limited by the attachment of specific tools, fixtures and feeding mechanisms;
- (v) lack of software integration flexibility.

Modular robot systems [5-7] have been introduced with a view to overcoming these limitations. However, the use of modular robots does present some difficulties. Engineering effort is required to design and construct a suitable configuration of axes and possibly a controller [8,9]. The evolution of task programming facilities for the generic machines is by no means a trivial exercise [10]. However, to facilitate the more widespread use of flexible machines, it will be necessary to promote concepts of modularity and universality in the design, implementation, programming and integration of manufacturing machines [11]. One approach can be the introduction of modular distributed manipulator systems, which can offer both operational and configuration flexibility. These manipulators may not be mechanically connected together but may be electrically sequenced and synchronised [12,13]. Such universal modular machines require a configurable computer-based hierarchical control architecture to manage and handle them so that a combination of flexibility and mechanical optimisation can facilitate wider applications in industry than those served by conventional industrial robots. It is highly desirable to evolve a family of control system primitives (or software and hardware modules) to provide this control architecture so that modules can be selected and aggregated to meet the need of future programmable machines, operating within Computer

Integrated Manufacturing (CIM) environments [14,15]. Modularity concepts for software programming have been implemented for many years [16,17]. The disciplines embodied have also begun to be applied to construct robot software [18-21]. More recently, control system architectures for distributed manipulators and modular robots have been investigated [22]. The functionalities and interfaces of these drive elements can be arranged in a hierarchical modular form whereby hydraulically, electrically or pneumatically driven actuators can be employed. In the design and development of any manipulator system it is of great importance to minimise the systems engineering content since this usually accounts for a very large part of the total installed cost. It is therefore vitally important that the elements in a modular system are easy to integrate and reconfigure as possible both during initial build and for post installation modifications and extensions [23].

In comparison with hydraulic and electric counterparts, pneumatic drives can offer the ability to operate from a simple stored energy system while at the same time providing adequate performance at potentially lower capital cost [24]. Further advantages can be identified as a wide temperature range capability, simple mechanical structure, safe and clean working condition when compared with hydraulic drives [25]. The potential industrial application market of servo pneumatic drives is estimated to be wide [26,27]. Present servo pneumatic drives have adequate performance to satisfy a wide variety of positioning tasks in systems ranging in configuration from a single axis through axis groups to large distributed manipulators [28,29,30]. Often simplicity and cost

advantage can be gained from using direct pneumatic drives without adding extra transmission elements.

The foregoing discussion illustrates the background conceptual thinking to this research study and highlights the need to evolve new methods of achieving the programmable control of pneumatic servo drives, whereby they can find a wider application base in manufacturing industries.

### 2.3. The Design of Pneumatic Servo Drives

Pneumatic drives can be loosely classified into on-off motion control (to mechanical stops) and servoed motion control. Most industrial pneumatic drives today use on-off motion control (commonly to two mechanical end-stops, i.e. can be considered to be a binary drive) and have been widely used in controlling pick-and-place robot manipulators and dedicated manufacturing machines with a long-standing history. The use of pneumatic binary control has been thoroughly reviewed by Moore [31], thus this review will concentrate on the previous and present work on the servo control of pneumatic drives.

It is not easy to find a single theme which can give a simple, coherent and comprehensive view of servo pneumatic drives. This is because there are many ways to categorise servo pneumatic drives. There are many aspects involved which can affect the features of the modelling and control of servo pneumatic drives. In terms of actuator types, for example, there are linear devices (e.g. symmetric or asymmetric cylinders) and rotary devices (e.g. piston or vane motors). With



respect to the control valves used, there are on-off, servo and/or proportional valves, which again can be classified into flow control and pressure control valves. With respect to power source, high pressure (e.g. in missile systems) or low pressure (e.g. in industrial application environments because of safety factors) air supplies can be used. With respect to application areas, servo pneumatic drives can be used for simple to advanced purposes (e.g. low accuracy, low speed positioning to high accuracy, high speed positioning) as single degree of freedom workhandling units or as the building elements of industrial robots. A classification of modelling and control techniques alone will not be adequate and sufficient without referring to individual features of specific drives. In this review, methods for achieving the position and velocity motion control of pneumatic drives prior to this research study will be considered. The review will illustrate how the servo control of pneumatic drives has been studied and control schemes implemented with the aid of analogue computing elements before the advent of microprocessor control technologies. This traditional work done in the field of pneumatic servo drives, including both high pressure and low pressure cases, will provide clues for the following study. Subsequently a survey is conducted around the use of microprocessor-based pneumatic servo mechanisms.

### 2.3.1. Studies of Pneumatic Servo Drives

#### 2.3.1.1. Early Foundation Work on Hot Gas and High Pressure Actuators

Frontier work on the control of pneumatic servo mechanisms, using servo control valves, was founded on the work of Shearer [32-34] during the

1950s. Shearer set up the fundamental differential equations for gas flow through a narrow clearance and methods of damping of the effects of drive resonance. The results of his work have been successfully applied in spacecraft and missile engine controls, where hot, high pressure gas (1000 bar) is used as the working medium. Most of the latter publications refer to the work of Shearer J.L.

Vaughan D.R. [35] in 1965 derived a linearised model of a hot-gas position control system. Three time constants were defined in this model: "cylinder change time", "acceleration time" and "slewing time". The mass flow rates into the two ram chambers were assumed to be equal and opposite. The model obtained was thought to be the simplest possible to describe the features of the system response. This model was used to characterise both the linear and non-linear operations of the system. However, Vaughan also pointed out that any system which always operated in the linear domain would be overdesigned (and this fact represents a limitation in the use of his model for control purposes). The results presented indicate that significant improvements in response time can be achieved through the use of non-linear feedback compensation. The maximum mass flow and pressure limits were considered to constitute the most important nonlinearities involved. In this study, position, velocity and pressure feedback signals were used.

The effects of a number of system variables including saturation, inertial load, stiction and coulomb friction, valve laps and leakage, were investigated by L.R.Botting [36,37]. This study was conducted using a high pressure rotary actuator. The equations derived were non-

dimensionalised. Useful terms were defined like "valve gain", "valve pressure sensitivity", "compressibility" and "load ratio". Botting pointed out that the stiffness and accuracy of a servo is very sensitive to stiction and leakage, and that sufficient bandwidth is necessary to facilitate the inclusion of compensation signals. The conclusions drawn were helpful in the design of pneumatic servo-mechanisms.

#### 2.3.1.2. Previous Work on Low Pressure Drives

Mannetje J.J. explained the success of using pneumatic servos at high pressure is due to the fact that the servo response in such a high pressure system can equate to that of hydraulic responses [38]. For low pressure industrial applications, he concluded that the comment that the control strategy (when compared with high pressure cases) must be different while the formulas developed by Shearer are still valid. Having calculated that the pressure wave propagation speed in a hydraulic line is normally only four times better than that in the pneumatic line, he concluded that the response of pneumatic systems should not be particularly inferior to that of hydraulic systems (in the example quoted by Shearer and Lee [39], it had been shown that the hydraulic system responds approximately 50 times faster than the pneumatic system for the same supply pressure and load mass). Mannetje concluded that the fact that pneumatic systems are indeed inferior is attributable to the use of flow control valves. He suggested that pressure feedback instead of velocity feedback should be applied to minimise the order of the describing equations. Mannetje described a method of controlling a pneumatic servo motor. A high gain differential pressure feedback loop was designed to alter valve characteristics. He

suggested that the final system obtained would have a closed loop bandwidth of approximately 24 times better than that obtained simply by applying methods commonly used to control hydraulic systems. It was declared, that the system could be made very stiff with a high sensitivity and with position accuracy of less than  $1\mu\text{m}$  even in the presence of Coulomb friction and that the larger the load mass used, the better the system would behave in terms of stability. However, a limitation would be the acceleration force applied to the mass.

Burrows and Webb [40] in 1967 described an analogue computer simulation of an on-off pneumatic servomechanism comprising a polarised relay with deadzone and hysteresis, power relays, and an on-off four-way spool valve. The effect of varying the friction forces and the magnitude of position, velocity and acceleration feedback signal was studied. The effect of using stabilising tanks was also considered (the idea of using the tanks to improve stability had been proposed by Shearer in his early work). It was suggested that on-off valves are cheaper and present less sealing problems. Burrows and Webb suggested that an on-off pneumatic servo can be used where high accuracy is not required. Their work demonstrated the feasibility of constructing a low-pressure on-off pneumatic servo-mechanism and showed that transient pressure feedback is the most effective method of achieving system stabilization. Simulation studies of on-off pneumatic servo-mechanism have also been described by other researchers [41,42].

Experimental work was conducted to verify the computer simulation results [40] published by Burrows and Webb in 1969 [43]. Step and

frequency tests were conducted. Both the experimental and computer results indicated that the motion was more oscillatory about the midstroke position than about any other position. An alternative statement was that the motion of the system about mid-stroke position is less damped than about any other position. Burrows [44] extended their simplified mathematical model into a fourth order equation (the position control system including loop closures based on position, velocity and quasi-pressure feedback) to study the effect of position in stroke on the positioning stability of pneumatic servomechanisms. The motion about the mid-stroke position was also described by Burrows and Webb's, through the use of root loci analysis methods [45], and by Shearer [32]. The position-dependent damping effect had been shown to arise not only in on-off [46] but also in proportional low pressure pneumatic servomechanisms by Cutland [47].

Chitty A. and Lambert T.H. [48,49] described experiments on a low-pressure pneumatic actuator. Assuming that the supply pipes (mass flowrates) to the actuator chambers were sealed, they derived a model including viscous friction and leakage terms to analyse the natural frequency and damping ratio. Impulse tests (by giving a sharp thump by hand) and drift tests were conducted under conditions where the chamber connected to the supply line was sealed but the other chamber was maintained at constant pressure. Piston leakage was evaluated with the assumption of laminar flow. It was shown that the leakage had a small but sometimes important damping effect on the free vibration of the actuator. An interesting friction characteristic curve obtained by Rashid [50] at low velocities (less than 0.15 m/s) was included. The

polytropic value was evaluated as 1.28 in their tests and hence had a value between the two extremes 1.0 and 1.4 described by J.L. Shearer [32]. However, gas expansion and gas compression have a very complex nature under real working conditions [51].

Em T. [52] (1982) considered the stability and positioning accuracy of a pneumatic on-off servomechanism. His analogue control system utilised input and feedback potentiometers. The two signals were compared and an electric relay used to switch the appropriate control solenoid. Stabilisation methods included the use of stabilising tanks and velocity feedback. The deadband of the relay had an influence on both the system stability and the positioning accuracy. He concluded that improving the stability might require a large deadband but on the other hand deteriorated the positioning accuracy.

A digital simulation programme was developed by Brown D.E. and Ballard R.L. [53] to obtain the transient performance and stroking time of a pneumatic actuator and valve combination. The major difficulty in the simulation was found to be in the assessment of suitable values for the seal friction of the actuator. Factors of significant influence could be the dwell time between strokes, the lubricant variability, changes in ambient temperature and variations in load conditions (which even varied along the stroke of the same actuator). The friction effects on the performance were discussed by Kato etc. [54].

Mariuzzo etc. [55,56] (1980) described a simplified nonlinear mathematical model for pneumatic actuators. The objective of this study

was to develop a model which would enable the selection of optimised parameters for use when designing pneumatic actuators in terms of response speed, torque to weight and gas consumption. The simplified non-linear model was linearised. Subsequently, the linearised model was compared with the non-linear one. Transfer function and basic design relations were also derived. Digital simulation (using a 4th order Rung-Kutta algorithm) and experimental tests were carried out with D.C. and PWM inputs. The dead-time and dead-zone were considered in the simulation study and their relationships with input conditions were investigated. One of the conclusions drawn was that the effect of the dead-zone could be eliminated by imposing a D.C. bias on the input signal.

Nazarczuk K. [57] presented a mathematical model for simulating pneumatic drives, which uniformly describes all the periods of rectilinear motion of a pneumatic actuator (the movement was divided into preparatory, motion and terminal periods). Coulomb friction and elastic impacts were taken into account of the two-mass (the cylinder and the piston). This model was a general description valid to any number of degrees of freedom in the system.

#### 2.3.1.3. Discussions

Anderson [58] in 1967 stated that publication in the field of pneumatics often showed tendencies either to oversimplify or to overcomplicate the analysis of pneumatic systems. Anderson continued that the oversimplified approach assumes that everything is linear and that the overcomplicated approach to pneumatic analysis is to treat all equations

in their bare, most general nonlinear form. He explained the basic mechanics of compressible flow and tried to define general procedures for evaluation and design analysis of pneumatic components and systems and the essential tools required.

McCloy D. and Martin H.R. [59] in 1980 (revised edition) discussed the two main differences in performance between pneumatic and hydraulic servos: (i) hydraulics have a rapid initial response while pneumatics exhibit a time delay when the valve is opened to build up the pressure; (ii) pneumatic servo exhibit a lack of stiffness, especially with respect to external load disturbance. Burrows defined the load sensitivity as being the change in output position per unit change of external load (i.e. the inverse of output stiffness). He continued that the hydraulic systems could be shown to have a stiffness approximately 400 times that of the equivalent pneumatic system [25]. McCloy and Martin developed the linear approach used by Shearer [33,34]. The load sensitivity and the stiffness of the fluid in the cylinder were analysed. It is interesting to note that the open loop transfer function for the valve-cylinder combination is similar to that defined by Botting L.R. (third-order) [37]. They pointed out the likely reason that the analysis of the pneumatic servo had received relatively little attention in comparison to the hydraulic servo was due to the fact that the basic equations of pneumatic systems are non-linear and tend to be more complex than those of incompressible systems and was also due to the fact that the pneumatic servo is not so popular as its hydraulic counterpart.



However, since the late 1970s, new initiatives have emerged in designing pneumatic servos with the widespread availability of VLSI enabling technologies. Such a trend is obvious from the papers in the next section. Emphasis is given to the work in designing pneumatic servos using proportional or servo valves. The reason for using on-off valves by some designers was mainly due to the lack of the commercial availability of servo/proportional valves or economic consideration. This does not necessarily dictate the use of different control strategies for on-off or servo/proportional valves.

#### 2.3.2. Pneumatic Servo Drives for Industrial Robots

Drazen P.J. etc. conducted a range of research studies in the late 1970s aimed at using pneumatic drives to actuate multi-axis robots. The result of their work was a conventional pedestal robot initially consisting of three main degrees of freedom. The robot was powered by pneumatic motors activated by on-off solenoid valves, capable of point-to-point positioning [60]. The controllability of pneumatic drives was investigated in terms of hardware and computer software facilities [61,62] and also in terms of system stability [63]. A non-linear numerical model was developed for simulation purposes and as a control algorithm development tool [64]. The model was used to test the system under different control conditions and to simulate transient responses. A pressure transducer was used to determine the mass flowrate characteristics through the valve nozzle under the case of a constant actuator volume. The experimental data was fitted by a polynomial series (weighted least-square polynomial approximation being applied).

Valve resistance and time constants were introduced in a linearised analysis. Subsequently, they used the numerical model to develop two control approaches: a single piece-wise linear SISO representation and a state space model [65]. The effect of the system nonlinearities was shown by plotting families of system root-loci families. The state space model was used to identify such parameters of the controller which would retain the system roots in the desired locations and largely eliminated their migration caused by the nonlinearities. Velocity and acceleration feedback information were added to the position loop in the controller. Velocity and acceleration were not measured directly on the robot but estimated by using averaging algorithms resident in the microcomputer. Initially hydraulic servo valves were used but on-off valves were used later because of the unavailability of suitable low cost pneumatic servo valves. However, the use of on-off valves could present additional problems for the final control region when positioning and the on-off actions of the valves could produce the phenomenon of limit cycling (the air compressibility and the solenoid delay could make the limit cycle unstable). The previous work of Drazen etc. was further investigated recently by Taha [66] with the replacement of the on-off valve by pneumatic servo valve. Taha adopted the model presented by McCloy and Martin [59]. He decoupled the system by separately modelling the components contained in the control loop.

When state variables are not accessible for measurement it is usual to include an observer in the control scheme. The observer is a subsystem, based on plant parameters, which is designed to generate state variables from mathematical manipulation of the plant output signals [67]. The

possibility of using an observer model for the servo control of an industrial pneumatic robot was considered by Kraynin etc. [68]. The study was based on the earlier work of Barker H.R. [69]. Two methods of linearisation were considered to derive a model for designing the observer. Relationships were obtained which permitted the selection of parameters when designing the observer. It was shown, with the help of a specific example, that the introduction of a negative feedback in terms of velocity, obtained via the observer without a differentiation operation thus reducing the noise level in the control system, made it possible to eliminate oscillation when positioning. The use of an observer model would enable the system to dispense with supplementary sensors, thus greatly simplifying the design of the installation, increasing its reliability and reducing the cost of the servo actuator. However, as many of the system parameters of each individual drive are required for designing the observer, in the author's opinion, optimisation of the system performance is unlikely because of the difficulties in correctly estimating these parameters.

In about 1985, a research programme in the design of computer-controlled pneumatic servo drives was carried out at Loughborough University of Technology (LUT) [70]. A family of pneumatic servo controlled robot modules were released into the world market place. Weston and Moore etc. [71-73] derived a minor-loop compensation model (velocity and acceleration feedback loops) based on Burrow's linearised analysis [44]. Stable system behaviour was obtained by manipulating software loop gains to alter the inherent gain coefficients of the linear model thereby partially masking the effect of system non-linearities. Closed loop

gains were found to be functions of piston position in stroke. Position accuracy was studied with respect to changes in the forward path gain and the deadzone (hysteresis) of the system. Velocity and acceleration information were derived in the controller from measured position information. The acceleration feedback loop was found to be noisy [74] and gain and algorithm scheduling found to be necessary in improving both static and dynamic characteristics.

Their approach was improved with the introduction of a so-called front-end control strategy by Nagarajan and Weston [75] which improved the point-to-point positioning time especially for short moves. The approach was based on an outer decision loop which modifies the command issued to an existing closed loop drive. The outer loop was designed to generate a set of initial (front end) control sequences at the start of any 'point-to-point' move. A learning procedure was also incorporated in the scheme to obtain an optimised set of control parameters which could produce a specified quality of response. The control scheme was suggested with reference to the work of Astrom [76] for stable linear systems with monotone step responses. This research interest was further examined by Moore and Ssenkungo [77].

The Loughborough University pneumatic servo research was based on the use of a single stage 5-port valve which was designed specifically for the control of low cost pneumatic servo drives. A low cost rotary encoder with rack pinion transmission was used again with economical considerations in mind. The modular concept was embodied in both the design of the mechanical and the control system hardware and in the

software design. Their success has led to on-going research activities in the design of "Universal Machine Control" systems [4].

Extremely interesting implementation and design work was also conducted at RWTH Aachen with a number of publications resulting. Nauyen etc. [78] described a pneumatic servo drive system for position control. A simplified model was presented in order to provide a theoretical tool to guide the designer of the control system. Velocity and acceleration feedback loops were used in addition to the position loop. Velocity and acceleration were derived in a digital controller from the measured position signal. A rodless cylinder, a single stage servo valve and a high precision linear incremental transducer were used for the construction of the positioning system. Experiments were conducted to study the dynamic properties of the system.

Schwenzer [79] described an analogue control system using a 5-port servo valve. The static and dynamic behaviour of the system was detailed according to experimental observations. The whole control path was divided into four stages. The relationships between the positioning precision, valve hysteresis, drift in the analogue system, drive stiffness, forward path gain and the noise level in the feedback circuit were explained in a simple "intuitive" manner. With respect to deriving velocity and acceleration feedback, the use of a differential circuit and an "observer circuit" was compared. The noise level in the observer circuit was relatively low but the complexity (cost) of the circuit design was increased. In practice, however, a differential circuit was preferred, which corresponds to the derivation of velocity

and acceleration information from the position signal measured in the forementioned digital control system.

Backe W. in 1983 [80] described a pneumatic servo system based on the use of two 3-port valves. His control approach was similar to those reported by the other workers from RWTH Aachen. He presented the results of dynamic responses relating to output position, velocity, acceleration, the desired input and command signals giving an explicit picture of the system dynamic behaviour. Backe W. in 1986 [29] eventually reviewed the main work conducted at RWTH Aachen. The design of two-stage pneumatic servo valves and their main performance characteristics were explained with reference to hydraulic servo valves. Experimental profiles were presented to illustrate the effect of piston position and transient velocity amplitude on the system nonlinearities. It is interesting to observe the data reported with respect to damping ratio and natural frequency. The development of a control scheme was briefly described around the implementation of position loop with velocity and acceleration feedback. The various commercial layouts of the control system designed and their possible applications in industry were also illustrated.

The study of pneumatic servo drives for robot applications were also conducted by other researchers. Hammett G.G. [81] described a low-cost pneumatic servo drive. The design employed a pulse width modulated poppet valve. Feedback signals from a position transducer were processed by the control microprocessor which generated the control signal to the solenoid operated poppet valve. Wikander [82] described a

development scheme for the position control of pneumatic cylinders through use of a microprocessor. He described the components of a servo system based on a pneumatic cylinder, using an interesting symbolic model (which is very simple) by equating it to a soft spring. Position, velocity and acceleration feedbacks were used. However, generally speaking, these approaches more or less fall within the LUT and/or RWTH's in terms of both hardware component features (valve and actuator) and the control software.

Recently, Dunlop [83] described a torque motor with microprocessor control, which would challenge the electric drives in the low power robotic market place. The motor comprises four radial pistons acting on a three-lobed cam with a rubbing plastics interface. With the use of proportional valves, the microprocessor not only allows the motor shaft rotation to be controlled, but also the sequence of operation to be memorised and repeated when required. Computer simulation using a mathematical model was conducted to predict the motor performance accounting for air compressibility, pressure losses, friction, valve timing and rate of valve opening. The model also allows the motor passage and the valve dimensions to be optimised (the model assumes that the motor rotates at constant angular velocity while the pistons are always in contact with the cam). Programming of the motor can be achieved through a keypad, a computer terminal or via an integrated computer network.

The choice of the control valve will affect the main features of the performance characteristics and the cost of any pneumatic servo system

to be designed. The direct use of hydraulic servo valves is expensive as the industrial pneumatic pressure is much lower than hydraulic counterpart and hence arguably the low values of flow forces do not necessitate the use of a two-stage valve. The author concluded that the choice of on-off pneumatic valve does create problems in the final phase of positioning. Thus, the necessity for a dedicated pneumatic servo valve can be argued in order to achieve significant economic benefit over alternative drive technologies and promote the increased applications of intelligent pneumatic servo machines. Aiming for medium power applications, the first generation of intelligent pneumatic servo machines represented by UK and German products have become commercially available worldwide. Their valving arrangements have indicated such a requirement.

The effects of valve configuration on the frequency response of pneumatic cylinders were studied by Araki K. [84]. The expressions "mean lap", "unevenness of the laps" and "port width ratio" were used to describe the valve configuration of an underlapped or an overlapped valve. A describing model was derived with consideration of friction forces and load inertia. His latest work [85] was to analyse the frequency response when asymmetric cylinders of five different values of asymmetry ratio (i.e. ratio of piston areas on either side of the cylinder). Again a describing model for simulation was used. In these studies, simulated results and experimental data were compared. The behaviour of the system was theoretically analysed using the so-called Runge Kutta Gill method and the frequency response was obtained from a Fourier analysis of the wave forms. His early work [86] discussed the



modelling and compensation of a pneumatic servo-valve employing force feedback. The valve comprised two stages, a flapper-nozzle arrangement at the first stage followed by a spool and sleeve stage. Methods of compensation were compared, which included the use spring centralising of the spool and stabilising volumes. The best results were obtained when the two methods were used together. With respect to the development of two-stage pneumatic servo valves, interesting work is reported by Taft [87] and Viersma [88]. Referring back to the pneumatic servo robots developed at RWTH Aachen and those developed at LUT, the former took the two-stage valve control approach while the latter was based on the use of a low-cost one stage servo valve. It can be seen that high performance pneumatic servos may use two-stage valves while medium and low performance drives may utilise an one-stage (most likely 5-port) valve.

### 2.3.3. Velocity Control of Pneumatic Drives

The velocity of a pneumatic cylinder is often determined by the cylinder size, valve orifice diameter, and air pressure [89]. PERA [90] reported in 1965 on the determination of the effects of inlet port size, exhaust port size and various external load systems on the dynamic behaviour of double-acting cylinders. This investigation was conducted using analogue computer studies and practical tests. Exhaust port size was found significantly to affect the stroking time (increasing the effective exhaust port size will decrease the stroking time). A stability coefficient was introduced to verify the mode of piston motion. \*It was concluded that the magnitude of the load mass affects both the equilibrium velocity and the proportion of the stroking time

occupied in achieving dynamic equilibrium and that increasing the mass delays the onset of dynamic equilibrium and increases the magnitude of the equilibrium velocity. \*It was also stated that increasing the magnitude of the opposing force (with or without simultaneous increase in the magnitude of the mass) advances the onset of dynamic equilibrium and reduces the equilibrium velocity. \*It was further recommended that stroking speeds should always be controlled by restricting the Inlet Port, with or without simultaneous restriction of the Exhaust Port. It should be pointed out that these conclusions marked with '\*' will be referenced to some of the conclusions drawn from the modelling studies presented in Chapter 4 of this thesis. The author of this thesis has different interpretations of the experimental data presented by PERA. (The definition of the terms used here are those recommended by the British Compressed Air Society [91] and British Standard 2917:1957 [92] and those added by PERA).

When referring to hydraulic counterparts, McCloy D. and Martin H.R. [59] compared the behaviour of meter-in and meter-out velocity control circuits for pneumatic cylinders. The mathematical model of a meter-out velocity control circuit was presented to display the steady state velocity (equilibrium velocity termed by PERA), the natural frequency and the damping ratio of the system. It was concluded that the steady state piston velocity using meter-out control is less sensitive to external loading forces.

Conventionally, the primary controlled variable of a pneumatic cylinder is the velocity of the piston through bang-bang control with the

adjustment of a pressure regulator by using binary valve (master and directional valves of a pneumatic circuit [93] (Shimada K.)). A PWM mode feedback method was used to achieve a low and constant driving speed for a pneumatic cylinder [94] (Moritsugu T. and Hanafusa H. 1983). Salihi [95] investigated one case where pneumatic positioning mechanisms were used in the forming of glass containers by controlling the stroke time and cushioning.

However, very little published work is available so far with respect to control of velocity when using servo pneumatic drives, except where velocity information of the pneumatic drive is referenced in cases where the primary aim is that of point-to-point positioning (position servo control) of payloads and only the maximum velocity of the servo is specified or controlled. The use of pneumatic servo drives has not been generally applied with respect to contouring applications (such application areas being largely dominated by electric drives).

#### 2.4. The Control Techniques

There is no general standard approach to solving the control problems when using an inherently nonlinear system [96], this being the case for pneumatic servo control. Various mathematical models of pneumatic servos have been derived and their linear counterparts have been used to obtain the linearised transfer function. Most of the linear models are derived for small perturbations around the mid-piston position. The control analysis techniques used cover both classical and modern control approaches, i.e. transient response in the time domain, frequency

response, root locus, phase-plane, state space, and model-reference observer, with the aid of both analogue and digital computers.

However, despite the diversity of modelling approaches and control syntheses applied, almost all of the controller designed, digital or analogue, have used velocity, acceleration or pressure feedback compensation for point-to-point positioning. Pressure feedback has proved to be the most effective approach to system stabilisation. This reduces the nonlinearities of the system and eases the problems of system identification.

As those used in most of the present industrial robots, error-based control techniques (i.e. proportional control on with minor-loop compensation) [97] are very common and have been applied to achieve the servo control of pneumatic drives as well. They are most commonly used by engineers not because they offer the best control but they are the first and the only one which is theoretically well established and can be easily used. Small perturbation analysis techniques can be used effectively in transforming a non-linear system into a linear system for analysis purposes, but for the analysis to represent the practical situation this presupposes small changes in system variables. By imposing such constraint a control system may behave in a linear manner but this may lead to the design of an expensive and less desirable system than with a properly designed non-linear one [96,98]. Moreover, in most practical situations, the assumptions made in producing a linear model of a highly non-linear system will ignore fundamental factors which play an important role in determining characteristics of the

drive: in practice pneumatic drives are highly non-linear and it is very questionable whether an adequate linear model can be obtained. Linearised analysis can be used in gaining an understanding of the behaviour of pneumatic servos to enable classical control theory to be employed when evolving control strategies. However, it is not necessary to implement conventional linear control system techniques when controlling pneumatic servos (or even hydraulic or electrical drives) through the use of digital computers.

Exciting developments in integrated circuit technologies have provided cheap and powerful microprocessors with fast execution times. Such a trend still continues. With the widespread application of VLSI elements, artificial intelligence has been introduced as an enabling tool in control engineering and enhanced control system performance is possible through the use of a learning. Optimal control is based on the knowledge of the control plant. Unfortunately in many cases, it is difficult or impossible to obtain a model which can provide the necessary knowledge required. However, limited system identification may be possible which can yield "rules" which may be used by a learning algorithm. Thus a knowledge-base for the plant can be built up before carrying out the real control task and used to accomplish a measure of optimisation or improved performance characteristics (the concept of "knowledge base" has been receiving considerable attention in many industrial application areas [99,100]).

Thus "pure" linear control almost certainly will not be satisfactory, but "non-linear" control can be introduced to "optimise" the use of a

linear control loop. Such "non-linear" methods can be achieved through the use of "gain scheduling" or "algorithm scheduling" and this "scheduling" can be "adapted" through learning procedures. In fact the work of Nagaragan and Weston has shown that improvements can be achieved by utilising a combination of linear and non-linear real time control algorithms running concurrently (so called front-end control) which can reduce positioning time: a step forward in the optimal control of positioning time [75].

The non-linear control concepts were implemented by Saffe [101] to control a hydraulic positioning servo system. He used a so-called "switching integrator" to improve the positioning accuracy of the actuator. "Self-learning" procedures were also used to adjust the position loop gain and to generate a nonlinear compensation curve to improve the dynamic behaviour of the position control system. The use of modern control concepts was discussed by Backe etc. [102] to achieve improved behaviour of hydraulic servo and pneumatic servo drives, which had shown the necessity for using "learning" or experimental identification to overcome or reduce the effect of the system nonlinearities. Taha [66] used experimentally-based learning to identify the valve characteristics of a servo pneumatic drive to eliminate the inaccuracy caused by hysteresis in the system.

## 2.5. Summary

Although pneumatic devices have been used for a long time, it is in this decade that new initiatives have emerged in the use of microprocessor-

based pneumatic servo drives. Commercial products were released into the world market around 1985. However, their use is still limited largely to prototype machines designs and experimental investigations. There is great potential for such drives and it is timely for a new generation of pneumatic drives, particularly those having continuous motion control capabilities, to be evolved which can compete and/or complement the use of their hydraulic and electric counterparts. The perceived cost advantage of using pneumatic drives can be loosely identified with medium power industrial application areas according to the survey made above. Cheap electric motors are available at low powers (order of 10 to 100 kw) while hydraulic drives will continue to dominate in high power (100 kw) applications.

The LUT modular robotics research-group has been instrumental in advancing the design of digital pneumatic servos and has a longstanding research interest in this area. This interest led to this research study, where the impetus for the work lay in the identified limitations of present-day pneumatic servo drives as summarised below:

(i) There is a lack of formal basis for the final positioning of pneumatic servos, especially with respect to special null control conditions that occur when using low cost single stage 5-port proportional valves.

(ii) Continuous pneumatic positioning has been achieved, but in general optimisation of performance characteristics is very difficult to achieve.

(iii) Good positioning accuracy is relatively easy to achieve but the good control of velocity is not. In fact, often the positioning errors can be relatively large in many industrial application environments (better than 0.5mm). If high precision is required, pneumatic drives can still be used but long positioning times may result. However, significant advantage would accrue if pneumatic drive can achieve simple velocity control even with relatively large following errors.

(iv) The performance displayed by present servo pneumatic drives is encouraging but the need for complex control algorithms necessitates the tuning of many control parameters so the set-up procedure is normally complex and often poor performance characteristics result. This can defeat the "userability" of pneumatic drives and require highly skilled installation personnel.

(v) Various mathematical models have been proposed but very rarely can be applied effectively to solve control problems. Simple but effective models which can help in the manual or automatic formulation of control schemes are highly desirable.

So far, many of the major research studies of pneumatic servo drives which are related to this project have been outlined. However, some of the important references are not mentioned or not detailed in this chapter but will be referenced or detailed as appropriate in the corresponding chapters. Other valuable reference materials concerning the design of servo pneumatic drives are omitted from this thesis but can be found in reference material relating to work done in the Department of Manufacturing Engineering at Loughborough University.



During this project, the author has been involved in two training areas: (i) Structured or modular software programming; (ii) design of modular robots or modular machines. These activities inspired the author to conceptually consider areas of "similarity and modularity". This builds on previous interests and knowledge of the author in other fields of natural and social sciences. Although the study is not directly related to this project, it can be of general use in a widespread range of science and engineering disciplines. Thus a summary of the author's work is included in Appendix II as a reference document for interested readers. (This is a stand-alone piece of work).

## CHAPTER 3 EXPERIMENTATION DESIGN: METHODOLOGY AND FACILITY

### 3.1. Introduction

As many other research projects do, an experimental environment is required to be set up for evaluating the ideas formed theoretically. Modelling alone is not tenable in gaining a sufficient understanding of pneumatic drive systems so that they can be evolved into servo mechanisms which can find general industrial application, hence necessitating a combination of experimental and modelling studies. Such an approach has been found to be essential in evolving control strategies for pneumatic motion control where interaction between experimental and theoretical approaches can yield advances in control techniques coupled with opportunities for gaining an understanding of the underlying physical mechanisms. The inherent properties of pneumatic drives acting against the achievement of high performance servo control in comparison with their electric or hydraulic counterparts relate to:

- (i) compressibility of the working fluid;
- (ii) non-time varying non-linear properties of pneumatic valves, actuators, transmission elements, etc.; and
- (iii) variation of certain parameters as a function of time resulting from factors including load variation, drift, lubrication, wear and the supply pressure.

Through the body of this thesis, the effect of these properties will be considered and where possible quantified. It is also important to note that similar difficulties can exist in controlling hydraulic and electric drives but the magnitude of the problems encountered are usually of an order of lower significance.

The difficulties involved in accomplishing advanced position and velocity control of pneumatic drives inevitably raises the question "Why should we attempt to evolve drive systems using components which have unsatisfactory inherent properties?" However, this question is simply answered by realising that (1) Economic advantages accrue from utilising low cost industrially accepted components. Many industrial position and contouring application areas can be served by drive system demonstrating limited performance characteristics (viz accuracy, robustness, cycle times). The relationship between performance and cost is, however, a vitally important issue in assessing the potential market-place for servo pneumatics. (2) The continuing trend towards the availability of increased processing power at stable or reducing cost can be harnessed as witnessed recently by the commercial availability of servo pneumatic drives and their industrial usage. This factor is a central theme in this thesis offering opportunities for novel control approaches which would utilise the learning procedures.

In this chapter, an experimental evaluation facility will be described which was evolved during the period of the research study. Essentially the facility represents an enhanced version of earlier prototyping facilities provided at LUT to evolve microprocessor-based controls for

pneumatic drives which subsequently became UK licenced commercial products. The enhancements were included to improve opportunities for generating and testing new control methodologies offering a basis for statistical evaluation of the results obtained.

The result of work of this nature should be statistically valid. However, we should point out that statistical results can overestimate or underestimate the actual behaviour of the control system. It is difficult to tell and compare their validity with respect to industrial application standards or with the results claimed by other researchers. The statistical results are to be used only to illustrate the improvements made by the control methods evolved.

### 3.2. Hardware System

Although there had been an experimental rig available built around Texas 9900 microprocessor from the previous work in the Department, its functionalities were limited both in hardware and software which would impact the flexibility and quality of the experimental system (see Table 3.1.). It was decided that a new experimental system should be designed and constructed to evolve new control methods.

Table 3.1. Limitations of 9900 microprocessor development system

Computer system	Limitations
microprocessor (hardware)	low running rate, limited memory space, dependence on MDS (microprocessor development system) thus, possible interaction with other users
programming (software)	inadequacy of combining low-level and high-level computer languages

The evaluation system produced during the project is shown schematically in Figure 3.1. The kernel of the control system is the MPE 9995 personal computer chosen in 1984 as it provided adequate software development facilities, fast execution of code (4 times faster over the 9900 microprocessor), the ability to integrate code derived from both high level and low level language sources (viz Q-BASIC and TEXAS assembly language) and offered a stand-alone prototyping machine. It should be mentioned that MDS used in earlier LUT research did not offer many of these advantages. Communication between this microprocessor based controller and the interface electronics of the pneumatic drive is achieved through a parallel bus (E-bus) which was introduced by the TEXAS Instruments Corporation around 1980 [103]. As a bus system capable of multiprocessor operation, E-bus offers a very flexible interrupt structure, a variable 8 or 16 bit wide data bus and address range of up to 1 mbytes, as well as three different types of input/output facility (DMA, memory mapped and CRU microprocessors). In

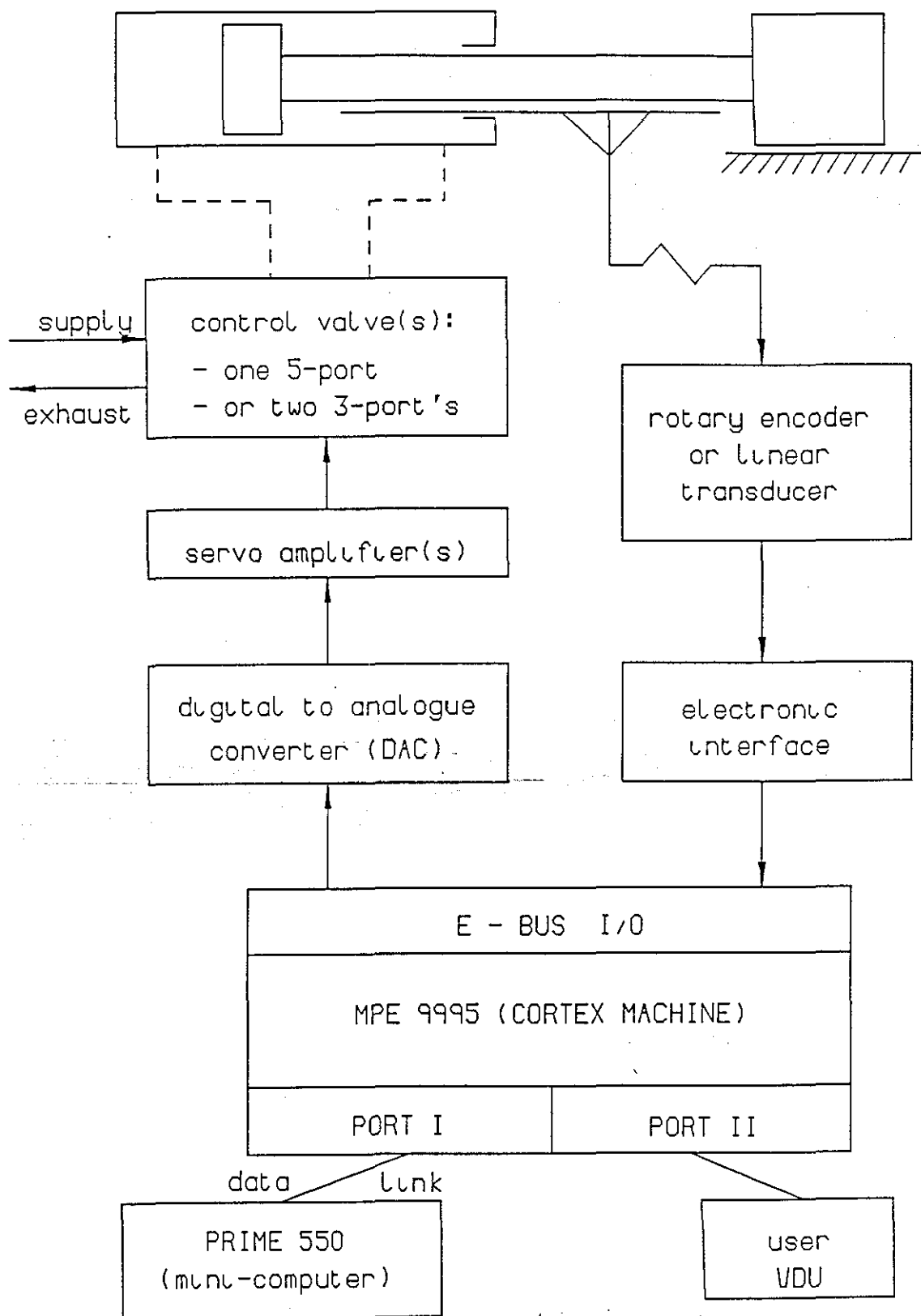
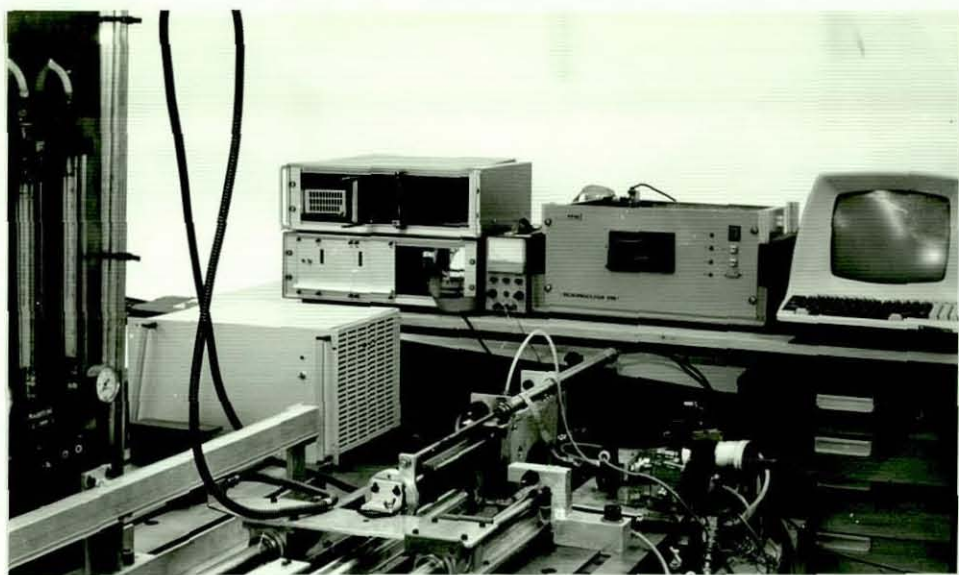
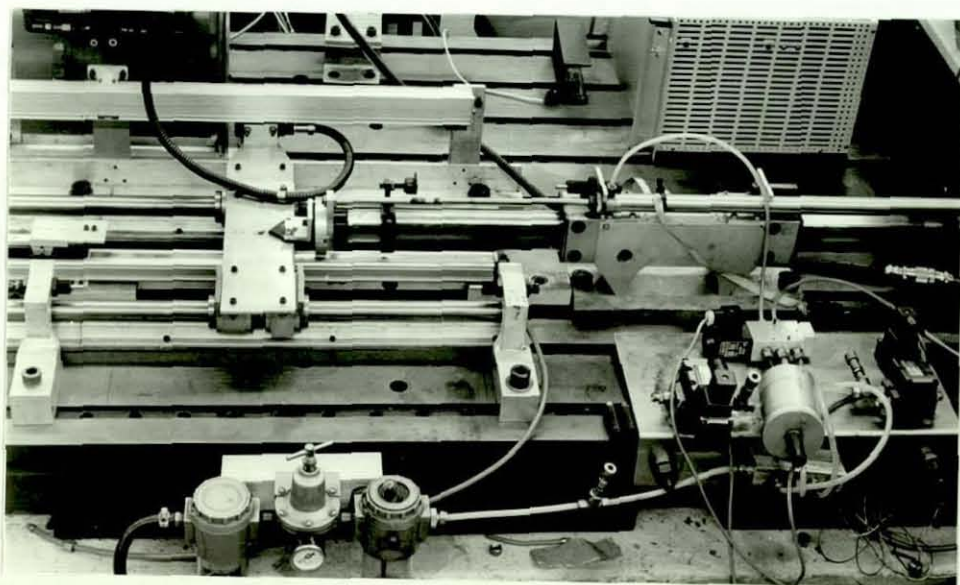


Figure 3.1 Experimental System



Experimental Environment



Actuator and Valve(s)

this case the CRU format is employed to interface DAC with the microprocessor and memory mapped I/O is used for position (and/or velocity) feedback.

The command signal is converted by the digital to analogue converter (DAC) into an analogue one, which is then amplified by the servo amplifier so that the output signal takes the form of a solenoid current. The current thus produced is supplied to the solenoid which in turn generates a magnetic force at one end of the valve spool to offset the spring. Thus the mass flowrate across the valve(s) (two 3-port or one 5-port valve) can be controlled by manipulating the spool offset i.e. through controlling the current supplied to the solenoid. The load is moved as the result of a pressure difference between the two chambers of the actuator. Position and/or velocity information relating to the position/load is sensed by a transducer (rotary or linear) and fed back to the controller. The pulses produced are encoded and interfaced to the microprocessor via the position and/or velocity encoder interface cards, which are mounted on a standard E-bus backplane. A link between the MPE 9995 machine and a PRIME 550 mini-computer are also established so that the motion and command data can be transferred to the PRIME where off-line data storage, graphic display and hard-copy documentation can be carried out.

The characteristics of the actuator and control valve(s) are listed in Table 3.2a. The electronic interface components are described in Table 3.2b. The pressure supply and the load conditions are shown in Table



3.2c. Components in square bracket "[ ]" implies that they are only used, when necessary, for modelling studies or to provide an alternative system configuration. These tables give a description of the experimental conditions (those not in square brackets) under which the control methods are going to be evaluated.

Table 3.2 Data Relating to Experimental Facility

(a) Actuator and valve

---

actuator	asymmetric cylinder: stroke = 400 mm; bore diameter = 25 mm; piston rod = 10 mm. [symmetric cylinder (rodless cylinder) stroke = 600 mm; diameter = 25 mm]
valve	single stage; proportional; solenoid-spring activated;

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(b) Electronic interface components

---

transducer	rotary encoder: 0.025 mm/pulse [linear transducer: 0.01/pulse]
servo amplifier(s)	dither signal frequency: 100 to 200 Hz adjustable
DAC(s)	12 bit(digital): 0 to +10V (analogue)
encoder interface card(s)	16 bit [32 bit available]

---

(c) Pressure supply and load conditions

---

supply pressure	experimental range: 60 to 100 (p.s.i) or 4 to 7 bar; (1 bar = $10^5$ N/m <sup>2</sup> )
exhaust pressure	open to atmosphere
cylinder arrangement	horizontal
friction force	estimated between 30 to 50N
load piston + slideway	= 25 (kg)estimated

---

For the poor commercial support of the Cortex microprocessor and the lack of expertise time and experience in the Department on the E-bus system much time and effort had been spent in debugging and modifying the E-bus circuits on board and the interface cards bought (they were new products when the experimental system was constructed around the end of 1985). Signal conditioning circuits were made to filter the noise and to square and amplify the sensor signals from the sensor to its interface card to the microprocessor. PCB backplane for the servo amplifiers was also built. However, in order to avoid redundant explanation on those electronics, the circuits and modifications involved will not be provided in this thesis. Nevertheless, the author's knowledge in microprocessor hardware and bus system had also been enhanced when doing this practical work.

### 3.3. Software Environment

The hardware architecture adopted (Figure 3.1.) provides support facilities commensurate with the control or evaluation task to be accomplished. The PRIME 550 minicomputer can facilitate off-line data storage and processing and graphical display whereas the Cortex machine can achieve the real time control of the plant and conduct supervisory/administrative task in aiming to automate where possible the repeating of tests whereby statistically valid performance criteria could be obtained.

The necessary software functionality was decomposed into modules and arranged in an organised hierarchical manner. Thus the evaluation

software has a main module (high order primitives) which accesses other modules, which in turn access other modules, etc., until base modules (low level primitives) are initiated and serviced. Individual modules, when needed, provide a man/machine interface which is menu driven, which can be configured in flexible form. In consideration of the research nature of this study (i.e. variation in real time control strategies and different system configurations) special attention to providing a flexible software architecture was desirable and achieved through modular independence, i.e. the functionality of a module should be independent of other modules where possible. Top-down and bottom-up design techniques are both used here. Structured programming has also been implemented, especially when high level languages were used, Q-BASIC on the Cortex and PASCAL on the PRIME computer where the speed of execution is not critical. Assembly language is used in deriving code for the realtime control task and certain administrative tasks to ease parameter passing between software derived from assembler and Q-BASIC sources. To facilitate the efficient and convenient production of software, most of the administrative software for the real time control was written in Q-BASIC.

Complete software environment was organised hierarchically as shown in Figure 3.2. Detailed software flowcharts are provided in Appendix III.

Data transfer from Cortex machine and PRIME minicomputer was accomplished via an RS232 serial link. Here the author was required to produce a low level (assembler derived) driver and message handler (see Appendix III). At the PRIME 550, data are evaluated in two ways: either

statistically calculated and printed on a printer and/or graphically displayed (with hardcopy option) either locally (in the Department) or remotely (at the Computer Centre of the University).

Control programmes on Cortex machine can be classified within two groups of modules: viz (i) the real time control modules which are frequently changed, as various control methods were evolved (see Chapters 5,6 and 7) and (ii) the administrative modules providing facilities for defining initialisation procedures, movement sequences, control parameter selection, etc. and to display and transfer the data to the PRIME.(see Appendix III).

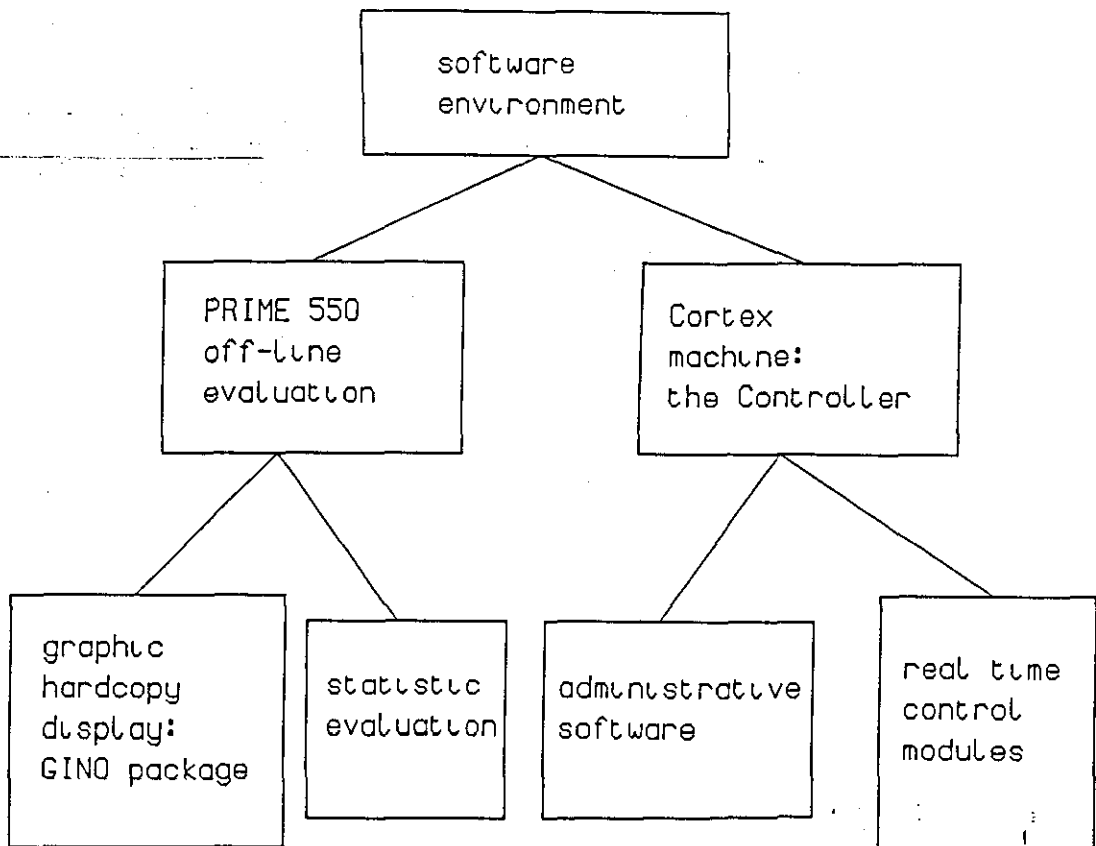


Figure 3.2 Software Environment - Module Schematic

## CHAPTER 4 MODELLING STUDIES

### Nomenclature

$A, A_a, A_b$	ram area
$A_o$	orifice area
$c_1, c_2$	friction component coefficients
$C_d$	discharge coefficient
$c$	total stroke length
$F_e$	external force
$F_f$	total friction force
$F_c$	Coloumb friction (sliding friction)
$K_s$	load stiffness
$K_a, K_b$	simulated air spring coefficients
$k$	ratio of specific heats (for air:1.4)
$k_v, k_{va}, k_{vb}$	flow parameters
$\ell$	half length of the stroke
$M$	piston mass (including load)
$m_a, m_b$	gas mass
$\dot{m}, \dot{m}_c, \dot{m}_a, \dot{m}_b$	mass flow-rate
$N, N_a, N_b$	flow parameters
$n$	polytropic exponent
$p_1$	up-stream pressure
$p_2$	down-stream pressure
$p_{atm}$	atmospheric pressure
$p_a, p_b$	chamber pressure

$P_e$	exhaust pressure
$P_s$	supply pressure
$R$	gas constant
$T_1$	up-stream temperature
$T_a, T_b$	absolute chamber temperature
$V, V_a, V_b$	chamber volume
$x$	piston displacement from the mid-stroke position
$\bar{y}, \bar{y}_a, \bar{y}_b$	normalised spool displacement from the null position of the valve
$\omega$	natural frequency
$\zeta$	damping ratio

Subscript:

a	denotes control chamber on the major side
b	denotes control chamber on the minor side
s	denotes steady state (when used together with another subscript

#### 4.1. Introduction

The severe nonlinearities introduced when using compressed air as the working medium have been described in the literature survey and can easily be demonstrated through experimentation. These raise much greater difficulties in modelling pneumatic drives in comparison with their electric or hydraulic counterparts. Mathematical models for computer simulation of servo pneumatic drives are complex and it is questionable whether they can be used extensively for evolving control strategies. Few other models for such a purpose have been derived in the literature. Small perturbation linearised modelling approaches (which cover most of those models available for servo control) have only a very limited range of validity and do not offer much insight or provide practical value with respect to the particular control problems of pneumatic drives. It is highly desirable to have simple but penetrative models as stated in Chapter 2. The models derived in this study have attempted to highlight the problems when designing the control system of pneumatic drives.

Two particular studies are aimed at (i) providing a simple but comprehensive way of understanding pneumatic servo control systems and (ii) deriving simple but penetrative models which can be used to evolve control strategies, the operation of which can easily be understood and implemented. This somewhat intuitive approach is based on two reasons. Firstly, it is unlikely that a complex and not well understood control algorithm can be used successfully in a practical industrial environment. This reasoning is not difficult to accept if we realise

that most industrial robots use relatively simple control techniques. The more complicated and badly understood, the more likely is it to lead to unsatisfactory performance. It is wiser to use simple and robust control techniques rather than complicated but badly tuned ones. Even if complicated control algorithms can achieve improved or even optimal performance they often do not retain such advantages as reliability, use-ability and maintainability. Secondly, the previous evolution of controllers for servo pneumatic drives (as described in Chapter 2) used simple strategies based around conventional position loop closure, with velocity, acceleration or pressure feedback compensation. The basic structure of those controllers are fundamentally similar to those used for electric and particularly hydraulic drives.

Although some of the control algorithms derived in the main body of this thesis are somewhat complicated, and have been used in this form during the evolution and implementation stages, they have subsequently been simplified into a 'generic' form.

Various modelling approaches or control syntheses have then been used in designing the controller. However, the author is of the opinion that often those models have complicated rather than simplified the problem. Complicated theoretical analyses have been 'used' to draw apparently obvious conclusions, without really addressing the particular control problems inherent in pneumatic drives. Therefore, in this study, no attempt has been made to derive complicated mathematical models for design. Efforts were directed towards studying the specific characteristics of pneumatic servo controlled systems.



#### 4.2. Basic Differential Equations and Important Assumptions

Figure 4.1. shows a schematic model of a pneumatic actuator and Figure 4.2. shows two possible valve arrangements: ports A and B being supplied via either a 5-port valve or two 3-port valves.

Arguably the construction of a mathematical model for pneumatic drives involves three major considerations:

- (1) a determination of the mass flowrate through the valve(s)
- (2) a determination of the pressure, volume and temperature of the air in the actuator (cylinder in this case)
- (3) a determination of the load conditions and resulting motion.

Equation (4.1) describes the mass flow rates [32,58,108]

$$\dot{m} = \dot{m}_c f \left( \frac{p_2}{p_1} \right) = \begin{cases} > 0 & \text{when mass flows into a chamber} \\ < 0 & \text{when mass flows out of a chamber} \end{cases} \quad (4.1)$$

where

$$\dot{m}_c = \frac{C_d A_o p_1}{\sqrt{T_1}} \sqrt{\frac{k}{R} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} \quad (4.1a)$$

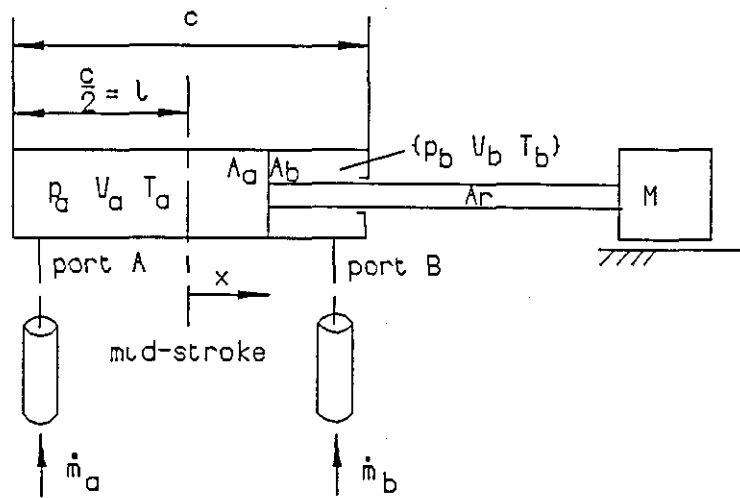


Figure 4.1 Schematic Model of the Actuator

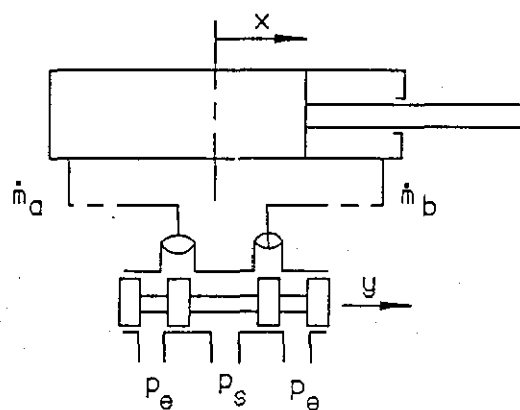


Figure 4.2a One 5-port Valve Arrangement

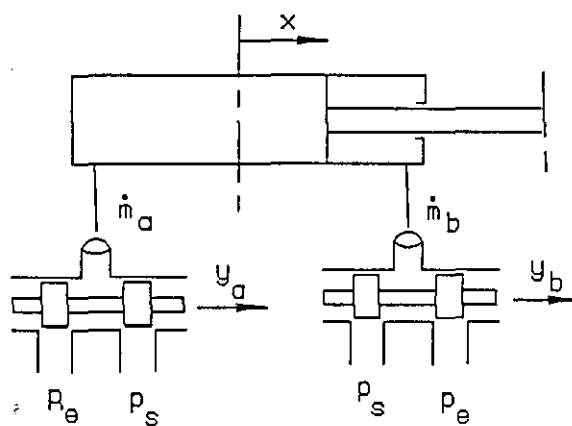


Figure 4.2b Two 3-port Valves Arrangement

$$f\left(\frac{P_2}{P_1}\right) = \begin{cases} 1 & \text{for } \frac{P_2}{P_1} < 0.528 \\ & \text{(choked flow)} \\ \sqrt{\frac{2}{k-1} \left[ \left(\frac{P_2}{P_1}\right)^{\frac{k}{k-1}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k-1}} \right] \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{2(k-1)}}} & \text{for } \frac{P_2}{P_1} > 0.528 \\ & \text{(subsonic flow)} \end{cases} \quad (4.1b)$$

$$\text{for } \frac{P_2}{P_1} > 0.528$$

(subsonic flow)

note: for air,  $k = 1.4$ , so that  $\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.528$

Equation (4.2) assumes the use of a perfect gas and describes a relationship between pressure, volume and temperature of the air in the cylinder.

$$P_a V_a = RT_a m_a$$

(4.2)

$$P_b V_b = RT_b m_b$$

The geometric relationships are

$$V_a = A_a(\ell + x) \quad (4.3)$$

$$V_b = A_b(\ell - x)$$

Equation (4.4) describes the motion equation of the piston and indirectly the load connected to the piston rod.

$$P_a A_a - P_b A_b - P_{atm} (A_a - A_b) = M \ddot{x} + F_f + F_e(t) \quad (4.4)$$

where  $F_e(t)$  is the load force which can include a constant load force and/or a variable load force e.g. a sinusoidal input of a functioning force or any external force disturbance and  $F_f$  is a friction force. The third term on the left-hand side of equation (4.4) is included to account for the fact that the pressures are absolute. However, this third term will be assumed to be negligible in subsequent derivations.

The friction force  $F_f$  can be highly non-linear, and is formed primarily from load, guideway and seal friction components, one formulation being [59]:

$$F_f = F_c \operatorname{sgn}(\dot{x}) + c_1(\dot{x}) + c_2(P_a - P_b) \quad (4.5)$$

For pneumatic drives it is often difficult to predict with accuracy the effect of the friction components as they are influenced significantly by operational factors such as dwell time, ambient temperature,

lubrication, dryness of the air, etc., all of which can vary with time and/or duty cycle.

#### Important Assumptions

Assumption 1: isothermal or adiabatic heat transfer

There are two extreme cases of a polytropic expansion [32].

(a)  $n = 1.4$ , if the cylinder is perfectly insulated

(adiabatic conditions);

(b)  $n = 1$  for isothermal conditions, i.e. if sufficient heat is

transferred to maintain the chamber temperature at a

constant value regardless of the rate at which compression or expansion effects take place in the chamber

where  $n$  is the polytropic exponent.

In the first case (adiabatic conditions), equation (4.2) will take the form [25,59]:

$$\dot{m}_a = \frac{1}{RT_a} \left( p_a \dot{V}_a + p_a \frac{\dot{V}_a}{k} \right) \quad (4.6)$$

$$\dot{m}_b = \frac{1}{RT_b} \left( p_b \dot{V}_b + p_b \frac{\dot{V}_b}{k} \right)$$

whereas for isothermal condition, the counterpart of equations (4.6) will be:

$$\dot{m}_a = \frac{1}{RT_a} (\dot{p}_a \dot{V}_a + \dot{p}_a \dot{V}_a)$$

(4.7)

$$\dot{m}_b = \frac{1}{RT_b} (\dot{p}_b \dot{V}_b + \dot{p}_b \dot{V}_b)$$

or

$$\frac{\dot{m}_a}{m_a} = \frac{\dot{V}_a}{V_a} + \frac{\dot{p}_a}{p_a}$$

$$\frac{\dot{m}_b}{m_b} = \frac{\dot{V}_b}{V_b} + \frac{\dot{p}_b}{p_b}$$

Equation (4.7) can be directly derived when assuming  $T_a$  and  $T_b$  to be constant. For simplicity, isothermal condition is assumed in this modelling study.

Assumption 2: Choked flow or subsonic flow

According to equation (4.1b),  $f(\frac{p_2}{p_1})$  is a complex nonlinear function of a pressure ratio. However, limiting conditions apply as follows:

$$0.8 < f\left(\frac{p_2}{p_1}\right) < 1 \quad \text{when} \quad \frac{p_2}{p_1} < 0.8 \quad (4.8)$$

Therefore, with reference to equation (4.1b), we can conclude that

$$\dot{m} \approx \dot{m}_c \quad \text{when } p_2 < 0.8 p_1 \quad (4.9)$$

That is, the condition of the mass flow can be assumed to be of choked flow nature when equation (4.8) is satisfied. This assumption can be used without loss of significant accuracy when compared with the assumptions made when choosing between equations (4.6) and (4.7). Equation (4.1) can then be approximated as follows:

$$\begin{aligned} \dot{m} &= k_v \bar{y} p_1 \\ &= N p_1 \quad \text{when } p_2 < 0.8 p_1 \end{aligned} \quad (4.10)$$

where

$$N = k_v \bar{y} \quad (4.10a)$$

$$\text{and } k_v = \frac{\dot{m}_{\max}}{p_1(\max)} \quad (4.10b)$$

$$\text{and } \bar{y} = \frac{y}{y_{\max}} \quad (0 < \bar{y} < 1) \quad (4.10c)$$

note that  $y$ , the opening of the valve, is normalised with respect to the corresponding maximum valve displacement,  $y_{\max}$ .

#### 4.3. Characteristics of the Drive System when stationary

##### 4.3.1. Load Stiffness

When the two actuator ports are sealed and the load is stationary (i.e. if the position is constant and the velocity and higher order of position derivatives are zero), then the load stiffness (external force disturbances) can be derived. In such a steady state position, equation (4.4) becomes

$$F = A_a p_a - A_b p_b$$

and a substitution of equations (4.2) and (4.3) leads to

$$\begin{aligned} F &= \frac{RT_a m_a A_a}{V_a} - \frac{RT_b m_b A_b}{V_b} \\ &= \frac{RT_a m_a}{l+x} - \frac{RT_b m_b}{l-x} \end{aligned} \quad (4.11)$$

Assuming that there is no leakage between the two chambers (i.e. the gas mass in each chamber is constant) the load stiffness can be then derived from

$$K_S = F'_x = \frac{dF}{dx} = \lim_{\Delta x \rightarrow 0} \frac{\Delta F}{\Delta x} = - \left[ \frac{Rm_a T_a}{(l+x)^2} + \frac{Rm_b T_b}{(l-x)^2} \right] \quad (4.12)$$

This equation can also be written in the form



$$K_s = - \left( \frac{m_a T_a A_a^2}{V_a^2} + \frac{m_b T_b A_b^2}{V_b^2} \right) R \quad (4.13)$$

or

$$K_s = - \left( \frac{p_a A_a^2}{V_a} + \frac{p_b A_b^2}{V_b} \right) = - \left( \frac{p_a A_a}{\ell+x} + \frac{p_b A_b}{\ell-x} \right) \quad (4.14)$$

At the mid-stroke position of a symmetric cylinder

$$A_a = A_b = A \text{ and } V_a = V_b = A \ell = V$$

In such circumstances equation (4.14) becomes

$$K_s = - \frac{A^2}{V} (p_a + p_b) \quad (4.15)$$

Equation (4.15) takes the same form as the 'model' presented by D. McCloy and H.R. Martin [59] for the stiffness of the fluid in the cylinder with polytropic exponent equal to 1.

$$\text{If } \frac{dK_s}{dx} = 0, \text{ i.e. } \frac{p_a A_a}{(\ell+x)^2} = \frac{p_b A_b}{(\ell-x)^2} \quad (4.16)$$

the load stiffness  $K_s$  will reach its minimum value.

By solving equation (4.16), we have

$$x_{\min} = \frac{\sqrt{\frac{p_a A_a}{p_b A_b}} - 1}{\sqrt{\frac{p_a A_b}{p_b A_b}} + 1} \quad (4.17)$$

$$\text{where } \frac{p_a A_a}{p_b A_b} = \frac{F}{p_b A_b} + 1$$

In the absence of any external force

$x_{\min}$ . stiffness = 0 (i.e. at the midstroke)

However, equation (4.17) indicates that the minimum stiffness is not necessarily at the midstroke position but will be load force (F) dependent.

#### 4.3.2. Equilibrium Point: Hysteresis and Drift Variables

A rearrangement of equation (4.11) leads to

$$F x^2 - (RT_a m_a + RT_b m_b) x + (RT_a m_a - RT_b m_b) \ell - F \ell^2 = 0 \quad (4.18)$$

Thus when  $F = 0$

$$x = \frac{T_a m_a - T_b m_b}{T_a m_a + T_b m_b} \quad (4.19)$$

otherwise,

$$x = \frac{RT_a m_a + RT_b m_b - \sqrt{(RT_a m_a + RT_b m_b - 2F\ell)^2 + 8F\ell T_b m_b}}{2F} \quad (4.20)$$

According to equations (4.19) and (4.20), we can see that there are three types of variable which can cause drift and hysteresis of the piston position:

(1) Temperature variation

Temperature changes (e.g. due to environmental heat transfer) in the chamber will cause gas expansion or contraction and hence pressure changes which could result in drift of the piston away from its previous stationary position.

(2) Friction and external force variation

As previously mentioned friction forces have a complex nature, the effect of which is difficult to predict in quantitative form. Friction forces vary in the working stroke (this is particularly the case with low cost drive elements). Unexpected external force disturbances can also occur. The effect of such problems can introduce hysteresis which will drift as the forces vary with temperature.

(3) Leakage effects

Leakage between the two chambers can lead to a variation in the mass of gas in each chamber. This can cause a drift of the piston position and

indirectly hysteresis problems. It is better to use a symmetric cylinder to reduce the effects of leakage as the difference between the two chamber pressures will be zero if there is no other external force.

Drift and hysteresis can have a significant effect on the positioning accuracy of the drive, particularly for low cost drive elements where significant leakage and friction force variations occurs. However, compensation for or reduction of the effect of drift and hysteresis phenomena can be achieved by using the methods discussed in other chapters of this thesis.

#### 4.4. Steady-state Models under Specified Working Conditions:

- Air-spring model, meter-in and meter-out models

To achieve position and/or velocity control, it would be instructive to consider the dynamics of the drive in response to small perturbations around a defined position or a defined velocity (the parameter defined can be an initial point, a steady-state or equilibrium value or a terminal point, or some operating point). Because of the complexity involved in analysing the control system when using compressed air as the working medium, some extreme but representative cases will be considered to simplify the analysis before analysing the real situation. To appreciate the effect of compressibility in the gas, a simulated air-spring model is derived based on the assumption that both actuator ports are sealed. This model can be used to demonstrate *the effect of* compressibility on position and velocity responses by comparing the model with conventional mass-spring systems. For velocity control, a

meter-in model and a meter-out model are presented. The natural frequency and/or damping characteristics are considered for specified working conditions.

#### 4.4.1. A simulated air-spring model

Combining equations (4.2), (4.3) and (4.4) yields

$$M \ddot{x} + R \frac{m_a T_a + m_b T_b}{\ell^2 - x^2} x = R \ell \frac{m_a T_a - m_b T_b}{\ell^2 - x^2} - F$$

$$\text{or} \quad M \ddot{x} + R \frac{A_a A_b (m_a T_a + m_b T_b)}{V_a V_b} x = R \ell \frac{A_a A_b (m_a T_a - m_b T_b)}{V_a V_b} - F \quad (4.21)$$

(Note: the two actuator ports are sealed, i.e.  $m_a$  or  $m_b$  constant)

Small perturbations with respect to the working temperature, the control volume and the piston position will result in a second-order mass-spring system (see Figure 4.3).

Hence, the equivalent simulated "spring coefficient" of the drive will be given by

$$K'_s = K_a + K_b = \frac{R(m_a T_a + m_b T_b)}{\ell^2 - x^2} \quad (4.22)$$

$$= \frac{R A_a A_b (m_a T_a + m_b T_b)}{V_a V_b}$$

$$\text{where } K_a = \frac{R m_a T_a}{\ell^2 - x^2} \text{ and } K_b = \frac{R m_b T_b}{\ell^2 - x^2}$$

i.e. the drive actuator can be considered to comprise two gas-filled springs, corresponding to the two control chambers, the springs (of chambers) being connected in parallel.

Note that the first term on the right-hand side of equation (4.21) results from a displacement of simulated springs ( $K_a$  and  $K_b$ ) from their original positions at  $x = -\ell$  and  $x = +\ell$  respectively.

Note that equation (4.12) and equation (4.22) are different. Equation (4.12) involves a first derivative while equation (4.22) does not. Equation (4.12) gives an accurate description of the load stiffness whereas equation (4.22) provides a physical characteristic meaning through comparison with an equivalent mass-spring system. However, at the midstroke position of a symmetrical cylinder, equations (4.12) and (4.22) will reduce to the same function as described by equation (4.15). Figure 4.4. provides a schematic representation of equation (4.22).

#### 4.4.2. Meter-in and meter-out models

##### 4.4.2.1. The concept of meter-in and meter-out control

The meter-in or meter-out pneumatic control philosophy is of similar nature to that used in the control of hydraulic drives. The feasibility of those approaches can be considered with reference to the following extremes:

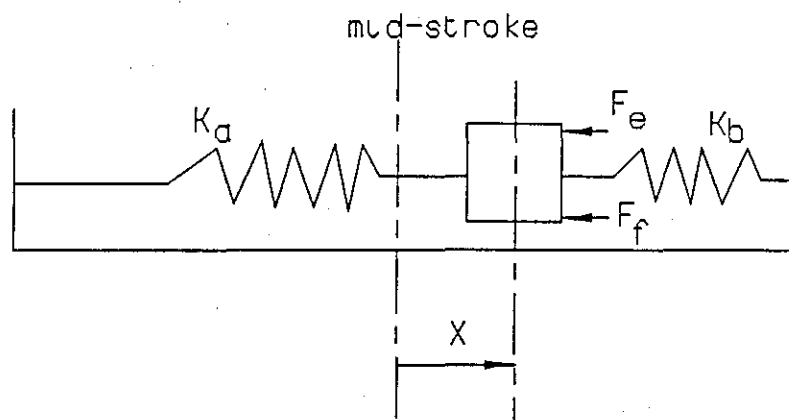


Figure 4.3 An equivalent non-linear mass-spring system

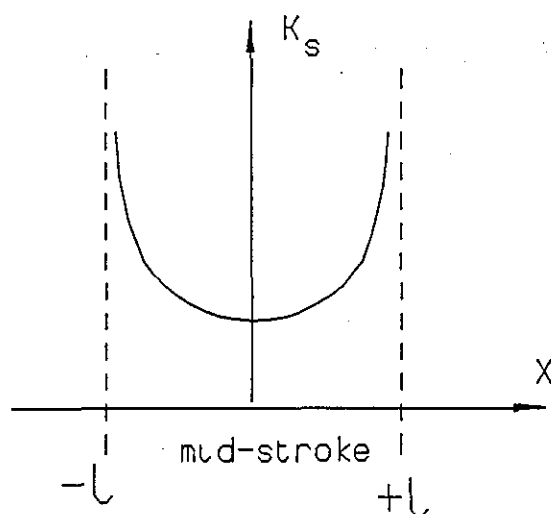


Figure 4.4 Simulated gas spring coefficient

Assume that the piston (and hence the load) is fixed at a particular position (e.g. via a powerful brake), then the control volume on each side of the piston will remain constant. In such a case, the pressure  $p_a$  will build up becoming equal to  $p_s$ , whereas  $p_b$  will fall becoming equal to  $p_{atm}$  (when chamber "a" is connected to the supply pipe and chamber "b" is connected to the exhaust pipe). Then imagine that (the brake is switched off) the piston is released initiating movement. Providing that the rate at which the chamber pressures  $p_a$  and  $p_b$  rise to  $p_s$  and fall to  $p_{atm}$  respectively is relatively large with respect to the velocity of the piston, thus either  $p_a$  or  $p_b$  will be more or less constant. The principle involved in using meter-in or meter-out control is that the pressure in one of the two actuator chambers remains constant while the other chamber pressure is allowed to be changed or controlled. Figure 4.5a and Figure 4.5b respectively illustrate the meter-in and the meter-out control approaches.

#### 4.4.2.2. Meter-in Model (refer to Figure 4.5a)

Equations (4.4), (4.7) and (4.9) are used to derive the meter-in model.

They are re-written as follows:

$$p_a A_a - p_{atm} A_b - F = \frac{M}{A_a} \ddot{V}_a$$

$$\frac{\dot{m}_a}{m_a} = \frac{\dot{V}_a}{V_a} + \frac{\dot{p}_a}{p_a}$$



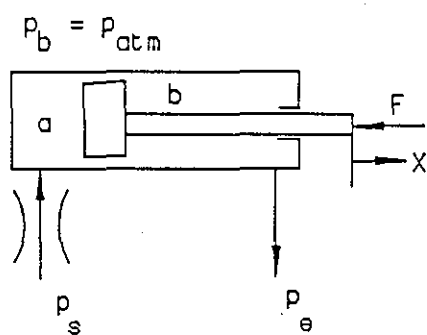


Figure 4.5a Meter-in

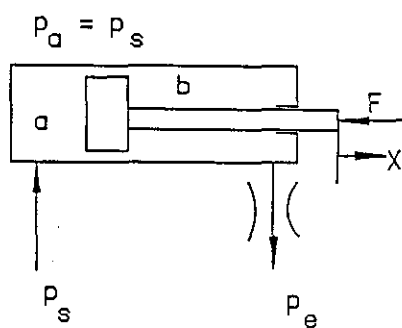


Figure 4.5b Meter-out

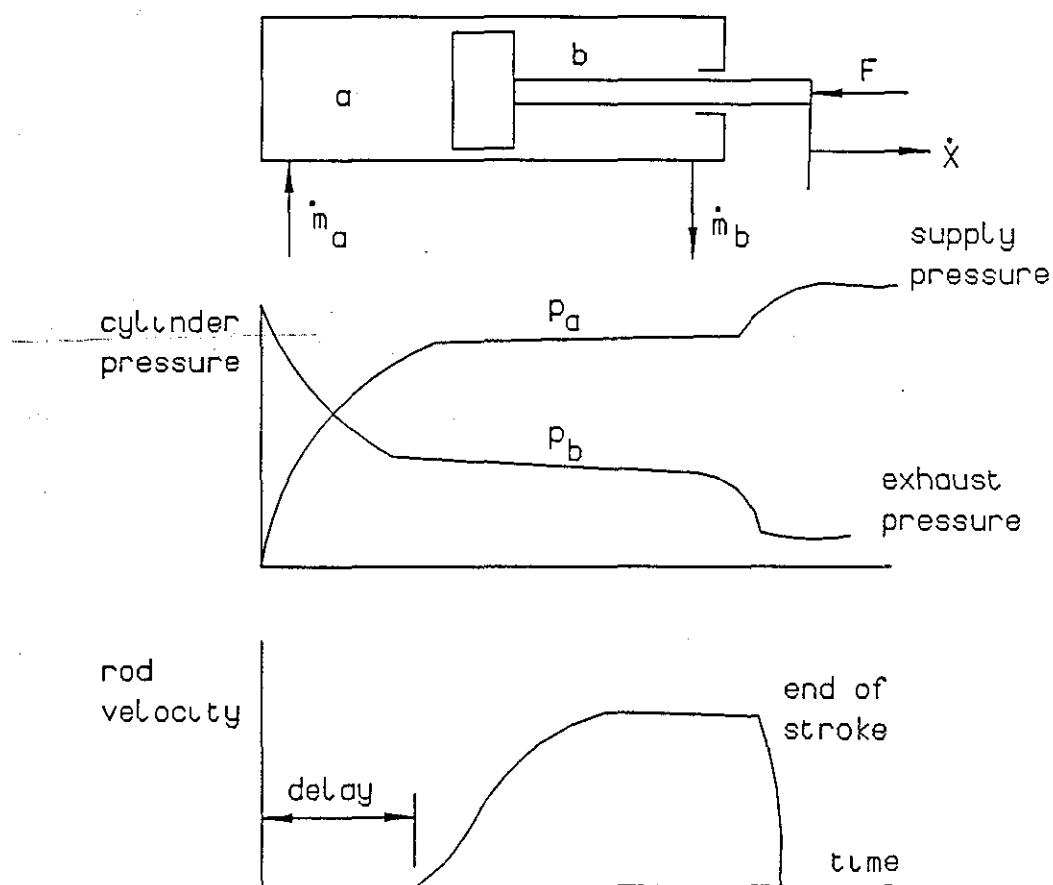


Figure 4.6 Ideal response of a pneumatic cylinder loaded with a constant force

$$\dot{m}_a = k_{va} \bar{y}_a p_s \quad (\text{assuming choked flow})$$

From the above equations, we can conclude that

$$\frac{M}{p_a A_a^2} \ddot{V}_a + \frac{1}{V_a} \dot{V}_a = \frac{k_{va} \bar{y}_a p_s}{m_a} \quad \text{or}$$

$$\ddot{V}_a + \frac{p_a A_a^2}{M V_a} \dot{V}_a = \frac{A_a^2 k_{va} p_s p_a}{m_a M} \bar{y}_a \quad (4.23)$$

For the case where  $\ddot{V}_a$  is assumed to be negligible, then the "mean" steady-state velocity would be

$$\dot{V}_{as} = \left( \frac{p_s}{p_a} \right) R T_a k_{va} \bar{y}_a \quad (4.23a)$$

$$\text{Note } p_a = \frac{1}{A_a} (F + p_{atm} A_b) \quad (4.23b)$$

Then, assuming small perturbations, the linear counterpart of equation (4.23) will take the form

$$\ddot{V}_a + \omega_a^2 \dot{V}_a = \omega_a^2 \dot{V}_{as} \quad (4.23c)$$

where, the natural frequency of meter-in drive

$$\omega_a^2 = \frac{p_a A_a^2}{M V_a} \quad (4.23d)$$

and the natural damping

$$\zeta_a = 0 \quad (4.23e)$$

#### 4.4.2.3. Meter-out Model (refer to Figure 4.5b)

Equations (4.4), (4.7) and (4.9) are also used to derive the meter-out model. This derivation is based on the approach taken by McCloy and Martin [59].

$$p_s A_a - p_b A_b - F = -\frac{M}{A_b} \ddot{v}_b$$

$$\frac{\dot{m}_b}{m_b} = \frac{\dot{v}_b}{v_b} + \frac{\dot{p}_b}{p_b}$$

$$\dot{m}_b = -k_{vb} \bar{v}_b p_b = -N_b p_b$$

From the above equations, we can have

$$\ddot{v}_b + \frac{N_b p_b}{m_b} \dot{v}_b + \frac{p_b A_b^2}{M v_b} \dot{v}_b = \frac{N_b p_b A_b}{M m_b} (F - p_s A_a) \quad (4.24)$$

Equation (4.24) is called meter-out model.

If a constant terminal velocity occurs, i.e.  $\ddot{V}_b = \dddot{V}_b = 0$ , then according to equation (4.24) the steady-state velocity of meter-out drive will be

$$\dot{V}_{bs} = -R T_b k_{vb} \bar{V}_b \quad (4.24a)$$

$$\text{Note, } p_b = \frac{1}{A_b} (p_s A_a - F) \quad (4.24b)$$

Then, assuming small perturbations, the linear counterpart of equation (4.24) will take the form

$$\ddot{V}_b + 2 \zeta_b w_b \dot{V}_b + w_b^2 V_b = w_b^2 \dot{V}_{bs} \quad (4.24c)$$

where, the natural frequency of meter-out drive is

$$w_b^2 = \frac{p_b A_b^2}{M V_b} \quad (4.24d)$$

and the damping is

$$\begin{aligned} \zeta_b &= \frac{1}{2} \frac{N_b p_b}{m_b} \frac{1}{A_b} \sqrt{\frac{M V_b}{p_b}} \\ &= \frac{1}{2} \frac{N_b R T_b}{A_b} \sqrt{\frac{M}{p_b V_b}} \end{aligned} \quad (4.24e)$$

which can also be written in the form

$$\xi_b = -\frac{1}{2} \frac{\dot{V}_{bs}}{A_b} \sqrt{\frac{M}{p_b V_b}} \quad (4.24f)$$

#### 4.4.2.4. Comparison between meter-in and meter-out control

##### Natural frequency

The natural frequency of both a meter-in drive and a meter-out drive is piston-position dependent. Each of them tends to increase as the control volume ( $V_a$  or  $V_b$ ) is reduced.

##### Damping

Based on the assumptions previously described, the meter-in drive displays no damping but the meter-out does. The damping using meter-out control will depend upon the magnitude of the steady-state velocity of the piston; as the steady-state velocity increases so too will the damping.

##### Steady-state or terminal velocity

Firstly, meter-in velocity control is load-dependent and meter-out velocity control is less so. Secondly, it would be instructive to consider the difference between the terminal velocity of meter-in drive and that of meter-out drive. Assume that

$$k_{va} = k_{vb} = k_v \text{ and } \bar{y}_a = \bar{y}_b = \bar{y}, \text{ then,}$$

with reference to equations (4.23c) and (4.24c), we see that

$$\frac{\dot{V}_{\text{meter-in}}}{\dot{V}_{\text{meter-out}}} = \left( \frac{P_{\text{supply}}}{P_{\text{meter-in}}} \right) \left( \frac{T_{\text{meter-in}}}{T_{\text{meter-out}}} \right) \quad (4.25)$$

(Note that, for clarity, we use the complete words corresponding to the subscripts in equation (4.25)).

As  $P_{\text{supply}} > P_{\text{meter-in}}$  and  $T_{\text{meter-in}} > T_{\text{meter-out}}$ , then the magnitude of the piston velocity for meter-in control is greater than that of the piston velocity for meter-out control (it can easily be double in magnitude) with an equal port opening. This indicates that the entry port can play a more important part in determining the stroking speed of the drive. The control modes for meter-out and meter-in will be explained later.

#### 4.4.3. Discussions

The meter-in model and the meter-out model indicate that the terminal velocity of the piston can be controlled either by manipulating the opening of the entry port or the opening of the exit port while the pressure in the other chamber is maintained at a constant value via some means. This conclusion also implies that neither the manipulation of the openings of the entry or exit ports will be sufficient to control the piston velocity without limiting the pressure variation of one of the two control chambers. This point can be emphasised further by considering the simulated air-spring model with respect to a simple experiment:

Assume that the two actuator ports are completely sealed and there is no leakage between the two chambers. Use the hand to pull the piston/load away from its stationary position (say at the mid-stroke), then release it. Oscillation of the piston would be expected as in any mass-spring system. A maximum velocity can be observed when the piston is returned to its original stationary position (i.e. the origin of a spring system). However, this motion has nothing to do with the opening of the entry port or the exit port (they are sealed). It will be instructive to appreciate the relative importance of the opening of the two actuator ports by examining the motion of the piston when one port is opened while the other is sealed. Such a "mass-spring" oscillation motion component can be monitored and stored when performing open-loop valve-actuator tests.

The foregoing illustrates the importance of the relative openings of the entry and the exit ports. However, for a given actuator, the opening of one port (e.g. the entry port) can be more effective than through opening the other port. The magnitude of the mass of the load can also be a significant factor in affecting the transient velocity response before a steady-state velocity is reached (this steady-state condition may not in fact exist with a given size of valve and actuator drive system). These phenomena can explain certain experiments conducted by PERA [90]. Nevertheless, it is difficult to believe that the conclusions drawn by those researchers are generally valid (refer to the relevant literature survey in Chapter 2).

#### 4.5. Choked-flow Model

To analyse the dynamic behaviour of pneumatic drives without considerably restricting their working conditions, neither meter-in, meter-out, or simulated air-spring models are generally appropriate. Ideally the assumptions should closely reflect the actual working conditions of servo pneumatic drives. A choked-flow model will be developed for such a purpose.

##### 4.5.1. Operating conditions to achieve a constant piston velocity

Figure 4.6.(on page 63) shows a possible velocity curve and pressure responses of a pneumatic cylinder loaded with a constant force [59]. However, the actual velocity response is more oscillatory (see Figure 4.6a). Pressure changes may depend on the cylinder size, the valve opening, the supply pressure, and the initial conditions of chamber pressures, etc.

The necessary operating conditions to achieve a constant piston velocity have been investigated by PERA [90]. Some relationships concerning the port opening (entry or exit) and cylinder size were defined in aiming to define those working conditions. These definitions are meaningful. However, it is difficult to comment conclusively on the validity of their experimental results and conclusions as no mathematical evidence is provided.



In order to achieve acceptable velocity control (see Chapter 7) we will see that it is important to establish a relationship between a step valve opening and the corresponding velocity response of the piston. The meter-in model and meter-out model have already shown such a relationship, particularly between a steady velocity and the opening of a single port. However, in servo pneumatic motion control, the manipulation of both the entry port and exit port of the actuator can be important. Let us now consider the necessary operating conditions to achieve an ideal constant terminal velocity of the piston (see Figure 4.6) when manipulating the two actuator ports.

Remembering that for a constant terminal velocity of the piston,

$$\ddot{x} = 0; \dot{p}_a = 0; \text{ and } \dot{p}_b = 0$$

then according to equations (4.1), (4.7) and (4.10), we can rearrange them to give

$$\bar{y}_a = \frac{\frac{p_a A_a}{RT_a} \dot{x}_{\text{const}}}{k_{va} p_s f\left(\frac{p_a}{p_s}\right)} \quad (4.26)$$

$$\bar{y}_b = - \frac{\frac{A_b}{RT_b} \dot{x}_{\text{const}}}{k_{vb} f\left(\frac{p_{\text{atm}}}{p_b}\right)}$$

When  $\dot{m}_a$  and  $\dot{m}_b$  are both choked, equation (4.26) can be rearranged as equation (4.26a)

$$\dot{x}_{\text{const}} = RT_a k_{va} \bar{y}_a \frac{1}{A_a} \frac{p_s}{p_a} \quad (4.26a)$$

$$\dot{x}_{\text{const}} = RT_b k_{vb} \bar{y}_b \frac{1}{A_b}$$

Equation (4.26) or (4.26a) indicate that  $\bar{y}_a$  and  $\bar{y}_b$  must be independently controlled, when achieving a family of open-loop responses with constant terminal velocity, as  $p_a$  and  $p_b$  can vary (which makes  $\bar{y}_a$  and  $\bar{y}_b$  unequal unless by chance). Combining equation (4.26a) will give the relationship

$$\frac{k_{va} T_a \bar{y}_a}{k_{vb} T_b \bar{y}_b} = \frac{p_a A_a}{p_s A_b} \quad (4.26b)$$

Note that equation (4.26b) will be referenced later when deriving the choked flow model.

#### 4.5.2. Choked-flow model

Assume that the mass flow across both entry and exit ports is choked. The following equations are used for deriving choked-flow model (the reader may like to compare these with those used for deriving meter-in model and meter-out model).

$$p_a A_a - p_b A_b - F = -\frac{M}{A_b} \ddot{V}_b$$

$$\frac{\dot{m}_a}{m_a} = \frac{\dot{V}_a}{V_a} + \frac{\dot{p}_a}{p_a}$$

$$\dot{m}_a = k_{va} \bar{p}_s \quad (\text{assuming choked flow})$$

$$\frac{\dot{m}_b}{m_b} = \frac{\dot{V}_b}{V_b} + \frac{\dot{p}_b}{p_b}$$

$$\dot{m}_b = -k_{vb} \bar{p}_b = -N_b \bar{p}_b$$

Rearrangement of these equations gives:

$$\begin{aligned} \ddot{V}_b + \frac{N_b \bar{p}_b}{m_b} \dot{V}_b + \left( \frac{p_a A_a^2}{M V_a} + \frac{p_b A_b^2}{M V_b} \right) \dot{V}_b = \\ = \frac{A_b^2 \bar{p}_b}{M} \left[ \frac{N_b (F - p_a A_a)}{A_b m_b} - \frac{N_a \bar{p}_s p_a A_a}{m_a p_b A_b} \right] \end{aligned} \quad (4.27)$$

Equation (4.27) is the choked-flow model. (This model is a modified form of the meter-out model. The following conclusions assume that the variation of  $p_a$  is relatively smaller than that of  $p_b$ )

Assume that a constant steady (terminal) velocity can be achieved (as determined by equations (4.23a) and (4.24a)), then

$$\dot{V}_{bs} = - \frac{(N_b R T_{bs} + N_a R T_{as} \frac{p_s A_a V_{bs}}{p_{bs} A_b V_{as}})}{1 + \frac{p_{as} V_{bs} A_a^2}{p_{bs} V_{as} A_b^2}} \quad (4.27a)$$

If equation (4.26b) is satisfied, i.e.

$$\frac{N_a T_a}{N_b T_b} = \frac{p_a A_a}{p_s A_b}$$

then,

$$\dot{V}_{bs} = - N_b R T_{bs} \quad (4.27b)$$

which is the same expression as that described by equation (4.24a) which was derived from the meter-out model.

For symmetrical cylinders (i.e.  $A_a = A_b$ ), if

$$N_b R T_{bs} = \frac{p_s}{p_{as}} N_a R T_{as} \quad (4.27c)$$

(which is a simplified form of equation (4.26b) relating to symmetrical cylinders), then

$$\dot{V}_{bs} = -\dot{V}_{as} = -\frac{P_s}{P_{as}} N_a R T_{as} \quad (4.27d)$$

These results show that the steady-state terminal velocity of the choked flow model will be the same as that for the meter-out model if the valve opening of the entry port is set to value satisfying equation (4.26b). Moreover, the expression will be the same as that for the meter-in model (according to equations (4.27c and 4.27d)) if symmetrical cylinders are used. This result indicates that the opening of the exit port should be larger than that of the entry port as  $p_s > p_{as}$  according to equation (4.27c).

Assuming small perturbations, the linear counterpart of equation (4.27) would take the form

$$\ddot{V}_b + 2 \zeta_n \dot{V}_b + \omega_n^2 V_b = \omega_n^2 V_{bs} \quad (4.27e)$$

where,  $\omega_n$  is the natural frequency in the choked flow model

$$\omega_n^2 = \frac{P_a A_a^2}{M V_a} + \frac{P_b A_b^2}{M V_b} \quad (4.27f)$$

Equation (4.27a) can also be written as

$$\omega_n^2 = \frac{K_s}{M} \quad (4.27g)$$

where  $K_s$  is the load stiffness derived in section 4.3.1 (see equation (4.13)).

The damping will be

$$\begin{aligned} \zeta_n &= \frac{N_b p_b}{2 \dot{w}_n m_b} \\ &= \frac{1}{2} \frac{N_b R T_b}{A_b} \sqrt{\frac{M}{V_b \left[ \frac{V_b}{V_a} \left( \frac{A_a}{A_b} \right)^2 p_a + p_b \right]}} \end{aligned} \quad (4.27h)$$

which takes a similar form as the expression for damping in meter-out model. In other words, the damping of the choked flow model is determined by conditions relating to the manipulation of the exit port.

#### 4.5.3. Discussions

It has been mentioned in section 4.2. that the mass-flow can be approximated as a choked flow if

$$p_2 < 0.8 p_1$$

Blackburn etc. [104] and Mannetje [38] estimate that the mean pressure level in the control volumes is given by

$$p_{\text{mean}} = 0.8 p_{\text{supply}}$$

if a 5-port (or 4-way) control valve is used, which has equal size for the inlet and outlet ports. Normally the pressure in the control

chambers will be less than the supply pressure, except in conditions where there is a violent oscillation or cushioning in the control chambers. Therefore, the approximation made is close to the actual working condition. In other words, the choked flow model derived can present the real system fairly accurate especially when compared with other linearised models.

Further conclusions on the validity and use of choked flow model will be discussed in the next section where experimental results are presented.

#### 4.6. Experimentation and Conclusions

Some of the results of the experimental tests conducted by PERA [90] can be used to verify some of the conclusions drawn from the mathematical models derived. For example, the "typical recorder trace" for cylinders operating with restricted inlet and exhaust ports support the discussion in section 4.4.3.3. A set of velocity response curves for pneumatic cylinders are provided along with damping and natural frequency data by RWTH [29] providing evidence which supports the conclusion. Despite the fact that Backe suggested that "the degree of damping and the natural frequency increase with increasing amplitude" of movement (the input current to the valve solenoid is increased, hence, the terminal velocity), the data given in his paper shows that the damping increases with "amplitude" (damping ratio: 0.055, 0.155, 0.25, 0.35) but the natural frequency does not increase accordingly (21, 20.5, 36, 63). However, the first set of data agrees quite well with the damping ratio determined by the meter-out/choked-flow model derived in this study

(refer to equations (4.24f) and (4.27f)). The damping in the system is significantly altered by the terminal velocity whereas the natural frequency is not. The natural frequency might be indirectly affected by the velocity amplitude, as changes in the terminal <sup>velocity</sup> will cause pressure variations in the chambers but is more dominantly piston-position dependent (see equation (4.27f)). Moreover, both the damping and the natural frequency are piston-position dependent. Actually, the piston-position at which the results obtained by Backe would be likely to be significantly different ("increases as the velocity amplitude increases"): the piston position will be different after the same period of time when the piston is moving at different velocities (starting from the same position). The modelling studies have shown that any information about the damping and natural frequency of pneumatic drives will not be adequate enough without specific reference to the piston position. The damping of the system will be significantly increased when the chamber volume connected to the exit port is reduced.

Both the literature survey and the foregoing mathematical modelling have shown that the dynamic behaviour of pneumatic drives is complex and highly non-linear. This implies that theoretically identified conclusions with arguable assumptions should be experimentally verified. Thus certain experimental tests were conducted to verify the mathematical conclusions presented in this thesis. Further observations are classified as follows:

Observation 1: The concept of meter-in and meter-out phases.



The velocity response (see Figures 4.7a and 4.7b) can be roughly divided into two phases: in the first region meter-in control phenomena dominate so that the velocity response is largely determined by the entry port conditions; in the second region meter-out control phenomena dominate so that velocity response is determined by exit port conditions. The first phase will be referred to as the "meter-in" phase and the second phase the "meter-out" phase.

In Figure 4.7a, the opening of the entry port is fixed. The velocity will be approximately constant in the first phase. However, the variation of the opening of the exit port causes a change in the second phase. In Figure 4.7b, the opening of the exit port is fixed. Relatively, the velocity response in the second phase remains more or less constant. However, variation of the opening of the entry port causes a change in the first phase. The author believes that this is an important observation and will utilise this concept in the following observations.

The phenomena demonstrated during the meter-in and meter-out phases may be partly caused due to the changes in the size of control volumes or piston position so that the pressure changing rate of one chamber is relatively small in comparison to that of the other chamber. Normally, for example, pressure variations in the supply chamber relative to the control volume can be bigger in the first phase when compared to the second phase. Obviously, the existence of the meter-in and meter-out phases depends not only on the stroke position but also on the condition of the chamber pressures. The initial conditions of the

NOTE: ( from Figure 4.7 to Figure 4.14)

The actual valve displacement is not necessarily proportional to the magnitude of a DAC command;

i -- denotes initial pressure condition;

% -- indicates normalised DAC command:  $|d|/d_{\max} = |d|/2048$

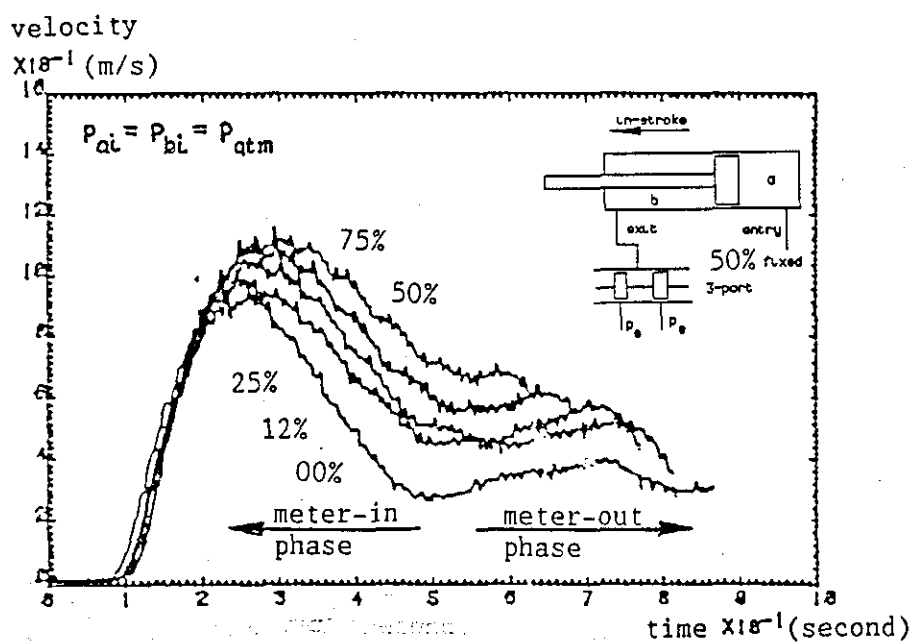


Figure 4.7a Adjusting the opening of exit port with the entry port fixed

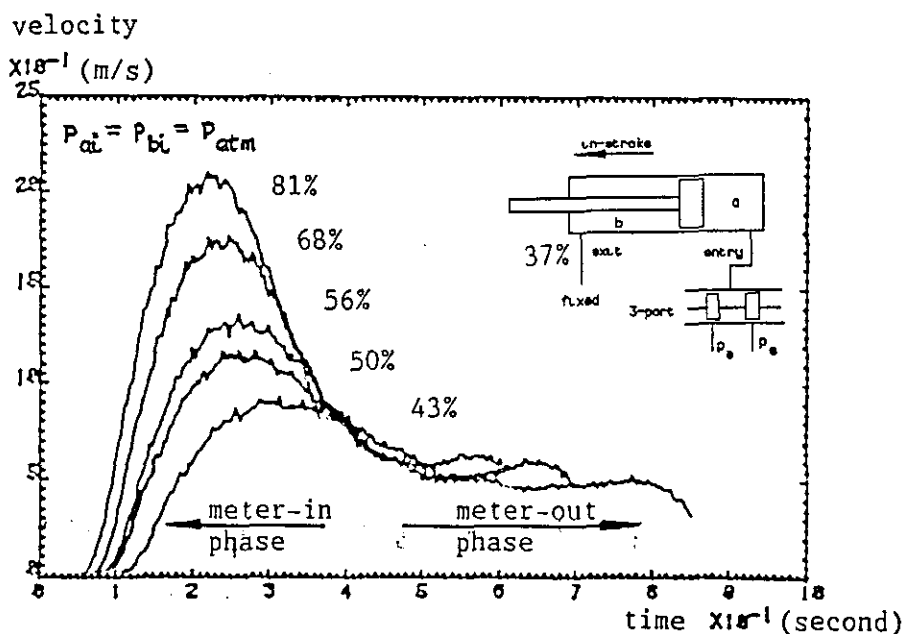


Figure 4.7b Adjusting the opening of entry port with the exit port fixed

chamber pressures can be important as pressure variables will affect both the natural frequency and the natural damping of the system (see equations (4.27f) and (4.27h)). The meter-out phase will be extended when the quiescent pressure in the meter-out chamber is increased. (See Figures 4.7a and 4.7b and Figures 4.8a and 4.8b). The information presented in Figures 4.7a and 4.7b was obtained where the initial chamber pressures were at atmosphere pressure. Figures 4.8a and 4.8b illustrate results corresponding to Figures 4.7a and 4.7b where the initial entry chamber pressure was at the atmosphere pressure and the initial exit chamber pressure was at the supply pressure. However, in servo pneumatic positioning the initial pressure conditions of Figures 4.8a and 4.8b do not apply whereas the information represented by Figures 4.7a and 4.7b is more typical.

#### Observation 2: Functionality of the entry and exit ports

The meter-out model, the meter-in model and the choked flow model (see equations (4.23e), (4.24e) and (4.27h)) indicate that the manipulation of the exit port provides damping while manipulation of the entry port does not (step valve opening). If the characteristics of the entry and exit ports of the valve are similar, then manipulation of the entry port would more significantly affect the stroking speed of the piston than would manipulation of the exit port (see equation (4.25)). Figure 4.9 shows velocity responses when the entry port opening was varied and the exit port was fully open (here two 3-port valves of the same type were used for testing). Furthermore, comprehensive experiments were conducted by PERA to verify the effect of the inlet port size, exhaust

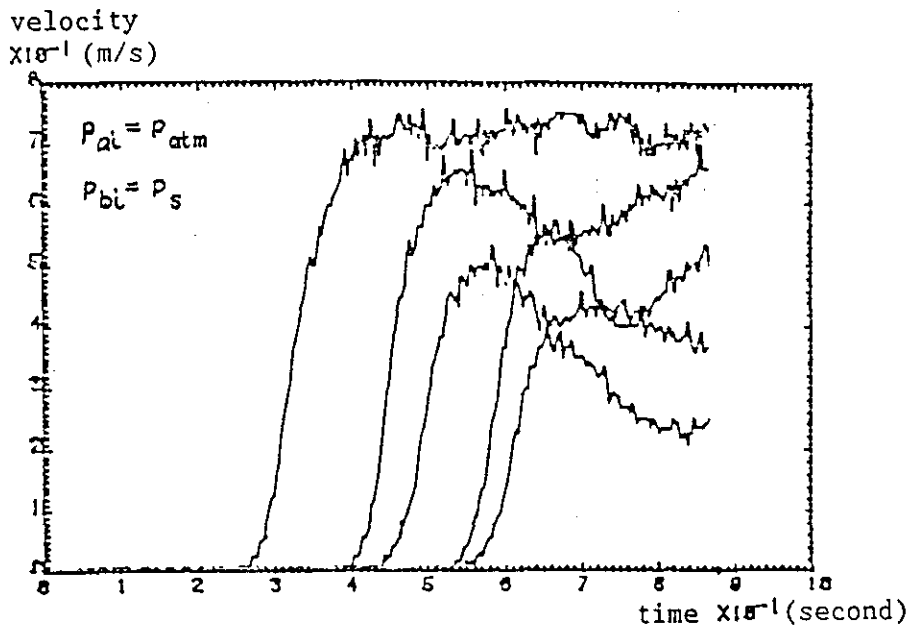


Figure 4.8a Effects of initial chamber pressures on velocity responses of the piston (a)

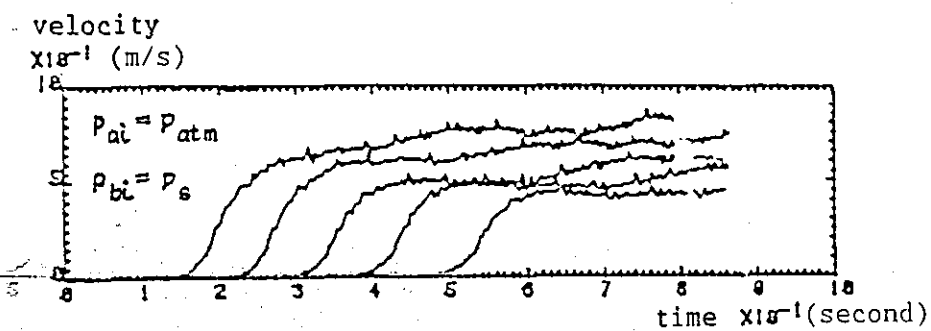


Figure 4.8b Effects of initial chamber pressures on velocity responses of the piston (b)

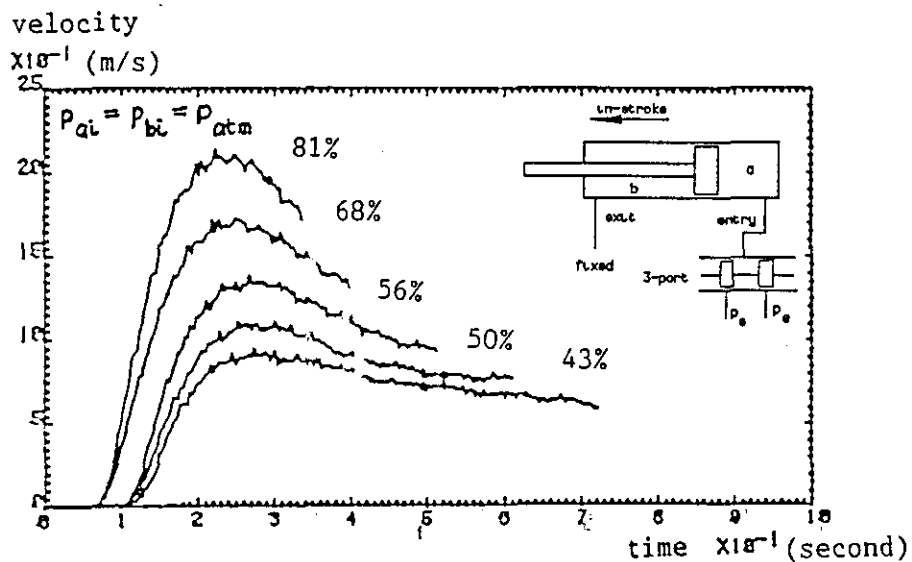


Figure 4.9 Adjusting the opening of entry port with a full opening of exit port ("meter-in" circuit)

port size and external load system on the dynamic behaviour of pneumatic actuators, from which other detailed information about the functionality of the two actuator ports can be obtained.

Observation 3: How to achieve a constant stroking speed with respect to relative openings of the entry and exit ports.

Although manipulation of the entry port has greater effect on controlling the stroking speed, manipulation of the exit port will determine the terminal velocity in the second phase (meter-out phase). To achieve a constant stroking speed, the opening of entry port should be adjusted to match the second phase terminal velocity determined by the exit port conditions. Equation (4.26b) shows that the relationship between the opening of the two ports would depend on the load condition as well as the asymmetry of the actuator. For symmetric actuators, the magnitude of the opening of the entry port should be smaller than that of the exit port, according to equation (4.27c).

Observation 4: Load sensitivity in the meter-in and meter-out phases

It is known that meter-out control is less sensitive to load variation than is meter-in control[59]. This property can also be witnessed in the meter-in phase and the meter-out phase illustrated in Figure 4.10 where two families of curves with loading (solid lines) and without loading (dotted lines) are shown. It can be seen that the velocity amplitude in the meter-in phase has been significantly reduced when

loads are applied whereas in the meter-out phase the effect has been much reduced.

#### Observation 5: Symmetrical or asymmetrical cylinders

Choice between symmetrical and asymmetrical cylinders would raise a question as to whether this makes a difference in terms of stroking speed. Apart from their mechanical characteristics (determined primarily during manufacture) there will be a difference in the velocity behaviour of piston in the two directions (in-stroke and out-stroke motions). Equations (4.27c) and (4.27d) indicate that actuator asymmetry will affect the necessary relationships determining manipulation at the entry and exit ports. A fundamental question concerns "which of the two piston areas will determine the stroking speed?", the drive side, the exhaust side or both and whether their relative importance will depend upon other operating conditions. We can conclude that the piston areas of the exhaust side and the drive side are both important depending upon which stroking phase they are, on the meter-in phase or the meter-out phase. However, it appears that the piston area of the exhaust side is likely to play a more dominant part in determining the velocity amplitude than that of the drive side. Let us consider the steady state velocity behaviour of the piston in the meter-out and meter-in phases.

(a) In meter-out phase (see equation (4.24a)), the steady-state piston velocity is inversely proportional to the piston area of the exhaust side.

(b) In meter-in phase (see equation (4.23a) and (4.23b)) the steady-state piston velocity will be load-dependent. Through combining equations (4.23a) and (4.23b), it can be seen that the larger the piston area of the exhaust side, the smaller the steady-state velocity of the piston. The piston area of the drive side does not affect the steady-state velocity of the piston.

Again, the conclusions derived above are not absolute for two reasons: (i) the analysis is based on the steady-state results not for the transient or dynamic process; (ii) the actual working conditions of pneumatic servo actuators will not strictly meet the assumptions of meter out/in models. Thus, the above analysis only provides guidelines for studying the asymmetry effects of the piston. The cylinders used in this project were not adequate enough to verify the conclusions made above. However, some evidence could be provided through running the asymmetric cylinder in two directions (in-stroke and out-stroke) with the same valve opening. Figure 4.11 indicates that the velocity amplitude of the piston in the meter-in running is higher than that in the meter-out running. (Bear in mind the maximum velocity achieved within the stroke will be position dependent).

#### Observation 6: Supply pressure and load conditions

With respect to observation 4, we have concluded that the meter-out phase is less sensitive to load variations. Furthermore we can conclude that the meter-out phase is less sensitive to the variations in the supply pressure as well. Figure 4.12 indicates that the variation of

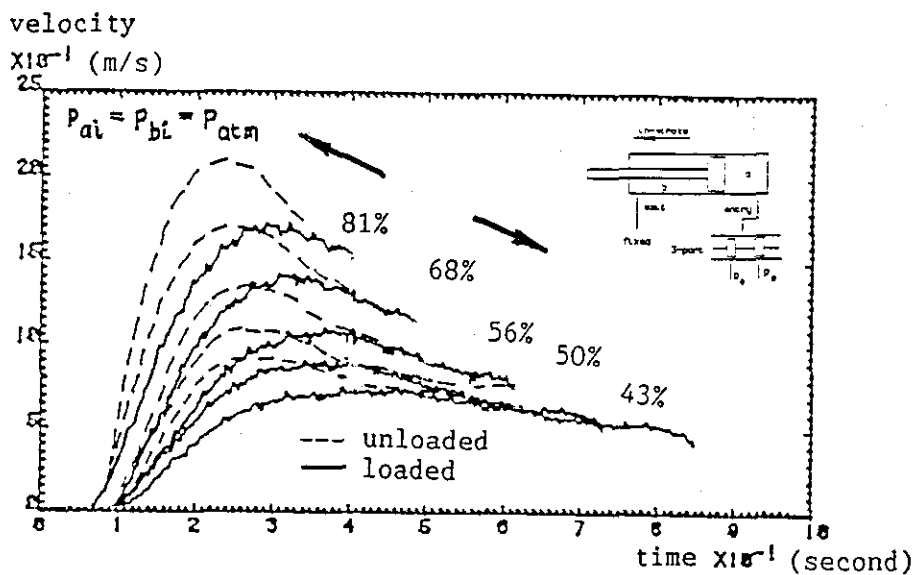


Figure 4.10 Load sensitivity:  
 during both meter-in and meter-out phases

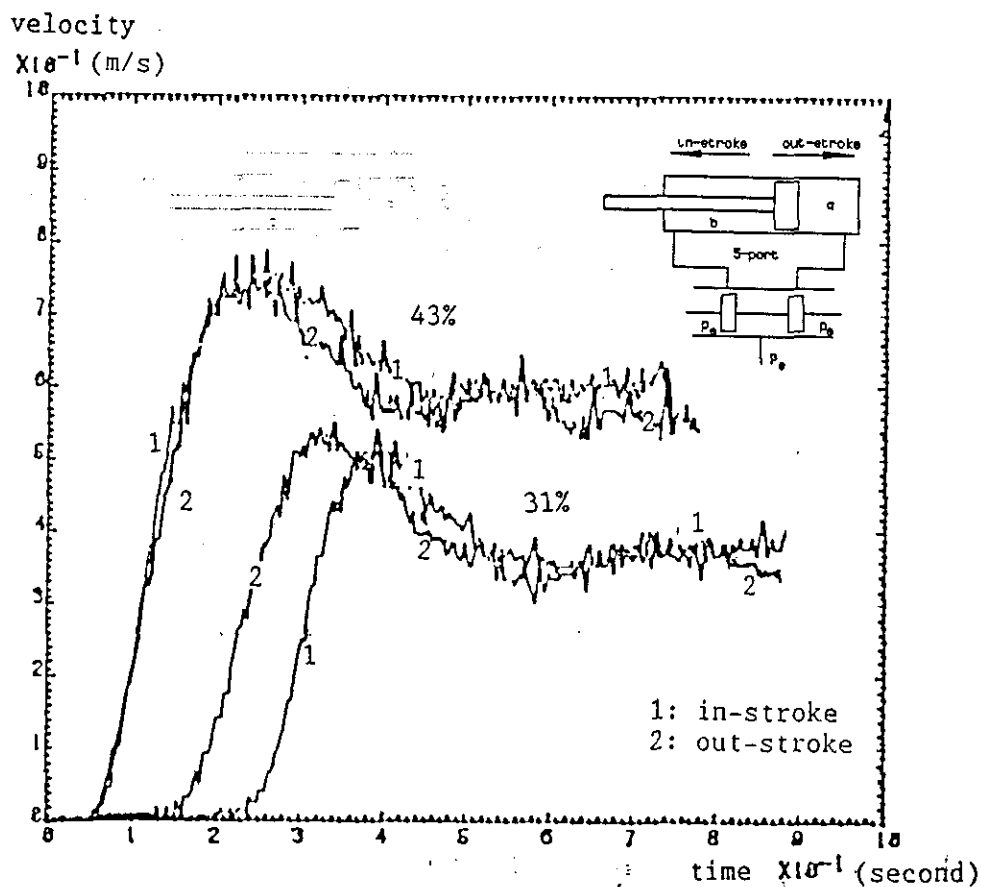


Figure 4.11 Asymmetry effects:  
 for in-stroke / outstroke movements



supply pressure causes a bigger change in the acceleration and/or the meter-in phases than in the meter-out phase.

#### Observation 7: Nonlinearity with initial position of the piston

Both the damping and natural frequency of pneumatic drives is piston position dependent but it is difficult to quantify this conclusion. The experimental results presented by Backe [29] using a combination of 5-port and symmetrical cylinder illustrate that a time delay is introduced because of variations in the initial piston position. However, the effect of initial piston-position on the transient behaviour of the actuator is of a complex nature. Figure 4.13 shows a family of velocity responses when the piston motion begins from different initial positions and with different valve displacements. It can be observed that variation of the initial piston position will cause a change in the time-lag before motion commences which eventually affects the existence of the meter-out and/or the meter-in phases.

#### Observation 8: Choice between the use of two 3-port valves or a single 5-port valve

Choice of valving is an extremely important decision to make as the characteristics of the resulting servo pneumatic control systems will be significantly affected. The observations made above can aid in making the choice between the use of a single 5-port valve as opposed to that of two 3-port valves. There are various advantages and disadvantages of using either of these two approaches:

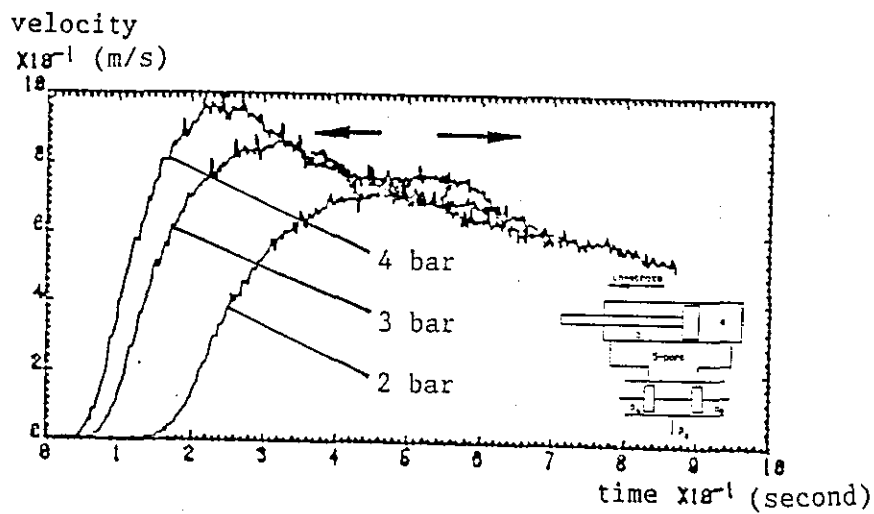


Figure 4.12 Velocity "sensitivity" with respect to supply pressure variations during both meter-in and meter-out phases

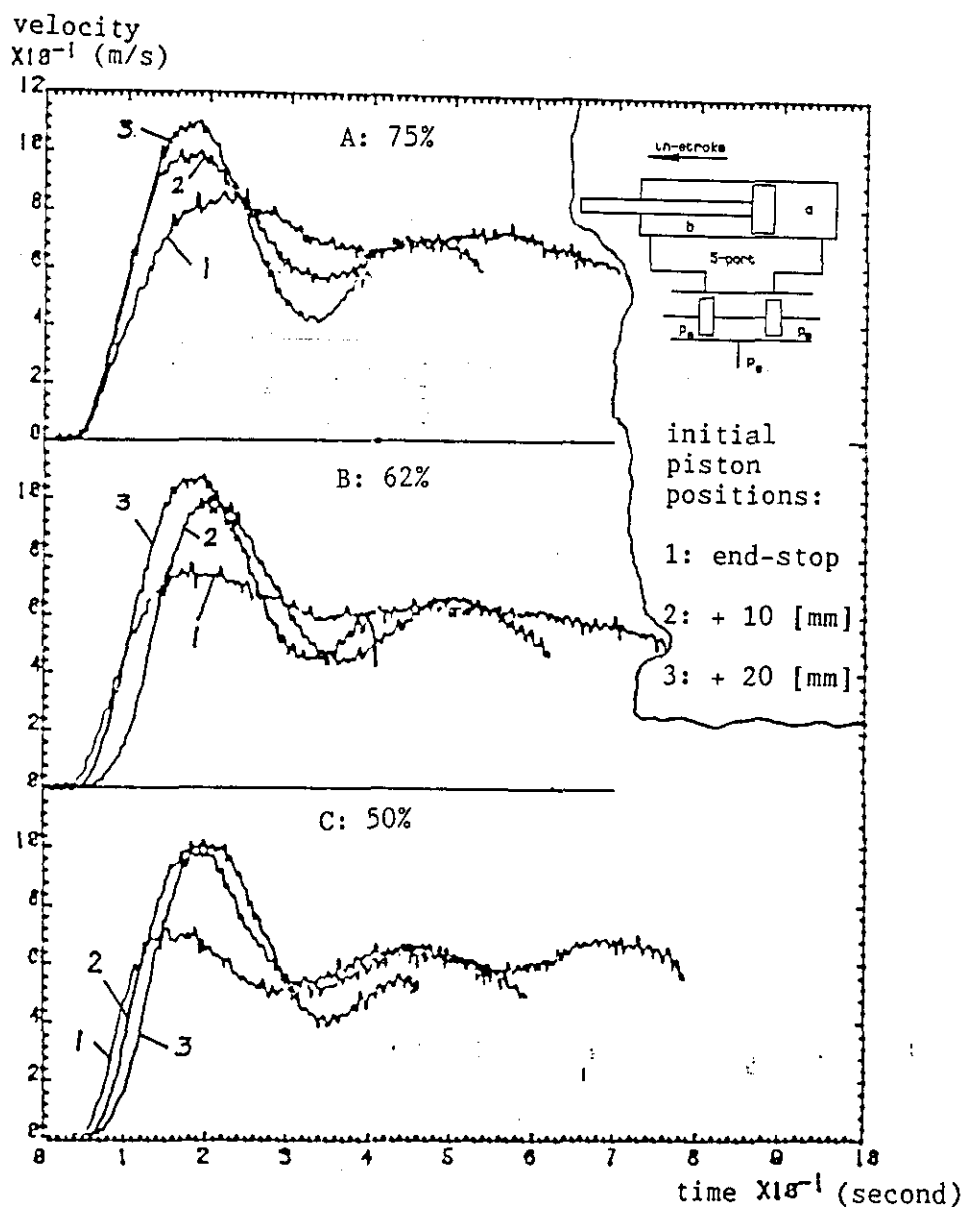


Figure 4.13 Velocity responses for various initial positions of the piston

(i) When using two 3-port valves, one advantage is that they can be integrated within the two ends of the cylinder. Such an arrangement will reduce the response delay (keeping the length of the supply pipes to a minimum) even if the two 3-port valves are used in a similar way to that of a single 5-port valve. Another advantage of using two valves is that independent control of the entry and the exit ports to the actuator can be achieved, the benefits of which have been discussed in the foregoing modelling studies. However, this approach can also be associated with a disadvantage, i.e. the cost of the control system can be significantly increased when compared with its counterpart single 5-port valve system. This fact begs the question "Can the use of a single 5-port valve provide a workable alternative to that of using two 3-port valves?"

(ii) We must remember that most of the previous modelling studies are based on a study of the open loop response with a step change in entry and/or exit port conditions and that often the conclusion relates to the steady-state response. When using a single 5-port valve (where the valve openings to entry and exit ports are approximately equal or at least the spool displacement is the same), it is still possible to achieve a mean terminal velocity (even though this velocity will not be a constant, see Figure 4.14). Moreover, as the dynamic behaviour of the pneumatic actuator is complex, inappropriate or non-optimised manipulation of the two 3-port valves may result so that the actual improvements in performance may be marginal.

A further argument for using a 5-port valve approach is that this has been used successfully to achieve point-to-point positioning previously at Loughborough University where significant capital cost advantages over the use of electric and hydraulic drive systems have been demonstrated. However, the field testing of the previous Loughborough University pneumatic servos had highlighted the need to incorporate some measure of velocity control.

Note, importantly that in obtaining the results presented in this chapter the spool displacement of the 5-port valve and the spools displacement of the two 3-port valves were controlled indirectly through controlling the voltage supplied to a solenoid driver amplifier circuit. This implies that the actual opening of the valve spool is not monitored. The relationship between the command voltage output via a DAC to drive the amplifier, and the actual spool displacement will depend on the linearity, symmetry and stability of the valve(s) activated. In fact, experiments conducted by the author indicated that the characteristics of the 5-port valve used change significantly when the supply pressure was above 4 bar (i.e. the flow forces increased) especially when the valve offset approached its maximum value. However, alternative experiments on the two 3-port valves used in this study showed the 3-port valves to be stable up to 7 bar which was the maximum supply pressure used in this study. (The 5-port valve and the two 3-port valves were in fact from different manufacturers and any generic conclusions herein should be made). The experimental results presented in Figure 4.14 were recorded by setting the supply pressure to 7 bar and studying features of the 5-port valve. This characteristic is of

importance in the context of the rest of this thesis as it represents conditions often used when deriving subsequent control strategies. However, the results presented in this chapter were mainly obtained using a supply pressure of 4 bar to reduce the spool offset and avoid the introduction of inherent errors through changes occurring in the valve.

Before finishing this chapter, the author would like to emphasise that in general the complexity embodied in the operation of pneumatic drives makes it very difficult to quantify interrelations between variables. The conclusions developed from a consideration of the extreme cases (meter-in and meter-out) do not apply at all times but provide guidance and a level of understanding of the behaviour of the system which will be vital in evolving and explaining the operation of control strategies.

velocity

(m/s)

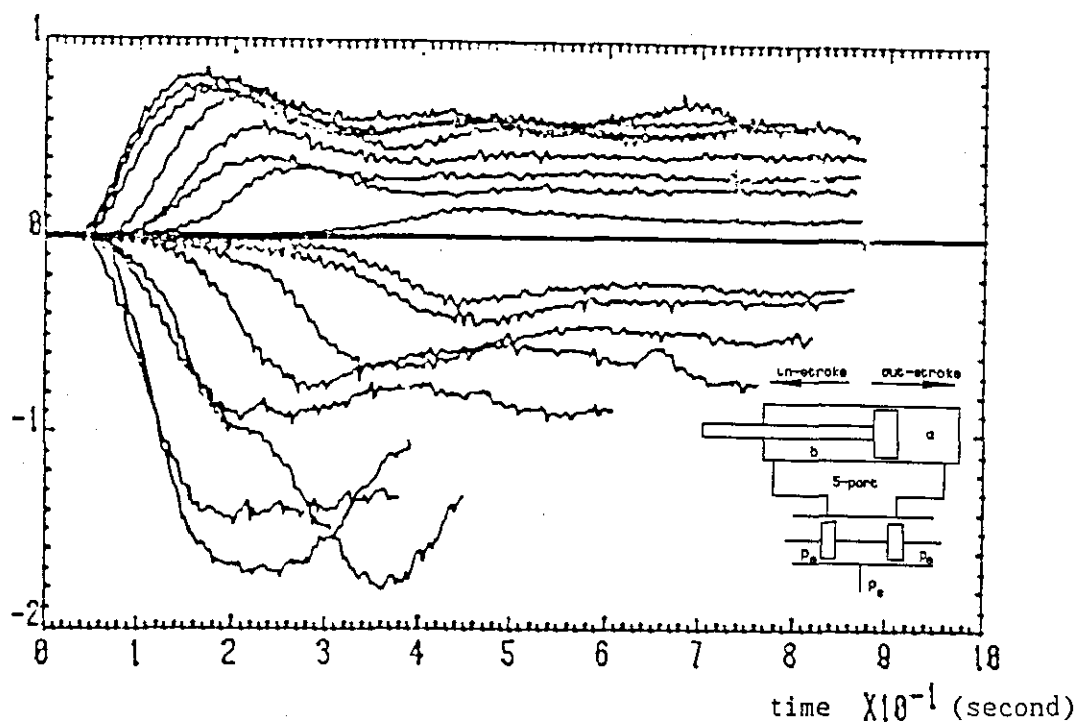


Figure 4.14 Piston velocity versus DAC command:  
step responses of a servo pneumatic open-loop  
( 5-port valve-cylinder arrangement )

## CHAPTER 5

### REDUCING HYSTERESIS AND DRIFT EFFECT BY AUTOMATING NULL CONDITIONS

#### 5.1. Introduction

Drift and hysteresis are common phenomena in many kinds of system components, e.g. electrical devices like transistors and coils and mechanical devices like gearboxes and cams. Any control system comprising individual components of this type will also exhibit drift and hysteresis. Hysteresis is observed as a "deadzone" in the input which causes no change in the output whereas drift is used to describe the changes in the output with time while the input remains unchanged. There also exist various methods to overcome hysteresis and drift on different types of system. For example, the hysteresis in hydraulic/pneumatic servo/proportional valves is overcome by applying "dither" signals (an oscillating signal superimposed upon the control signal) [22]. The "jerking" motion of an actuator driven by an electric motor at a low speed is improved by utilising dither signals [105]. The limit cycling of a hydraulic positioning actuator is eliminated by using "switching integrators" [101].

For pneumatic drives, the experimental study conducted by Araki [85] indicated that a "biased" signal would be necessary in order to maintain the piston at the midstroke so that the piston position would not "drift" away during the testing process. Simulation study by Mariuzzo [56] showed that the hysteresis in a pneumatic motor could be reduced by using "biased" driven signals thus positioning accuracy increased. The

"offset" techniques of using "biased" signals are also applied to overcome the hysteresis in electric drives and other servo mechanisms which may have similar effects.

The directional effect of static friction forces is one of the common sources which can introduce hysteresis where mechanical systems are involved. The presence of significant friction forces, coupled with leakage effects in pneumatic drives (valve and actuator) will introduce hysteresis and drift when positioning. The problems of friction and leakage have been investigated by various researchers [32,36,46]. The phenomena of hysteresis and drift in pneumatic drives also have been considered in previous work as mentioned above. However, to the author's knowledge, the relative importance of leakage and frictional effects as well as drift and hysteresis on static performance have not been evaluated or formalised. For pneumatic drives, from a control view point a well-founded systematic solution is lacking due to the problem of achieving a final or steady state position. This chapter will describe the work of the author on identifying the effects of leakage and friction on the null condition of a pneumatic positioning mechanism and how automated approaches to deriving the null condition are developed to improve the static performance of the system.

In order to gain a good understanding of the behavioural characteristics of a pneumatic drive mechanism with significant levels of friction and leakage, the system has conceptually been disassembled, isolated and reassembled using the following procedures.



Firstly, let us imagine that the load resides on the surface of a table where friction (denoted as  $F_f$ ) (see Figure 5.1a) always opposes the motion of the load with or without a driving force (denoted as  $F_d$ ). Secondly, assume that the driving force is applied via two air-driven "soft" springs (see Figure 5.1b). Let the mass flowrates through port A and port B of the air springs be regulated by their control valves. In fact this conceptual model resembles features of an actual pneumatic drive.

Friction force will oppose motion so that a load will be kept at a stationary state when

$$F_d < F_f . \quad (5.1)$$

where  $F_f$  is the critical friction force when the load starts to move, which is bigger than the friction while the load is moving (Coloumb friction). In later discussions, we shall not distinguish the difference between these two friction forces, which both sometimes are simply referred to as "friction force".

In such a stationary state, the real friction force opposing motion will be between zero (when no external force applied) and the static friction force. Therefore, there will be a "deadzone" or hysteresis in the drive force which causes no movement of the load. (Correspondingly, there will be a "deadzone" in the DAC output which causes no movement of the load using pneumatic drives resulting from both friction and leakage).

Two approaches can be undertaken for identifying the two boundary values of the "deadzone" of the driving force (see equation (5.1)). (The motion of the pneumatic drive system about a stationary position can be described by an equivalent electronic circuit as shown in Figure 5.1c).

(1) The value of the driving force which is just large enough to make the load begin to move will be the boundary value of the driving force or the critical value of the static friction force.

(2) The driving force which keeps the object moving at a constant velocity will be the Coloumb friction force, particularly when the constant velocity is approaching zero so that for pneumatic positioning, the corresponding DAC signals will be one of the two boundary values of the deadzone (for one direction). It is clear that a positioning error will be introduced if the load comes to a stationary state before reaching its commanded position, i.e. the magnitude of the control signal will not result in a drive force  $F_d$  greater than the opposing friction force  $F_f$ .

Thus, we can classify the control methods mentioned above for overcoming drift and hysteresis by a combination of two kinds: "dither" methods and "biased" or "offset" techniques. The use of "biased" or "offset" techniques will be emphasised in the following study.

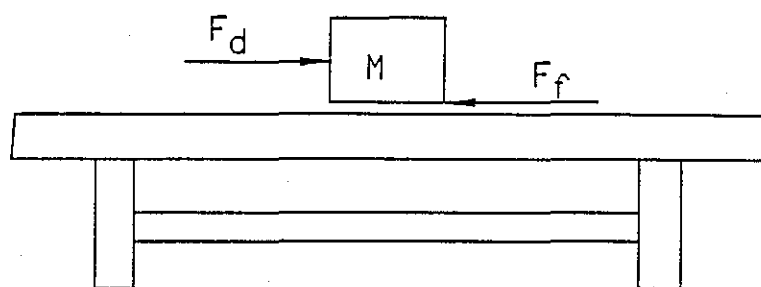


Figure 5.1a

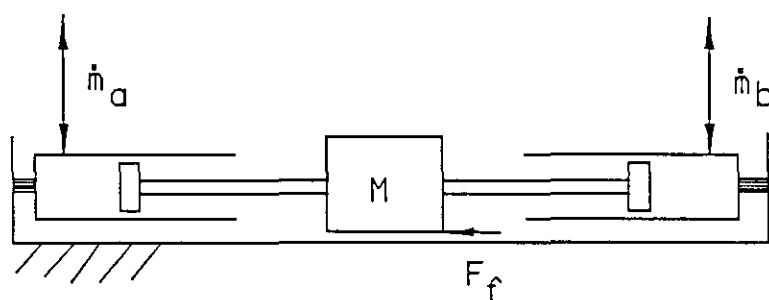


Figure 5.1b

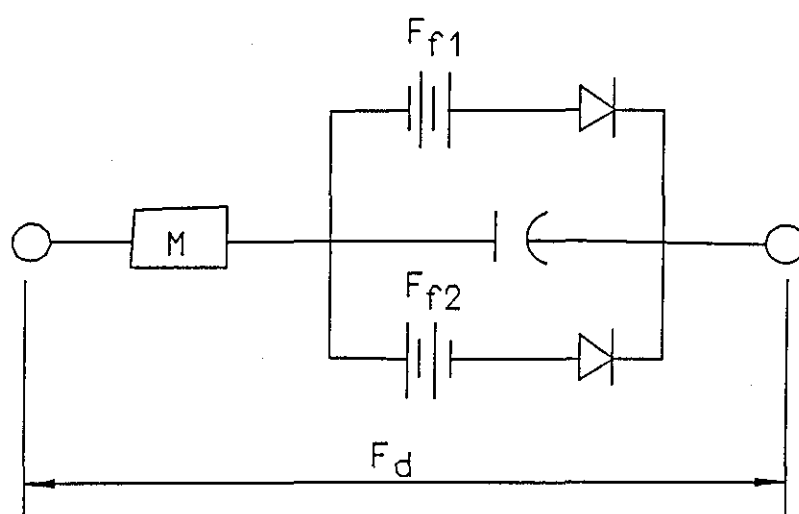


Figure 5.1c

## 5.2. Definition of the Null Conditions and their Evaluation

Here the features of null conditions for a pneumatic drive system are studied with reference to the use of a single stage 5-port servo valve. Two types of null value for the control system are specified:

(i) the null value of the 5-port servo valve, denoted as  $n_v$ , which is the digital number transmitted from the microprocessor exit port to the DAC which in turn corresponds to a null position in the spool displacement; and

(ii) the null value for the complete system with a given payload, denoted by  $n_s$ , which is the digital number transmitted from the microprocessor exit port to the DAC causing no movement of the payload in a specified period of time.

According to the definition of null value it can be observed that the system null value is referred to the DAC value within a deadzone being characterised by two boundary values  $n_1$  and  $n_2$ , which keeps the load in a stationary state (see Figure 5.2). The absolute difference between  $n_1$  and  $n_2$  is considered to be the deadband of null condition. The "drift" of the system null condition at a given piston position is referred to as the variation of the mean value of  $n_1$  and  $n_2$  with time.

Whereas they are controlled from the same microprocessor I/O port, the deadbands associated with  $n_v$  and  $n_s$  are different as shown in Figure 5.2.

The deadband of the system will always be larger than that of its system components. The "natural" hysteresis in the solenoid valve is overcome in this study by using an oscillatory "dither" component (i.e. by adding a sinusoidal signal to the drive signal of the valve [22].) In general, the magnitude of the deadband for the valve can be assumed to be negligible when compared to that of the system deadband.

Figure 5.3a is a hysteresis curve of the boundary null values  $n_1$  and  $n_2$  against various load/piston positions (changes in hysteresis are ignored during the experimental tests). (The experimental test procedures to obtain these results are described in Appendix IV). Further experiments have shown that drift in the system null condition is normally small but that the variation in hysteresis with time can be significant. Figure 5.3b shows the system null value at different supply pressures in the mid-stroke region of the actuator, from which we can see that the deadband varies significantly but variations in the mean value of  $n_1$  and  $n_2$  are relatively small. (Again Appendix IV describes the experimental procedure followed in obtaining these results).

The load will remain stationary within the working stroke of the actuator when

$$|(p_a - p_b)A_a + (p_b - p_{atm})A_r - F_e| < F_f \quad (5.2)$$

where  $(p_a - p_b)A_a + (p_b - p_{atm})A_r$  is the drive force and  $F_e$  is a constant external force opposing motion. The friction force  $F_f$  can have a very complex nature (see equation (4.5) in Chapter 4).

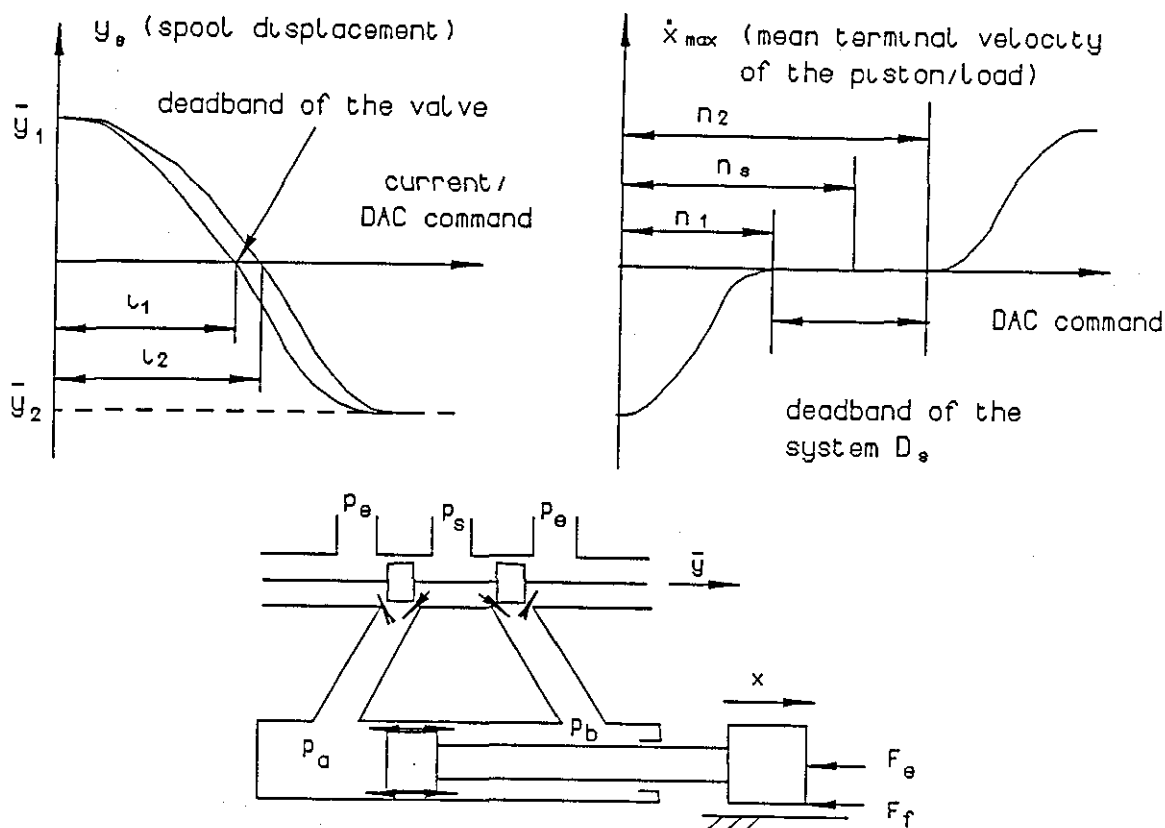


Figure 5.2 Illustrating "null conditions"

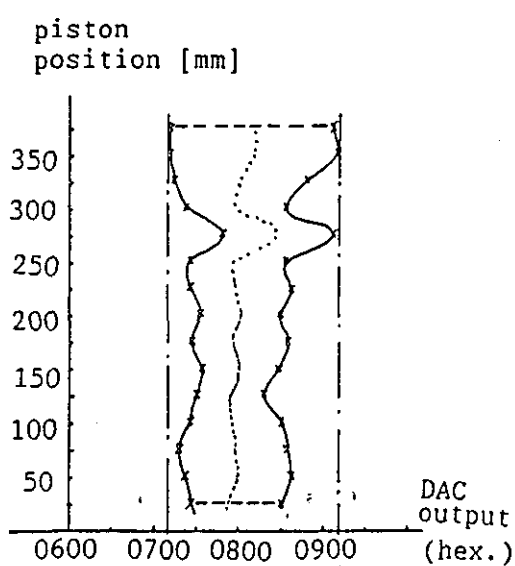


Figure 5.3a System null value versus piston position

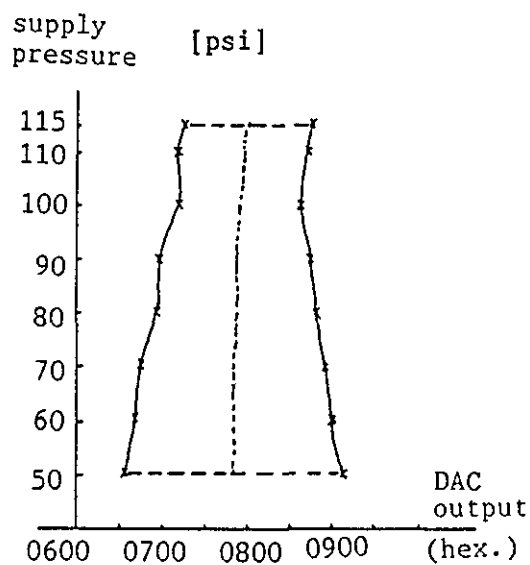


Figure 5.3b System null value (at mid-stroke) versus supply pressure

Let us now consider how friction and external forces can affect the null value,  $n_s$ , and the deadband,  $D_s$ , of the system by anticipating the effect of leakage in the valve ( $L_v$ ) and between the two control chambers of the actuator ( $L_a$ ). (Port A is assumed to be the entry port connected to the supply).

A. Without leakage  $L_v = 0$  and  $L_a = 0$

In the absence of fluid leakage in the control chambers or across the spool of the valve, over a period of time the pressures in the control chambers can become equal to the supply pressure ( $P_s$ ) and the exhaust pressure ( $P_e$ ) respectively when the spool is offset from its null position. Consequently the load will be moved if

$$|(P_s - P_e)A_a + (P_e - P_{atm})A_r - F_e| > F_f \quad (5.3)$$

by replacing  $p_a$  and  $p_r$  with  $p_s$  and  $p_e$  respectively in the left side of equation (5.2). Assuming the exhaust pressure is equal to the atmospheric pressure, we have

$$|A_a (P_s - P_{atm}) - F_e| > F_f \quad (5.4)$$

A delay period will result before the working pressures are built up. That is to say that the magnitudes of the null values and the deadband values will be the same for both the system and the valve.

B. When  $L_v = 0$  and  $L_a \neq 0$

If there is leakage between the control chambers, a fraction of the driving pressure difference between  $p_a$  and  $p_b$  will be lost: the fraction being dependent upon the leakage condition. The spool position may need to be offset from its null position to overcome the external force  $F_e$  thereby compensating for the leakage. In general, the magnitude of the friction force will be dependent on the direction of motion thereby imposing a hysteresis component on the null value of the system. Furthermore, the null value for the system will not correspond to that for the valve with the offset being a function of the external and friction forces as well as the direction of motion.

C. When  $L_v \neq 0$  and  $L_a = 0$

In this case, the result will be more or less the same as in B except that the system dynamics will be more complex as the leakage across the valve will occur at a much greater rate due to the smaller volumes involved.

D. When  $L_v \neq 0$  and  $L_a \neq 0$

This can be considered to be a combination of cases B and C above and corresponds to the practical situation. Again the null value for the system will not correspond to the null value of the valve due to the



presence of leakage and the difference between the two values will be a function of the friction and the external forces.

Obviously, variation in the supply pressure and the exhaust pressure will have a significant effect on the system null value and deadband (see Figure 5.3b). As factors such as leakage, friction, external force, and the null values of the valve change, so too will the null value and the deadband of the system. The complexities involved in theoretically formalising this analysis led to a series of qualitative tests being carried out to verify the conclusions drawn from the analysis presented above. Furthermore, early experimental tests illustrated the complex nature of null conditions and the fact that the null conditions were not repeatable. Generally speaking the drift of mean null value was found to be relatively small whereas the drift in the magnitude of the deadband or hysteresis of the system null value can be significant. However, it will be shown that improvements in reducing positioning errors still can be achieved in software by automating the null conditions when significant drift in system null values exists.

### 5.3. The Importance of Null Values and the Deadband

The DAC command output which activates the solenoid of the servo-valve comprises two parts, viz: the null value  $n'_s$  (defined as being the estimated DAC command for which the system is expected to be stationary) and the demand offset  $d'$  which offsets the spool of the valve to increase the flow-rate thereby motivating the load. Thus

$$d = n'_s + d' \quad (5.5)$$

when the load is in a stationary condition (i.e. velocity and acceleration are zero) the effect of  $d'$  is governed by the nature of the control laws implemented within the controller. For simplicity let us consider the case of a proportional control algorithm

$$d' = k_p e \quad (5.6)$$

where  $e$  is the position error and  $k_p$  the position-loop gain. Some features of the effect of forward path gain are discussed by Weston et al [71].

In the case of proportional control, equation(5.5) can be re-written in the form

$$d = n'_s + k_p e \quad (5.7)$$

From equation (5.7), we see that when the system is in a stationary state, the DAC command  $d$  will actually be a system null value  $n_s$  according to the definition of null condition, therefore,

$$n_s = n'_s + k_p e \quad (5.8)$$

Note however (with reference to Figure 5.2), that  $n_s$  will not be uniquely located and  $n_s$  will in fact be somewhere within the deadband, i.e.

$$n_1 < n_s < n_2 \quad (5.9)$$

where  $n_1$  and  $n_2$  are the lower and upper limits of the null value for the system respectively so that the deadband of system null condition is

$$D_s = n_2 - n_1 \quad (5.10)$$

From equation (5.7) we see that

$$e = \frac{n_s - n'_s}{k_p} \quad (5.11)$$

which leads to the following conclusions.

- (i) When  $n'_s$  is ill-defined, an unnecessary position error is introduced.
- (ii) The greater the value of the proportional gain  $k_p$ , the smaller the error  $e$  when the stationary condition is reached.
- (iii) Position errors can still be introduced for a fixed  $n'_s$  even if  $n'_s$  is selected so that it lies within the deadband.
- (iv) To reduce the static error we can either increase in software the proportional gain  $k_p$ , or reduce the difference  $(n_s - n'_s)$ . In fact,  $(n_s - n'_s)$  can be made equal to zero so that through use of low values of

$k_p$  improved dynamic characteristics (i.e. better damping) can be achieved while maintaining good accuracy when positioning.

The load will be moved if the DAC command lies outside the deadband of the system i.e. if  $d < n_1$  or  $d > n_2$ . This fact and the foregoing analysis suggests that static position errors can be overcome by choosing a null value  $n'$  (Figure 5.4) which is equal to either  $n_1$  or  $n_2$  (i.e. one of the two boundary values), choice between  $n_1$  and  $n_2$  being made with reference to the sign of the position error as illustrated by Table 5.1. When using a null value selection scheme in this way, equation (5.5) will assume the form

$$d = \begin{cases} n_2 - d' & \text{when } e < 0 \\ n_1 - d' & \text{when } e > 0 \end{cases} \quad (5.12)$$

When the load moves within the deadband region the DAC command can be "switched" between  $n_1$  and  $n_2$  thereby keeping the static position error small by introducing a form of "dither" to reduce the system hysteresis. Our studies have shown that for pneumatic drives, very encouraging results can be obtained by carefully controlling null conditions in this manner.

It should be emphasised that the idea of using equation (5.12) is to define the assigned system null value,  $n'_s$ , out of the deadzone of the actual system null value  $n_s$  in order to keep enabling the load to be moved. In fact, the actual value assigned to  $n_1$  or  $n_2$  can be smaller or larger than the real boundary null value (see equations (5.13) and

(5.14)), as long as such an "excess",  $(n_1 - n'_1)$  or  $(n'_2 - n_2)$  will not cause instability and/or limit cycling [109].

$$d = \begin{cases} n'_2 - d' & \text{when } e < 0 \\ n'_1 - d' & \text{when } e > 0 \end{cases} \quad (5.13)$$

$$\begin{aligned} \text{where } n'_2 &> n_2 \\ n'_1 &< n_1 \end{aligned} \quad (5.14)$$

However, the difference  $n_1 - n'_1$  and  $n'_2 - n_2$  will not be quantified in this study as it is not easy to derive the real boundary null values  $n_1$  and  $n_2$ : When using low-cost pneumatic mechanisms, the repeatability of the system behaviour is poor; the results obtained will not be reliable. Thus, the use of  $n_1$  or  $n'_1$  and  $n_2$  or  $n'_2$  will not be distinguished. Further study of this problem will not be considered herein.

#### 5.4. Algorithm for Automatically Selecting Null Conditions

To implement such an approach we must obtain values for  $n_1$  and  $n_2$ . With reference to equation (5.6),  $n'_s$  can be initialised with a rough guess and the drive system moved to a commanded set point position (thus saving the tuning work in selecting the correct system null value during initialisation). As long as the initial guess for  $n'_s$  does not cause the drive to reach an end-stop, a stable position will be reached: having reached this stable position a null value within the deadband will be obtained and it is possible to calculate one of the boundary null values using the relationships described in equation (5.12) and Table 5.1.

Appendix IV documents the algorithm flowcharts of the system null condition automating procedures.

This illustrates the philosophy behind automating choice of null, but a practical implementation can be more complex than this. Difficulties lie in process identification when obtaining boundary null values for the system null conditions have a very complex dynamic nature (drift). By making the system over-damped, however, a simple but effective approach has been devised, which is illustrated by the flow chart in Appendix III. When using such a procedure the chosen null values iterate towards the appropriate null value ( $n_1$  or  $n_2$ ) in a monotone manner as the number of trials proceed. However, environmental perturbations may lead to false values of  $n_1$  or  $n_2$  which in turn can make the system unstable, hence a diagnostic algorithm may be necessary (can be manually manipulated). Depending on drive conditions, software algorithms can be devised to automatically optimise the choice of null. Although  $n_1$  or  $n_2$  may not be correctly obtained in a first trial, the accuracy with which  $n_1$  or  $n_2$  can be obtained will be gradually improved (as shown in Figure 5.5). Table 5.3 is the results obtained with this approach. To avoid unacceptable static positioning error an integral of error control term might be added to the applied control rules. One effective way is to constantly update the null value (integrating positioning errors as shown in Appendix IV). The correctness of  $n_1$  and  $n_2$  is affected by the "damping ratio" when the load comes to the stationary point where  $n_1$  or  $n_2$  is derived (see Table 5.2). (The degree of damping of the same point-to-point positioning is quantified in this test by the periods of positioning time required).

Table 5.1 (see Figure 5.4)

Sign of Previous Error in Position ( actual position - required position )	Chosen Value of Null for the Next Move of the Drive System
+	$n_1$
-	$n_2$

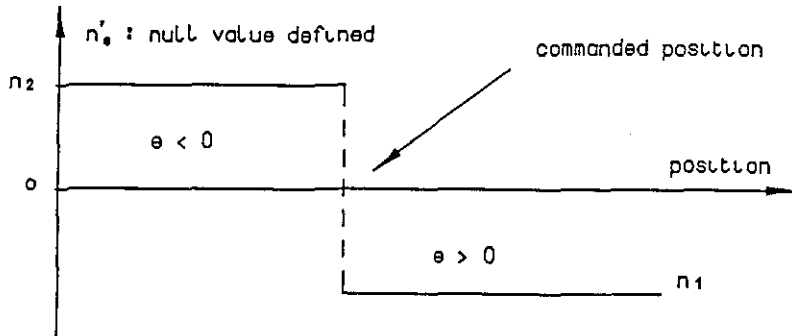


Figure 5.4 Null "switching" strategy at a single point

Improvement on Positioning Precision with a Number of Automating Trials

No. of trial	positioning error unit: [pulse]	
	stroke in	stroke out
1	+540	-556
2	-85	-752
3	-1	0
4	0	0
5	-1	0
6	-1	-1
7	0	-1
8	0	0
9	-1	-1
10	0	-1

Deadband of the drive system:

$$D_s = n_2 - n_1 = 2258 - 1825 = 433 \text{ [pulse] approximately}$$

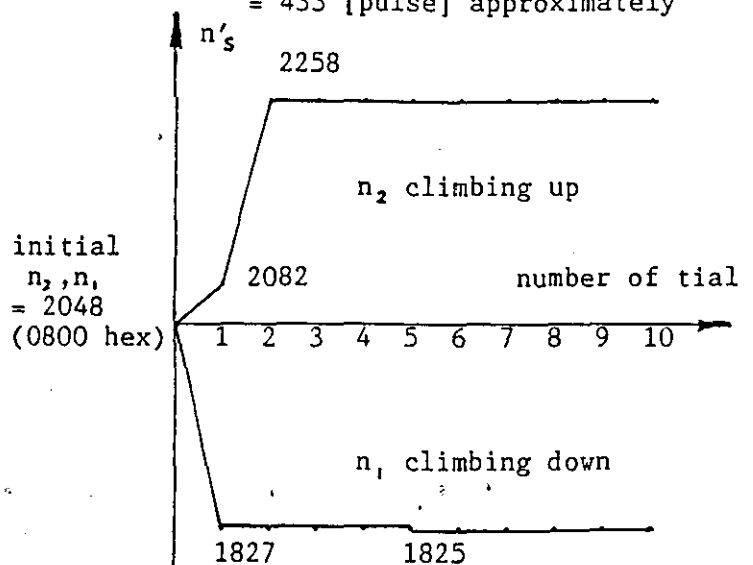


Figure 5.5 Schematic of the Result of the Null Automating Procedure

Table 5.2 Positioning accuracy versus positioning speed

Positioning Speed	Direction of Movement	Accuracy		
low	in stroke	0.037 mm	0.037 mm	0.034 mm
	out stroke	0.038 mm	0.038 mm	0.036 mm
high	in stroke	0.618 mm	0.500 mm	1.099 mm
	out stroke	1.346 mm	1.472 mm	1.515 mm
Move Magnitude		50 mm	100 mm	150 mm

Note: assuming normal distribution

Table 5.3 Positioning accuracy: comparing the use of the null automating method with that of using a fixed estimate for system null value

static phase	control law	null algorithm	direction of the movement	move size [ mm ] (step input)		
				50	100	150
excluding the integral term	proportional control (only position loop); Ts=1 [s]	fixed null 07F0 (hex.) best result	out stroke	9.793	9.445	9.443
			in stroke	10.549	11.876	10.536
		automating Gp=1/16	out stroke	0.037	0.037	0.034
			in stroke	0.038	0.038	0.036
	NOTE Ts: staying time ; Gp: software position loop gain					
including the integral term	The integral term is inserted to overcome any extra static positioning error. When applied, the best accuracy can be up to +0.025(i.e. one encoder pulse); but the prolonged positioning time, the side effect of including the integral term, can be a terrible problem, sometimes can be intolerable.					



As indicated earlier, the null values and the deadband of the system depend partly on the friction and load forces and also upon leakage conditions so that values for  $n_1$  and  $n_2$  can vary with position along the stroke of the drive. For a given sequence of movements, different values for  $n_1$  and  $n_2$  (at the various points where positioning is required) can be stored in a table format. By scheduling the boundary null values  $n_1$  and  $n_2$  for each individual point at which positioning is required an improved static performance can be obtained (see Appendix IV). This approach could be defeated if the changes of the system null conditions are so great that the validity of the boundary null values obtained in the previous cycle had been lost, which is likely to be the case due to the significant changes in friction forces when the drive system has not been used for a long time.

#### 5.5. Discussions

The features of drift and hysteresis relating to the system null conditions have been considered so that this study provides a formalised base of understanding on the null control problems of pneumatic servos, which has not, to the author's knowledge, previously been attempted. The approaches investigated in this study can be generalised and applied to any similar system which demonstrates significant drift and hysteresis.

This study shows that leakage and friction are the primary causes of drift and hysteresis for the system null condition (refer to 5.3. section A). If, however, the system exhibited no leakage then friction would not increase the hysteresis of the system. (The pressure would eventually

rise high enough to overcome friction). In real systems leakage always exists and friction is therefore an important consideration.

If the drift of system null conditions is small but the hysteresis is large, the algorithms illustrated above can be applied effectively to reduce the static positioning errors. When high positioning speed is required a "knowledge-base" (e.g. look-up table) can be built up prior to control through storing the null values for various individual positioning points. Subsequently the "knowledge-base" can be used in achieving real time control of position. Importantly, the null automating algorithm can be run on-line while the task is being carried out. When the system has been working for a reasonably long period of time, the system null conditions repeats well. (This is the case with the experimental rig constructed). When the system null conditions change dramatically, however, there is no reliable algorithm derived in this study which seems to work satisfactorily (this can be observed through deliberately introducing sharp friction variations within the stroke). An adaptive algorithm was derived but only a normal system null value can be updated to cope with the drift of system null conditions but not the two boundary null values. To avoid considerable work in this respect further studies in the area were not carried out. It seems, in such a case, that improvements in hardware (the actuator and the control valve) could be made to reduce the effect of drift: for such a situation the control approach has been shown to work very well.

## CHAPTER 6 POSITIONING TIME OPTIMISATION

### 6.1. Introduction

Experimental studies by the author have showed that when positioning a payload in point to point mode, position errors of one or two encoder pulses (0.025 or 0.050mm) can be achieved by (i) making the system overdamped and choosing a low position loop gain and (ii) through automating the system null conditions as described in Chapter 5. However, high speed positioning is required in manufacturing applications whereby the cycle time of machines can be reduced. To improve positioning speeds for pneumatic drives, Nagarajan and Weston [75] devised a front-end control scheme which operated as an outside control loop around a conventional linear real time control algorithm. This approach was motivated by the mathematical analysis by Astrom [76] involving "stable time invariant linear systems with monotone non-negative step response", and led to significant improvements in positioning time. However, tedious tuning work was required and the positioning time was not optimised.

Previous control laws applied at Loughborough University for pneumatic positioning were based on the so-called minor-loop compensation technique [71-73] with the form

$$d = k_p (x - x_d) + k_v \dot{x} + k_a \ddot{x} + n' s \quad (6.1)$$

where gain scheduling of  $k_p$ ,  $k_v$  and  $k_a$  had also been suggested to reduce the positioning time and increase the system stiffness. However, using this approach considerable tuning time was involved as a variety of control parameters need to be tuned resulting in high complexity of the set-up procedures thus introduced. As a result satisfactory performance characteristics were commonly difficult to achieve when the LUT/Martonair commercial single axis controllers were used in practical situations (this conclusion was drawn through discussion with the research assistant and students, and other users from outside the University, illustrating that usability is a very important property of any industrial drive system).

One solution to these problems is to simplify the number of control parameters tuned by the user and/or find methods of making the parameters effectively independent of each other. Thus, new control strategies were studied in an attempt to optimise the positioning time without adversely affecting the accuracy/repeatability achieved. The approach evolved divided the control path of any specific move into three control phases. Phase 1 corresponding to full opening of the servo valve, Phase 2 to full reversal of the servo valve and Phase 3 to the use of easily-tuned control algorithms (involving the previously devised auto null approach).

## 6.2. Control Scheme

For point-to-point positioning, the shortest possible positioning time would occur if maximum acceleration of the payload is achieved from its rest position followed by a maximum deceleration of the payload so that

it is stopped at the desired point. However, necessary and sufficient conditions for the payload being positioned without oscillation require that the acceleration and other higher order terms are also zero. For this reason, the use of a proportional control algorithm is more appropriate as the payload approaches the desired position. In motion control systems whether they are pneumatically, hydraulically or electrically driven, high values of acceleration will correspond to a large command signal, this necessitating a large offset of the spool of the valve or a high value of current in the case of an electric motor.

It has been shown in the previous discussions that good accuracy can be achieved when positioning by adopting "up-to-date" null values. A high position-loop gain can further improve the accuracy, increase the stiffness of the drive and increase the rise time. However, such an approach is likely to introduce instability, especially for pneumatic drives involving a series of different stroke lengths and movements. Thus the following control strategy was formulated which includes the use of three dynamic phases and two specially designed control algorithms when steady state conditions are approached.

#### 6.2.1. Dynamic Phases

The following describes the dynamic phases adopted after considerable investigatory experimentation:

- (i) phase one: full opening of the valve to reduce to a minimum the delay time at the commencement of motion and to accelerate the payload as quickly as possible.
- (ii) phase two: full reversal of the valve to reduce overshoot and to

decelerate the payload as quickly as possible.

(iii) phase three: the use of control laws previously employed in Loughborough research: typically minor-loop compensation and gain scheduling or only position loop control with gain scheduling.

The three dynamic phases adopted are shown schematically in Figure 6.1a.

In the experimental procedure reported here only a single performance index is optimised, i.e. that of minimising positioning time, but a constraint imposed was that there should be no overshooting of the final position (i.e. critical or close to critical damping). To achieve optimisation the following parameters values were modified as a result of applying a learning procedure:

- (1) the position at which switching between phase 1 and phase 2 occurs  $\chi_r$ , and
- (2) the position at which switching between phase 2 and phase 3 occurs  $\chi_n$ .

The mathematical relationship between the normal control region ( $L_n$ ) and the front-end control region ( $L_f$ ) (refer to Figure 6.1a) is illustrated by Figure 6.1b, where  $|\chi_d - \chi_s|$  specifies the complete control path which is then divided into two parts, i.e.  $L_n$  and  $L_f$ . Experimental work has shown that it is more appropriate to determine the extent of the normal control region manually with a few trials but to select  $\chi_r$  in the front-end control region through use of a self-learning loop. There is a limit on the minimum value of the normal control region ( $L_n$ ). If  $L_n$  is too small, it is not possible to select a value for  $\chi_r$  which can make the

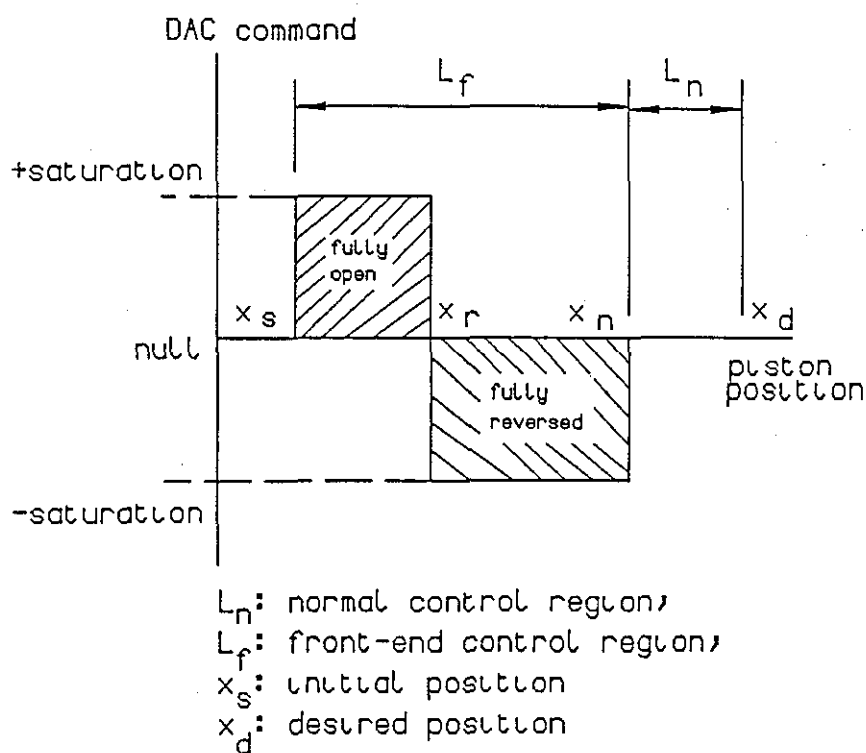


Figure 6.1a Illustrating 3-phase control scheme

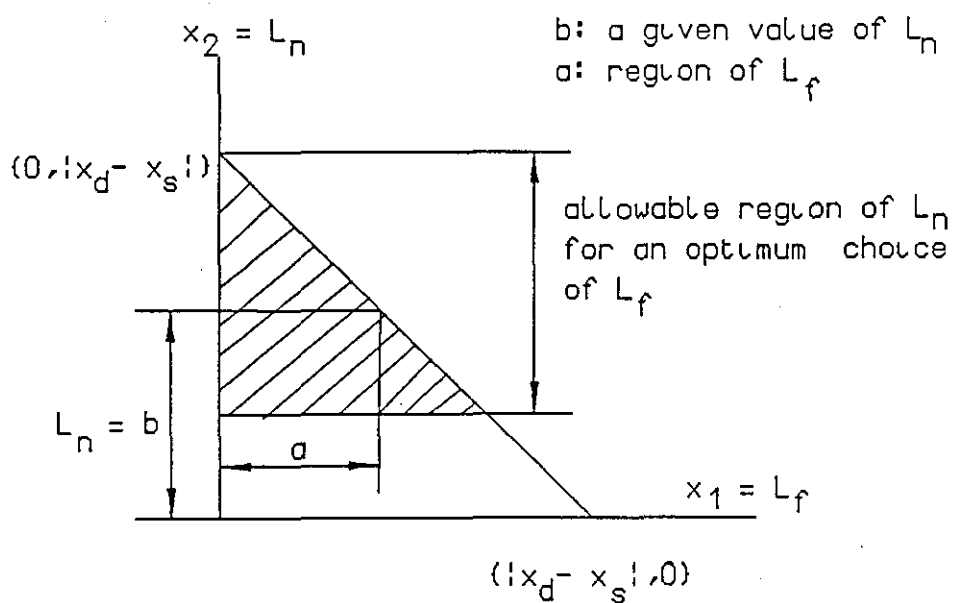


Figure 6.1b Relationship between the normal control region and the front-end control region

system overdamped or critically damped, i.e. either overshoot and/or reverse movement will occur.

#### 6.2.2. To Improve the Steady-state Performance

(i) When the drive approaches the required steady state position the null-updating algorithm (as described in Chapter 5) is applied to reduce the static position error.

(ii) When the drive is in the neighbourhood of the final position, a higher proportional gain is used to increase the drive stiffness.

The control and learning strategies are illustrated by Figure 6.2.

#### 6.3. Learning Procedures Applied

To optimise positioning time there are various possible approaches which could be used to optimise the choice of  $\chi_r$  and  $\chi_n$ . Appropriate learning methodologies [110] can be classified into two groups:

(1) build well-established mathematical models which present the knowledge of the controlled plant. However, it is extremely difficult to do this for pneumatic drives due to the highly non-linear nature of the drive and the fact that this model can change as a function of time (i.e. drift, lubrication changes, etc.). It is possible, however, to use an approximate model which is simple enough to be implemented without imposing a significant processing load on the microprocessor.



(2) Trial and error method. With an initial guess, the system can assume a self-learning mode and by adopting certain learning rules (or learning algorithms) a level of optimisation can be achieved. There are two concepts worthy of note in this respect:

(i) initial guess (at which the search action is started) which is made according to previous experience and/or some qualified knowledge. If an optimal initial guess is made, the system would have been well modelled. A bad guess can mislead the "learning march".

(ii) learning step, which is the search step being taken in the learning procedure. It can be constant or a function of some process parameters. If the learning step is too big, the optimum point may be missed or misled, while if it is too small a long learning time will result. A constant learning step is simple and easy to implement but to speed up learning without losing quality it is better to include functionality which regulates the learning step especially when the neighbourhood of the optimum location is approached.

For pneumatic drives if the valve is fully open for too long overshoot may occur while if the valve is reversed for too long reverse movement of the payload can occur. Thus over-opening or over-reversal of the control valve would lengthen the positioning time, and could be worse than that with conventional control approaches. Thus a reasonable period of "buffer control" (where a conventional control algorithm is used near the

desired position) is necessary to achieve final positioning without overshooting of the final position. This "buffer control" can be a proportional control approach as mentioned before, or may be proportional control plus minor-loop compensation (Moore [31]). Having included a phase 3 conventional control algorithm, if overshoot still occurs this can be overcome by either compressing the full opening control phase or extending the period for which phase 3 control is used. If there is any reverse movement during positioning this can be overcome by either extending phase 1 control or compressing phase 2 control.

In this study a learning (search) procedure is carried out by using both a constant learning step and successive approximation search methods as shown in Figure 6.3. There can be various approaches in initialising the learning procedure (manually or automatically): initial values for the two "switching" points  $X_r$  and  $X_n$  can be chosen as being some function of the payload position. Detailed software documentation is provided in Appendix V. Table 6.1 compares the positioning time obtained when using only proportional control for complete moves with that of using the three control phases and subsequent optimisation.

#### 6.4. Discussion

Theoretically, this manual tuning procedure can also be implemented by including software-based learning to achieve self-tuning procedures. Some of the control parameters are time dependent while the others are not. Correspondingly, learning can be classified into two groups: (1) dynamic learning (adaptive) and (2) static learning (time independent).

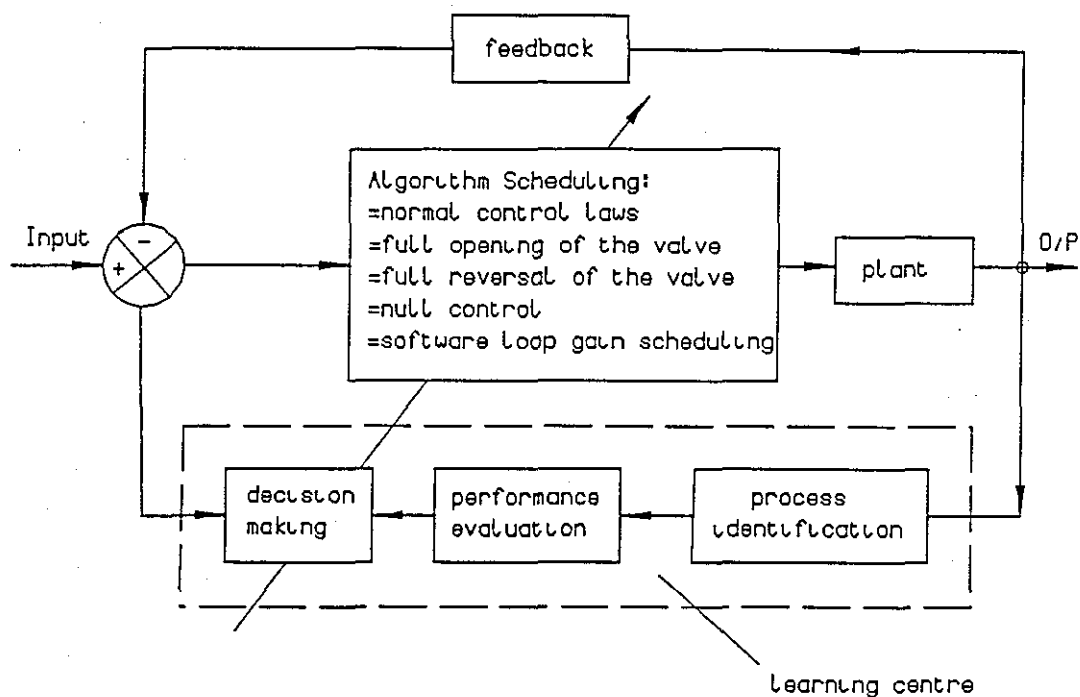


Figure 6.2 Learning scheme

O: reverse movement/increase full opening period  
 R: overshoot/increase full reverse period

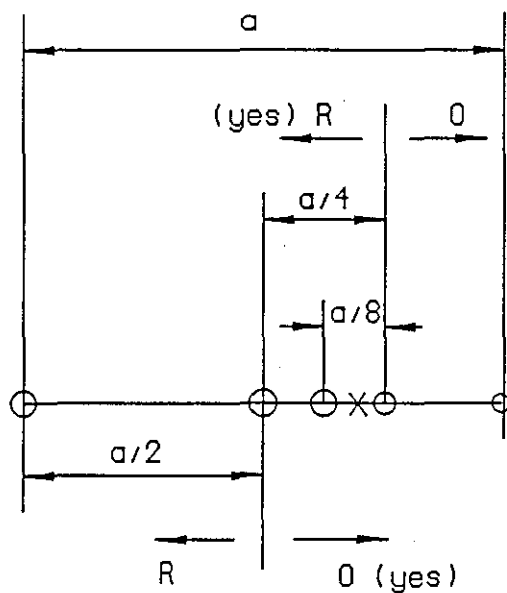


Figure 6.3 Successive approximation selection of the learning scheme

Table 6.1 Improved positioning time [ms]  
through using a learning method

direction of movement	control strategy	move size (step input) [mm]		
		50	100	150
in-stroke movement	learning	626	824	815
	original RTC	1457	1490	1720
out-stroke movement	learning	892	985	1169
	original RTC	1363	1917	1925

Static learning can be done manually or automatically but dynamic learning may require automation where man's slow or incorrect response to parameter changes may occur. In certain instances parameters may not be tunable.

The sequence followed when tuning the control parameters can fundamentally affect the quality of optimised results. Basic rules which should be followed when parameter tuning are:

- (a) tune the parameters which are most independent of others and leave those most dependent for subsequent tuning stages, then
- (b) tune the parameters which have the greatest effect on the performance index.

For the 3-phase control scheme, the control parameters in phase 3 are tuned first, the switching point ( $\alpha_r$ ) in the front-end control region is then optimised. The initially set-up values in phase 3 may be modified again as a fine tuning exercise. The advantages of the 3-phase control are:

- (i) the number of control parameters to be tuned are small (2 switching points of the 3 control phases, and 2 or 3 parameters in phase 3 to be tuned).
- (ii) the effect of control parameters is explicit so that a right decision can be made in adjusting the parameter value.

As parameter tuning and robustness of a control system is not only a problem for positioning but a universal one, the author would like to leave any further discussion on this topic to the following chapters.

## CHAPTER 7 VELOCITY CONTROL OF PNEUMATIC CYLINDERS

### 7.1. Introduction

The application areas for pneumatic servo-drives will be limited if some measure of velocity control cannot be achieved despite the fact that successful point-to-point high precision positioning is possible [29,38]. The use of pneumatic servo drives will be significantly increased if even very limited velocity control is incorporated in the controller [27]. So far, there is very little knowledge in the public domain available in this field. This chapter is a continuation of Chapter 4 - the modelling of the velocity behaviour of pneumatic drives - and is presented to extend knowledge in the field.

As for hydraulic drives, meter-in and meter-out velocity control of pneumatic cylinders has traditionally been used in industry [59]. The approaches have shown that it is feasible to achieve a certain level of velocity control. Instead of a typical meter-in or meter-out pneumatic circuit, the flow characteristics of a single 5-port proportional valve, when controlling a pneumatic cylinder, can be regarded as restricted inlet and outlet circuits as Figure 7.1a. The analytical and experimental results obtained in Chapter 4 have shown that it is not possible to achieve a constant terminal velocity in response to a step displacement of the valve spool from its null position but that it is possible to obtain a mean steady-state velocity with limited (but acceptable) variations (see Figure 7.1b). This observation was utilised in constructing control methods for achieving velocity control of

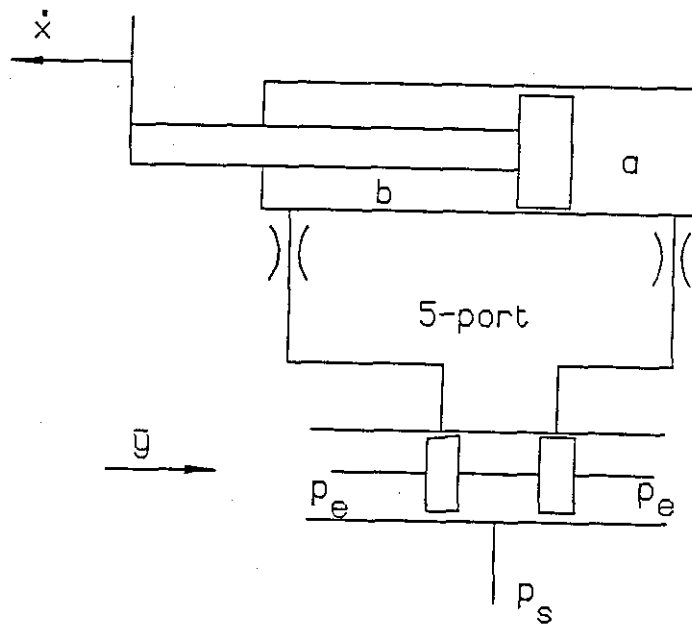


Figure 7.1a 5-port valve control: restricted in-let and out-let with equal size

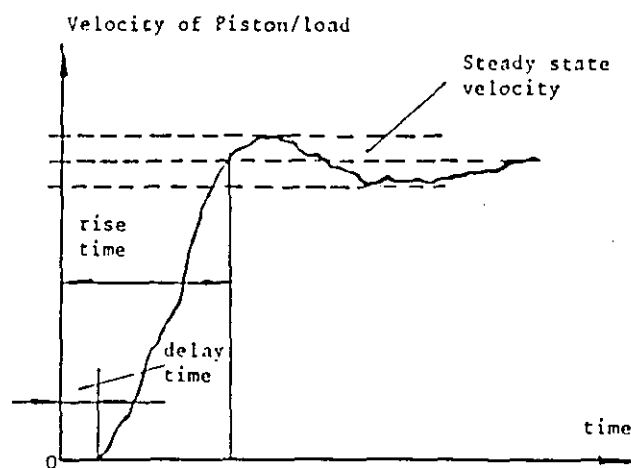


Figure 7.1b Transient response to a step change in DAC command

pneumatic drives. It is important to stress at this point that the use of digital (microprocessor-based) control offers significant flexibility (when compared with the use of analogue controls) and opportunities to implement complex control strategies.

## 7.2. Possible Valve Arrangement and Step Tests Using a 5-port valve

As suggested in Chapter 4, two alternative control valve arrangements can be used which involve either (i) the use of two 3-port pneumatic servo valves to independently manipulate the mass flowrates through the entry and exit ports of the actuator or (ii) using one 5-port valve. Only a mean terminal speed can be achieved through using a 5-port valve but such an arrangement allows reduction in the cost of hardware. However, if improved speed control is required it is necessary to use two 3-port valves and independently control supply and exhaust flowrates: where it is especially helpful to utilise meter out/in velocity control.

The approach of using two 3-port valves has not been the centre of detailed study in this work where the aim has been to investigate the performance achievable from low-cost pneumatic servos. However, the use of two 3-port proportional/servo valves could be investigated in following research studies. The tests conducted in this study were mainly based on the use of one single stage 5-port pneumatic proportional valve in order to minimise the cost of the control system with software being evolved to overcome many of the difficulties introduced through such a valve choice.



Figure 7.2a shows the ideal characteristics of a 5-port proportional valve. A series of experimental tests were conducted to determine the relationship between the DAC command (to the valve) and the maximum velocity of the piston/load (see Figure 7.2b from which a deadzone can be observed). It should be emphasised that maximum velocity was used instead of mean terminal velocity in this study. Such a choice offers two advantages: (i) it is easy and reliable to derive the maximum velocity in software through experimentation; (ii) the maximum velocity (which is normally the first "overshooting" velocity) contains the information about the rise time of the transient process (which is related to the degree of damping of the system). Actually, a "steady" terminal velocity may not be achieved or "observed" within the stroke but a "maximum" velocity can always be obtained. It is true that the information provided by the "maximum" velocity may not be sufficient to give a "solid" relationship between a step displacement of the 5-port valve and the piston velocity (the maximum velocity can vary when the piston is initiated from different positions within the stroke with the same valve opening and does not always increase as the valve opening is increased even from the same initial position as was illustrated in Chapter 4), but it does provide a "practical" compromise between the "overshooting" velocity and the mean terminal velocity. As it is not easy to find a reliable way to determine the linearity of the spool displacement of the valve versus the DAC output to the valve, the mathematical formula derived in Chapter 4 is not utilised in the control algorithm to be introduced.

### 7.3. The Control Laws Applied

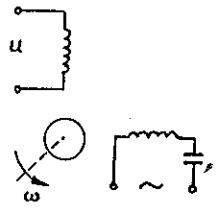
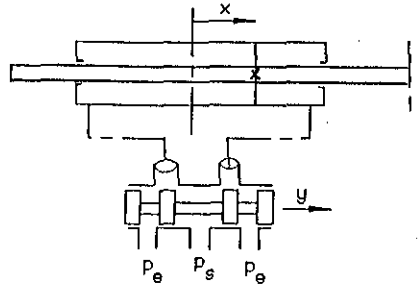
As discussed in Chapter 4, it is extremely difficult or even impossible to obtain generally applicable non-linear differential equations governing the motion of a pneumatic servo. This fact makes it difficult to define suitable and robust control strategies by using a theoretical approach which dictates the use of a largely experimentally-based approach as previously stated. In embarking on an experimental approach it is sensible however to consider parallels in the control of hydraulic and electric actuators so that the control laws for the control of pneumatic actuators might embody similar principles where and when they might be appropriate.

Table 7.1a compares behavioural characteristics of electric, pneumatic and hydraulic drives whereas Table 7.1b gives the expressions of transfer function of these drives with specified assumptions involved when being linearised (each selected drive element is only an example of its drive type).

Table 7.1a. Comparison of behavioural characteristics between electric,  
pneumatic and hydraulic drives

drive type	null/condition	steady state condition for a step command	assumptions
electric	idle if power switched off	constant velocity	no friction
pneumatic	oscillation if actuator ports closed	mean velocity with variation	no leakage and friction
hydraulic	still if actuator ports closed	constant velocity	no leakage and friction incompressible fluid

Table 7.1b Comparison of transfer function between electric, pneumatic and hydraulic drives

drive type	transfer function	assumption	schematic
electric servo motor	$\frac{W(s)}{U(s)} = \frac{K_m}{s(T_m s + 1)}$ <p>[Hu:106] where,  U - input voltage  W - motor angle  <math>K_m</math> - transmission coefficient  <math>T_m</math> - time constant</p>	no friction	
hydraulic servo motor	$\frac{X(s)}{Y(s)} = \frac{K_q/A}{s(\frac{MK_1}{A^2}s+1)}$ <p>[Hu:106] where  A - piston area  X - piston displacement  Y - valve displacement  <math>K_q</math> - valve flow gain  <math>K_1</math> - flow/pressure coefficient  M - load mass</p>	no friction no leakage incompressible symmetry	
pneumatic servo mechanism	$\frac{X(s)}{Y(s)} = \frac{K_f}{s(s^2 + 2\zeta_m \omega_m s + \omega_m^2)}$ <p>[Burrows:25] where  <math>K_f</math> - forward path gain  <math>\zeta_m</math> - damping ratio  <math>\omega_m</math> - natural frequency  (detailed information in the reference)</p>	no friction no leakage small perturbation around mid-stroke	<p>ideal valve characteristics symmetry</p>

Present-day manipulators (e.g. robots and other manufacturing machines) which commonly utilise electric or hydraulic drives are usually controlled with very simple control laws which are error driven using a control equation (7.1) [67,106].

$$\text{error command: } d = k_p e + k_v \dot{e} + k_a \ddot{e} + k_i \int e dt \quad (7.1)$$

where  $k_p$ ,  $k_v$ ,  $k_a$  are the position loop, velocity loop and acceleration loop gains respectively and  $k_i$  is the integral term gain. For electric drives, satisfactory positioning and contouring can be achieved by applying equation (7.1) in some form to follow velocity profiles of the type shown in Figure 7.3a [67,107]. However, in attempting to achieve contouring with pneumatic drives such an approach is not at all appropriate unless high quality pneumatic elements are used (engineered specially to reduce non-linearities) and more complex valving arrangements are devised which would defeat the LUT objective of devising low-cost servos which can have widespread industrial application. By referring to Table 7.1a we see that the pneumatic actuator will tend to oscillate when a single 5-port valve is nulled so that the use of a controlling equation such as equation (7.1) would fail in following a velocity profile. However, rather than use a constant null value the DAC command can be made equal to that value established through open-loop testing i.e. a value which corresponds to the mean velocity defined by the velocity profile being followed (refer to Figures 7.2 and 7.4). (Note: the mean terminal velocity is in reality replaced by the maximum velocity in the control algorithm to be developed). (The reason for doing this has been previously mentioned). Thus when contouring with

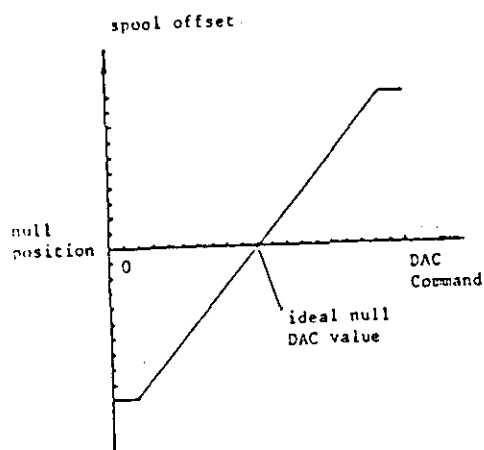


Figure 7.2a Idealised spool displacement versus DAC command

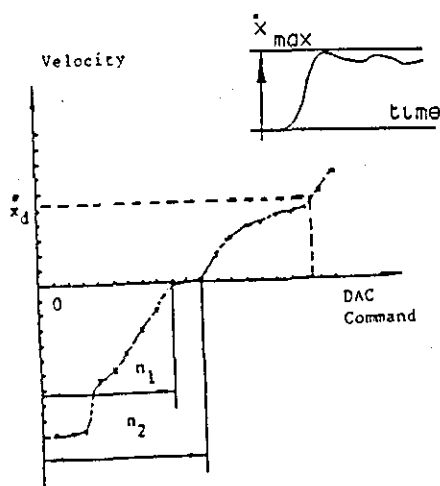


Figure 7.2b Maximum velocity versus step DAC command change

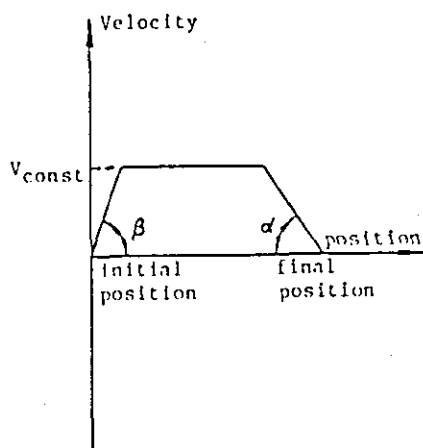


Figure 7.3a Desired velocity profile

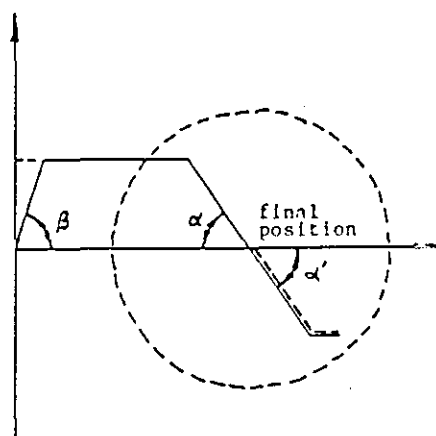


Figure 7.3b Extended reference velocity profile

pneumatic drives a modified version of equation (7.1) has been used as described by equation (7.2).

$$\text{error command: } d = k_p e + k_v \dot{e} + k_a \ddot{e} + C \quad (7.2)$$

where  $C$  is obtained from a knowledge base (e.g. Figure 7.2) and can be referred to as a Null Compensation Value or Function. Such an approach is critical in accomplishing contouring when utilising pneumatic drives.

Choice of  $k_p$ ,  $k_v$ ,  $k_a$  will not only have significant effect on the dynamic response and the stability of the system but will also affect the static performance characteristics (positioning accuracy) and time lags when following velocity inputs. Here, decision-making will be required to determine the weights associated with  $k_p$ ,  $k_v$  and  $k_a$ . A technique of gain scheduling has been incorporated here. Figure 7.5 shows the overall control strategy evolved. Detailed information about control equation (7.2) will be presented in the next section.

#### 7.4. The Control Algorithm (See Appendix VI for details)

There are various methods by which a "knowledge-base" for the Null Compensation Value  $C$  can be established. One practical approach is to build up an experimentally-based look-up table through initialisation and learning procedures involving a limited number of step tests. This approach has proved to be reliable. Either the maximum or mean terminal velocity can be evaluated by processing the terminal velocity data. The desired velocity can be related to the velocity values in the look-up

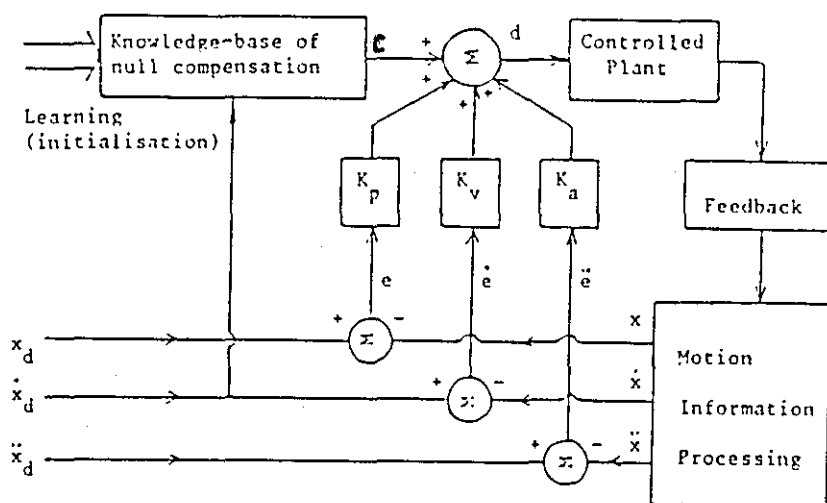


Figure 7.4 "Null" compensation control strategy

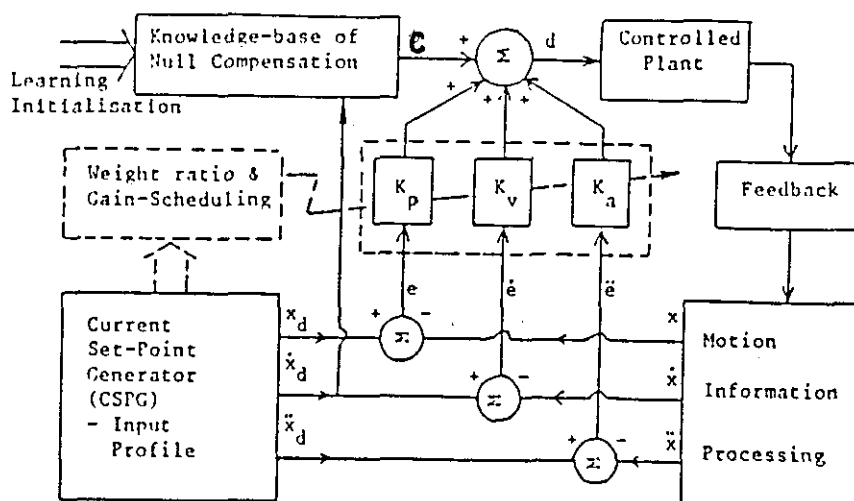


Figure 7.5 Velocity control scheme



table being derived according to its nearest data. (Alternatively the desired data can be derived between two nearest data e.g. by a proportional relationship). Actually, the terminal velocity can be calculated somehow according to the formula derived in Chapter 4 for the velocity response under a step displacement of the valve spool within a limited number of tests to reduce the number of initialisation tests.

To facilitate position and velocity control, the velocity profile (as shown in Figure 7.3a) is extended into the opposite direction around the final position (as shown in Figure 7.3b) so that the null compensation value is always used when in the neighbourhood of any final position irrespective of the system damping. Thus, if overshooting of the final position occurs, C adds an extra "returning force" acting like a "spring" returning the piston/load to its commanded position. In this way, the stability and robustness of the control algorithm increases leading to increased drive stiffness.

System hysteresis and drift, resulting from variations in friction and leakage can be significant. To reduce static positioning errors, one of two boundary values  $n_1$  and  $n_2$  (see Figure 7.2) can be chosen with reference to the sign of the position error rather than through using a single null value as detailed in Chapter 5. However, the success of this approach can be reduced by the occurrence of non-zero velocity and acceleration terms: see equation (7.2). To gain a clear understanding of the effect of  $k_v$  and  $k_a$  on the static position errors, the following analysis outlines the effect on the system null value:

At any static position, the actual velocity and acceleration of the piston/load are zero. Equation (7.2) becomes (referring the notation in Chapter 5 on automating the system null value)

$$n_s = k_s e + k_v (\dot{x}_d - 0) + k_a (\ddot{x}_d - 0) + C \quad (7.3a)$$

(note: the input velocity profile is derived according to the input position profile in this study).

By rearranging equation (7.3a), we have

$$e = \frac{n_s - [k_v \dot{x}_d + k_a \ddot{x}_d + C]}{k_p} \quad (7.3)$$

When  $k_v \dot{x}_d + k_a \ddot{x}_d = 0$ , equation (7.3) becomes

$$e = \frac{n_s - C}{k_p} \quad (7.4)$$

Equations (7.3) and (7.4) are a more generalised form of the static position error equation which was derived in Chapter 5 (equation 5.12) for a proportional error control law. According to equation (7.3), we know that not only  $k_p$  and  $C$  but also that the definition of  $k_v$ ,  $k_a$ ,  $\dot{x}_d$  and  $\ddot{x}_d$  will affect the static position errors. However, in this study the main effort is concentrated on the ability of pneumatic drives to follow a velocity profile.

### 7.5. Tests Results and Discussion

A series of experiments were conducted involving the tuning of parameters such as  $k_p$ ,  $k_v$ ,  $k_a$  and  $\alpha$  to evaluate the control scheme implemented according to equation (7.2). A software algorithm was also used to generate the position and velocity profiles on a sampled-data basis which were stored for analysis purposes. To compensate for the asymmetry of the actuator, here two values of  $k_p$ , correspond to instroke and outstroke motions respectively, were used (see Figures 7.6, 7.7 and 7.8).

Having introduced a null compensation value  $C$ ,  $k_v$  can be appropriately chosen to compensate for velocity errors albeit with an inherent time lag (see Figure 7.6).

The introduction of an acceleration-loop through appropriate choice of  $k_a$ , achieves smoothing of the achieved velocity profile (see Figure 7.7). The magnitude of both  $k_v$  and  $k_a$  affect the system stability and the accuracy with which final position is achieved. The "uprising" in the velocity output at the "corner point" is due to the sudden change in the input acceleration defined which causes a sudden change in the DAC command (referring to curve 1 in Figure 7.7).

Here responses for three desired terminal velocities are illustrated in Figure 7.8. It is interesting to compare the results presented in this chapter with those by Rogers and Weston in achieving motion control of electric drives (trapezoidal velocity profile) [107]. It indicates that

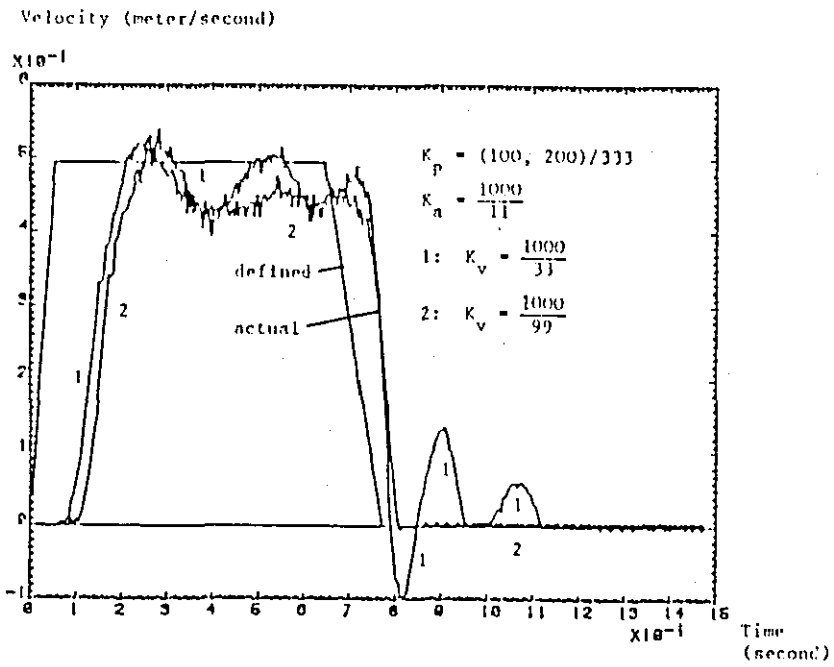


Figure 7.6 Effects of  $K_V$  on velocity profile output

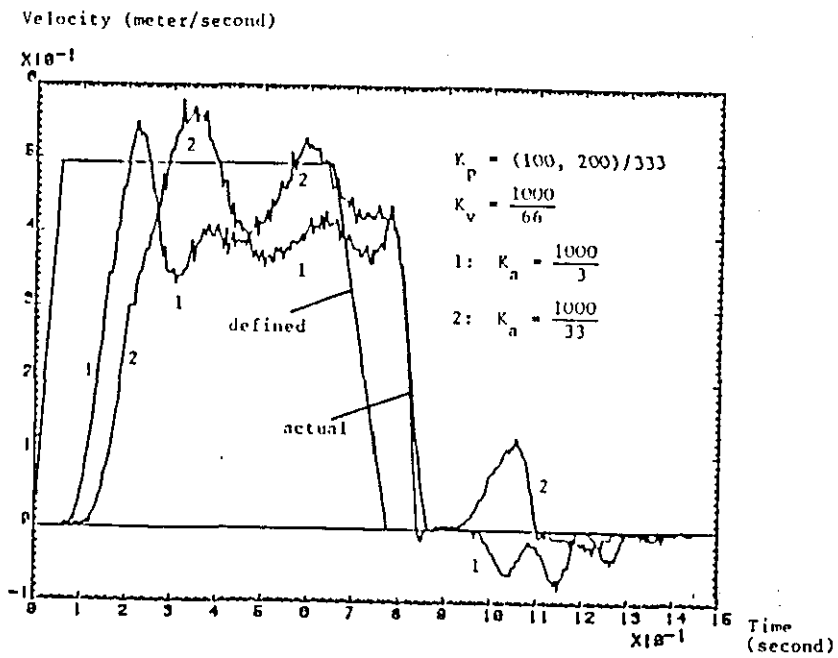


Figure 7.7 Effects of  $K_a$  on velocity profile output

it is easier to achieve low velocity control of pneumatic drives. One reason could be due to the reduction in natural damping when the desired velocity level is increased: the system will be quicker to "uprise" and follow the demanded velocity but also easier to oscillate. (The modelling and experimental studies in Chapter 4 have indicated that the natural damping of the system is not only a function of the piston/load position but also the velocity amplitude: the degree of natural damping increases as velocity amplitude increases (spool displacement of the exit port)).

It seems that low-cost pneumatic drives would be more appropriate for velocity control purposes where high precision positioning is not required (hysteresis and drift tend to cause more problems around the null position of the valve in final positioning). Compromised requirements in respect to both velocity and position control when using control equation (7.2) can be manipulated through the tuning of  $k_p$ ,  $k_v$  and  $k_a$  and the choice of parameters  $\dot{x}_d$ ,  $\ddot{x}_d$  and  $\alpha$  (this point will be continued in Chapter 8).

Because of the time constraints, no further evaluation of the control methods for the velocity control of pneumatic drives was conducted. The author realises that some other evaluation could be important, e.g. the choice of  $\alpha$  and the reference velocity profile for control purpose (see Figure 7.3b) (which is not necessarily the same as the desired velocity profile (see Figure 7.3a)). However, this study, with specific reference to the modelling study presented in Chapter 4 has demonstrated that an

acceptable level of velocity control for some industrial application areas can be achieved through the use of pneumatic drives, involving readily available low-cost components, which has not, to the author's knowledge, previously been demonstrated.

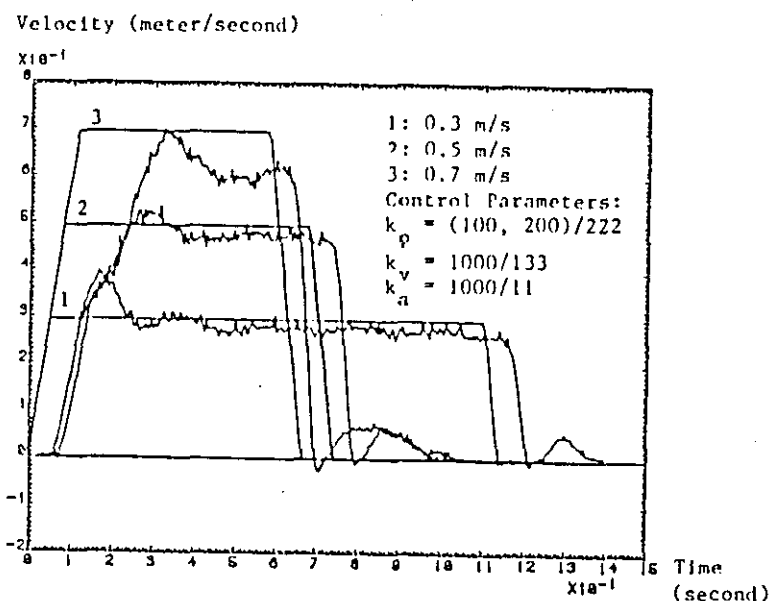


Figure 7.8 Trapezoidal profile control: velocity response

CHAPTER 8  
PARAMETER TUNING AND THE GENERAL PURPOSE MOTION CONTROL  
OF PNEUMATIC DRIVES

8.1. Introduction

In the previous chapters, modelling studies and various control methodologies have been proposed and evaluated. The information provided so far is not sufficient for evolving a "user-friendly" product. This chapter deals with the problems encountered when designing a general purpose motion controller. The aim of so doing is to set up an architectural framework which could be the basis of a commercial motion controller which would be sufficiently robust to accomplish the control of a range of drive mechanisms with various operating conditions. The availability of such a control architecture should enable future generations of pneumatic servo drives to find much wider industrial application than those serviced by first generation controllers; i.e. the Martonair single axis controllers (MES 38 ) evolved by earlier LUT research studies. Much of the discussion will make specific reference to practical knowledge gained from tuning the Martonair SACs by other researchers in the Department of Manufacturing Engineering.

"User-friendly" parameter tuning and algorithm robustness are two major properties which should be available from motion controllers which use sophisticated control strategies. Thus use-ability (i.e. the incorporating of these properties) is an important attribute which can

facilitate the control of various pneumatic drive arrangements thereby reducing the need for complex tuning procedures.

Concepts of parameter tuning will be discussed first. Then, a discussion of the factors effecting algorithm robustness will be presented with specific reference to the derivation of a general control equation which would combine the individual control schemes previously suggested. Finally, features of a general purpose pneumatic motion controller will be outlined.

## 8.2. Parameter Tuning

Parameter tuning involves the adjustment of parameters, while studying features of the actual output in response to desired input, so that satisfactory performance characteristics can be achieved (this can be done manually or automatically). Normally, the control parameters governing the operation of the control algorithms, within the controller, can in fact be assigned within a range of values which will enable the required performance characteristics to be achieved. However, the range of acceptable values for each parameter can vary considerably with properties of the various individual control systems. Moreover, it may not be easy to identify this range of values. The ease with which tuning of the control parameters can be accomplished will be referred to as the "tune-ability" of the control system. If the best tuned control system still cannot achieve the performance required, we will consider the control system to be untune-able.



The tune-ability of a control system cannot easily be quantified and compared with other alternative systems. It is not easy to derive those essential criteria to carry out such a analysis. However, a qualitative assessment of necessary criteria can be outlined and a few concepts introduced to identify the problems involved.

#### 8.2.1. User Tuning Ratio

As discussed in earlier chapters, the motion control of pneumatic drives necessitates the use of complex control algorithms for which a range of parameters should be selected (or tuned) to achieve acceptable performance criteria. Parameter tuning is not only related to the structure of the control algorithm but also to the "tuner's" experience and knowledge of the algorithm. Generally, a user will be less qualified than the designer or manufacturer. For these reasons, it is highly desirable to reduce tuning work for the user, otherwise the successful application of pneumatic motion control may not be achieved in operational circumstance where it could be.

Similarly, use-ability is desired when the manufacturer installs such products, so that installation times/costs can be reduced. In practice, some of the tuning work must be done by the user (due to the changes in operating conditions or to manufacturing processes as product variations occur) while other parameter tuning will usually be carried out by the manufacturer (in either pre-installation or installation phases of a project). It is convenient to identify some of the problems involved by introducing the concept of user tuning ratio.

User tuning ratio will be conceptually defined as

$$\text{user-tuning-ratio} = \frac{\text{user tuning}}{\text{total tuning involved}}$$

Parameter values can be stored in RAM ("soft" parameter storage) or in ROM ("hard" parameter storage). "Hard" tuning could be done at pre-installation and installation stages and "soft" tuning by the user at post-installation and installation stages.

Parameter tuning includes both manufacturer tuning and user tuning. It is important to distinguish between the user tuning and the manufacturer tuning as the first will be related to the perceived useability of the motion controller whereas the second will have cost implications: although it may not be necessary to distinguish between the two in the research and development stage where the control strategies are both evolved and tuned by the researcher(s).

#### 8.2.2. Number of Control Parameters and the Sequence in which those Parameters are Tuned

When defining a tuning procedure, it is necessary to consider:

- (a) the number of control parameters to be tuned;
- (b) the interaction of the functionality (or crosstalk) between these parameters.

For example, there will be 51 (or 240) possible tuning sequences if there are 5 independent control parameters. If these parameters are

independent (i.e. they do not functionally interact) then it does not matter how the tuning sequence is arranged, all 240 sequences can work. However, if certain parameters are not independent then the order in which those parameters are tuned will be important.

Thus, ideally the functionality of each control parameter should be clearly and independently defined so that the tuner will know which parameter to tune so that a particular change in the system behaviour can be achieved. In this respect, major problems exist with the present Martonair controller which could have severely limited their effectiveness and widespread application. These first generation controllers can be identified as inherently possessing:

- (i) too many tuning parameters (about 15), and
- (ii) an interrelated functionality between many of the control parameters and often a poor understanding of their functionality.

It is important to minimise the number of tuning parameters, and define the order in which those parameters should be tuned, to improve the useability of the control system and yet facilitate sufficient flexibility to achieve good performance for most pneumatic drive systems. Ideally, theoretical analysis is required here to anticipate changes in the behaviour of the control system in response to changes in the control parameters. However, it is extremely difficult to isolate and define the functionality of each control parameter and hence quantify the optimisation process in regard to number and choice of the control parameters to be tuned. Thus in this study a largely experimentally-

based approach has been used. However, a computer-based automation of the experimental approach, through the use of learning methods, has been devised which should facilitate an advance in the application of pneumatic motion control.

#### 8.2.3. Diagnostic tuning procedure: following error display

The tuning process, be it automatic or manual, must reference the system output and the following errors in order to decide if action in adjusting certain control parameters is required. A comprehensive display of the output and following errors should be available to the tuner:

(a) When manual tuning, the tuner can compare the actual results obtained with those desired (or possibly with previously obtained results) then accordingly attempt to improve the characteristic errors and responses. Unpredictable tuning cycle times can result if the tuner does not have sufficient information on the functionality (or effect on the characteristic errors and/or responses) of the parameters to be tuned.

(b) When automatic tuning is used, the user can monitor the correctness of the self-tuning process, e.g. in a case where the self-tuning process is misled due to unexpected disturbances.

The availability of comprehensive diagnostic displays can help somewhat in overcoming these difficulties. However, the display of following errors available with the present Martonair controllers is poor. This problem should be avoided in the next generation controller, otherwise it

will only find successful application where "smart technicians" are employed.

### 8.3. The generalised control equation and algorithm robustness

The structure of the control algorithm will ultimately determine the procedure for parameter tuning. Parameter tuning is also related to the user's familiarity with the algorithm (or the control methods). Thus, the controller designed should possess two properties to promote its usability and a successful application in industry. Firstly, the controller is tunable for a range of control mechanisms and a range of operating conditions, i.e. the control strategy should be robust. Secondly, the control methods should be fully evaluated and formalised so that the manufacturer and the user can easily understand the principles involved. Consider electric drives as an example. There are a number of factors which contribute to their successful application in industry: (i) good linearity of the governing differential equations, so that conventional linear control theory can be effectively applied, (ii) as a result of (i) simple and successfully applied control strategies can be used, involving relatively simple tuning procedures and (iii) for the foregoing reasons electric drives have found widespread use in industry so that essentially standard codes of practice have been evolved which are followed when designing and installing the drive elements. Correspondingly, the significant effects of the non-linearities, inherent in the use of pneumatic drives, arguably dictates the use of relatively unfamiliar and unproven non-linear control methods. Control methods which are common to electric and/or hydraulic drives can be applied

directly to pneumatic drives but this thesis has illustrated that this is by no means the best answer. Unfortunately, however, a disadvantage associated with the use of non-linear control methods is that the number of control parameters will be high, many of which are interrelated and their functionality not well understood (this is certainly the case with the Martonair controller). Furthermore, the control methods developed by the author (Chapters 5,6 and 7) can deal with point-to-point positioning in terms of accuracy and positioning time, and yet achieve velocity control with acceptable following errors. However, the diversity in the control methods (or concepts) involved adversely affects the usability, particularly when use of the three control methods is combined. However, it is highly desirable to integrate the control methods to establish a generic control equation for various forms of pneumatic motion control. There are two direct advantages of so doing: (a) the robustness of the controller will be significantly increased; (b) a theoretical formalisation of the control methods may be advanced.

For convenience, we will duplicate control equation (7.2) which shall incorporate the control strategies described in Chapters 5 and 6 and be rearranged to form equation (8.1).

$$\begin{aligned}
 d &= k_p e + k_v \dot{e} + k_a \ddot{e} + C \\
 &= k_p (x - x_d) + k_v (\dot{x} - \dot{x}_d) + k_a (\ddot{x} - \ddot{x}_d) + C
 \end{aligned} \tag{8.1}$$

$\dot{x}_s \quad \text{-----} \quad d_s$

where

$d$  is the actual DAC command signal to activate the control valve;

$C$  is the null value compensation term, which corresponds to a terminal velocity given by  $\dot{x}_d$ ;

$x$ ,  $\dot{x}$  and  $\ddot{x}$  are the actual position, velocity and acceleration of the piston/lead;

$x_d$  is the desired position;

$\dot{x}_d$  is the input velocity which determines the value assigned to  $C$ .

$\ddot{x}_d$  is the input acceleration;

$k_p$ ,  $k_v$ ,  $k_a$  are the position, velocity and acceleration loop gains which in the controller implementations used in this study can easily be manipulated in software.

The key part to be manipulated in equation (8.1) is the relationship between  $C$  and  $\dot{x}_d$  such that the control methods established can be integrated into one control equation. The input velocity  $\dot{x}_d$  corresponds to a steady-state velocity  $\dot{x}_s$  described in the meter-out/in and choked flow models.  $C$  will be the steady state DAC command, which corresponds to a valve opening at the open loop terminal velocity (refer to the modelling studies in Chapter 4).

The front-end control scheme for optimising the positioning time (as described in Chapter 6) can be obtained by manipulating the velocity control profile  $\dot{x}_d$  (and hence the compensation term  $C$ ) along with the software loop gains  $k_p$ ,  $k_v$  and  $k_a$ . The principle previously suggested when automating the system null conditions (see Chapter 5) can also be incorporated by manipulating  $C$ . The reader should remember that  $C$  can

have two boundary values dependent upon direction of motion. A more detailed explanation is given as follows:

#### A. Front-end control scheme for positioning time optimisation

A typical manipulation of the velocity control profile  $\dot{x}_d$ , to generate the front-end command signal (full opening or full reversal of the valve) is illustrated by Figure 8.1a. Figure 8.1b shows a simplified relationship between a terminal velocity of the piston in response to a step opening of the valve or change in the DAC command. According to Figures 8.1a and 8.1b, the compensation term  $C$  can be derived (see Figure 8.2). Thus in the front-end region,  $C$  is generated as  $C^-$  and  $C^+$ . By setting the three software gains to zero, the control equation (8.1) becomes

$$d = 0 e + 0 \dot{e} + 0 \ddot{e} + C = \begin{cases} C^+ \text{ full opening} \\ C^- \text{ full reverse} \end{cases} \quad (8.2)$$

It can be seen that equation (8.2) is the front-end control DAC output. The switching point between  $C^+$  and  $C^-$  can be selected as described in Chapter 6 (see Figure 8.1a). When approaching the normal control region, the software loop gains  $k_p$ ,  $k_v$ ,  $k_a$  can be restored.

#### B. Automating the system null condition to reduce the positioning error



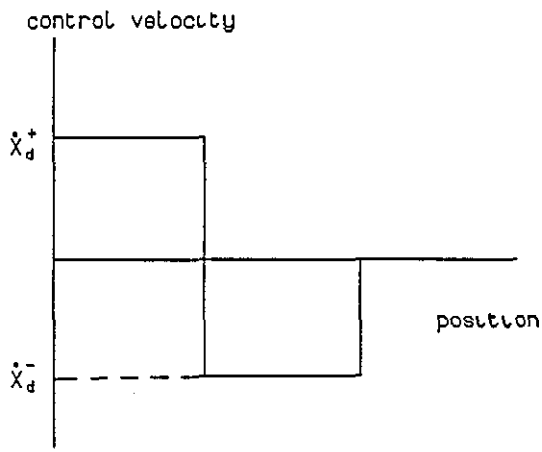


Figure 8.1a  
Command velocity  
profile for  
front-end control

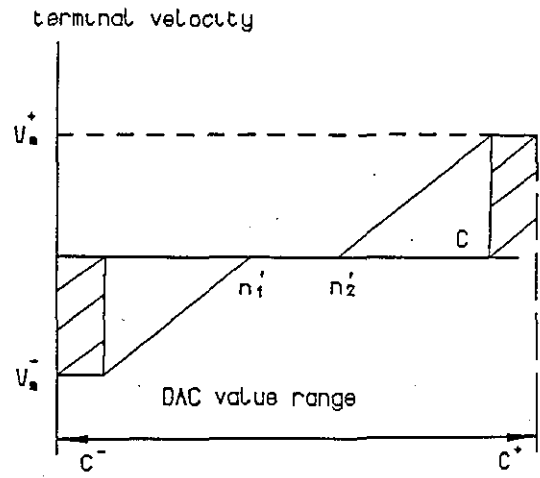


Figure 8.1b  
Terminal  
velocity  
versus step  
DAC command

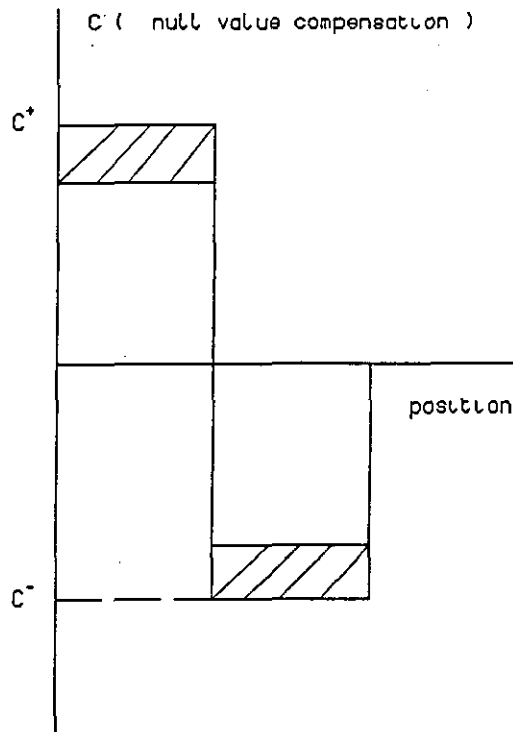


Figure 8.2 "Null" compensation profile  
for front-end control

A typical velocity control profile, after completion of the front-end control region is illustrated by Figure 8.3a. This involves selection of the two boundary values  $n'_1$  and  $n'_2$  at zero velocity dependent on the direction from which the static position has been approached. Figure 8.3b is the corresponding output of C derived according to Figures 8.3a and 8.1b. Here,  $n'_1$  and  $n'_2$  are of the same nature as the two boundary null values  $n_1$  and  $n_2$  described in Chapter 5.

#### C. Velocity control scheme

As far as the velocity control is concerned, no further discussion will follow here as the operation of the control method has been detailed in Chapter 7.

It should be noted that in the final positioning region, equation (8.1) will take the form

$$d = k_p e + k_v \dot{x} + k_a \ddot{x} + C \quad (8.3)$$

where  $e = x - \dot{x}_d$

and  $\dot{x}_d$  and  $\ddot{x}_d$  are both set to zero. This is a result of the "residual" input velocity and acceleration terms which may introduce positioning errors as explained in Chapter 7.

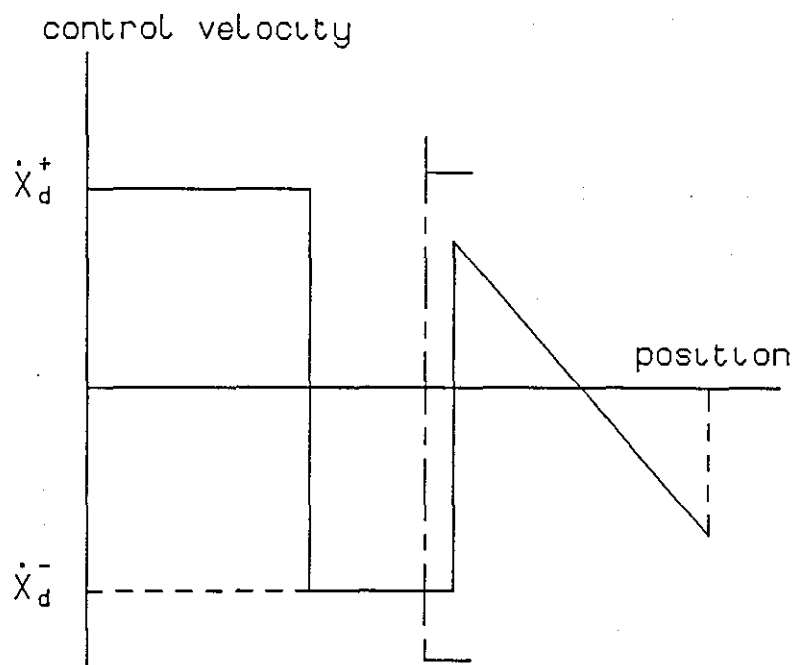


Figure 8.3a Velocity profile  
for normal control  
(system null condition)

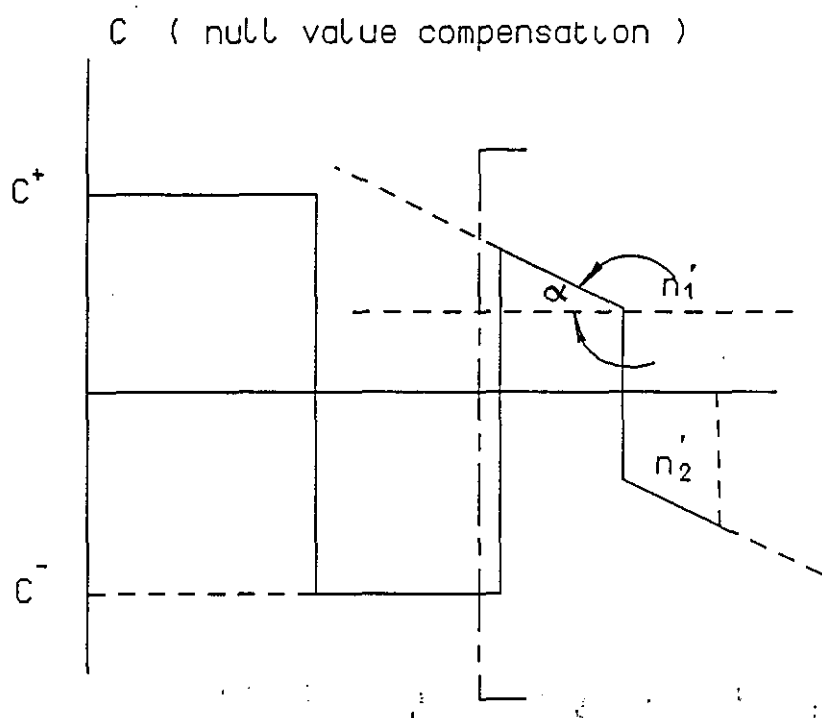


Figure 8.3b Null value compensation  
command profile

#### 8.4. Use of Modern Control Concepts

It can be seen that the successful manipulation of the general control equation will depend on a form of algorithm and gain scheduling which implies the use of a digital controller so that sufficient "intelligence", or decision making capabilities, can be included. This is also true in respect to implementing self-tuning and learning procedures. The use of modern control concepts based on the use of microprocessor technologies can thus be considered to be an integral part of the success of designing servo pneumatic drives.

The following concepts will be introduced to illustrate the use of such a methodology.

##### 8.4.1. Front-end system identification

When using compressed air as the control medium, a truly *quantitative* solution is ~~difficult~~ due to the extraordinary nonlinearities in the relevant governing differential equations. Mathematical models can be used to derive information, useful in the system identification, but the results obtained are often used only as "indicators" because of the difficulty in deriving a model which can accurately estimate the associated control parameters. To overcome such difficulties, "front-end" learning procedures can be applied before the real motion control task is executed. The knowledge obtained can be stored in computer memory (e.g. in the form of a look-up table). Such a "knowledge-base" can subsequently be used for realtime control. For example, a knowledge

of the necessary null value compensation can also be obtained by applying front-end learning to find the relation between the step DAC command (valve displacement) and the terminal velocity of the driven piston/load. This approach can also be applied to other similar system identification problems.

#### 8.4.2. Optimisation software

The modelling studies have shown that the damping and natural frequency at any operating point is dependent on the position and velocity and the load of the drive system. This implies that the optimal value of each control parameter will be different for different motion types. However, it would be tedious, time-consuming and probably impractical for the tuner to identify the optimal values for each of the control parameters for every motion type whether manual or learning tuning procedures are followed. It may also be impossible to achieve optimisation in some cases because of "drift" in the system behaviour. However, a fully optimised solution may not be required by the user. The alternative is to adopt a compromise solution: the control approach being partially optimised to provide acceptable performance for a range of operating points which correspond to a specific application. Such a semi-optimised solution can significantly reduce parameter tuning work and thus improve the usability of the control system (i.e. the number of the tuning parameters could be reduced).

However, it is probably easy to tune the algorithm at a few representative positions (or velocities). Further reduction in the complexity of the tuning process can be achieved by automating the tuning

of some parameters, with software algorithms included to accomplish this purpose. This "optimisation" software can be particularly useful to the manufacturer with pre-installation tests being carried out to find the optimised values of the control parameters and their valid regions for specific drive mechanisms. The results obtained and the relevant operating conditions could be documented (possibly automatically) and issued to the user. This information will be helpful at installation or post-installation phases.

As the design of pneumatic motion control drives is likely to be largely experimentally-based, the concept of optimisation software is considered to be vitally important in achieving advanced performance characteristics and usability of pneumatic servos in a commercially available product form.

#### 8.4.3. "Repeatability" of the system behaviour

For low-cost pneumatic control systems, the "repeatability" of the control system can be poor due to drift and hysteresis, especially where low-cost system components are used. When this happens, the results obtained through learning procedures may lose their validity and the frequency with which tuning should be carried out will be increased, having consequences with respect to product quality and equipment downtime in a manufacturing environment. Thus it is highly desirable that the system behaviour can repeat well.

The effects of drift and hysteresis in the system can be reduced through automating the system null conditions but this compensation will fail in

certain circumstances. Hardware improvements will be necessary if sophisticated performance characteristics are required. Possible improvements can be made (a) by supply pipe delays to a minimum; (b) by minimising the friction and leakage in the system and (c) by using better quality control valves which offer more acceptable valve characteristics: e.g. higher frequency response, linearity, repeatability, stability and/or robustness to flow forces. For example, when point-to-point positioning, high quality hardware arrangements will be necessary in order to achieve both high accuracy positioning with good speed control over a wide range of traverse velocities. However, such pneumatic servos will then be in direct competition with their electric and hydraulic counterparts.

In conclusion the use of software based learning methods can significantly extend the application areas of low-cost pneumatic servos but ultimately, in high performance application areas, hardware improvements must be obviated.

#### 8.5. Architectural Layout of the Controller and Conclusions

Various considerations when designing the pneumatic motion controller have been outlined in the above sections. A schematic architectural layout of a controller is shown in Figure 8.4. which summarises those observations. The concepts embodied here have been discussed above and in the body of this thesis. Industrial/Government agency funding is expected to utilise the framework devised through this thesis as a basis from which a second generation commercial product can be evolved.

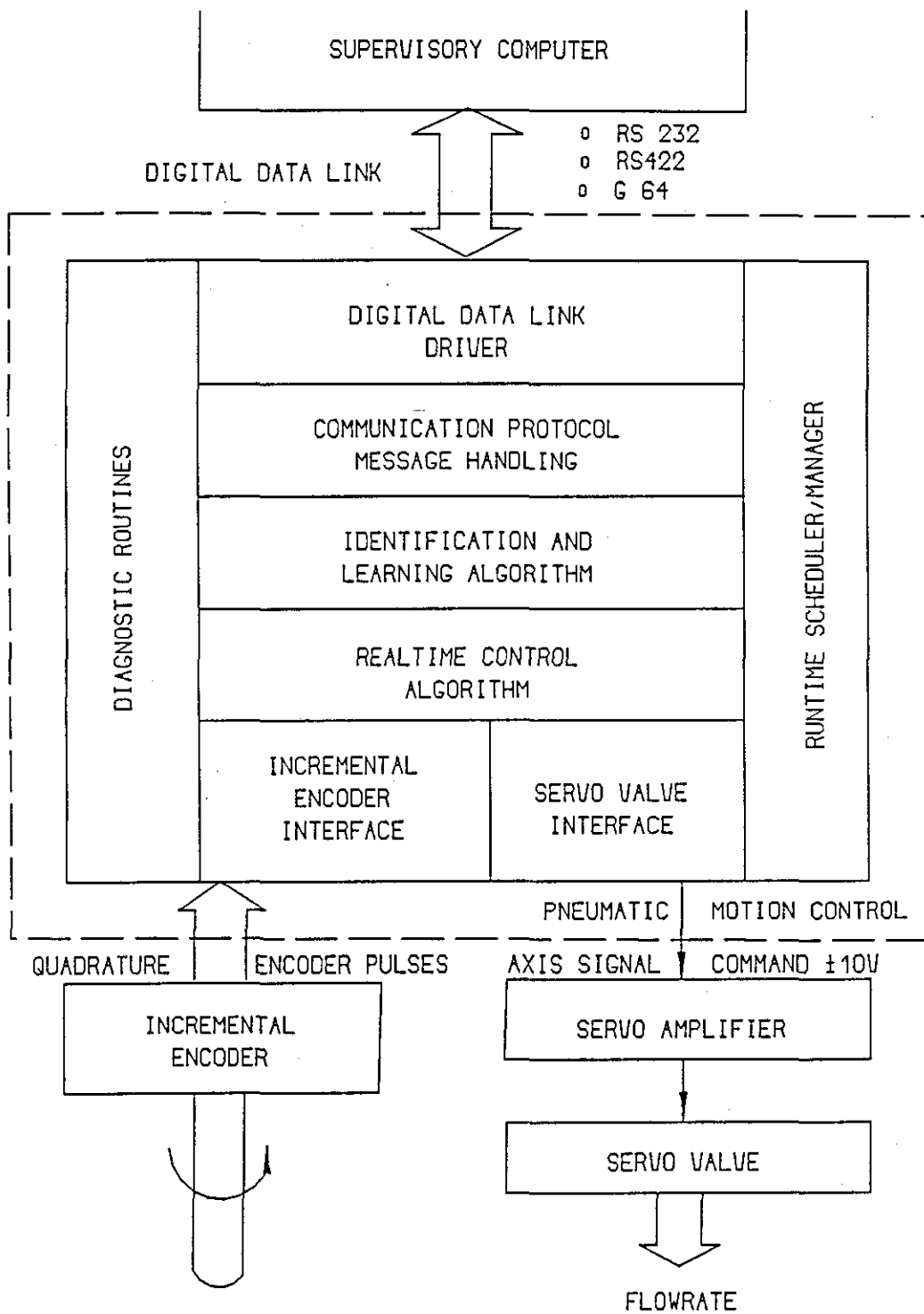


Figure 8.4 General Purpose Pneumatic Motion Controller



## CHAPTER 9 CONCLUSIONS AND RECOMMENDATIONS

### 9.1. Conclusions

(1) It is impractical, with the current level of available knowledge and tools, to derive a universal approach for the design of continuous pneumatic motion control. In order to achieve advanced performance characteristics, it is necessary to use specified hardware components (with known valve and actuator characteristics) and to have defined user requirements and operating conditions in order to limit the uncertainties and non-linearities in the control system of pneumatic drives. Generally the methods of overcoming the excessive nonlinearity of the control system is specific and requires "specific" solutions. The control methods and design strategies discussed in this thesis are universal in a sense, but the control algorithm can be optimised only when the hardware system is specified.

(2) When using compressed air as the control medium, it is difficult to derive an accurate analytical model, based on governing differential equations. It is natural to utilise "regional modelling" methods specific to "local" phenomena. To increase the accuracy of regional modelling, specific characteristics of hardware components should be known. Furthermore, the evaluation of any pneumatic drive system model should be experimentally based. This conclusion is closely related to conclusion (1).

Through a comparison between the meter-in model derived in this thesis and the meter-out model, studied with reference to previous work, the functionality of the entry port and exit ports has been precisely described and their relative importance and relationship has been evaluated. The concept of the meter-in phase and the meter-out phase has been introduced through analytical and experimental observation. Furthermore, the derivation of the meter-in model and the meter-out model has been extended, in this research study, into the so-called choked flow model which represents closely the actual system behaviour of a combination of valve and actuator. The mathematical formula concerning the steady-state terminal velocity, the natural damping and the natural frequency have been provided to support the experimental control studies. Thus, the modelling studies advanced in this thesis can be used, in a simple but effective and comprehensive manner, to analyse the dynamic behaviour of pneumatic drives and to evolve control strategies in a way which is not provided by previous modelling studies.

(3) System null conditions can significantly alter performance characteristics (e.g. the accuracy of pneumatic servo positioning). A minimisation of positioning errors can be achieved by automating the system null conditions, i.e. through reducing the effect of drift and hysteresis on the existing drive mechanisms. Very importantly, the automation of system null conditions simplifies the tuning work at initialisation stages and the algorithm derived is robust, i.e. being able to be applied to a variety of similar drive systems.

A software approach can be advantageous but ultimately has limited effectiveness so that if high performance characteristics are required, improvements in the hardware components may be necessary, e.g. reducing the friction and leakage in the valve and the actuator to limit inherent system drift and hysteresis (the use of a symmetric cylinder can reduce the leakage across the piston area for positioning when no external load force is applied as the pressure difference between the two actuator chambers will be eliminated). Important variables which may introduce inherent drift and hysteresis have been outlined in this study (see Chapters 4 and 5). Such a formalisation of the system null conditions could also provide a guideline in hardware arrangements.

(4) In order to minimise positioning time in the point-to-point mode, full displacements of the valve opening (saturation) is necessary during part of the positioning period. Such a saturation can be achieved using a conventional proportional position loop with minor-loop compensation by increasing the position loop gain and selecting appropriate levels of velocity and acceleration damping. However, such an approach can only provide a compromise solution and cannot optimise the positioning time. Thus a front-end control scheme, involving binary-switching of the valve between saturation limits can be of great benefit. The thesis results have shown how between 3 and 5 parameters can be manually selected or tuned automatically using self-learning loops. Although it is still difficult to achieve true optimisation, the positioning time can be significantly improved.

(5) Null value compensation is a vitally important concept for successfully achieving the velocity control of servo pneumatic drives.

The use of null value compensation can be considered to be similar to that of a meter-out and/or a meter-in velocity control method for pneumatic actuators (as conventionally used in industry): the function of null value compensation is equivalent to a series of valve openings which correspond to a range of steady-state velocities (which in practice, is replaced by the use of the maximum velocity or a mean terminal velocity through experimentation); thus every valve opening equates to one adjustment of the entry and exit ports of a meter-out and/or meter-in pneumatic circuit which results in the velocity required. This explains the success of the use of the null value compensation approach presented in Chapter 7, which in fact is a natural extension of the concept of conventional velocity control using pneumatic drives.

By implementing the approach in digital form, the resulting null value compensation approach can be manipulated in an advanced way so that automation of the system null conditions and front-end binary-switching of the valve can be incorporated into control equation (8.1) as discussed in Chapter 8.

(6) It is possible to achieve contouring with pneumatic actuators with moderate values of following error, at a level which would be acceptable in many manufacturing application areas. In such application areas high values of performance/cost ratio can be facilitated through using low-cost pneumatic actuators and valves. In fact, in velocity control

application areas, inherent hysteresis effects are less troublesome than when positioning.

(7) Experimentally based on-line front-end learning is potentially an extremely important concept (see Chapter 6). Such a knowledge-based approach is not only important for experimentally based system identification but also for the self-tuning of control parameters.

## 9.2. Recommendations

Microprocessor-based continuous motion control in the field of pneumatics is still new and novel in many aspects. There is a lack of public awareness in this area, industry does not yet generally perceive this to be servo-drive technology, and the formalisation of the problem lags behind that of electric and hydraulic servo-drive technologies. Despite the fact that much frontier work has been done and advancements made in this study in terms of model formulation and in evolving control strategies, a range of research and development investigation are required to enable pneumatic servos to be widely exploited. Some recommendations thus required are listed below:

(1) Two independent 3-port valves can be employed in order to reduce time delays and achieve independent manipulation of the mass flowrates in the two control chambers. Such an approach would facilitate greater flexibility when implementing the meter-out/in null-value-compensation velocity control approach.

(2) Pressure transducers can be used so that the process or system parameters associated with the models derived in Chapter 4 would be more easily identified and ultimately the model-reference control approach might be utilised to accomplish more advanced motion control tasks.

(3) Further evaluation studies, relating to the general control equation for pneumatic drives evolved during this study, could yield significant benefits. For example, manipulation of the null compensation function could be accomplished with reference to a model-based velocity control profile (which is different from the user defined velocity profile) to overcome the inherent time lags.

(4) Formalisation of the performance characteristics of the drive components and the control system parameters would result in benefits both for manufacturers and users. Promotion of public knowledge on servo-drive technology using pneumatics is highly desirable and often could lead to increased efficiency/profit levels.

## Appendix I

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## Appendix II

### THE PERVASIVENESS OF SIMILARITY AND MODULARITY

#### SUMMARY

The words "SIMILARITY" and "MODULARITY" are used by us all in a variety of contexts. Their use of these words can imply abstract but useful concepts and principles. In this appendix we shall consider the nature of the underlying concepts and provide a formalised basis of understanding which could be used to describe many phenomena. To avoid misunderstanding and confusion, the concepts of "similarity" are discussed and defined first in a simple manner ( as the reader may have a different "picture" in mind ), then an introduction to the philosophy behind these concepts is given. Finally, the key points of the methodology thus derived are presented and one example given for illustration.

#### 1. INTRODUCTION

Deductive and inductive reasonings (1) are the two common methods for thinking with two opposite processes, i.e. reasoning from the general to the individual and in the other direction from the individual to the general. However, either reasoning method inherits two attributes in common: the GENERAL conclusion and the INDIVIDUAL case holding the general. This paper is a study of these two attributes translated and focussed by the author into the concepts of SIMILARITY and MODULARITY (1).

With the diversification and growing complexity in various branches of science and technology it would be interesting and beneficial to study the similarities between these branches especially those with firmly established scientific and technical principles which have been accepted to be true or not proved to be false. Then, consider whether these principles be generalised and applied to other "similar" systems. With this motivation, the author reviewed some general laws of nature ( mainly in physics, e.g. the principles of relativity, the third law of Newton, the conservation laws of mass and energy ) and used well established scientific knowledge ( e.g. the concept of molecular formula in chemistry, transfer function in control theory, modular programming in computer science, truth table of mathematical logic, etc. ) to illustrate and formulate ideas presented under the headings: the principle of similarity; the logic of similarity equilibrium; the model of similarity field; the hierarchical modular structure and the similarity and modularity methodology. Although they are introduced through the use of logical reasoning rather through analytical proof, they provide axioms which can be developed further and applied to the natural world, science, engineering and our society.

## 2. CONCEPT OF SIMILARITY

Many phenomena, not only in the natural world but also in the social world are observed being "similar". The use of EXAMPLE is a common method of illustrating idea, formula, structure or something which is general or "similar" to the "family" in which the example belongs. Those phenomena can be seen as the EXAMPLES in a "similar

family". By understanding the meaning underlying the example the general can be learned. The following example is used to explain features of "similarity":

In chemistry, physical matter which consists of molecules is found to exist in three states: solid, liquid, and gas. For example, ice, water, and the water steam have the same molecular structure  $H_2O$ . Therefore, we say that ice, water, and the steam embody similarity in molecular structure.

From the above example, the following terms are introduced:

#  $H_2O$  will be termed the "SIMILARITY ATTRIBUTE (SA)" of ice, water and the steam.

# parameters such as temperature, pressure etc. which determine the state of the similarity attribute(  $H_2O$  ) are termed "STATE VARIABLES".

# "real bodies" will be referred to as "ENTITIES" and similar entities( such as ice, water and steam ) will possess one or more SA.

"SIMILARITY" can be simply defined as the property of having the same inner structure ( like molecular formula ) in various entities. Depending on its state variables, the state of SA can be varied as the molecular formula  $H_2O$  can exist in solid, liquid or gaseous states.

### 3. THE PRINCIPLE OF SIMILARITY

Entities possessing the same SA can be interpreted having an equivalent discription of the SA. For example, ice, water, and steam can be said to hold an equivalent discription of their molecular formula  $H_2O$  ( SA ). This interpretation will be related to the principle of relativity.

In physics, the principle of general relativity in Einstein's relativity theory (2) is stated as " All bodies of reference K, K', etc., are equivalent for the description of natural phenomena ( formulation of the general law of nature), whatever may be their state of motion.". Consider "the general law of nature" as a SA and the "reference bodies" as entities. It could be conceived that the concept of similarity is embodied in the principle of general relativity. The physical laws characterising light propagation, dictate that the velocity of light does not depend on the motion of the source producing it (2), i.e. the velocity of light is constant with respect to all bodies of reference K, K', etc.. In the view of individual observers at any of the bodies of reference, the velocity of light can be seen as a SA. The the compatibility between the principle of relativity and laws of light propagation lead to a conclusion that space and time ( four-dimensions ) is relative (2).

The phenomena so far discribed are consistent with the PRINCIPLE OF SIMILARITY which is defined by the author as follow:

" A SA, CONSIDERED AS AN INNER STRUCTURE, SHOULD HAVE AN EQUIVALENT DESCRIPTION FOR ALL ENTITIES ( OR BODIES OF REFERENCE) SO THAT ALL TYPES OF OBSERVATION MADE OF THE SA WOULD BE EQUIVALENT OR RELATIVE."

With respect to natural phenomena, the following rules are hypothesized and listed here for the reference of interested readers:

- (i) all dimensions of an entity should be relative;
- (ii) all levels of the state variables of an entity should be relative, e.g. velocity, acceleration, etc..
- (iii) and the state variables, like temperature, pressure, etc. should possess "constraints" like the velocity of light, which should be independent from the bodies of reference ( entities ). These "constraints" can be seen as being SA of a single entity or common SA of a number of entities.

The conservation laws for energy and mass can also be seen to support the principle of similarity , if we consider the quantity conserved to be a SA of either a single entity or a common SA of a number of entities. To allow our arguments to develop further, we can integrate the concepts "energy conservation", "mass conservation", and the "natural laws" with a single concept of "EXISTANCE" (1) ( simply understood as " it is there ") and combine these principles to define the "LAW OF EXISTANCE CONSERVATION":

"EXISTANCE BY ITSELF IS NEITHER CREATED NOR DESTROYED BUT CAN BE CONVERTED OR TRANSFORMED."

#### 4. THE LOGIC OF SIMILARITY EQUILIBRIUM

Entities can only be in a neutral state i.e. any entity must be in an equilibrium state. Whenever there is an internal "positive" effect there must also be an equal and opposite "negative" effect. Otherwise, the law of existence conservation will not be observed. If we denote any "positive polarisation" within entity as P the opposite "negative polarisation" N will have equal magnitude i.e.

$$P = - N \quad \text{and} \quad |P| = |N|$$

Here, "=" means "equal"; "-" means "opposite".  $|P|$  or  $|N|$  are called the similarity modulus of the entity. Or, we can call  $|P|$  or  $|N|$  as the SA of the two opposite aspects of one entity.

For example, action and reaction forces F and F' existing between two objects K and K' will have the same magnitude but opposite directions according to the third law of Newton (2). By the second law of Newton the product of mass and acceleration of the object (mA) will equal to the accelerating force F to establish a state of dynamic equilibrium. Actually, if we consider K and K' as two reference bodies, from the theory of relativity, we see that the behavior of the action and the reaction forces F and F' on objects K and K' is consistent with the principle of similarity :

Let K and K' be two persons who use their hands to push or pull each other. If K experiences a pull from K', then K' will also experience a pull, similarly if K' is pushed to too is K. We can say that K and K' have the same ( or an equivalent ) description of the effect of the force existing between them. The same is true when K and K' describe

their relative velocities ( either towards or away from each other ).

Here, we define any two opposite effects of a completed entity as comprising a " Positive Similarity Pole " (PSP) and "Negative Similairty Pole" (NSP). We suggest that : "Similarity Poles" (SP) of the same "polarity" repel each other whereas SPs of opposite polarity attract each other. The action between similarity poles thus ruled above is listed in Table 1. The truth value discribed from these two opposite SPs about the action ( "attract" or "repel") existing between them is summarised in Table 2, where "T" denotes "true" and "F" denotes "false".

Table 1 Action between Similarity Poles

pole A	pole B	Action Effect
P	P	repel
P	N	attract
N	P	attract
N	N	repel

Table 2 Truth Value of the SA Discribed by the Acting SPs ( P and N )

PSP	NSP	Truth value
T	T	T
T	F	F
F	T	F
F	F	T

Table 1 and Table 2 together are referred to as the Logic of Similarity Equivibrium. Table 2 tells that the discriptions from PSP and NSP are considered to be true only when they have the same truth value on the action existing between them, i.e. they should provide the same answer either "true" or "false" when being asked whether they feel being

"attracted" or "repelled". Table 2 fits the same truth value table of the logical connective "if and only if" defined in mathematical logic (3).

## 5. MODEL OF SIMILARITY FIELD

The directional effects of SA can thus be considered to be the two extreme states ( PSP and NSP ) which provide necessary combinational effect to form an entity. An entity in total must be in an equilibrium state. When a complete entity is "split" into two parts, there will be the occurring of two poles ( SPs) with opposite functionality. For example, one event can be the result of the action of another event. A table is higher than the floor but lower than the ceiling. In this comparison the observed height of the table will also depend on the direction of the view. The south and the north poles of a magnetic field are an example of a PSP and NSP. Similarly, the phenomena of polarity can also be witnessed in mathematical operations like addition and subtraction, multiplication and division, integration and differential etc.. There are numerous other commonly accepted examples of SPs.

Imagine that there exist influences between and around any two SPs of an entity. Similarly, such influences can be imagined with respect to the action and reaction influences between complete entities, as previously described in the logic of similarity equilibrium. These influences should act along an infinite number of "operation lines", existing conceptually like mathematical operations or as found with lines of magnetic flux (4). According to the law of existence



conservation defined previously, these operation lines should form a "self-closed" system. The existence of such influences can be considered to form the directional effect of SPs and lead to the existence of a "Similarity Field". A magnetic field (4) is one example of such a similarity field.

Thus, an entity can exist as the result of a similarity field and can produce its own similarity field which will contribute to a larger similarity field, by the nature of its alignment with other entities, i.e. it influences its environment and is influenced by its environment.

Here we propose the existence of primitive entities from which higher order entities can be built up. When these low level entities are in a random order, they produce no external effects, in much the same way as for molecular magnets which are randomly oriented (4). As stated above, individual entities and their environments have a mutual influence or in other words these individual entities influence each other. Under certain conditions, previously "randomly oriented" primitive entities can be "oriented" in such a way as to produce an external effect. The process of taking "randomly oriented" primitives and organising them into "oriented" entities is called SIMILIZATION. The reverse process, i.e. to cause "similarised" entities to become "randomly oriented" entities which produce no external effect is called DISSIMILIZATION. The writing of an article can be said to be the similarization of "unorganised" words, phrases, sentences, etc., to convey information ( i.e. create an external effect ). Entities which have "oriented" external effects are called and used as MODULES in the

process of similarization or dissimilarization.

## 6. HIERARCHICAL MODULAR STRUCTURE

Having been similarized, entities will produce external effects and we suggest that such external effects can be considered to form SPs. An SP can be considered to be entry port of the entity and the other as the exit port. Thus an entity can be referred to as a MODULE which has entry ports and exit ports. When such modules are "linked" together, an "oriented" hierarchical structure is formed (which can be described by a tree structure as defined in discrete mathematics): entities in a hierarchy are not necessarily described as having only one entry port and one exit port, but those "oriented" having only one entry port and one exit port are referred to as in a "HIERARCHICAL MODULAR STRUCTURE". Although, in general, structures can be more complex than that of a pure tree structure, they can be simplified: e.g. a complex electric circuit can be transformed into an equivalent circuit in which elements are connected either in a serial or in a parallel way. The essential idea of hierarchical ordering still remains:

In a hierarchy, a module issues "instructions" to its nearest lower modules and provides "services" to its nearest higher module. Modules (entities) at the same level and/or at the different levels can demonstrate similarities in structure. By having a knowledge of the basic structure of one module (or entity) in a hierarchy, we may be able to infer knowledge concerning other modules (or entities) either vertically (at different levels in the hierarchy) and/or horizontally (i.e. for modules at the same level). For example, the celestial

bodies of the solar system comprise materials which themselves consist of molecules which in turn consist of atoms and so on. There are similarities between the kinematic and dynamic properties of our solar system and of a single atom: e.g. similarity between the movement of electrons and atoms and those of planets and solar systems in a galaxy. The general laws of physics can be seen as SAs for different levels of motion, i.e. uniform velocity, uniform acceleration, etc..

In the underlying principle of similarity, there are no fixed "markers" or "milestones" against which we can reference the absolute origin and/or the direction of of an operation. Therefore, the primitive modules in a hierarchy can only be defined in a relative manner (e.g. the international standard metric system).

A remarkable characteristic of the hierarchical modular structure is that a hierarchical modular structure retains after either modularisation (divide a module or tree into smaller modules or branches) or demodularisation (a reversal process of modularisation, i.e. combine smaller modules into one module). In either case, the hierarchical modular structure is still inherited. We can refer this as the RELATIVEITY or a SA of hierarchical modular structure which can be interpreted as being a natural extension of the effect of polarity of modules having only a single entry and exit port. The property of forming modules in such a similar or relative way is defined with the term MODULARITY.

Hierarchical modular structure is a universal phenomenon and an endless number of examples of such a structure can be found.

## 7. SIMILARITY AND MODULARITY METHODOLOGY

In the previous sections, we have introduced a series of concepts concerning similarity and modularity. We can refer to the use of these concepts as the methodology of similarity and modularity. Two aspects of this methodology are considered below.

(1) With respect to similarity, observations can be carried out in two ways: from the general to the individual and from the individual to the general. For example, deductive reasoning and inductive reasoning can be seen as the two SPs of the "reasoning field". Similarity can be demonstrated at different levels and/or at the same level.

(2) With respect to modularity, we have the following understanding. We see hierarchical modular structure as an essential ordering of the world. Having such an understanding we can discipline ourselves and make use of the structure to accomplish a particular function. For example we can structure any problem in hierarchical form, i.e. divide complex tasks into smaller component problems, then accomplish them one by one to approach the final solution. We also should notice that separated sub-problems or modules can have similar properties ( i.e. SAs ) and hence the solution of these problems may involve a similar approach. A design embodying a combination of methods (1) and (2) can thus offer advantages such as simplicity, flexibility, efficiency etc..

To further illustrate the ideas presented so far let us consider other common examples.

# In control theory, a transfer function can be used to describe SMAs of various physical systems or components of those systems, be they of electrical, mechanical, etc. in form. By studying the behavior of an electric system or component we predict the properties of a mechanical system or component with similar "transfer function". Many experiments are performed on imitation systems ( i.e. prototype systems) because of safety, cost, cycle time and the like. Computer simulation is of the same nature.

# Many scientific inventions are the result of imitating natural mechanisms.

# The theory of general relativity can be seen as a "similarity " generalisation of the theory of special relativity: a "structure" in a uniform motion is extended to be valid for all the types of motion at different levels.

# Our society is organised in a hierarchical modular form e.g. governments, armies, political parties, companies, schools and universities, and so on. Our natural language and artificial languages demonstrate a hierarchical modular structure. The same is also true of our knowledge systems.

## 8. MODULAR PROGRAMMING IN COMPUTER SCIENCE

Here, we will consider an example in greater depth to illustrate the use of the idea's introduced in this paper. Modular programming has

been accepted as a design principle by many software engineers and is an extremely important approach in the creation of large or complex computer programs.

One of the traditional methods used to control the complexity of a large program was modularisation, the division of a program into a number of independent modules. When this is done, each module or sub-program can be implemented independently of other modules. Thus, the work involved in writing the entire program is roughly the sum of the work required for each module, i.e. increases linearly with program size. Similarly, each module can be debugged, documented, and maintained individually.

In the view of a computer compiler ( interpreter or assembler ), the structure at the interface of each module will be the same or similar in form although the contents of each module will be different. Once a programmer knows how to design one module, he/she should be able to use the same principles in designing similar modules. In the example shown in Figure 1, modules { a, b, c, d, e } are the most primitive modules in a hierarchy and provide services to modules { f, g } which in turn provides services to module { h }. For example, functions and procedures defined in Pascal ( a high level computer language ) are used as modules to accomplish a specific task where at runtime low level software modules are called and executed. In computer programming and software engineering, top-down and/or bottom-up are the two commonly-used procedures. As numerous amount has been written on modular or structured programming (5), the author will not detail any further on this topic and the reader may like to read those related

articles to appreciate this programming method.

9. CONCLUDING REMARKS

The author believes that similarity, in terms of observation, operation, and the equivalence of discription, is an integral part of the nature of the world and of our every-day life. The develop- ment in both natural and social sciences has pointed to the fact that there exists these similarities which pervade the universe. Finally, the auther should state that topics of this nature are very likely to lead to discussions and conclusions which are too general, diverged or argurable. However, the author hopes that this paper is well "similarized" so that it can produce such a " similarity and modularity field" having external effects.

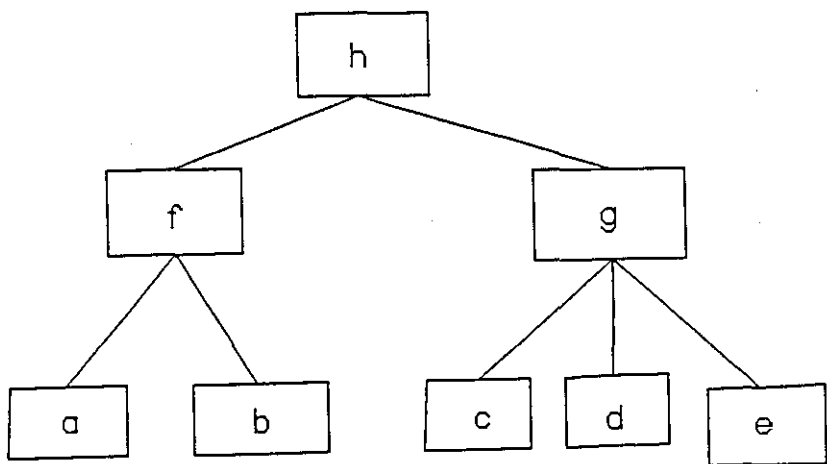


Figure 1

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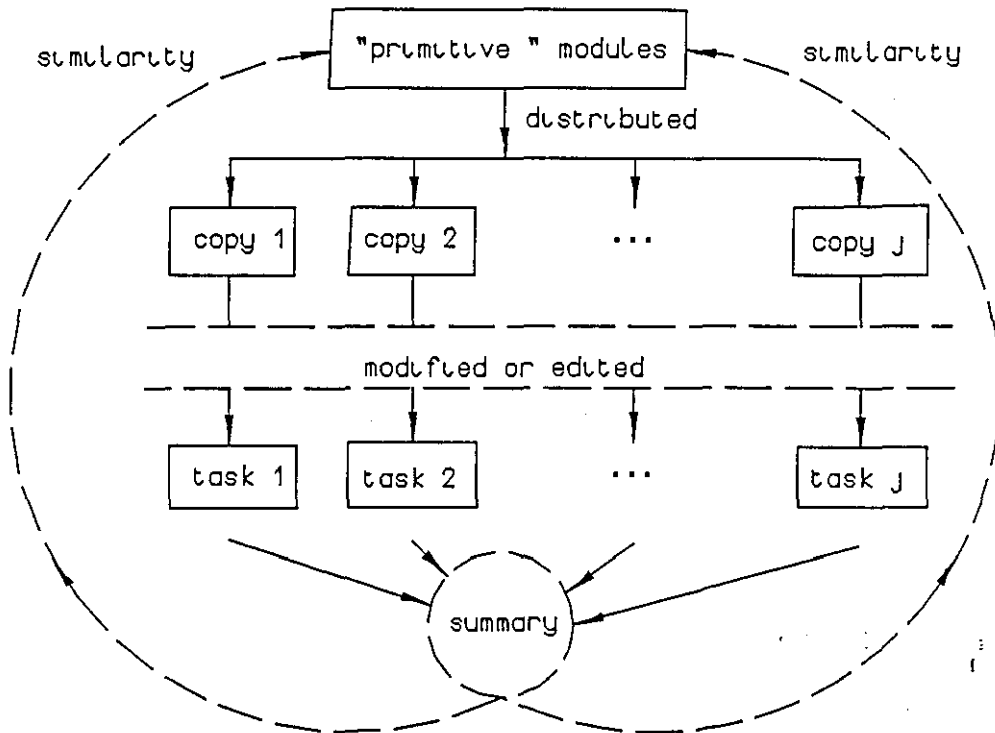


### Appendix III Evaluation and Administrative Software

(The complete software environment has been illustrated in Chapter 3)

#### III.1 "Distributed" Programming (method of "similarity" and "modularity"):

This method has been found helpful in the project and summarised by the author as follows (see the diagram below): Firstly, "primitive" programme modules are developed, then the "user" modules are obtained through editing the "primitive" modules. The process of editing the "primitive" modules is similar to that of manufacturing components from raw materials ("primitives") to produce finished products ("user" modules).



"distributed programming"

The content or structure of these "primitive modules" should contain the "interface" (between neighboring modules of the user program) and a "menu" of options for the specific task of the modules. Copies of these modules can be made as desired. These copies ("raw materials") may then be enhanced and edited to match a specific control scheme. Each final program should be "self-contained" and "dedicated". When linked together, they should be able to accomplish a specific task but their "primitives" are "general" or "universal". When developing software on a floppy disk based microprocessor the use of "primitive modules" offers the following advantages.

(1) In comparison with a "universal" program which contains all the development schemes (control tasks), (i) code execution is faster as the unnecessary code for a general program has been omitted; (ii) it eases the microprocessor memory loading capability (which can be a problem for a microprocessor having limited memory space).

(2) In comparison with a "dedicated" programme, the "primitive" software can be easily modified and enhanced to accomplish another control scheme, which has similar evaluation or administrative features.

It should be emphasized that the "primitive" software normally is not required to do a real job but to "simulate" or "idle" various control tasks. However, it contains the "generic frame for the tasks. The functionality of a software "primitive" is similar to that of the raw material for a machine component, which on its own will not do a useful job but can be used to produce various parts which have some "similarities".

The major problem is that it is not easy to have the right "primitive" software. If care is not taken the modules developed can be useless, thus wasting time. The natural solution is to develop a "dedicated" software first at the start of a project that can be later modified and enhanced to serve as "primitive" from which other programmes can be developed.

## III.2 Software Specifications

The complete software is written in a hierarchical menu-driven modular form

### Menu Headings:

#### 1. Initialisation

- .programme decrementor  
(interrupt driven)
- ."null" value initialisation
- .open-loop step tests
  - a.5-port valve control
  - b.two 3-port valve control

#### 2. Sequence and motion defined

motion	$x_d$	$\dot{x}_d$	$\ddot{x}_d$
1	$\pm$	$\pm$	$\pm$
2	$\pm$	$\pm$	$\pm$
.			
.			
m	$\pm$	$\pm$	$\pm$
cycle number: ---			

#### 3. Control options (refer to "distributed programming")

- 1> proportional control
- 2> minor-loop compensation
- 3> "null" evaluation
- 4> front-end control scheme:
  - a. manual
  - b. learning
- 5> velocity control: methods:
  - a. ---
  - b. ---
  - c. ---

#### 4. On-screen display

- a. real time data
- b. evaluation data

#### 5. Data transfer

CORTEX - PRIME

{ data handler:  
chips 9901 & 9902 }

data options:	data format
	data syntax
	evaluation
	realtime
1. real time	desired motion & DAC command
2. evaluation	
3. <1 & 2>	

## Appendix IV.1 Experimental procedures for evaluating system null values

system null value versus piston position

- tune the servo amplifier to set up the system null value roughly at the middle of DAC command range : 0800(hex)  $\rightarrow$  5 V
- locate the piston to a desired position
- increase the DAC command input gradually untill the piston starts to move then write down the last DAC command input
- decrease the DAC command input gradually untill the piston starts to move from its desired position, then write down the last DAC command input

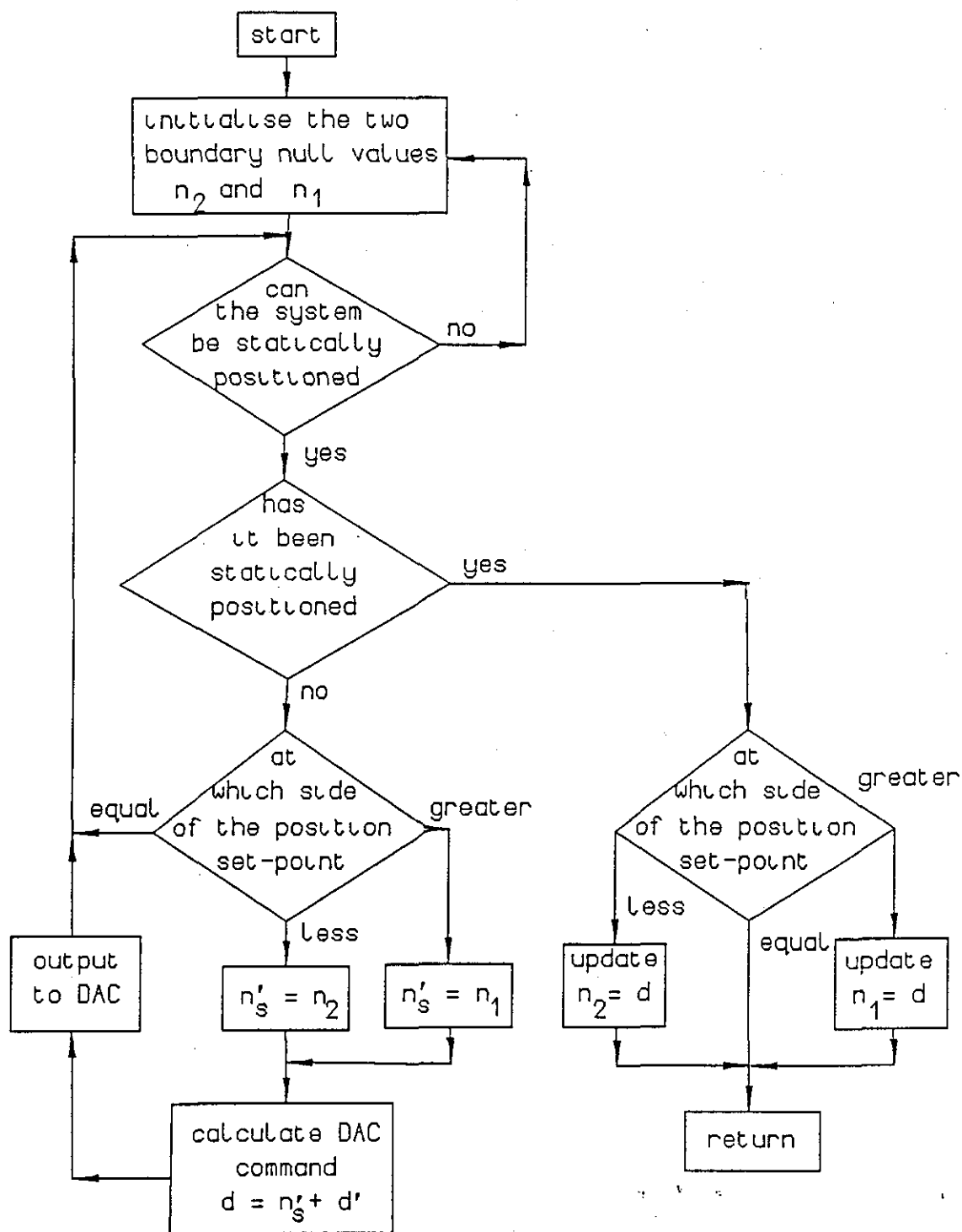
(see Figure 5.3a for results)

system null value versus supply pressure  
(at mid-stroke)

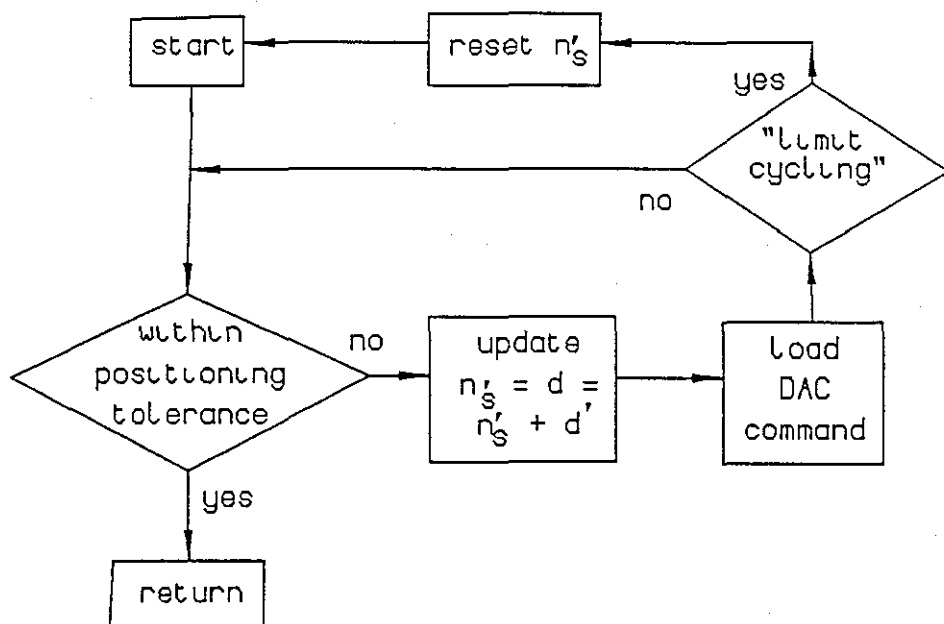
- locate the piston at mid-stroke
- set the supply pressure to desired value
- repeat the last two procedures described in the other table

(see Figure 5.3b for results)

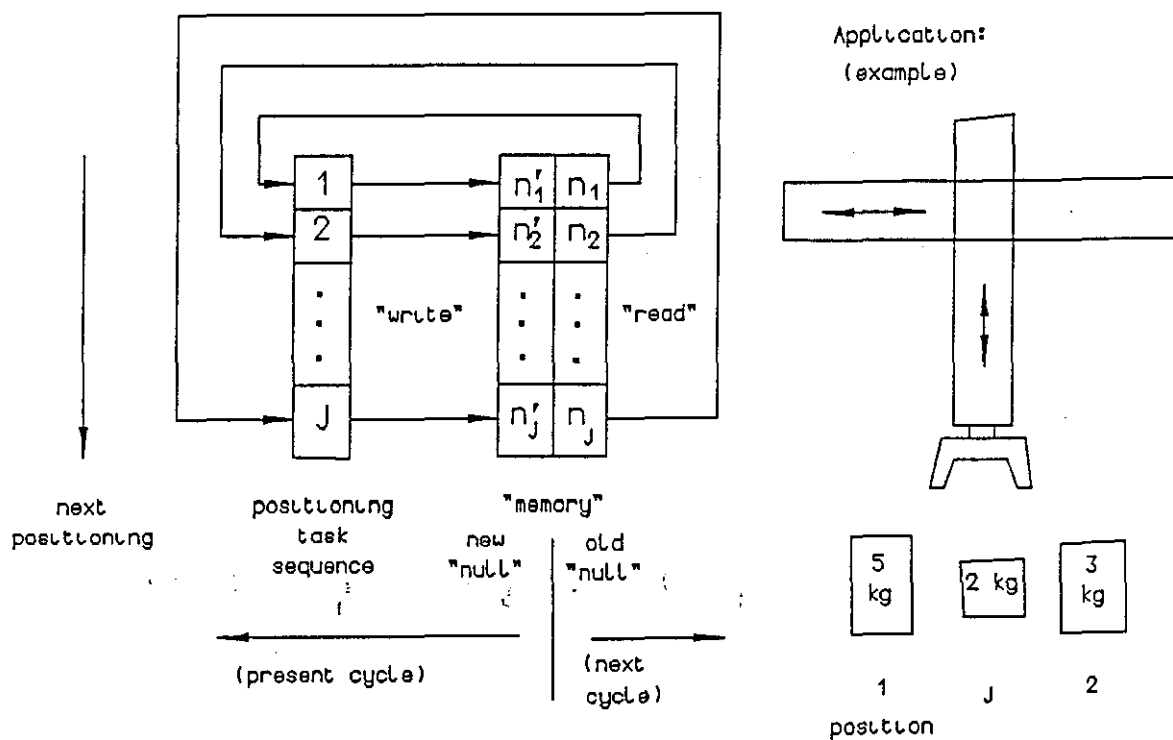
Appendix IV.2.1 "Null" automating Algorithm  
(see Chapter 5)



## Appendix IV2.2 "Null" error integrating strategy



## Appendix IV2.3 "Mapping" multi-position system null values



# Appendix V Front-end Control Scheme

(see Chapter 6)

