PNEUMATIC MOTION CONTROL SYSTEMS

FOR MODULAR ROBOTS

by

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Dedicated to my wife Jayne

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

This thesis describes a research study in the design, implementation, evaluation and commercialisation of pneumatic motion control systems for modular robots. The research programme was conducted as part of a collaborative study, sponsored by the Science and Engineering Research Council, between Loughborough University and Martonair (UK) Limited.

Microprocessor based motion control strategies have been used to produce low cost pneumatic servo-drives which can be used for 'point-to-point' positioning of payloads. Software based realtime control strategies have evolved which accomplish servo-controlled positioning while compensating for drive system non-linearities and time delays. The application of novel compensation techniques has resulted in a significant improvement in both the static and dynamic performance of the drive.

A theoretical foundation is presented based on a linearised model of a pneumatic actuator, servo-valve, and load system. The thesis describes the design and evolution of microprocessor based hardware and software for motion control of pneumatic drives. A British Standards based test-facility has allowed control strategies to be evaluated with reference to standard performance criteria.

It is demonstrated in this research study that the dynamic and static performance characteristics of a pneumatic motion control system can be dramatically improved by applying appropriate software based realtime control strategies. This makes the application of computer controlled pneumatic servos in manufacturing very attractive with cost performance ratios which match or better alternative drive technologies.

The research study has led to commercial products (marketed by Martonair Ltd), in which realtime control algorithms implementing these control strategy designs are executed within a microprocessor based motion controller.

CONTENTS

CHAPTER 1

INTRODUCTION

This thesis describes the design, implementation, testing and commercialisation of microprocessor based pneumatic motion control systems which can be used to achieve the actuation of single degree of freedom robot modules.

The research study was conducted in the Department of Engineering Production at Loughborough University of Technology as part of a Science and Engineering Research Council sponsored research programme working in collaboration with Martonair (U.K.) Ltd. The objective of this research study has been to design and implement pneumatic motion control systems which can provide high performance/cost ratios; these can be exploited as the drive elements of Modular Robots.

Pneumatic actuation was of particular interest in this research study as our industrial collaborators, Martonair, have a long standing association with this technology albeit in 'end stop' form. A survey of commercial Modular Robot Systems (Moore et al [1983]) had also indicated that in the present generation of proprietary modules their application will be limited by the unavailability of suitable servo-controlled pneumatically actuated modules and appropriate control system facilities.

Pneumatically actuated single degree of freedom modules can be used in combination to form custom designed configurations of multi-axis 'distributed manipulators' to suit application requirements, or alternatively they can be the building blocks of special automation equipment in which modules, which are not necessarily coupled together mechanically, can function in an integrated manner to perform specific manufacturing operations.

Pneumatic actuation is widely used throughout manufacturing industry as a simple rugged means of providing linear or rotary motion at low cost. In certain environments it is the only possible choice of actuation, eg for reasons of safety, or reliability (where inflammable environments are encountered, or contamination in food processing is a consideration). A double acting cylinder is usually operated by a five-port binary electro-pneumatic control valve which directs a supply of air to either chamber of the cylinder causing it to fully extend or retract. Apart from special 'positioner cylinders' which are sometimes used in process control industries, pneumatic cylinders are not generally used for point-to-point position control.

The simplest form of control is 'end-stop' or 'positive-stop', where position is set by a manually adjustable mechanical stop (a damper is often necessary to reduce impact forces to an acceptable level). Positional repeatability is often very good and typically 0.1mm. However, where an application requires several positions to be achieved within a fixed stroke length and/or requires that these positions are changed relatively frequently to meet changes in the related process (i.e. small to medium batch manufacturing) then end-stop positioning is not usually acceptable.

Where any degree of position selection flexibility is required then a motion control system is necessary. Previously this has implied the use of an electric or hydraulic position control system. Such requirements are characteristic of industrial robotic applications, which can be identified with flexible/reprogrammable automation (Engleberger [1980], Ottinger [1982], Owen [1985], Astrop [1982]).

The concept of modularity in the design of manufacturing/processing/automation equipment has generated increased interest as the trend towards flexibility in manufacturing systems has gathered pace with the widespread introduction of computer based controls (Riley [1982]). Indeed, the 1980's can be seen as the decade in which flexible automation schemes, where industrial robots participate, have found wide acceptance and application in manufacturing industry. Potential benefits from their implementation have been economic cost effectiveness, environmental safety, product/process development, quality assurance, and manufacturing system flexibility (Engelberger [1980], Ottinger [1982], Owen [1985]).

Unlike many areas of automation technology, (Riley [1982]), the manufacturers of industrial robots have not readily adopted the concept of modularity with respect to either mechanical configuration or control system design.

One small section of industrial robot suppliers that have in recent years adopted a modular philosophy in the design of manipulators are the manufacturers of pneumatically actuated pick and place units. Conveniently these handling systems are controlled by binary (off/on) electro-pneumatic valves and work between mechanically adjustable 'end-stops'. A number of manufacturers of modular robot systems have been identified (Moore et al [1983]) of which most are of a European origin. A few modular robot systems do employ hydraulic or electric actuation, but pneumatic actuation is the most popular (Weston et al [1986]). End-stop modules can be mechanically coupled to form low cost multi-axis manipulators, which can demonstrate short cycle times, good repeatability and reconfigurability. In late 1983 Martonair released into the U.K. market a family of single degree of freedom 'end-stop' modules of this type as the first stage of their entry into the robotic and work handling systems market. (Taylor [1984])

A pneumatic motion control system which can achieve point-to-point positioning has obvious potential in robotic and general work handling applications due to its cost advantages when compared with alternative drive technologies. However, previously pneumatic actuation has demonstrated distinct disadvantages when positioning payloads due to inherent time delays and highly non-linear static and dynamic characteristics. The problems introduced result primarily from the compressibility of the working fluid, which introduces significant time lags, so that the resulting system has low stiffness and little inherent damping and involves highly complex analysis when modelled.

For these reasons the design of position control systems for pneumatic drives has received considerably less interest from the research community than alternative drive technologies. Significant work has been carried out on the design of position controllers for hydraulic drives, (McCloy and Martin [1983], Burrows [1972], Turnbull [1976]) and in recent years the development of electric drive technology has superceded this (Weston et al [1986], Tal [1984]) for all but high power applications.

Due to the inherent problems associated with a pneumatic drive and the unavailability of a suitable proportional control valve (in commercial form) the initial work in designing a motion control system in this research programme concentrated on the use of binary control valves and pneumatically actuated pneumo-mechanical control elements (Moore [1986]). A microprocessor based control system allowed bang-bang controls to be produced using digital switching algorithms. A novel position control system was implemented, but the potential for commercial exploitation was limited by the additional complexity introduced by the use of pneumo-mechanical control elements (brake mechanism, sliding carriage, etc.) and associated control valving. Other researchers have developed bang-bang positioning systems (Rooks, Huhne et al, [1974] etc) but none of these to the authors' knowledge have been adopted in commercial positioning systems.

The obvious attractions of a conventional position control system with pneumatic actuation, (without the additional elements of pneumo-mechanical bang-bang systems), and the subsequent availability of a Martonair proportional control valve, (which has been developed specifically for this research study), resulted in this research study for the design and implementation of a digitally controlled electro-pneumatic servo drive.

Early research into the design of pneumatic servos used analogue control systems (Shearer [1984], Burrows [1967], Cutland [1969]). Attempts to compensate for the inherent time delays and non-linearities in the drive using various electrical, mechanical, and pneumatic devices have met with only limited success. However, the advent of low cost LSI elements provides an opportunity to implement much more sophisticated control strategies in software (Drazan [1978]), which allow the necessary compensation techniques to be applied. These developments in LSI chip technology, coupled with developments in actuator and sensor technology have allowed a low cost pneumatic motion control system to be produced which can find widespread industrial applications.

The use of microprocessors in motion control systems is expanding rapidly irrespective of the drive technology (Anon [1983], Tal [1984]). Most control systems currently being designed use microprocessors to perform all of the control functions of the system. When compared with traditional analog control system design involving the use of discrete components, the microprocessor based designs demonstrate fewer components, lower cost and improved control system design. A major advantage of the microprocessor based control system is the flexibility it offers, particularly significant in the design of pneumatic control systems. The microprocessor enables complex control algorithms to be performed such as the introduction of non-linear control functions, minor loop compensation (without the need for a large number of transducers) and in the use of adaptive control techniques: it can also provide a digital data link to other computers within a larger architectural structure. A further advantage is that the microprocessor, being a digital device, has no component variation, hence the characteristics of the control system generated by the microprocessor do not drift as they can with analog control systems.

As a result of this collaborative research programme, Martonair launched an electro-pneumatic motion control system in May 1985 (Morgan [1985]), which offers the capability for programmable positioning at a lower cost than competing technologies. This motion control system is the first of its type to become available commercially and promises significant exploitation within manufacturing industry. This thesis outlines the design, implementation and testing of the prototype drives which were produced at Loughborough University.

CHAPTER 2

MODULAR ROBOT SYSTEMS

2. INTRODUCTION

A popular concept of an industrial robot is that of a multi-axis mechanical structure with associated computer systems, which can manipulate components and tools about a defined working envelope (Engelberger [1980]). Programmed control of such manipulators enables flexibility to be introduced into manufacturing automation schemes. Many industrial robots conform to this concept and are supplied by an extensive range of vendors (Robotics World Directory [1984/85]). They can be associated with significant capital cost which is further increased by the addition of specialist tooling, non-standard grippers and complex sensors such as vision systems.

The definition of a robot formulated by the Robot Institute of America (RIA) is as follows:

"A robot is a reprogrammable multi-functional manipulator designed to move materials, parts, tools or specialized devices through variable programmed motions for the performance of a variety of tasks".

Within manufacturing industry, robots of all shapes, categories and size are being used to replace manual operatives in the same way as computers are taking the number crunching burdens from the mathematician. They are used to perform functions that are hazardous to the health of humans, eg paint spraying, adhesive application, arc and spot welding and fettling in foundries. They are similarly replacing human operatives in tasks that are repetitive and fatiguing as well as being used within environments that are impossible for humans (eg areas of radioactivity, extremes of temperature etc.). They are now finding ever widening application in assembly related tasks as the technological level of the robots increases (Owen [1984]).

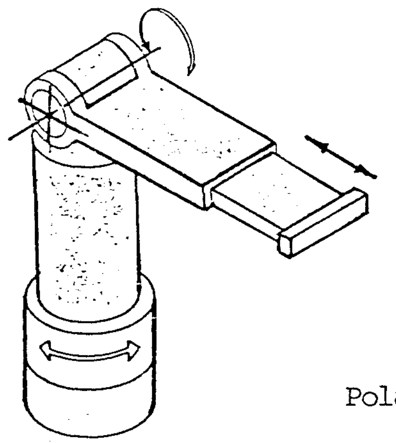
Within the RIA definition of industrial robots there are two distinct categories; non-servo and servo. Non-servo robots are those whose movements are set manually through adjustable stops on each axis. Servoed robots are those whose movements are totally controlled by a "control system" and as such there are many thousands of movements possible within one program. Both servo and non-servo robots now used are almost exclusively microprocessor controlled, although in a non-servo manipulator the control only relates to the sequencing of the motion. In a servo manipulator each function, action and task is initiated and supervised by the microprocessor.

There are two types of motion control; (i) "point-to-point", whereby the two extreme points are programmed and the motion between these two points is random (unless the sequence and actioning of each axis is done individually to obtain a desired motion), and (ii) "continuous path", whereby each point along the desired route is recorded and can be reproduced within the tolerance band of the robot. Motion is usually three dimensional and can involve the interaction of up to seven or eight degrees of freedom (ie the total number of axes of movement).

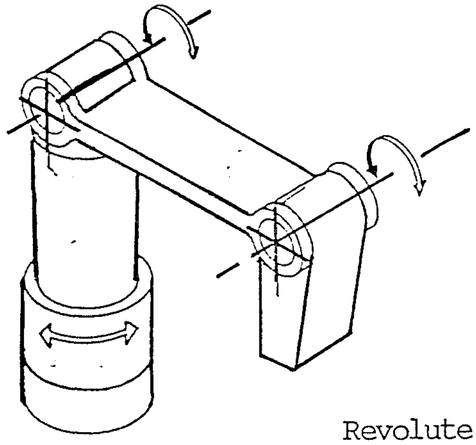
Industrial robots demonstrate a variation in axis configuration (such as revolute, cartesian, polar and cylindrical - see Figure [2.1]). They can be gantry or pedestal mounted, so that a mechanical structure which is particularly suited to the application can be selected (Warnecke et al [1982], Young [1973]).

Many manufacturers supply computer based control systems which enable the user to define, by either 'teach and follow' or 'textural' methods, the

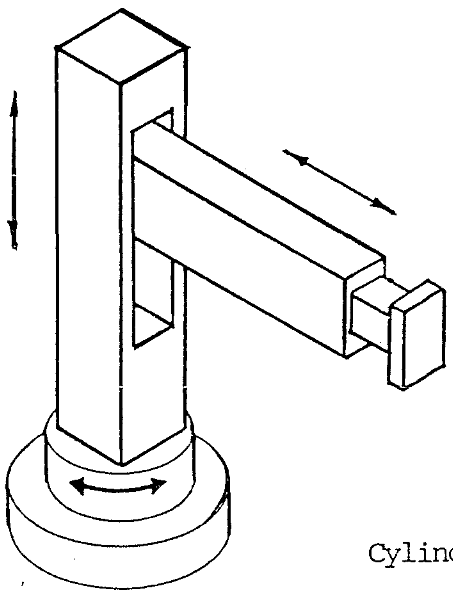
Figure (2.1) Industrial Robot Configurations



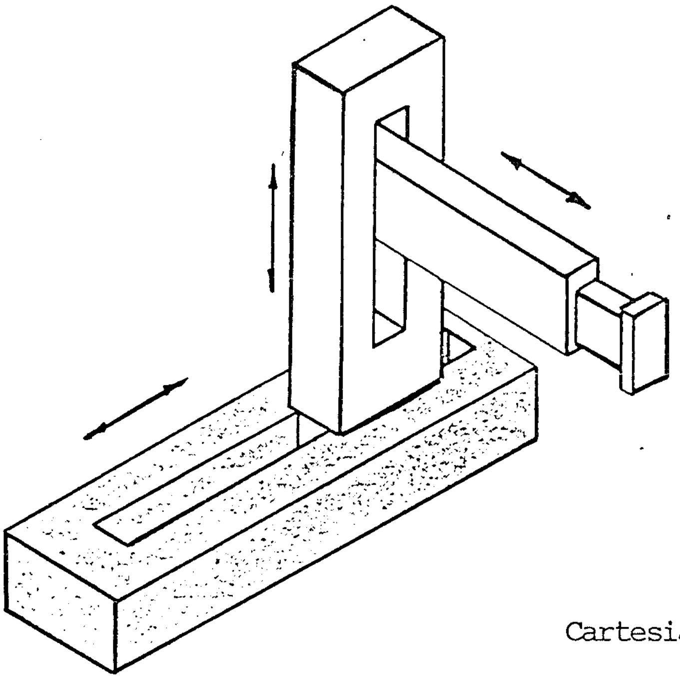
Polar



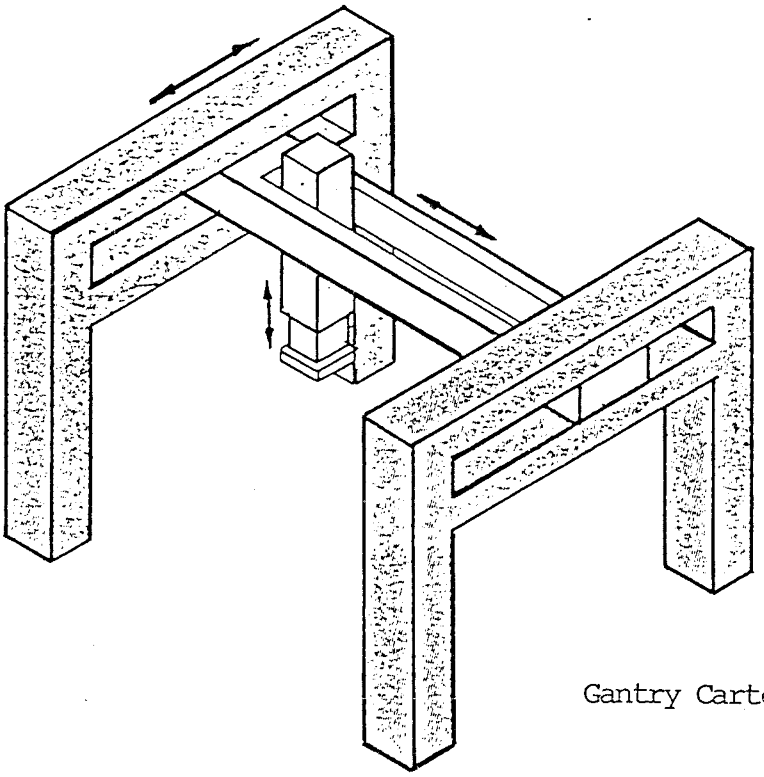
Revolute



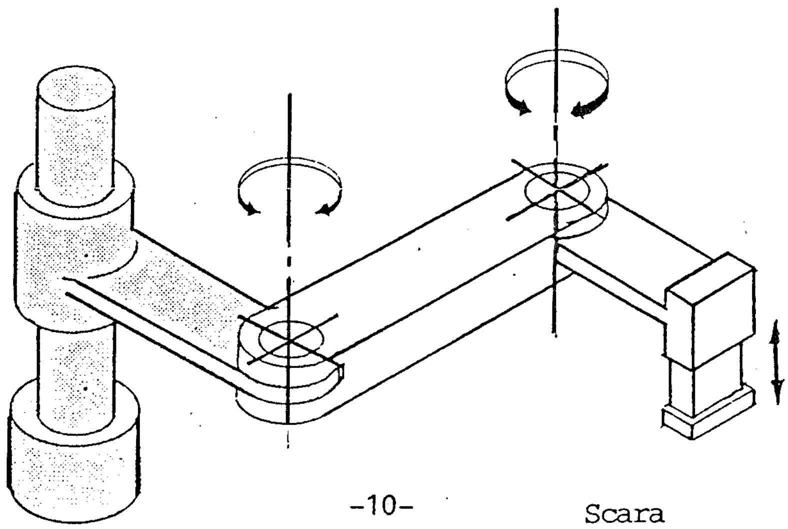
Cylindrical



Cartesian



Gantry Cartesian



Scara

robot articulations with that of associated equipment (Engelberger [1980]). In recent years the developments in commercially available robot technology have allowed improvements in the flexibility of the industrial robot so that the application areas in which they can and have been used has increased significantly (Ingersol Eng.[1980], Warnecke [1982]).

Even so the present generation of industrial robots have limitations which can restrict their application. The limitations can be associated with factors such as

i) high cost,

ii) limited flexibility in respect to kinematic structures,

iii) the limitations in performance of available drive systems,

iv) unacceptable dynamic performance,

v) insufficient control system flexibility, and

vi) problems of integration within the manufacturing environment.

The robot manufacturer would ideally like to produce an industrial robot which is sufficiently flexible to satisfy the requirements of a wide range of handling, processing and assembly operations. In attempting to maximise the flexibility of the 'universal manipulator' the manufacturer must produce a mechanical structure and control system to which it is not ideally suited for many of the applications/tasks in which it will be used (Yurevich [1981]); as a consequence it will incorporate redundant capabilities. An example being that of end user requiring an industrial robot exhibiting a large working envelope, up to six servo driven mechanical linkages could be provided, making the manipulator rather expensive due to the complexity of the mechanical structure and associated control system. However it may still not be able to achieve all the desired kinematic and dynamic characteristics.

A stiff mechanical structure can be formed if the linkages are massive (allowing high precision operations to be performed) but such an arrangement would be likely to exhibit unacceptable dynamic performance with long cycle times. In comparison with machine-tools, industrial robots are required to manipulate components and/or tools over a greater working envelope and at much higher speeds so that the performance of a robot will be constrained by the availability of suitable high performance drive systems. The problem is further compounded when linkages are mechanically coupled, with individual linkages supporting the weight of others.

When compared with specially designed cam operated automated systems, which are commonly used in high volume production assembly machines (Riley [1982]), industrial robots will be associated with longer cycle times and lower accuracy, and this will limit their application in large batch manufacture. However, industrial robots are finding more widespread application in automating processes in small batch manufacture where a significant level of flexibility is required under software control, although significant problems remain in automating small batch assembly and sub-assembly operations which require high speed, and high accuracy manipulation.

A generation of modular robot systems are now beginning to emerge which are designed largely to facilitate the automation of processes involved in medium batch manufacture, but could also find application in high volume batch manufacture as the desire for greater manufacturing system flexibility accelerates to match the ever changing market place (Yurevich et al [1981], Kozyrev [1978]).

2.1. THE CONCEPTS OF MODULAR ROBOT SYSTEMS

The axiom on which modular robots are based is that robots or manipulators can be constructed from modules to achieve user defined articulation.

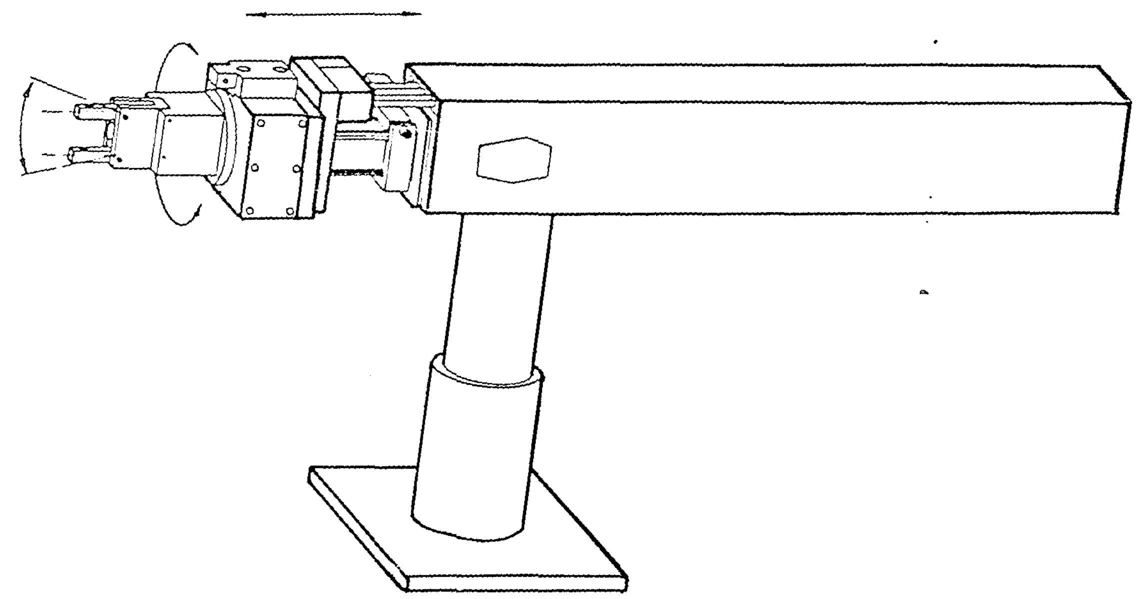
Ideally, the manufacturers of modular robot systems should offer a family of mechanical modules which exhibit varying kinematic and dynamic properties so that the user can select modules from that family to introduce automated systems which closely match the requirements of each application (Moore et al [1983]). Such a family should be complemented by appropriate control system elements which should also be modular so that control system features can be incorporated which provide a level of control and flexibility appropriate to the application requirement (Thatcher et al [1983], Weston et al [1984], Kozyrev [1978]).

Each mechanical module could provide a single degree of freedom, achieving either translation or rotary motion, so that such modules can be combined to form a very wide range of manipulator structures (Weston et al [1986], Moore et al [1983]). Figure [2.1.1] illustrates example combinations of mechanical units. Alternatively, a module could function to provide two or more degrees of freedom if sufficient demand for such a specialised mechanical unit were to exist.

It is important to illustrate that in constructing manipulators from modules significant advantage can be gained in relation to:

(i) the exclusion of redundant articulation within the mechanical structure of the manipulator as opposed to conventional robots with a pre-defined, and possibly overcomplicated structure. Thus significant cost

Figure (2.1.1)(a) Example Modular Robot Configurations



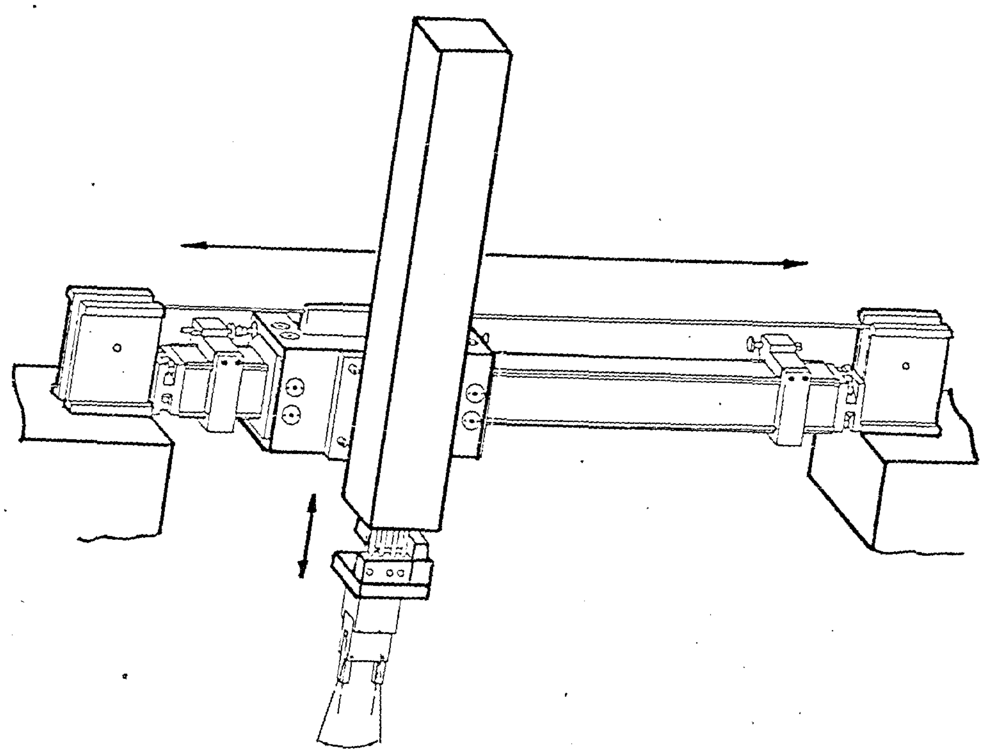
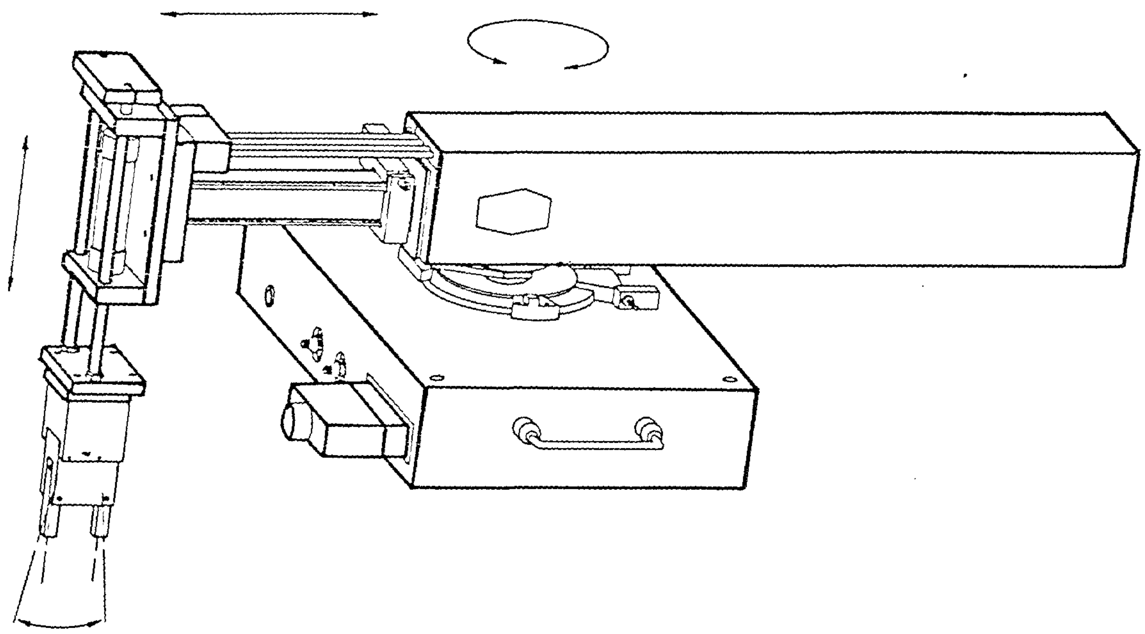
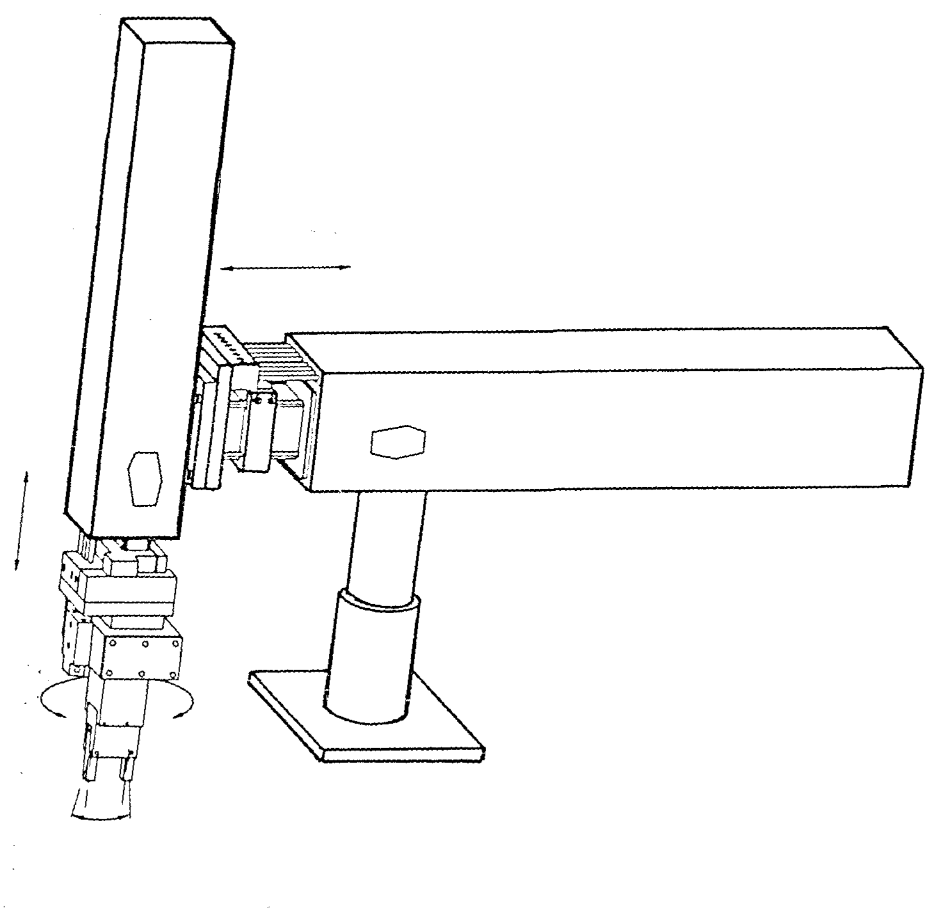


Figure (2.1.1)(b) Example Modular Robot Configurations





saving and possible performance enhancement may be achieved (Kozyrev [1978], Surnin et al [1978]);

(ii) the designer is not constrained by the need to use only mechanically coupled linkages. Mechanical modules may be arranged in more than one group of units (with no mechanical coupling between groups) which function as a 'distributed manipulator' to accomplish the task required (Weston et al [1984]). Thus the range of applications for which modular robots can be used is far wider than that for conventional robots. The use of 'distributed manipulators' also offers the possibility to achieve mechanical optimisation thereby improving mechanical stiffness and hence cycle times and/or accuracy and repeatability;

(iii) mechanical modules can be individually chosen with kinematic and dynamic characteristics which are appropriate for the function to be performed by the module. The module could be 'servo'controlled or be driven to manually adjusted end stops if only limited flexibility is required (Kreinin [1978]). Hence by selective choice of modules it may be possible to improve performance but reduce cost;

(iv) modular systems can be used to introduce automation schemes that do not require the full flexibility of a conventional robot but cannot be met by dedicated hard automation systems due to either capital cost, batch sizes or lack of flexibility (Riley [1984], Brown [1983], Richardson). They provide a half-way house between 'dedicated automation' and conventional robotic installations.

The major thrust of modular robot systems is to allow processes involved in medium batch manufacture to be automated more easily and efficiently. However, they will also find application in small and large batch manufacture where advantage can be gained. For example, re-usable/re-configurable equipment is an important consideration in small batch manufacture. The requirement for sophistication in control system elements will be closely related to the need for flexibility or the frequency with which job changes are required. In application areas, which conventionally require 'hard' automation the appropriate choice of modules may involve only mechanical units driven to preset endstops and may require only very simple control system elements which provide the necessary sequencing of axis motion in relation to the local production machinery (Kreinin [1978]). A manipulator and control system chosen for such an application may not fall conveniently into the generally accepted concept of an industrial robot. However, many modular robot systems used in automating medium and small batch operations will necessarily be complex and would be accepted by the purist as a robot. Whatever the application, potential benefit can accrue if control system elements are based on low cost L.S.I. devices which demonstrate both hardware and software modularity. Thus future generations of modular systems irrespective of complexity will be computer controlled (Moore et al [1983],Thatcher et al [1983]). Thus it is logical to consider modular robots as a class of Industrial Robots in which the level of complexity is related to the specific needs of an application.

2.2. COMMERCIALLY AVAILABLE MODULAR ROBOT SYSTEMS

It is now appropriate to consider modular robot systems and their associated control systems that are commercially available. These are reviewed in Moore [1986] and summarised in Table [2.2], from which conclusions can be drawn as discussed in the following sections.

Table (2.2)

Summary of Modular Robot Systems

FEATURES / VENDORS	DRIVE SYSTEM			MOTION CONTROL		ARTICULATION	
	PNEUMATIC	HYDRAULIC	ELECTRIC	END STOP	SERVO	ROTARY	LINEAR
AFAG	x			x		x	x
ASEA	x			x		x	x
BOSCH	x		x	x	x	x	x
EBOSA-ROBOLINE	x		x	x	x	x	x
ERACOND	x			x			x
FELSOMAT	x			x		x	x
FIBROMANTA	x	x		x	x	x	x
GEFIT	x			x		x	x
H.H. FREUDENBERG	x	x		x		x	x
I.N.A.	x			x		x	x
I.S.I.	x			x		x	x
KAUFELDT			x		x	x	x
MARTONAIR	x			x	x	x	x
METO-FER	x			x		x	x
MENZIKEN	x			x		x	x
MODULAR ROBOT SYSTEMS	x		x	x	x	x	x
MONTECH	x			x		x	x
NEFF			x		x		x
ROBOTIC SYSTEMS			x		x	x	
SCIAKY		x			x	x	x
TOLLO			x		x		x

2.2.1. SUMMARY OF ELECTRICALLY ACTUATED SYSTEMS.

Electrically actuated servo-controlled systems are produced by a limited number of manufacturers. These are either dc motor or stepper motor controlled with some form of position transducer feedback. These systems will provide 'point-to-point' control and 'velocity control' but a 'contouring' capability in a modular reconfigurable system is not currently available. The electric drive modular system appears ideal for providing high performance servo systems for 'point-to-point' and 'contouring' applications, particularly where high accuracy static performance and/or exceptional dynamic response is a requirement, for example in small component assembly where a number of Japanese systems are appearing (such as Pioneer, Panasonic, Silver Reed, etc..). A number of modular mechanical hardware systems are beginning to appear that will accomodate several types of electric drive (eg Neff, Tollo) and these should begin to find wide application once suitable proprietary axis motion control systems and supervisory controls have been identified.

The limited capabilities of the available control system elements, which compliment the present generation electric driven modules, have severely restricted the use of such systems in industrial application. Undoubtedly electrically actuated modular systems will become increasingly popular in automation schemes as their cost/performance ratio improves and as control systems elements become available with processing facilitites which equal those of conventional pedestal mounted robots. Such control system elements (hardware and software) must provide user friendly interface and ease of integration with other production equipment and computer systems in the manufacturing environment. Furthermore a control architecture, which matches the complexity of the manipulator constructed, must be configured for each specific application with a minimum of systems engineering yet must not incorporate redundant features which introduce high cost penalties. The most recent development is the use of direct drive modules (using linear motors) for cartesian assembly robots where high speeds are required.

2.2.2. SUMMARY OF PNEUMATICALLY ACTUATED SYSTEMS.

Pneumatic drive systems are the most widely used form of actuation for modular robot systems. The majority of the present generation of pneumatically actuated modules facilitate only 'end stop' positioning of payloads to adjustable mechanical stops. However, there are a limited number of manufacturers who supply modules which incorporate mechanical devices to allow limited intermediate stops. These devices often work uni-directionally and impose severe restrictions in terms of the pitch between successive stops.

A number of manufacturers, mainly European, provide a range of mechanical modules comprising translation, rotary, wrist, gripper and gantry/portal units with different performance characteristics. These modules are complemented by interface/adaptor plates, mounting stands and electrical and pneumatic couplings. Such modules can be used to configure various types and categories of manipulators at relatively low cost.

All the modular pneumatic systems reviewed position to mechanical 'end stops'. This obviously imposes limitations upon the applications that such a positioning system can be used for, allowing only limited flexibility. They are ideal for the relatively simple repetitive tasks, but are not suitable in applications which demand frequent position sequence changes (eg where a mix of products are to be manufactured) or several different positions are required on a single axis (eg for applications such as palletising and spot welding).

Clearly there is a need for low cost limited performance servo (or motion controlled) modules. Such low cost positioning systems could be best exploited if pneumatic actuation is employed as potentially significant cost advantages can be achieved when using pneumatic drives rather than hydraulic or electric drive systems. Pneumatic actuation lends itself to the design of modular single degree of freedom units; this is illustrated by the balance of companies manufacturing modular pneumatic systems compared to alternative drive types. Maintenance and operation are simple and industrially accepted, and most industrial sites have an established air line distribution network.

As the trend towards flexible automation schemes in manufacturing gathers pace, the demand for 'intelligent' drives will increase. For small and medium payloads, a low cost pneumatic servo-drive together with appropriate control system hardware and software facilities could be significantly exploited in such a market. However, any such system components will only find industrial acceptance if the control system provides for easy 'systems integration' and interface with other manufacturing equipment and can be accessed in a 'user friendly' manner. This thesis describes the evolution of, a now commercially available, family of pneumatically actuated servo driven modules and associated control system elements. These modules are the first of their kind known to the author to become available worldwide (Morgan [1985], Weston [1984]).

2.2.3. SUMMARY OF HYDRAULICALLY ACTUATED SYSTEMS.

Hydraulically actuated modular systems can be considered within two distinct categories; those which are 'servo' controlled and those controlled to mechanical 'end stops'. The number of modular systems that are designed specifically for use with hydraulic actuation is limited, but some of the 'end stop' pneumatic modular units can be adapted for operation at the higher pressures used in hydraulic systems. Hydraulic drives have found wide acceptance when used in conventional robot systems, particularly for the manipulation of large payloads (from 20kg to in excess of 2000kg). Similarly for modular systems requiring a high power to weight ratio or for the manipulation of large payloads hydraulic actuation would be the obvious choice (Anon [1983]) in either 'end stop' or 'servo' form dependant upon the application requirements.

Of the hydraulic modular robot systems, the Sciaky and Fibromanta systems are used quite extensively in the automotive and automotive components industries for parts handling and welding applications. The Sciaky modular system, which provides a full servo control capability, allows point-to-point positioning with velocity control to be achieved. The drive system is complimented by computer based controls which provide many of the facilities commonly available with conventional pedestal mounted robots, such as teach facilities, program editing and storage, user friendly operator communications interfaces, diagnostics capability, etc.

2.2.4. CONTROL SYSTEMS SUMMARY

The survey of commercial modular robot systems (Moore [1986]) indicates that the present generation of proprietary modules will be limited in their application by the availability of suitable control system facilities. The control system elements which are available do not reflect the level of sophistication and 'useability' provided by the programming and operating systems of conventional pedestal mounted robots.

It should be made clear to the reader that the control of manipulators which comprise of only end stop modules is relatively simple and is easily within the scope of most proprietary programmable logic controllers (PLC's). PLC's are functionally designed to allow the programming of sequencially operated systems. The programmable I/O facilities provided by the PLC normally allow for the switching of 24vdc devices (such as electro-pneumatic binary valves), sequencial control of the system is normally achieved by 'user friendly' programming facilities such as a variant of relay ladder diagrams.

However, the control system complexity must be increased considerably when manipulators, compromising one or more servo controlled modules, are to be controlled and programmed in a 'user friendly' way. This step change in control system complexity is accentuated for modular robotic systems where the kinematics and dynamics of the manipulator are determined by the application (Weston et al [1983]). Of the present generation of modular robotic systems this problem has not been resolved satisfactorily.

Ideally a manufacturer/vendor of modular systems should supply a wide range of control system elements which demonstrate both hardware and software modularity so that the user can select control system features in a cost effective way (that is without redundancy) and yet achieve the required degree of control and flexibility in operation.

A unique requirement of a supervisory control system of modular robots or 'distributed manipulators' is the ability to co-ordinate sequencially the motions of decoupled axis groupings. Such a facility can only be provided effectively with a true 'multi-tasking' software environment in the controller. In this way, the concurrent control of separate axis groupings of a manipulator can be achieved (Thatcher et al [1983]). A potentially significant advantage modular robots have over conventional manipulators is the ability to provide concurrent motions (particularly in assembly applications) and so the control system must have the capacity to exploit this feature.

The design of control system architecture adopted in the research study at Loughborough for 'modular robot systems' is illustrated in Figure [2.2.4]. Individual microprocessor based controls are assigned to each servo-controlled mechanical unit so that the problems associated with realtime control of such modules is decoupled from the supervisory controller. This design of control system architecture is implemented in the modular system now marketed by Martonair (Morgan [1985]).

A range of supervisory control functions have been investigated and implemented (Thatcher et al [1983]) to provide an operating and programming system for the supervisory controller which includes many facilities common to conventional robot operating systems such as;

(i) pendant controlled position teaching,

(ii) program specification and editing including conditional logic,

(iii) program driven interfaces to other digital and analog devices including end-stop modules,

(iv) overall system sequence control, and

(v) establishing a data link to other computer systems.

In addition, the following facilities unique to modular systems have been investigated;

(a) reconfigurability at pre and post installation

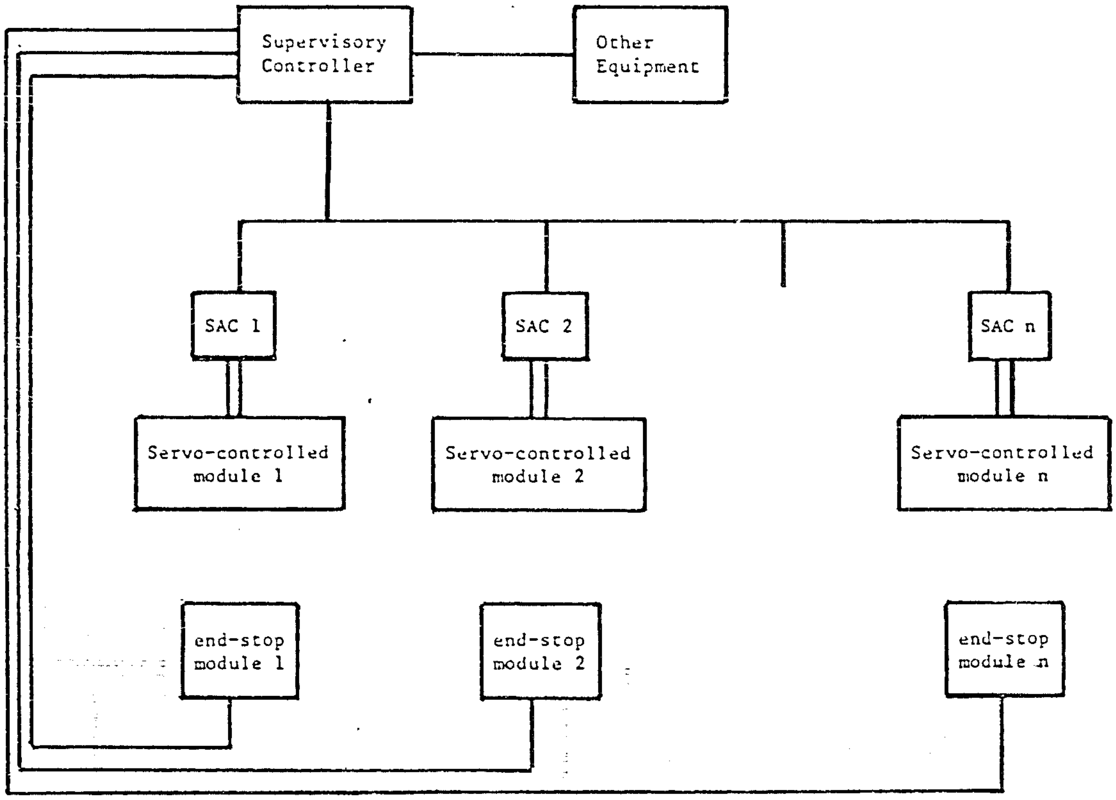


Figure (2.2.4) Distributed Computer Control Architecture phases, and

(b) programming and executing facilities for concurrent robot programs (Thatcher et al [1983]) so that more than one group of mechanically coupled modules can be controlled concurrently.

By adopting an architecture based on distributed processing, both hardware and software modularity can be established which is of primary importance in controlling manipulators which can range from the simple one or two degree of freedom arrangement to a complex automation scheme involving a number of mechanically decoupled multi-axis groups functioning concurrently to perform a given task. Furthermore, digitally controlled electrically and/or hydraulically actuated mechanical primitives can be included within such a control scheme.

CHAPTER 3

ELEMENTS OF A DRIVE SYSTEM FOR MODULAR ROBOTS

3. INTRODUCTION

In this Chapter the elements of a drive system for Modular Robots are considered. The choice of actuation and feedback system are discussed and the implications on the design of the module are considered. A description of the prototype module designs is then given.

The microprocessor based target system is briefly described, together with the necessary interface electronics for the proportional servo-valve and the feedback transducer.

3.1. ACTUATION

The choice of actuator/drive system for modular robots, as with any manipulator, is extremely important (Archer [1983]). The kinematic and dynamic properties and the cost of any manipulator will be directly related to the type of drive system used (Diaz [1980]). The state of the art in actuator technology implies a choice between Fluidic or Electric drive systems (Archer [1983], Walker [1983]). The choice of actuation within these two categories is summarised in Table [3.1.1].

TABLE 3.1.1

ELECTRIC ACTUATION	DC MOTORS AC MOTORS STEPPER MOTOR
FLUID POWER ACTUATION	HYDRAULIC PNEUMATIC

In selecting the drive for a modular robot system, whether linear or rotary motion is required, the following criteria should be considered;

(i) thrust: the actuator must be capable of reacting to the load and the intrinsic mass of the slideway indefinately and also be capable of providing additional accelerating and retardation torque/forces.

(ii) speed: the moving head of the module may be required to move at high speeds so that necessary cycle times can be achieved (this is often application dependant but any drive must be capable of reaching such high speeds).

(iii) backlash: bi-directional loads and positioning stability requirements would suggest that backlash in the drive should be as low as possible without significantly increasing cost (Archer [1983]). Direct drive is the optimal condition (assuming a stiff coupling between the actuator and slideway).

(iv) weight: the cumulative effect of actuator weight in a serially coupled modular robot demands that each drive slideway is as light and compact as is feasible. However it is possible with a modular system to "distribute" the elements of the manipulator (Weston et al [1986]).

(v) response: the actuator should have minimal inertia and a high mechanical stiffness to enable rapid accelerations to be obtained.

(vi) life: the actuator should operate with minimal maintenance and any maintenance should be simple and low cost, and

(vii)cost performance ratio: the actuator must meet a functional performance specification within an economic range (eg high performance at low cost is an essential feature of drive systems for modular robots (Moore et al [1983])).

3.1.1. ELECTRIC ACTUATORS

Where electrical actuators are chosen we are considering the final stage energy transfer from electrical energy into mechanical energy. Electric actuators for industrial modular robots are usually motors of the dc type with various armature and coil arrangements, with different excitation techniques that produce selected performance characteristics. Stepper motors are also finding increased numbers of applications particularly where open-loop control can be employed. A great deal of literature is available with tutorials on electric motors and their applications to servo controlled systems; these are summarised by Tal [1984] and Barber [1983].

In servo applications, typical dc motors offer superior torque characteristics when compared to ac motors (this definition of dc motors includes the 'brushless' type motor (Barber [1983])). The 'rare earth mineral' dc motors offer 30% greater torque when compared to dc motors with standard motors of the same size (Barber [1983]). This is a significant advantage because electric motors are in general high speed, low torque devices. Electric motors, therefore, require gearing with its inherent mechanical imperfections. Any improvement in available torque can lessen the actuators dependence on mechanical transmissions and reduce the amount of gearing necessary.

The advantages of electric actuation can be summarised as follows:

(i) high performance control achievable at reasonable cost,

(ii) availability of a wide range of motor types and sizes,

(iii) relatively low cost, high performance motors available,

(iv) low resolution position transducers can be used (gear transmission amplification),

(v) temperature stability,

(vi) high stiffness motor/transmission systems are possible,

(vii) linearity of system components, and

(viii) good reliability

The disadvantages of electrical actuation for modular robot systems are:

(i) relatively low power to weight ratio (compared with possible fluidic actuators),

(ii) need for low backlash reduction gearing,

(iii) limited choice of gearboxes,

(iv) gearbox/transmission cost,

(v) motor/gearbox inertia,

(vi) remote mounting of the actuator is not possible,

(vii) effect of mechanical linkage on module complexity and cost,

(viii) 'non-back driveable' (non-compliant),

(ix) need for careful matching of peak loadings to duty cycles to avoid thermal overload of motor, and

(x) cost of drive amplifiers becomes very significant as power requirements increase.

In selecting electrical actuation for modular robot drives, the problem is quite complex as a trade off has to be made between motor heating and transient load duty cycle and also an appropriate gearbox/transmission ratio has to be selected to give optimisation of the total drive inertia and volume. Reduction gearing is nearly always necessary with electrical drives because of the unacceptable penalty of size (and also weight and cost) when used in direct drives (Archer et al [1983]).

Few dc servo-drives were used in industrial applications before the late 1960's. However, with the appearance of the N.C. machine tools they became much more common, with a cost, reliability and performance that was acceptable. Particularly of significance for dc servo-drives for robotic applications have been the improvements in motor design resulting in much improved power to weight ratios and performance specifications.

The use of 'rare earth' motors results in lower inertia, lower electrical and mechanical time constants, and much less weight; hence they have rapidly found aceptance in robot drive systems. Rare earth motors have long thermal time constants and can tolerate the overload conditions they are often subject to in industrial robots. Rare earth motors are extremely reliable, have good brush life, are virtually impossible to de-magnetise and they give good performance for minimum weight and volume which is of paramount importance in modular robot systems (Duggan [1975]; Noodleman [1974]).

The next significant development of electrical drives has been the 'brushless' drive (Whited [1981]; Whited [1981]). Both brushless dc servo-systems and variable frequency ac servo-systems can be used as robotic actuators. The brushless dc variant lends itself well to applications requiring high power levels (Barber [1983]). In a brushless dc motor the power rating can be further increased by fluid cooling of the motor, which is not dificult as there is no problem with commutation. New magnetic materials currently under development should further raise the power to weight ratio of elctric motors and also result in a material cost reduction (as a result of using less Cobalt in the compound).

Motor drive amplifiers provide a source of current of a dc waveform for driving the electric motor. Over recent years the technology involved has changed considerably. The present trend is to use Pulse Width Modulated (P.W.M.) output stages, at a frequency of (typically) 5Khz (Barber [1983]).

As the magnetic materials technology and designs used in dc electric motors have advanced, so motor power to weight and power to size ratios have improved. An increase in drive amplifier voltage and current ratings has resulted in extended speed ranges and power ratings. The current trend of new designs of industrial robots that are appearing is to use electrical actuators of various designs. In particular where high speed and performance are required and the payload is typically less than 15kg then the use of dc electric motor drives is appropriate. The use of hybrid drives (eg electric drives with pneumatic counterbalancing) is also finding acceptance where higher payloads are encountered. Similarly in 'modular robot systems' where very high performance is required (eg assembly) or an ability to contour, then electric drives would appear to be the appropriate choice.

3.1.2. HYDRAULIC ACTUATORS

Hydraulic actuators are available in two basic forms, the cylinder (linear and rotary) and the motor, which can have several design variations (eg gear motors, radial piston motors and vane motors). Hydraulic actuators have been used since the introduction of the first commercial industrial robots exhibiting servo control. For high power applications they remain the most appropriate choice for 'end stop', 'point to point' and 'contouring' control. In some applications it is most convenient to use hydraulic actuation as the high working pressures (100 bar or more) allow very small actuators to be used (Walker [1983]; Diaz [1980]).

Hydraulic actuators are popular in robotic applications that require high power to weight ratios (Walker [1983]; Diaz [1980]). They are also used almost exclusively for paint spraying robots, due to the hazardous environment created by the inflammable nature of the paint compounds ("brushless" motors may be used in future robotic systems). Hydraulic actuation is normally used for payloads in excess of about 35 kg (Archer et al [1983]), below this point other forms of actuation tend to be favoured. The reasons for this are to an extent related to performance criteria, but more important is the relative cost and availability of components. In hydraulic systems, the servo-valves are a large proportion of the total actuation system cost. In general this is irrespective of flow capacity, so that as the requirement for increased loads in hydraulic systems is considered, the proportional increase in cost with load is much slower in this case. Hydraulic actuators are often used in a direct drive mode (particularly where linear motion is required). Due to the high forces/torques that can be developed it is not necessary to use a gear transmission. Components for producing small hydraulic systems tend not to be readily available, so the cost of such items is restrictive.

The advantages when using hydraulic actuation can be summarised as follows;

(i) high torque/volume permitting direct drive,

(ii) indefinite stall capacity,

(iii) low actuator inertia,

(iv) robust - abuse resistant,

(v) simple actuator design,

(vi) simplifies overall arm/module design,

(vii) low actuator cost,

(viii) back driveability, and

(ix) flameproof.

The disadvantages of hydraulic actuation can be summarised as follows;

(i) high cost of propriety servo-valves and proportional valves (closed loop control only),

(ii) need for high resolution position transducer (as a consequence of direct drive) when used in servo-controlled applications,

(iii) effect of oil temperature change on performance,

(iv) leakage,

(v) dirt sensitivity,

(vi) non-availability of proprietary small actuators,

(vii) need for a local supply of fluid power (this has considerable cost implications), and

(viii) non-linearities introduced by system components (eg valve lap, assymetry, leakage).

The choice of actuation for conventional industrial robots or modular robots is normally a necessary compromise between cost, operational and performance characteristics. Where the most critical factor in many instances will be minimum cost, the advent of low cost computer controls has allowed the cost/performance ratio of many drive technologies to be improved with the design of appropriate digital control strategies. Hydraulic actuators exhibit a much slower decrease in power to weight ratio, and power to cost ratio than electric drives (Walker [1983]). As for high payload applications the hydraulic drive is still the most appropriate selection in both 'servo' and 'end stop' modes.

3.1.3. PNEUMATIC ACTUATORS

In pneumatic systems, mechanical energy is used to pressurise a fluid which is then converted back to a mechanical force/torque by the actuator. Pneumatic systems, in general, are used for lighter loads than in hydraulic systems because the operating pressures tend to be much less (typically less than 10 bar), but high pressure gas servo-mechanisms have been investigated (Botting et al [1969]) and with 'end stop' control they are ideal where fast cycling is required (Anon [1978]). The variety of applications where pneumatic actuation has been used in industrial applications seems to be limited only by the designers' imagination.

Pneumatic actuators have found widespread use as drive elements for limited function robot systems providing 'end stop' motion to pre-set mechanical stops (Anon [1983]). At present the largest sector of the robot market is served by such pneumatically actuated robots which demonstrate relatively high traverse speeds (typically in excess of 1m/s), intrinsic safety in many hazardous environments, and relatively low cost (Heer [1981]). In terms of industrial control there are few devices that are as readily understood or have proved as reliable as the pneumatic actuator.

Pneumatic actuators are available in two basic forms; cylinders (rotary/vane actuators and cylinders) and motors (vane air motors, radial piston air motors, gear motors, axial piston motors, turbine motors, V-type and diaphragm motors).

Some advantages which can be identified with pneumatic actuation include;

(i) relatively high torque/force to volume permitting direct drive,

(ii) indefinite stall capacity,

(iii) relatively low actuator inertia,

(iv) robust (abuse resistant),

(v) simple actuator design (particularly where linear motion is required),

(vi) simplifies overall arm/module design,

(vii) very low actuator cost,

(viii) 'back driveability',

(ix) wide availabilty and acceptability in manufacturing plants,

(x) low cost of associated pneumatic components,

(xi) acceptability in hazardous environments (flameproof and non-toxic),

(xii) cycle rate can be very high (vent exhaust to atmosphere),

(xiii) operating temperatures tend to be low,

(xiv) no contamination problems due to leakage, and

(xv) intrinsic cushioning properties (particlarly useful when used to provide end-stop motion).

The disadvantages of pneumatic actuation are as follows;

(i) air lubrication is normally required,

(ii) 'sponginess' of response for servo controlled motion due to compressibility (and inherent time lags),

(iii) low natural frequency and hence limited bandwidth resulting from lack of drive stiffness,

(iv) need for high resolution position transducers (when used with a direct drive in servo applications),

(v) limited availability of servo-valves, and

(vi) highly non-linear system elements making servo control complex.

The availability of LSI elements and low cost powerful processing makes possible the use of digital compensation techniques. Consequently good cost/performance can be achieved and as such make a pneumatic drive an attractive proposition. The technical difficulties involved in achieving good performance from servo-controlled pneumatic drives have, until now, delayed their introduction.

Hence the decision was taken to use pneumatic actuation for motion control applied in a Modular Robot System. A pneumatic Modular Robot System should provide both 'end stop' and 'servo-controlled' capability (point-to-point control) at significantly less cost than other drive technologies. Such a drive technology, if exploited with appropriate control facilities (hardware and software) allowing easy systems engineering configuration and integration, should find wide acceptance as flexible automation schemes are introduced into the manufacturing environment.

3.2. OPTICAL ENCODERS

Encoders can take several forms and they can be 'absolute' devices (that is a transducer whose output information is a direct and absolute indication of its present position), or 'incremental' devices (that is a transducer whose output information represents only the incremental change in position). An example of an 'absolute' encoder is a 'digital shaft' encoder (eg Grey code device), an example of an 'incremental' encoder is a 'rotary optical encoder'.

The transducer that was selected for use with the pneumatic motion control was the 'rotary optical incremental encoder'. This device allowed good position resolution to be obtained (which was necessary as velocity and acceleration data was derived directly from this position data to minimise cost), at a reasonable cost and is easily interfaced using low cost LSI elements to digital control systems. Cost is a fundamental criteria in the design of a pneumatic motion control system of which Martonair could exploit commercially.

3.2.1. OPTICAL INCREMENTAL ENCODERS

The general arrangement of a 'rotary optical incremental encoder' is shown in Figure [3.2.1.1]. This shows that an incremental encoder usually consists of two disc gratings located side by side, with their centres concentric, one of which is fixed while the second is located on the rotating shaft of the encoder. Each grating is divided into a number of transparent and opaque radial lines, with a constant pitch between successive lines. A lamp (emitter) is positioned one side of the pair of discs, with a photosensor (receiver) positioned directly opposite on the other side of the pair of discs. The photosensor will convert the light pulse received into an electrical output signal of sinusoidal form which is then normally processed via a Schmitt trigger device to produce a square wave output (see Figure [3.2.1.2]). Hence a high or low signal is produced depending upon the relative position of the transparent and opaque radial lines of the two gratings.

There are various arrangements of the gratings in practice, and some

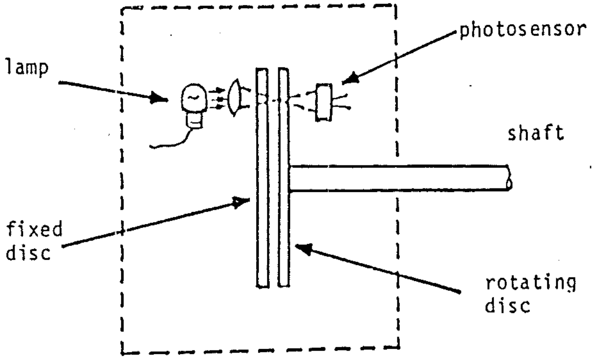


Figure (3.2.1.1) Optical Encoder

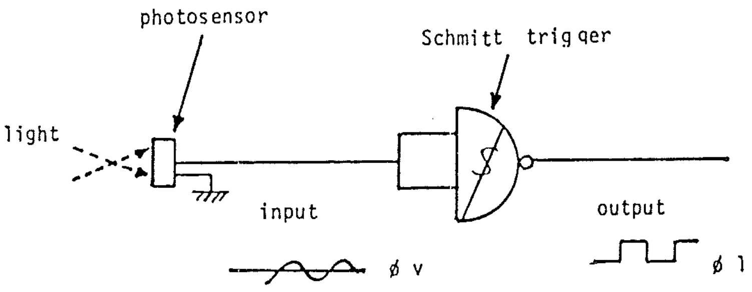
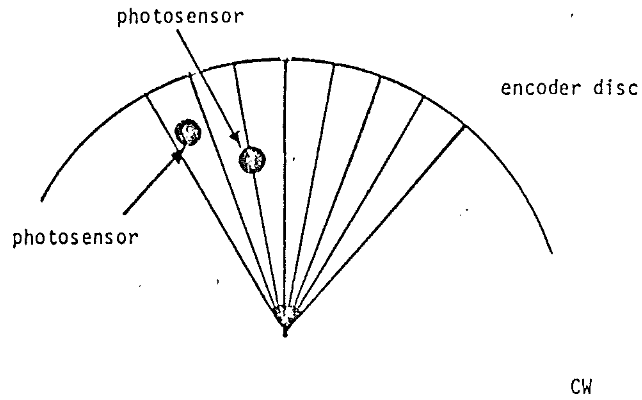
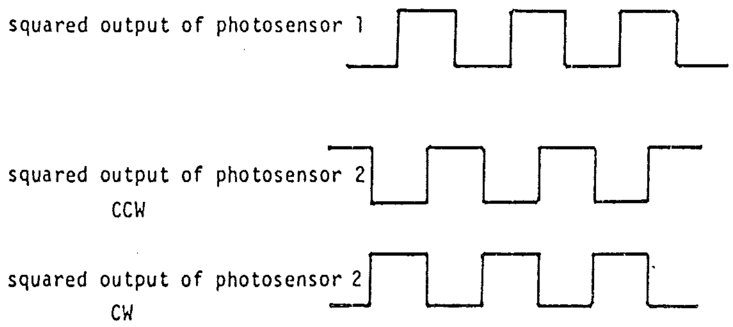


Figure (3.2.1.2) Encoder Photosensor Interface



a) arrangement of the two encoder photosensors.



b) squared output of encoder two photosensors.

Figure (3.2.1.3) Encoder Disc and Photosensor Arrangement manufacturers only make sections of the fixed grating and install it whenever there is a light emitter and photosensor assembly.

Direction of rotation of the transducer disc is obtained with the addition of a second photosensor which also allows the resolution of the encoder to be improved. The second photosensor assembly is positioned to produce a 90 degrees phase relationship (quadrature) between the two photosensors. The interfacing of the encoder and its use as a position measurement transducer is discussed appropriately in the thesis. The arrangement of the disc and photosensor assembly is shown in Figure [3.2.1.3], the output of the pair of photosensors is also illustrated for both clockwise and anticlockwise rotation, from which it can be seen that the relationship of the two pulse trains changes with a change in direction. Depending upon the method of processing the pulse trains, it is possible to obtain times two or times four resolution improvement.

3.3. PROTOTYPE MODULE DESIGNS

During the course of this research study it has been necessary to design and build a number of mechanical prototype modules. These modules have formed the basis for the evaluation system which is necessary for the design, implementation and testing of strategies for pneumatic motion control. In some cases the mechanical design of a prototype module has been influenced directly by the control strategy, and in other cases a general purpose 'mechanical unit' is designed to facilitate investigations of various forms of control strategy. Where it has been possible to purchase suitable 'off the shelf' mechanical units (designed for use in 'end-stop' modular systems) this has been done. In such cases it has been necessary to design and implement the conversion hardware for each module to incorporate the feedback transducer, lock mechanism etc..

A list of the prototype modules used in this research study is given in Table [3.3] and a description of these modules/primitives is presented within this section.

3.3.1. LUT LINEAR MODULE No.1

This prototype module was designed to investigate various pneumo-mechanical bang-bang position control strategies. It was necessary to incorporate features specific to the bang-bang techniques applied; these include a sliding carriage with hydraulic dampers, and a brake mechanism. The mechanical specification of the module is given in Table [3.3.1], and a general arrangement of the module is shown in Figure [3.3.1] and in Plate [3.3.1], which are presented in Appendix (3).

This prototype module was the first to incorporate the 'sliding carriage' which is operated with a binary control valve and lock cylinder. The sliding carriage consists of a frame which locates on the guideway of the module, onto which is mounted two hydraulic dampers (ACE [1983]) and centralising springs. This sliding carriage is located within the U-frame of the slideway, and the springs are used to centralise the carriage within the U-frame. When operated the carriage is locked onto the slideway, and as the motion of the module proceeds the dampers act against the U-frame, and induce a controlled retardation.

3.3.2. LUT ROTARY MODULE No.1

At the time of the design of the initial rotary modules, the only known available rotary vane actuators were marketed by Kinetrol, but these would provide only 90 degrees of traverse without the use of additional gearing, Table (3.3) Prototype Mechanical Modules

DESCRIPTION	MOTION	DIA (mm)	STROKE (mm)/(deg)
LUT Linear Module No 1	Translation	35	300 mm
LUT Rotary Module No 1	Rotation	-	270 deg
LUT Rotary Module No 2	Rotation	-	270 deg
M35 Arm Conversion	Translation	25	400 mm
Bosch Module Conversion	Translation	50	480 mm
LUT Rotary Module No 3	Rotation	-	270 deg
LUT Linear Module No 2	Translation	-	500 mm
Martonair M60110/600	Translation	25	600 mm
Martonair M60110/300	Translation	25	300 mm
Martonair M60120/300	Translation	50	300 mm
Martonair Gantry Conversion	Translation	50	900 mm

which was felt to be inadequate. Kinetrol rotary vane actuators have been used in the design of Placemate Robots (Pendar [1983]), where an articulated arm design means that 90 degrees of traverse about each joint is adequate. Modular robot systems tend to require a cartesian co-ordinate or cylindrical co-ordinate configuration when used in structures comprised of serial mechanical linkages.

This prototype module was designed to incorporate the necessary pneumo-mechanical control elements. A rotary module which incorporated direct drive and a rotary equivalent of the 'sliding carriage' and damper arrangement was required. The mechanical specification of the module is given in Table [3.3.2], and a simplified general arrangement is given in Figure [3.3.2] which are presented in Appendix (3).

The sliding carriage consists of a lever arm which is centralised between the two hydraulic dampers. When air at supply pressure is applied to the diaphragm it deflects and pushes the friction pad against the lever arm, the lever arm will follow the motion of the actuator shaft and operates against one of the dampers.

The vane actuator consists of an oval section aluminium housing, a fixed brass vane located in the housing with an 'O' ring seal against the shaft, and a brass vane with a perimeter 'O' ring seal arrangement. It was necessary to design and build the complete rotary vane actuator, as no proprietary commercial vane actuators were available at this time which provided in excess of 180 degrees of motion. The problem of producing a dynamic seal between the vane and chamber was never satisfactorily solved with this design, however the rotary actuator did operate at a reduced differential or standing pressure.

3.3.3. LUT ROTARY MODULE No.2

This prototype module was similar in design to the previous rotary module, where a vane actuator and a 'sliding carriage' and dampers are again incorporated. The most significant change was in the design of the vane actuator seal arrangement. The mechanical specification of the module is given in Table [3.3.3] with a general arrangement shown in Figure [3.3.3], which are presented in Appendix (3).

The rotary actuator consists of a fixed vane and a moving vane, with the seal between the vane and the housing chamber formed by the phosphor bronze insert to the vane. The seal is maintained in compression by the 'O' ring which is located between the phosphor bronze insert and the vane.

This second prototype rotary actuator was assembled, but never fully tested. The problem of maintaining a dynamic seal between the moving vane and the housing was never satisfactorily overcome. However, the problem was superseded when a proprietary range of rotary vane actuators were introduced by SMC. An SMC rotary actuator CRB80 was subsequently successfully incorporated into the design of the next prototype rotary module.

3.3.4. M35 ARM

The M35 arm is a linear module having an available stroke of 400mm. This module was originally designed as one axis of the M35 manipulator [Martonair Ltd [1984]), which was a three axis pick and place device, pneumatically actuated to mechanical 'end stops'.

Martonair produced the M35 arm conversion to a LUT specification as their first prototype module with position feedback. It was necessary to incorporate an optical incremental encoder (rack and pinion transmission system) and a wrap spring lock mechanism (located on the piston rod of the actuating cylinder). The brake is electro-pneumatically actuated with air being applied to disengage the lock spring.

The mechanical specification of the module is given in Table [3.3.4], and Plate [3.3.4] illustrates the M35 arm, which are presented in Appendix (3). This module was the first prototype design to mechanically resemble the production versions of the linear modules, and as such it was used extensively in the evaluation of new motion control strategies. The slideway, which consists of an aluminium extrusion of square section, with stainless steel surface inserts and pre-loaded roller bearings, is identical to the slideway used in the smaller production modules (M60110). The larger production modules (M60120) use a larger section aluminium extrusion with plastic bearings.

The use of rolling element bearings in the guideway resulted in consistent low friction characteristics of the slideway. When using a pneumatic drive the stiffness is relatively low due to compressibility and low operating pressures, and so frictional effects can be significant in influencing the dynamic and static performance. Evaluation results do show that the higher friction levels exhibited by slideways consisting of plain bearings combined with the friction characteristics of the actuator piston (or vane) seal can have a detrimental influence.

3.3.5. BOSCH MODULE CONVERSION

A Bosch pneumatic 'end-stop' module was purchased for conversion to a servo control unit. This linear unit was selected from the standard range of Bosch pneumatic modular handling equipment. The unit selected was a linear module, with a 480mm stroke and a 50mm diameter piston. The actuator consists of an aluminium extrusion and a through rod piston located in phospor bronze bushes.

Special design features incorporated in the module include; telescopic air feeds for other units mechanically coupled in an application, integral electrical connections for gripper/proximity sensor feedback etc., and end of stroke proximity sensors (which can be used for intialising the feedback system in the servo-control module).

3.3.6. LUT ROTARY MODULE No.3

This rotary module is shown in Plate [3.3.6]. It consists of; an SMC rotary vane actuator CRB-80 which will give 270 degrees of stroke; a Hewlett Packard encoder HEDS5000 (which has been adopted for use in the Martonair production modules); and a disc brake. The use of the SMC actuator has overcome all the problems associated with the earlier designs of rotary modules, and the use of rolling element bearings has resulted in a rotary module with good characteristics for control.

The full mechanical specification is given in Table [3.3.6], and a general arrangement of the module is shown in Figure [3.3.6], which are presented in Appendix (3). The use of a tooth belt and pulley wheels is an effective encoder transmission, and when correctly set should ensure almost zero backlash. The use of a pneumatic disc brake (GA5001) as the lock mechanism was also successful; it exhibited a rapid response and gave a good holding force.

3.3.7. LUT LINEAR MODULE No.2

This was the last prototype module completely designed and manufactured at the University. It was designed and manufactured by Mr. K. Seare as part of the LUT based design activity. The mechanical specification of the module is given in Table [3.3.7]. This module was used for the evaluation of early realtime control strategies. The slideway consists of a large diameter plain Glacier (graphite impregnated) bearing. The slideway even in the unloaded condition has a significant inertia mainly due to the intrinsic mass of the mild steel tubular guideway.

When used to evaluate the performance of a control strategy in parallel with evaluation studies carried out on the M35 arm, these two modules exhibit a complimentary diversity of operating conditions and module characteristics: from which the robustness of a control strategy to variation in mechanical primitive specifications can be determined. The general arrangement of this module is shown in Figure [3.3.7], which is presented in Appendix (3).

3.3.8. GANTRY MODULE CONVERSION

From the range of Modular handling units Martonair produced four prototype linear modules converted for servo-control, incorporating an incremental encoder and brake mechanim. Another module in this family, the Gantry/Portal module M/60400, was converted for servo-control at the University (Abdel-Gadir [1984]). The full specification of this module is given in Table [3.3.8], the details of the module are shown in Figure [3.3.8], which are presented in Appendix (3). The encoder transmission was a rack and pinion arrangement using a tooth belt laid out on one face of the slideway (held with an epoxy adhesive) and a pulley wheel pinion. The lock mechanism is a pancake piston arrangement in the main carriage, acting directly onto the slideway.

The larger extrusion from which the size (B) modules are configured is used with plain plastic bearings in the slideway. With the Gantry unit the plain bearings have caused some problems when used under servo-control, where slideway friction characteristics appear to be inconsistent. The replacement of the plain bearings with rolling element bearings has overcome these problems.

The Gantry unit is similar in design to a conventional cable cylinder. The actuating piston is located within the main extrusions, and it is connected to the slideway carriage via a steel cable running in pulley blocks at the ends of the slideway.

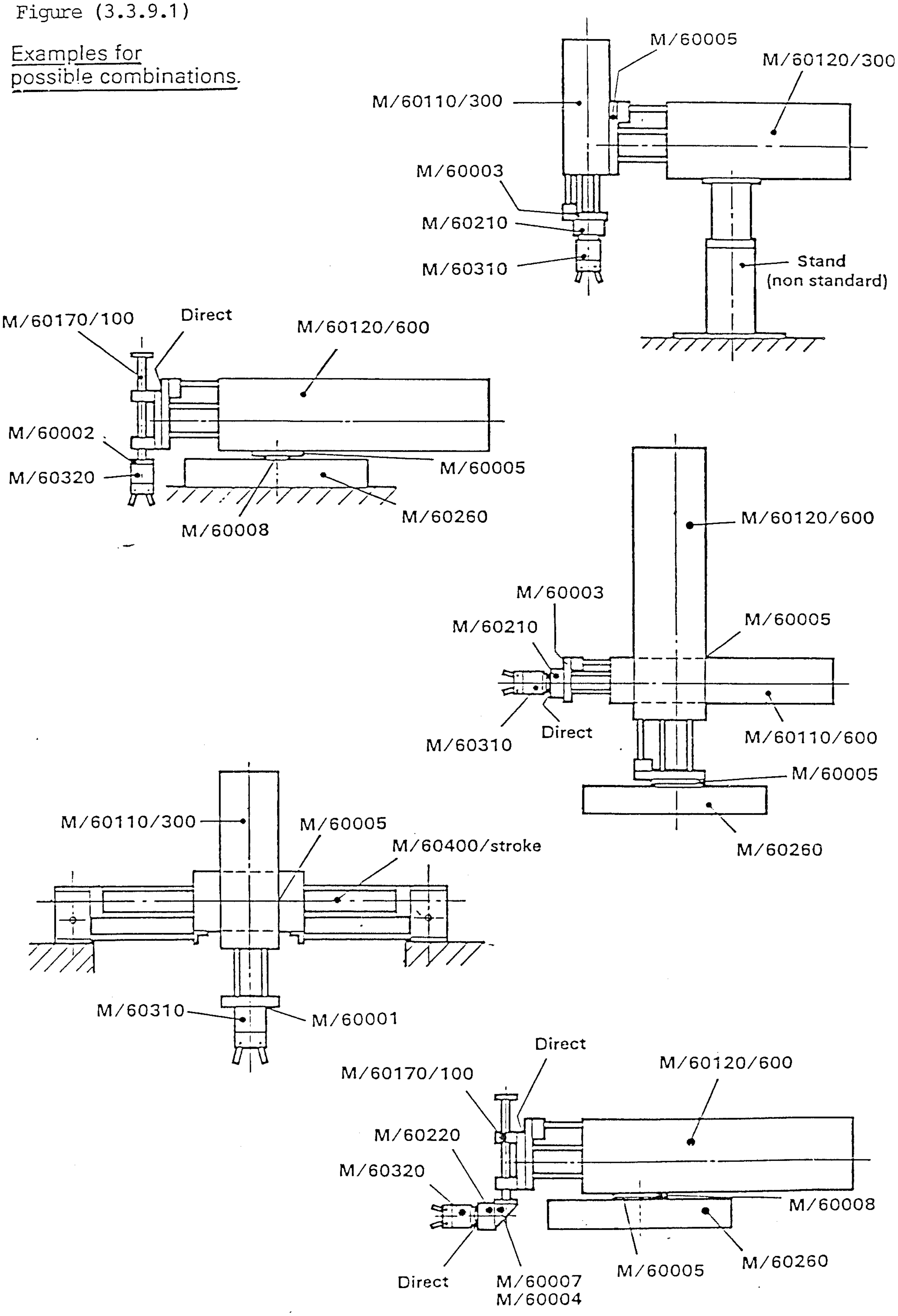
3.3.9. MARTONAIR MODULAR HANDLING UNITS

In November 1983, Martonair Ltd released onto the U.K. market a family of single degree of freedom modules which provide pneumatically actuated motion to preset mechanical end-stops. Such 'end-stop modules' can be mechanically coupled together to form low cost multi-axis manipulators. Figure [3.3.9.1] shows example combinations of these modules which can function in a similar manner to conventional multi-axis robots. A complete specification of the modules in this family is given in the Martonair publication [1984]. Alternatively these modules can be used as building elements of special automation equipment in which modules, which are not necessarily coupled together mechanically, can function in an integrated manner to perform given manufacturing operations.

Considerable flexibility is possible and user defined articulation can be

Figure (3.3.9.1)

Examples for possible combinations.



achieved with the resulting manipulator having kinematic properties which match closely the requirements of a given task. Each end-stop module is supplied with integral inductive proximity sensor feedback to allow ends of traverse to be detected. Both electrical and pneumatic supplies can be conveniently arranged for multi-axis configurations of the modules, whereby the supplies for modules further along the serial chain are accomodated within the module. For the resulting manipulator, sequence programming and control is achieved using solenoid operated binary pneumatic valves and any conventional proprietary programmable controller, which can provide a digital I/O capability.

The end-stop modules are finding wide application in many areas of automation, but the extent of their application is limited by the specific need for manipulators demonstrating servo-controlled motion.

In April 1985, Martonair Ltd launched a complimentary range of pneumatically actuated servo-controlled modules, thus considerably extending the versatility and flexibility of their range of modular handling units (Morgan [1985]). The design of this pneumatic position control system was a result of the collaborative research project with Loughborough University (Weston et al [1984]), whereby the specification and implementation of the complete motion control system were the result of this research study.

The linear end-stop modules require the addition of a feedback transducer (Hewlett Packard HEDS-5000 optical encoder) and transmission (tooth belt and pulley wheel), to provide a position control capability when used in conjunction with a 'proportional control valve' and 'single axis controller' (Morgan [1985]). The brake mechanism can also be included if the application requires a locking device (which takes the form of a pair of pneumatic disc brakes acting directly on the slide). These additional components are included in the module as shown in Figure [3.3.9.2].

The addition of a suitable displacement transducer to any pneumatic actuator, when used with the 'proportional control valve'(SPGB 18913) and 'single axis controller', will allow position control to be achieved. In this way the potential application of pneumatic position control is far wider than just Modular Handling Robot systems. Martonair are actively developing special actuators, (M8000 ISO Cylinders and M45000 rodless cylinders), with integral displacement transducers as "Intelligent Actuators" for position control applications. However any special machinery design could incorporate pneumatic motion control using standard actuators and an appropriate feedback transducer.

The standard range of Martonairs' Modular Handling units include;

(i) linear modules M/60110/300 - Size A
M/60110/600 - Size A
M/60120/300 - Size B
M/60120/600 - Size B

which come in two sizes and stroke variants, all of which can have servo-control,

(ii) short stroke modules
M/60160 - Size A
M/60170 - Size B

which are linear modules in two sizes, for providing end effector motion between mechanical stops,

(iii)rotary modules M/60210 - Size A
M/60220 - Size B

which are available in two sizes, for providing end effector wrist

Table (3.3.9)

MODULE DATA TABLE	LINEAR MODULE M60110/600
Actuator	Aluminium extrusion, Ø25 mm
Stroke	600 mm
Slideway	Aluminium extrusion with s/s face plates and rolling element bearings
Transducer	Hewlett Packard HEDS-5000 500 p/rev 0.028 mm/pulse
Transducer Transmission	Fixed tooth belt and pulley wheel
Brake mechanism	Direct acting pancake cylinders (2) with aluminium inserts
Carriage frame	Aluminium block
Additional features	Telescopic air feeds and integral electrical cable and connectors
Transmission	Direct drive

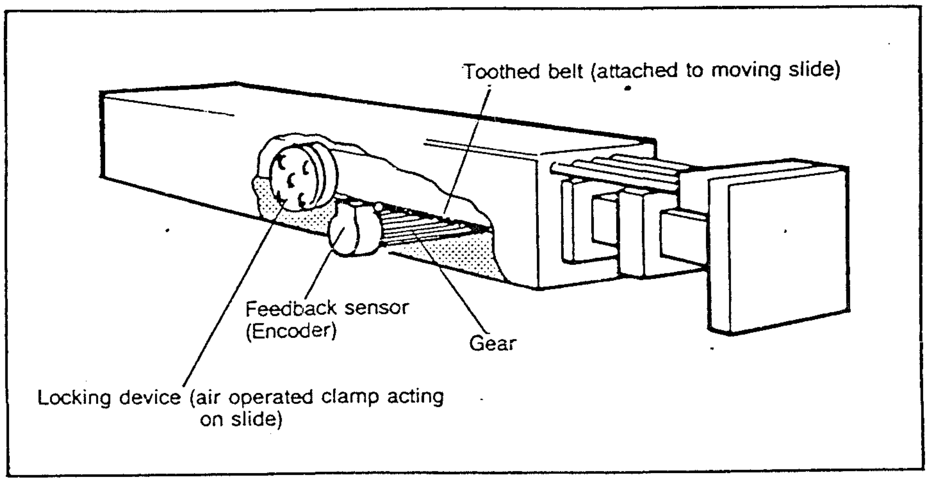


Figure (3.3.9.2) Martonair Modular Handling Unit - Servo Controlled Version motions between mechanical stops,

(iv) gripper modules M/60310 - Size A

M/60320 - Size B

which are available in two sizes, and are designed to locate and action finger inserts (application dependent),

(v) a portal module M/60400

this is a linear gantry module, the stroke length of which is variable, and

(vi) rotary module M/60260

this is a rotary base module, with rack and pinion actuation.

Other special design features of these modular handling units include;

(a) interface adaptor/mounting plates,

(b) integral telescopic air feed pipes,

(c) integral electronic signal connections,

(d) end of stroke inductive proximity detectors,

(e) hydraulic dampers at extremes of stroke with 'end-stop modules, and

(f) standard couplings and fitting.

Many of these design features are incorporated to simplify the integration and installation at the mechanical "systems engineering" phase when configuring modular handling systems. They allow easy configuration/re-configuration of the mechanical elements of the system.

The Martonair linear module M60110/600 in its prototype form was utilised for many of the latter control strategy performance evaluations. The mechanical specifications of this module are given in Table [3.3.9].

3.4. MICROPROCESSOR DEVELOPMENT TARGET SYSTEM

The selection of the microprocessor family for use in realtime control applications is important. In this research study the Texas Instruments (Texas Instruments [1976], [1977], [1980]) TMS 9900 processor was chosen. This decision was determined by the available hardware and software development tools and expertise at the outset of the research programme and its suitability for realtime control applications.

The TMS 9900 microprocessor is a single chip 16-bit Central Processing Unit (CPU), produced using N-channel silicon gate MOS technology. Target systems were configured using TMS 9900 based Single Board Computers (SBC), memory expansion boards, the T-bus backplane system, and additional proprietary and special purpose designed electronic interface boards. A detailed description of the Target System configured is given in Moore [1986].

3.4.1. TMS 9900 MICROPROCESSOR

The TMS 9900 microprocessor is a single chip 16-bit CPU capable of accessing 32K 16-bit words of memory and has a 16-bit wide data bus. The processor supports a total of 69 instructions including unsigned hardware, multiply and divide. There are 16 hardwired prioritised interrupts as well as three hardware registers referred to as the program counter (PC), the work space pointer (WP), and the status register (SR). It has a memory to memory architecture and a context switching feature (Texas Instruments [1976], [1977]).

3.4.2. TM 990 (100M/101M) SINGLE BOARD MICROCOMPUTER

For target system development work the TM990/100M or the TM990/101M SBC were utilised. The 101M SBC has more on-board memory than the 100M SBC and also has an additional RS232 serial port. These SBC's consist of the following elements;

(a) TMS 9900 Microprocessor

(b) TMS 9901 Programmable Systems Interface

(c) TMS 9902 Assynchronous Communications Control

(d) TMS 9904 Real-time Clock

(e) 2k words of 16-bit RAM

(f) 1k words of 16-bit EPROM - expandable to 2k or 4k words using on-board jumpers and appropriate EPROMS (TMS 2716).

A full description of the TM 990 (100M/101M) SBC's and their functions is given in Texas Instruments [1976].

3.4.3. ANALOG INPUT/OUTPUT RTI-1241

A capability to interface the Target System to analog I/O is provided by the Analog Devices [1980] Board RTI-1241. The RTI-1241 provides the capability to display, in realtime, control system data such as position, velocity, and command signal. The RTI-1241 board interfaces via the T-bus backplane, the board being configured as memory mapped I/O, where all control and data transfer operations are accomplished by writing to, or reading from, one or other of eight words located in the processors' memory map. Each word having a pre-assigned function, the eight words comprise the memory map of the RTI-1241.

3.4.4. CRU ADDRESS DECODE AND INPUT/OUTPUT EXPANSION

For each target development system it was necessary to provide additional input/output lines from the processor to provide the necessary interfaces to items such as solenoid drive cards, encoder cards, and proportional amplifier cards. The standard processor boards used have previously been described, and these had been designed and built commercially. The requirement for processor I/O expansion has led to the development of LUT designed and manufactured 'CRU address decode' and '16 line input/output' cards (these originally being designed by Mr G Charles).

The 'CRU address decode' card decodes up to eight blocks of I/O, each block having a boundary of eight (byte) bits or sixteen (word) bits, dependant upon the control line configuration. A block of eight dip switches, when set, determines the address lines that are decoded to produce the CRU Base address of the decode board. This enables the address decode card to reside at any undefined area in the processors CRU map, and this CRU Base address determines the CRU location of the subsequent eight blocks of I/O.

Once the CRU expansion has been decoded it is necessary to provide the blocks of I/O lines via the CRU input/output line expansion card. The card provides sixteen buffered input lines and sixteen buffered output lines which operate at TTL levels. The card is designed to allow individual I/O lines to be written, or read from, namely the I/O lines are bi-directional, but they can be used in a unidirectional mode as dedicated input or output. By designing interface cards using the CRU facility of the Texas processor the five specialised single bit and multi-bit manipulation instructions in the instruction set can be utilised. These make individual and multi-bit I/O control simple and efficient, which is one of the powerful features of the 9900 processor architecture.

3.4.5. SOFTWARE DEVELOPMENT FACILITIES

The capability of any microcomputer is dependant upon the software development facilities that support the processor. The choice of a suitable language for software development is primarily determined by the functional requirements of the process or task. For real time control systems to be designed on a sample-data basis, the digital algorithms must execute as fast as possible, typically one millisecond duration, thus the executable machine code must take a minimum form. Hence the execution speed of the algorithm is a function of the processor architecture and size of the assembled code generated. A comparison of language execution speeds for the Texas TMS 9900 processor is given by Salihi A. [1983]. From the languages supported within the department the only feasible choice for time critical real time control was Assembler. Interpretive languages by definition would have been too slow, and the various compiled forms of Pascal (Texas Instruments [1980]) could not match the execution speeds of Assembler. The Pascal was also limited by the requirement of a minimum operating kernel of about 16k words, which beyond the development phase could impose significant overheads upon the single axis motion controller design at prototype stage and in any commercial implementation.

3.4.5.1. TIBUG INTERACTIVE DEBUG MONITOR

TIBUG debug monitor is supplied by Texas Instruments [1980] to provide an interactive interface between the microcomputer and the user via any RS 232 compatible terminal. It provides facilities to load, execute, and trace the assembled code within the RAM environment of the microcomputer. On the TM 990/100M or 101M board, the TIBUG program usually occupies EPROM memory space from memory address >0080 through to memory address >07FE. It uses four workspaces of 40 words of RAM memory, of which part is also used for restart vectors which initialise the monitor following single step execution of instructions.

3.4.5.2. TERMINAL EXECUTIVE DEVELOPMENT SYSTEM (TXDS)

The Terminal Executive Development System (TXDS) is configured around the Texas Instruments' FS 990 microprocessor development system which consists of:

(i) 990/4 Computer CPU circuit cards

(ii) 48K bytes of dynamic RAM

(iii) 1024 bytes of ROM supplying a floppy disc loader, CPU self tests, and front panel utilities

(iv) dual floppy disc chasis and associated controller

(v) a 911 video display terminal

As an option, a TMS 9900/9980 emulator and buffer module, a 990 trace module, and a PROM programming module can also be added to the system to improve its versatility, all of which were available for this research programme.

The TXDS provides a comprehensive range of program development and testings facilities for assembler programming of the TMS 9900 processor. The TXDS facilities are fully documented in Texas Instruments [1977] and [1980].

3.5. PROCESSING ENCODER FEEDBACK SIGNALS

The rotary optical incremental encoder was selected as the position transducer for use in this study. The incremental encoder produces two digital outputs in the form of pulse streams when the shaft is rotating. From these pulses, the direction of rotation of the shaft and its position can be determined by using suitable signal interrogation, counting and data storage techniques.

The two different methods of achieving this can be summarised as follows:

(a) by using suitable software and utilising the microprocessor to perform the counting and storage of the values in registers in memory.

(b) by using suitable dedicated logic level electronic hardware capable of detecting the direction of rotation of the encoder shaft, count the pulses and store the right number of pulses in a buffer from which they can be accessed by the microprocessor when required.

3.5.1. COUNTING IN SOFTWARE

Counting in software can be achieved using interrupt controlled I/O to the microcomputer and a suitable interrupt service routine, where the processor can be made to perform the necessary counting and storage of encoder pulses. The output lines of the encoder can be directly interfaced to the processor via the TMS 9901 programmable system interface. This is achieved by connecting one encoder output to an interrupt port, and the second encoder output to an input port of the TMS 9901 interface (see Figure [3.5.1.1]).

The processor continues to execute the main program until an interrupt occurs. The processor will then complete execution of the current instruction, and then suspend execution of the main program and execute the associated interrupt service routine. The functional requirement for such an interrupt service routine (ISR) is shown in Figure [3.5.1.2]. Counting in software and its relative merits is discussed in more detail in Moore

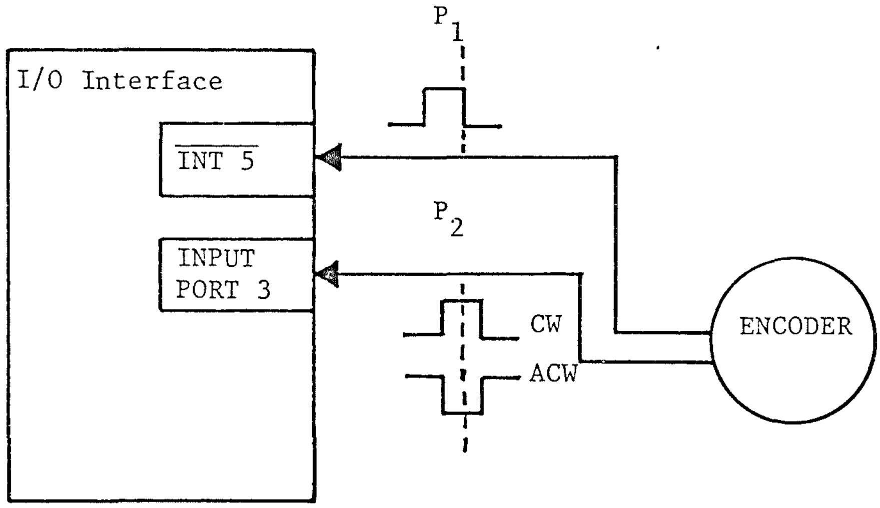


Figure (3.5.1.1) Direct Interface between Encoder and Computer

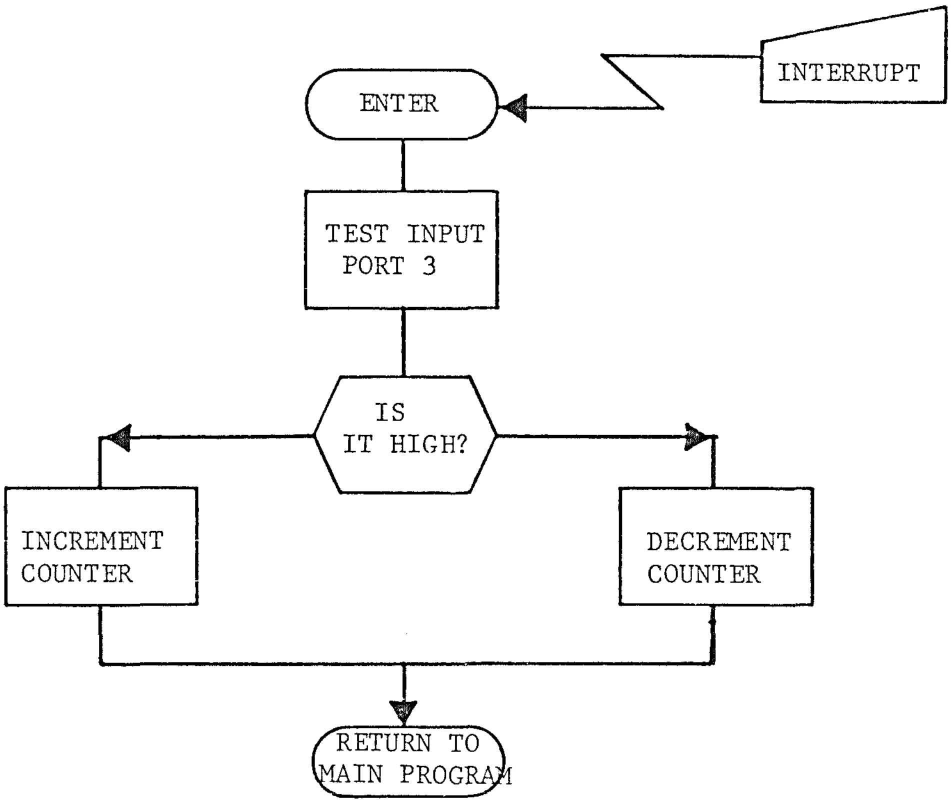


Figure (3.5.1.2) Flowchart of Count Interrupt Service Routine

[1986].

3.5.2. COUNTING IN HARDWARE

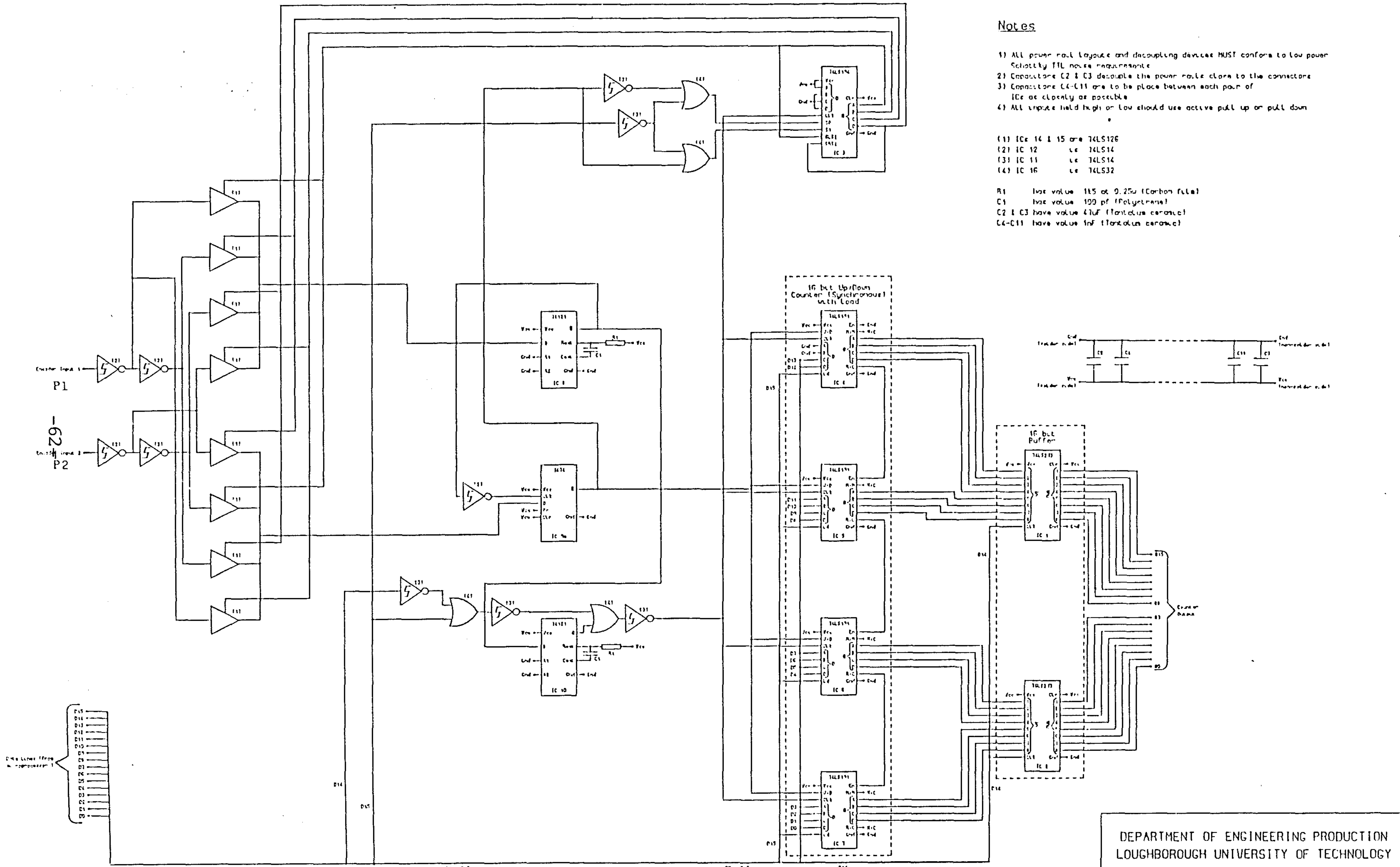
An alternative to the software counting method, is to use a suitable dedicated logic level electronic hardware interface to count and store the pulses. Counting using hardware techniques is more expensive than the software equivalent, but has the significant advantage of releasing the burden on the processor, which can be crucial in time critical real time control applications. Methods of counting in hardware and associated problems are discussed by Ditchfield [1984].

3.5.3. ENCODER INTERFACE BOARD

The encoder interface circuit board for processing quadrative pulse trains from position feedback transducers, used in this research study, is briefly described here. The full specification of the design and its operational characteristics are given by Ditchfield [1984].

The circuit diagram of the encoder board is given in Figure [3.5.3.1] and additionally, a schematic block diagram of the board is provided in Figure [3.5.3.2]. The encoder board provides a 16-bit bi-directional counter that interfaces to the microprocessor serially via the CRU of the TMS 9901.

Sequential edge enabling is achieved by the use of a ring counter and two sets of four tri-state buffers. The first set determine when to alter the counter, the second set determine wether to increment or decrement the 16-bit counter. The ring counter will enable only one pulse edge at any instant ensuring integrity. The counter is clocked by the leading and trailing edges of both pulse streams (P1 and P2), thus providing an increase in resolution of the encoder by a factor of four.



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LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY

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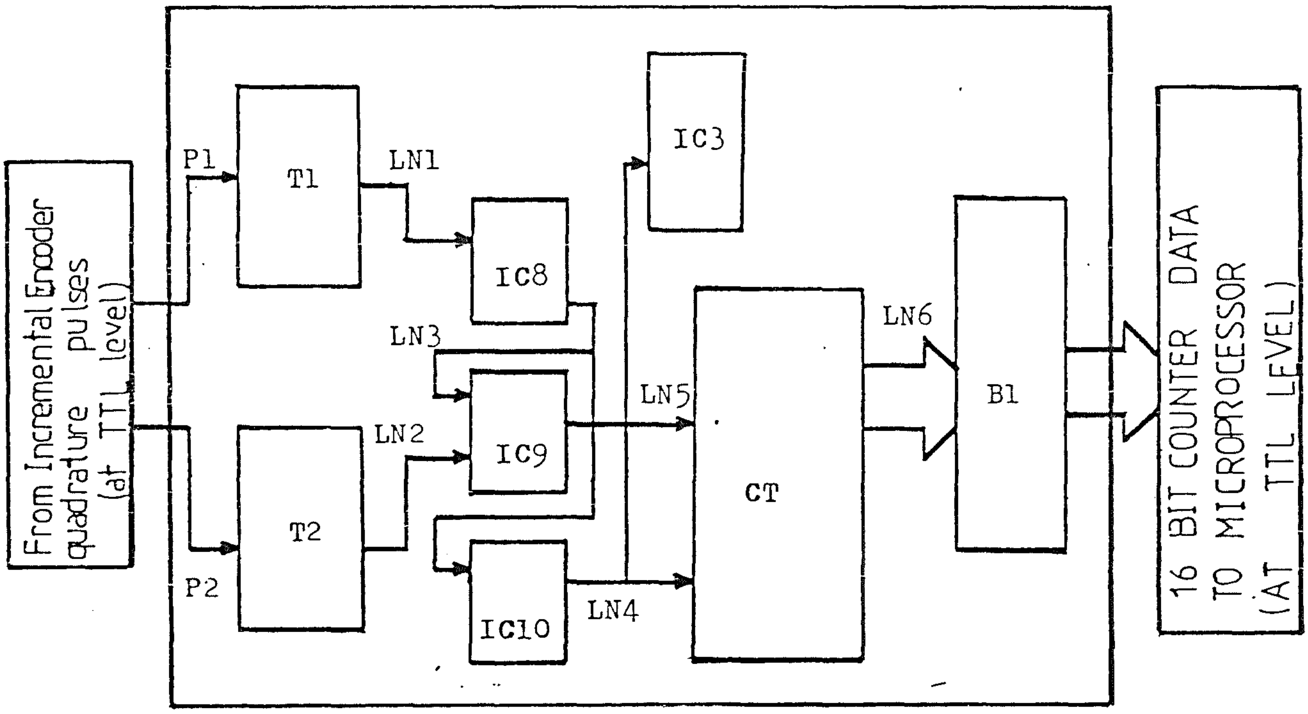
Figure (3.5.3.1) Encoder Interface Circuit



Line 1	This line provides a rising edge from section T1 of the edge enabling circuit to the first stage of the pulse delay circuit, IC8.
Line 2	This line provides a direction indication from section T2 of the edge enable circuit to the direction latch, IC9.
Line 3	This line provides a rising edge to the direction latch,IC9.
Line 4	This line transmits the rising edge coming indirectly from the encoder, via section T1 of the encoder enable circuit and pulse delay circuit (IC8,IC10), to the counter (CT).
Line 5	This line provides a direction signal to the 16 bit counter (CT) from the direction latch, IC9. High = count down, low = count up.
Line 6	These sixteen lines transmit the binary value of the sixteen bit counter (CT) to the buffer (B1).

Figure (3.5.3.2) Schematic Diagram of Encoder Interface Board

During the operation of the board, a clock pulse is transmitted to the ring counter (IC3) at each increment/decrement of the up/down counter. This clock shifts the ring counter one bit to the left or right, depending on the direction of the count. Subsequently, the next edge in the sequence is enabled along with its corresponding direction signal from the opposite pulse stream.

Interfacing is through the CRU to the processor. This requires additional circuitry in the form of an address decode board and an I/O board.

Before the board can be used to count the pulse streams from the encoder, an initialisation process must be followed. This consists of four stages:

(1) Reset the ring counter

(2) Enable and clear the 16-bit counter

(3) Enable and clear the buffer

(4) Disenable the buffer

The initialisation process is achieved by setting the relevant input ports to the board to the required state in software.

3.5.4. TEXAS INSTRUMENTS ENCODER INTEGRATED CIRCUIT

A functionally equivalent encoder interface to the system described previously has become commercially available during this research project which uses a Texas Instruments SN74LS2000 Direction Discriminator integrated circuit (Texas Instruments [1985]) and a number of discrete components. A functional circuit diagram is shown in Figure [3.5.4.1]. This circuit has now been adopted in the commercial version of the single axis controller.

Figure [3.5.4.2] shows the functional elements of the Direction Discriminator I.C., these include a sixteen-bit up-down counter, an

TYPE SN74LS2000
(DIRECTION DISCRIMINATOR)

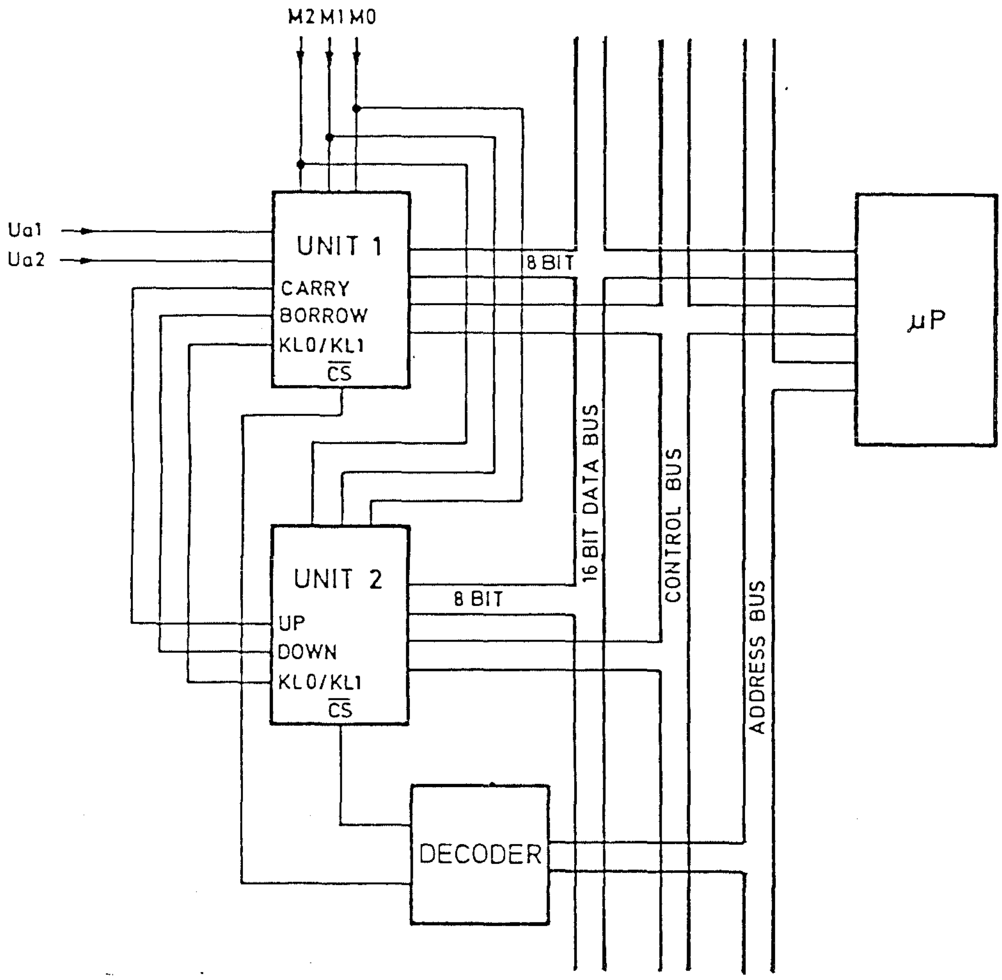


Figure (3.5.4.1) System Application

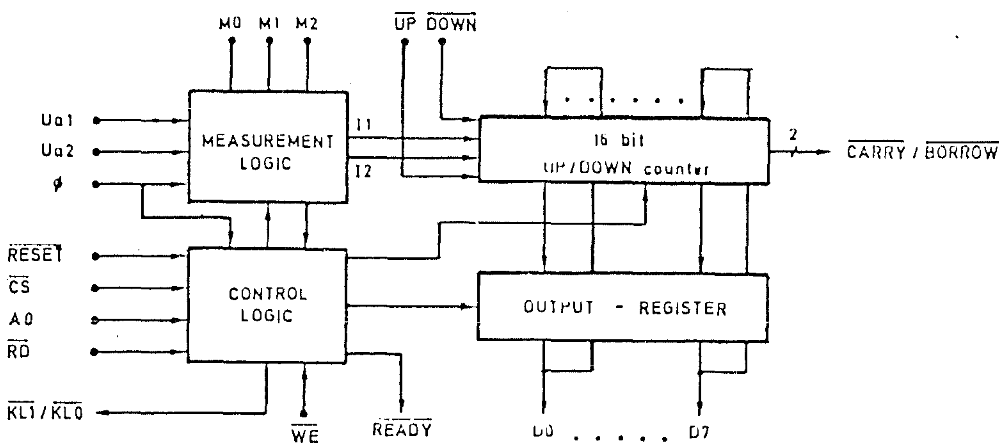


Figure (3.5.4.2) Functional Elements

eight-bit output register, measurement logic providing quadruple data from two quadrature inputs, and the necessary control logic. The control logic is the interface between the control signals of the processor and the SN74LS2000. The output register, which operates as a one word memory for the sixteen-bit evaluated data, is read by the processor in two bytes.

By appropriate selection of the input lines to the measurement logic M0,M1,&M2, the SN74LS2000 can be set in one of three basic modes. Direction discriminator mode, where it functions as an up/down counter for any incremental opto-electrical transducer. It is used in this mode to interface the position feedback transducer to the 'single axis controller'. As a direction discriminator it can be set up to count in single pulse, double pulse, or quadruple pulse state. The quadruple pulse state is used for the 'single axis controller' application so that the SN74LS2000 functions as an exact equivalent to the earlier design of encoder board.

The alternative modes for the SN74LS2000 are pulse width measurement and frequency measurement. In the pulse width/time measurement mode, the input pulse width is measured by dividing the input pulse during its high state into a number of counter clock pulse. The longer the input pulse high (mark) state, the greater the value loaded into the counter.

In the frequency measurement mode the input line is used as a frequency signal, the input pulses being counted in the counter register and being transferred to the output register at a set interval.

The data sheet on the SN74LS2000 provides a full operational specification and description (Texas Instruments [1985]).

3.6. INTERFACE TO THE DIGITALLY OPERATED PROPORTIONAL SERVO-VALVE

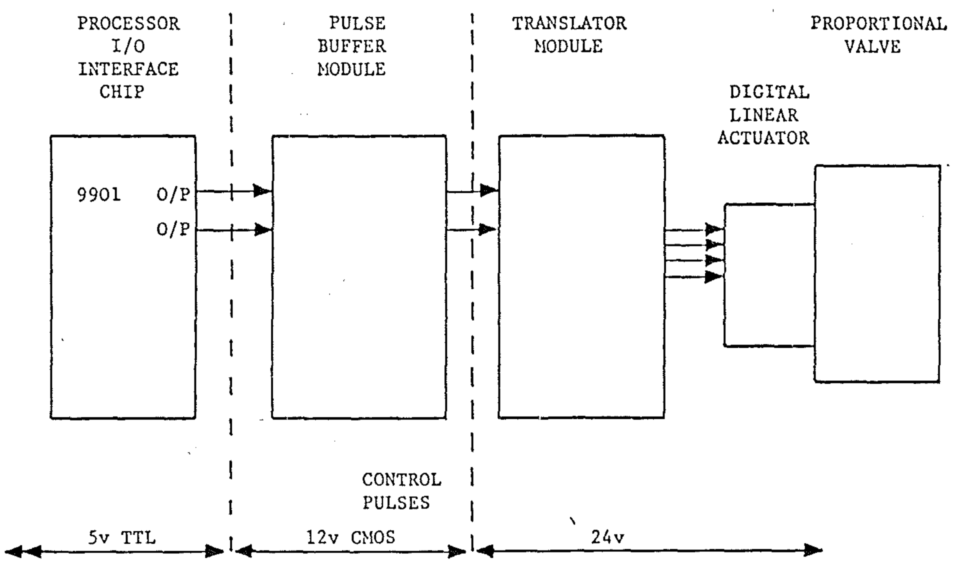
The proportional servo-valve developed by Martonair was evaluated using a 'digital linear actuator' to replace the proportional solenoid. This allowed a purely digital interface between the microcomputer and the actuation system. By extending the operational range of the valve manipulation system beyond 2mm, of which the proportional solenoid is capable, the full flow capacity of the valve spool and sleeve can be achieved. This has advantages when trying to achieve high acceleratons and velocities of actuators, particularly where large control volumes are encountered. A digital linear actuator when used for spool manipulation allows discrete linear increments of displacement to be obtained (the digital actuator is used in an open loop mode).

The interface circuit to the stepper motor is shown schematically in Figure [3.6.1]. When energised with a standard four-phase sequence (with each phase at 24v dc) bi-directional control of valve spool is obtained via the leadscrew and return spring with increments of 0.001"(0.0254mm) or 0.0005"(0.0127mm). The phase control of the stepper motor is obtained with the 'translator', the control pulses that drive the translator arrive via the 'pulse buffer module' which combines acceleration and deceleration pulse ramping techniques. The only interface to the processor necessary being two TTL level output ports which provide a direction signal and a set number of control pulses.

3.6.1. PULSE BUFFER MODULE (EM211)

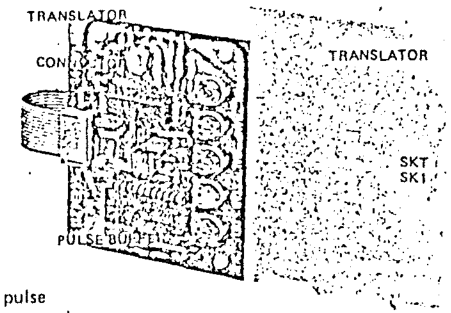
The 'pulse buffer module' manufactured by McLennan (EM211), is used to interface between the processor output ports and the translator. This enables step control pulses to be fed to the stepper motor at fixed frequencies up to the motors maximum step rate. The EM211 incorporates a ramping clock to provide automatic ramping of motor step rates and a 16K

Figure (3.6.1)

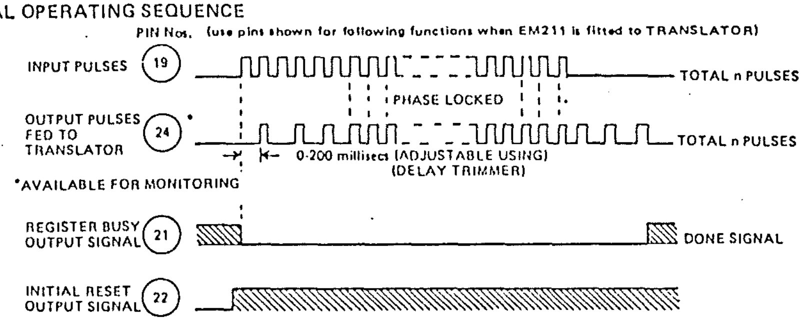


Interface to Proportional Valve With Digital Linear Actuator

The EM211 Pulse Buffer is designed for mounting on the EM160 Series translator modules. The unit enables step control pulses derived from a micro-processor or mini-computer to be fed to the stepper motor system at fixed frequencies up to the motor's maximum slew rate. The EM211 incorporates a ramping clock to provide automatic ramping of motor step rates and a 16K pulse buffer which stores pulses during the acceleration/deceleration programmes thereby ensuring that the motor moves one step for each pulse generated by the control source. On completion of the acceleration ramp which can be preset to suit individual applications the EM211 will phase lock the pulse frequency fed to the translator to the control source thereby making the system ideal for accurate crystal locked speed controlled drives. On completion of each sequence of incoming pulses the pulse buffer provides a programmed deceleration ramp to ensure that the exact number of steps is achieved thereby greatly simplifying high speed positioning control software.

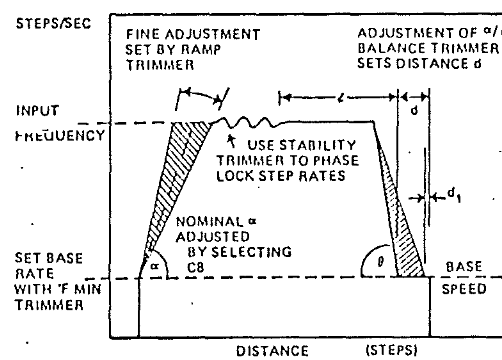


TYPICAL OPERATING SEQUENCE



TYPICAL MOVEMENT PROGRAMME

Below is shown the action of the trimmer controls.
Adjust stability trimmer to obtain phase locked rates as shown at 'ℓ'. Adjust balance to obtain approach as shown (d_1).



SPECIFICATION
INPUT/OUTPUT SIGNALS
LOW STATE : 0-1V } CMOS COMPATIBLE
HIGH STAGE : 10-12V }
MAX INPUT FREQUENCY : 25 KHz
BASE SPEED (ADJUSTABLE) : 150-1000Hz
RAMP ADJUSTMENT SET BY CAPACITOR C8

TYPICAL VALUES C8	RAMP RATES
0.1 micro F	50,000 steps/sec²
0.22 micro F	20,000 steps/sec²
0.47 micro F	10,000 steps/sec²
1.0 micro F	5,000 steps/sec² *
2.2 micro F	2,000 steps/sec²
* AS SUPPLIED	

Figure (3.6.1.1) Pulse Buffer Module

pulse buffer which stores pulses during the acceleration/deceleration phases, thereby ensuring that the motor moves one step for each control pulse generated. On completion of the aceleration ramp, which can be tuned to the system with trimmer controls, the EM211 will phase lock the pulse frequency fed to the translator to the control source. On completion of each sequence of control pulses the pulse buffer also provides a deceleration ramp to ensure pulse integrity and therefore simplifying high speed positioning control software.

The use of a pulse buffer module also enables the input pulse frequency to the motor to be increased by controlling the acceleration phases. The input and output control pulse relationship from the EM211 buffer module is shown in Figure [3.6.1.1]. This shows how adjustment to the ramp rates can be made and gives the specification of the device. The circuit diagram for the EM211 is shown in Figure [3.6.1.2]. The pulse input is equivalent to the number of steps of the motor that are required to produce a given spool displacement, these being transmitted to the buffer module from the TMS 9901 with the interface circuit, as shown in Figure [3.6.1.3].

3.6.2. TRANSLATOR (EM162)

The EM162 translator was selected as a suitable interface for the L92411 digital linear actuator. It provides a mode 2 type interface to the motor and also allows half step mode control, which then allows a resolution on spool displacement of 0.0005"(0.0127mm) to be achieved. A commercial implementation of the digitally operated valve would use a L92121 actuator which could be controlled by a single IC driver (ie type SAA 1027) which would result in a significant reduction in interface costs as compared to this prototype interface.

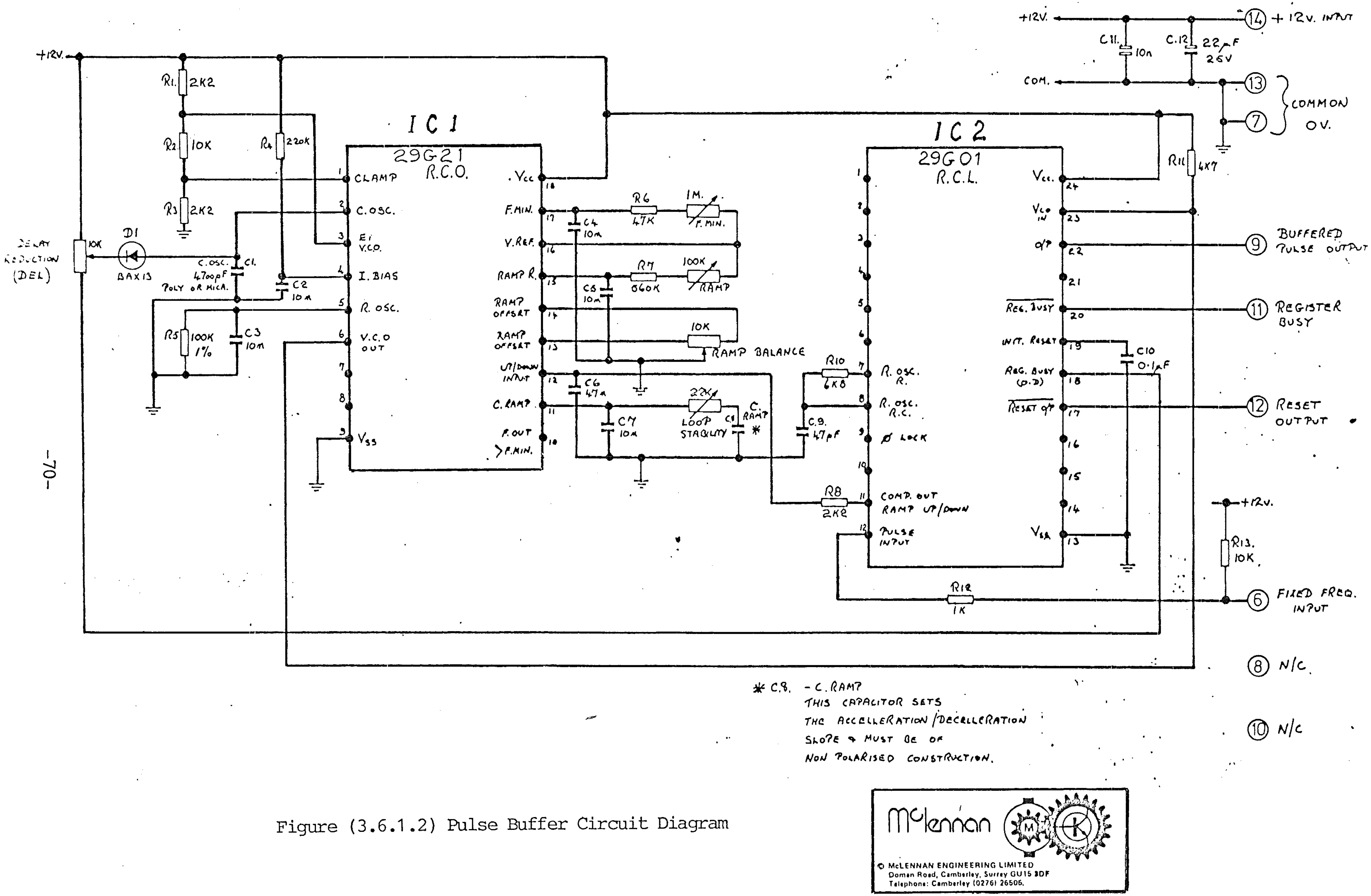


Figure (3.6.1.2) Pulse Buffer Circuit Diagram

Figure (3.6.1.3) Digital Linear Actuator - Microprocessor Interface

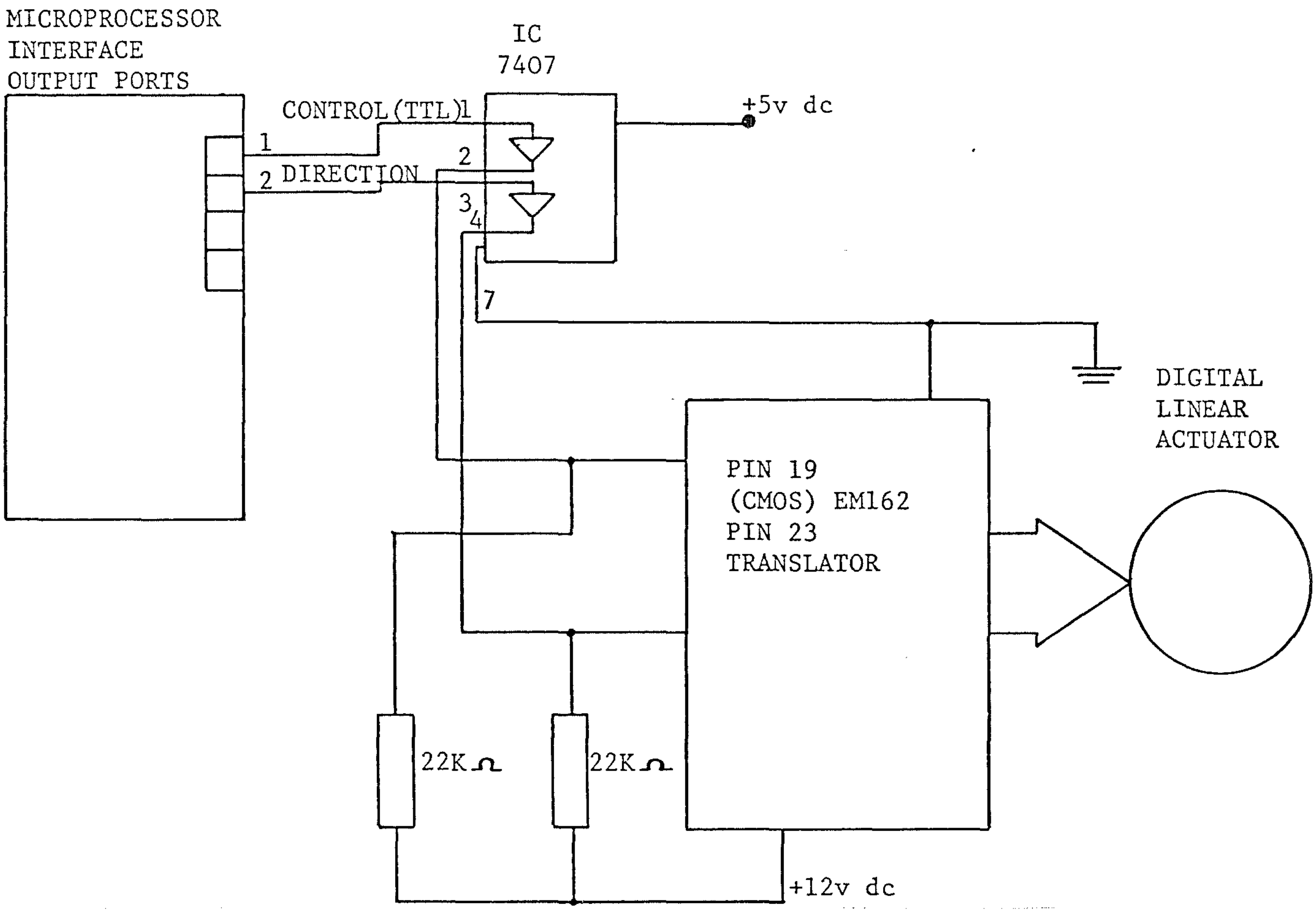
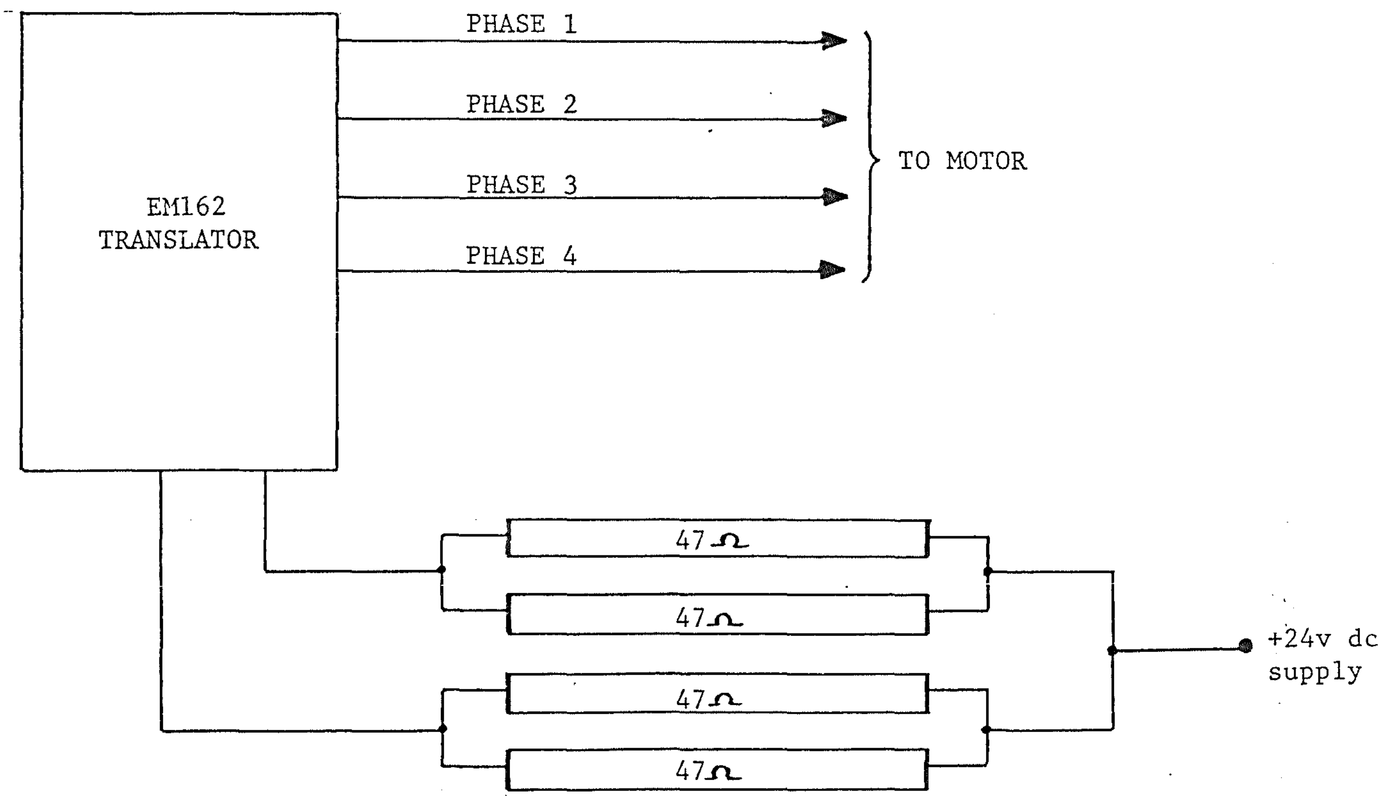


Figure (3.6.2.2) Digital Linear Actuator Interface Circuit



The circuit diagram of the EM162 translator is shown in Figure [3.6.2.1], and the interface circuit to the stepper motor is shown in Figure [3.6.2.2].

3.7. INTERFACE TO THE SOLENOID OPERATED PROPORTIONAL SERVO-VALVE

The Martonair proportional servo-valve used during this project was supplied without proprietary interface electronics. For the valve in its conventional form with analog operation, namely an MSM type GRF035 solenoid, it was necessary to provide a DAC and linear power amplifier. The DAC converts the digital 'command word' into a low power analog 'command signal', which is then amplified by the 'linear power amplifier' to drive the solenoid. The solenoid is a single acting device which provides a force in proportion to the applied current. It is rated to function at 24v dc with a maximum current of 1 ampere. The 'valve interface' is shown schematically in Figure [3.7.1].

The interface between the microprocessor consists of a 10 bit DAC, dither source, and a linear power amplifier. Several DAC circuit designs have been used for connection to the amplifier. One of these is illustrated in Figure [3.7.2], which uses a PMI DAC 1010.

The linear amplifier circuit diagram is shown in Figure [3.7.3], where the necessary inputs and power rails are also labelled. The only two control inputs necessary come from the DAC with the dither source which is generated off board and included at the summing junction in the circuit, as shown at point (A). A suggested design for the dither source is shown in Figure [3.7.4], a signal generator was used on the test rig. The circuit is a fairly conventional linear amplifier design, where the final amplification stage is provided by a darlington pair consisting of

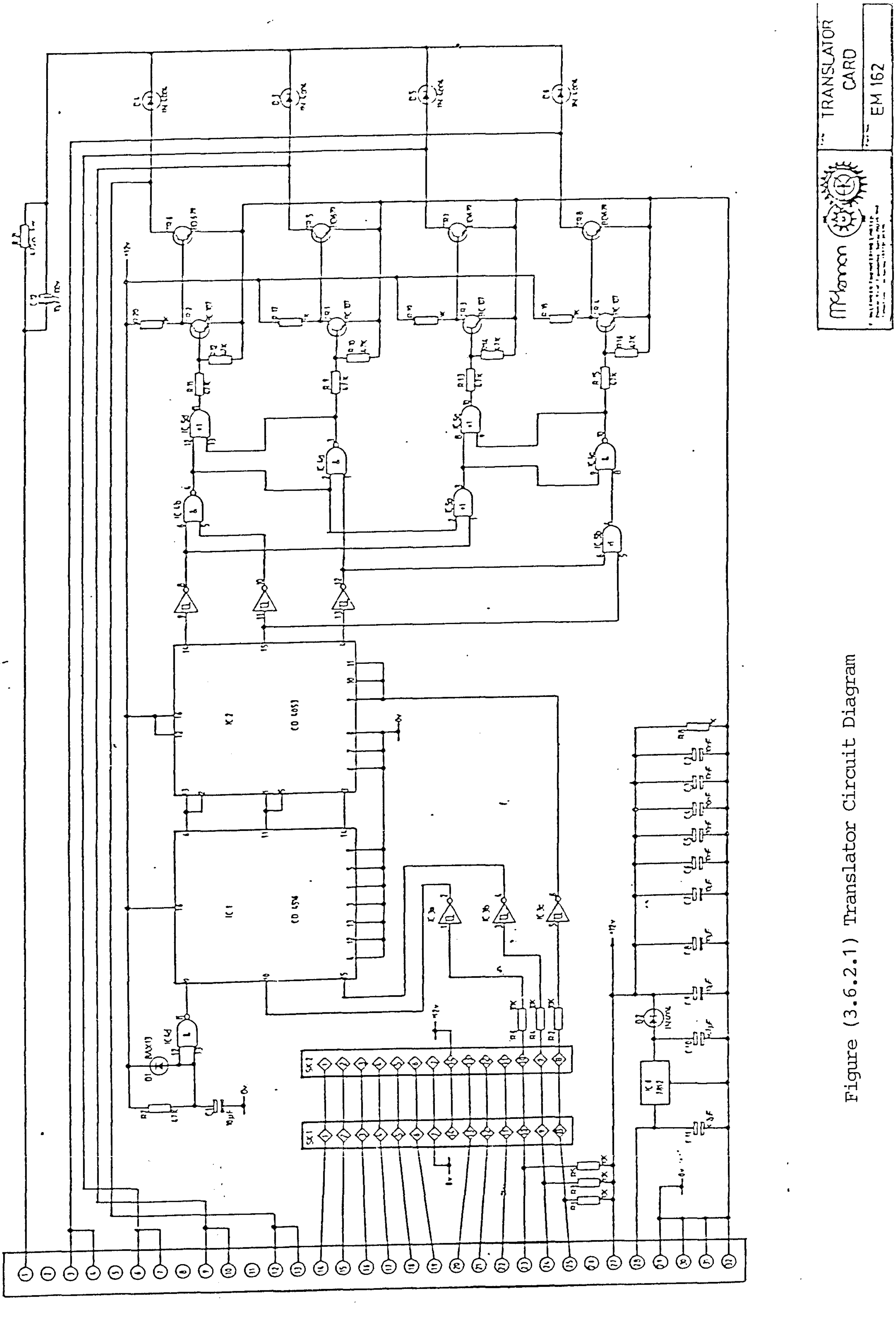


Figure (3.6.2.1) Translator Circuit Diagram

Figure (3.7.1) Proportional Valve Interface with Solenoid Actuation

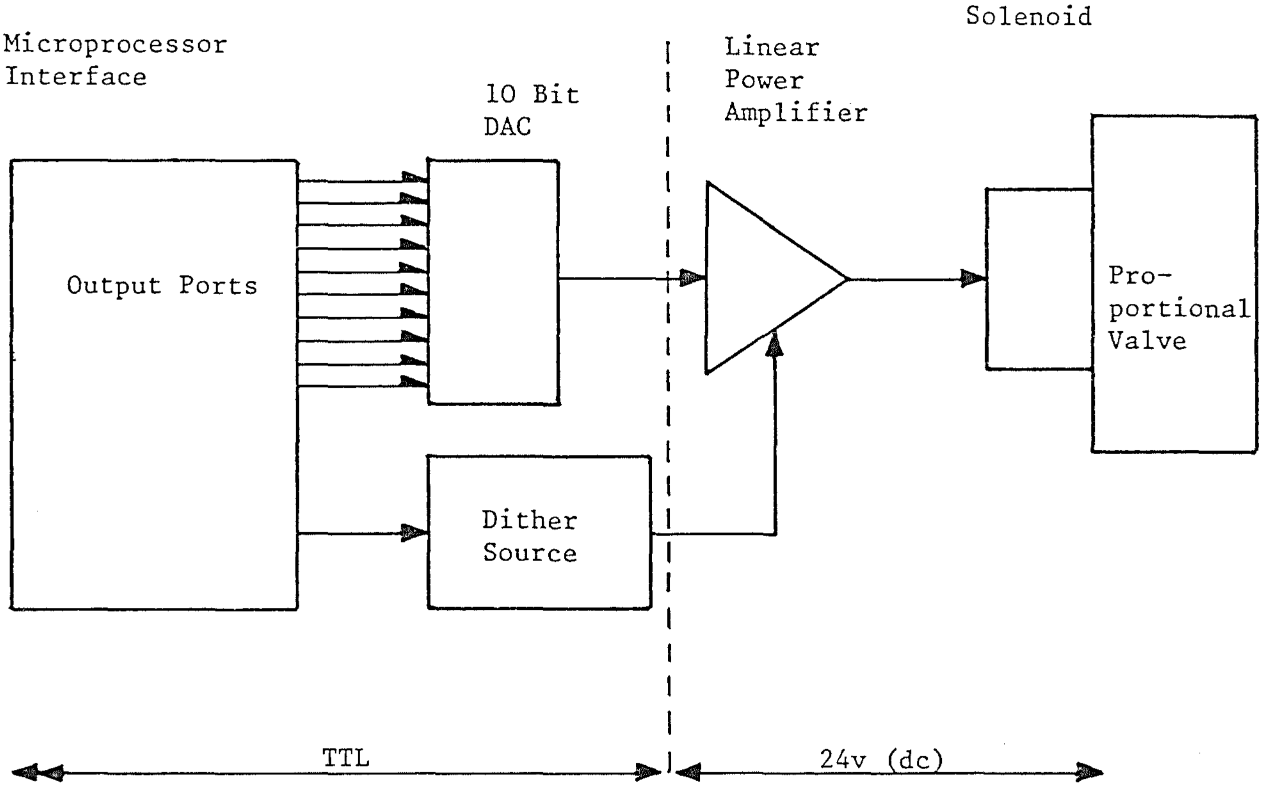
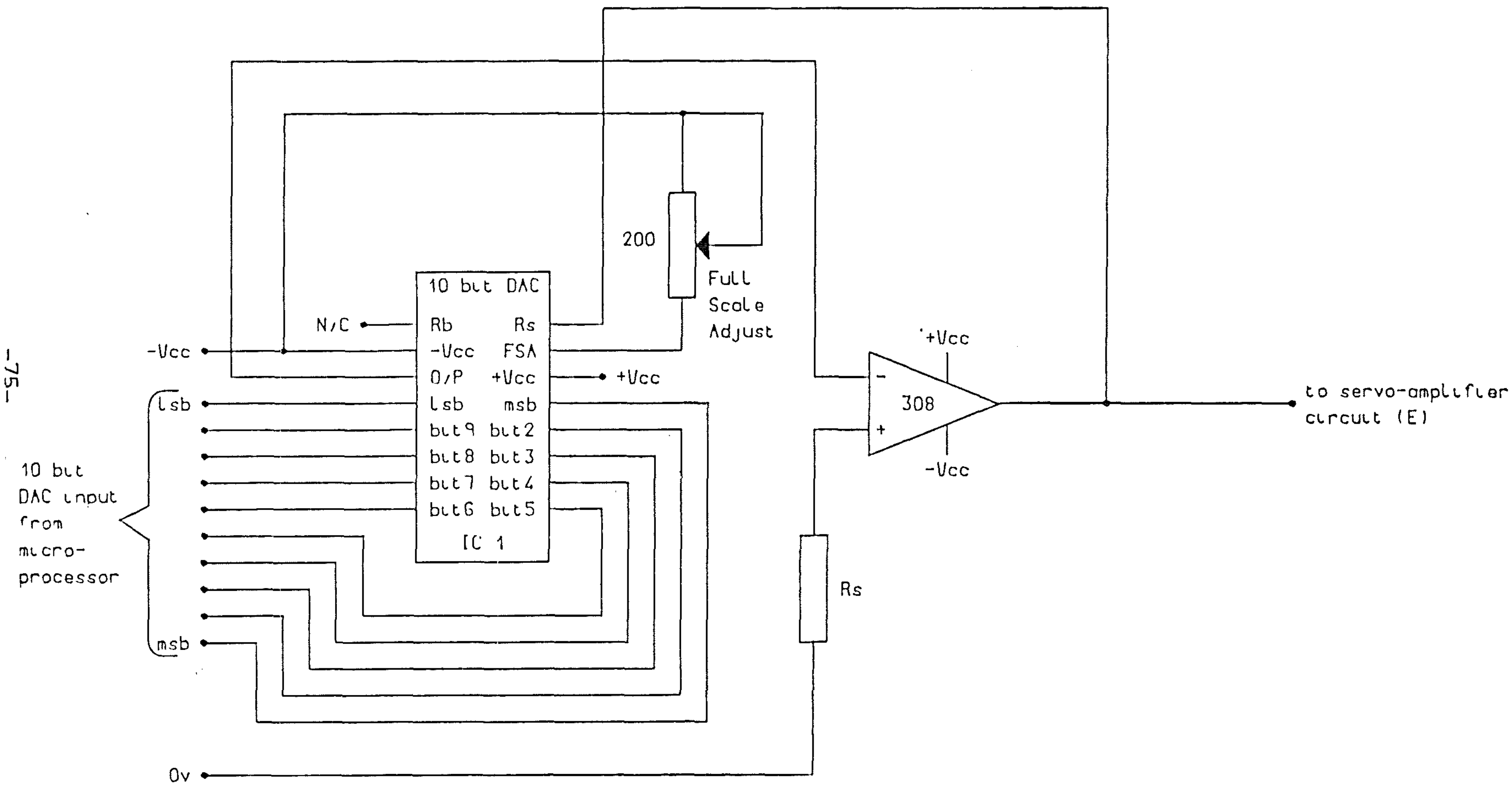


Figure (3.7.2)

10 BIT DAC CIRCUIT :



Note : IC 1 is suggested as PMI DAC 101Q

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transistors BC182 and a TIP 3055 (power transistor). The valve solenoid is connected between +30 volts and the power transistors. The 2K ohm variable resistor is used to provide 'zero' adjustment used to set the valve 'null'. The DAC input is simply inverted by the 308 op-amp which has unity gain. The 20K ohm variable resistor (RVAR) is used to adjust the 'range' of the amplifier.

A series of tests were carried out on the amplifier circuit at LUT, the points at which measurements were taken during the tests are indicated (A-E) on Figure [3.7.3]. The amplifier circuit went through a number of intermediate changes before the final design was produced, similar tests were carried out on the earlier designs, each of which exhibited certain failings, such as insufficient drive current and unreliability (Thatcher [1983]). The circuit described here represents (MK3.2) the up-to-date prototype design and will be the only amplifier design considered in this thesis. The series of tests carried out were;

(i) linearity,

(ii) frequency response, and

(iii) temperature stability.

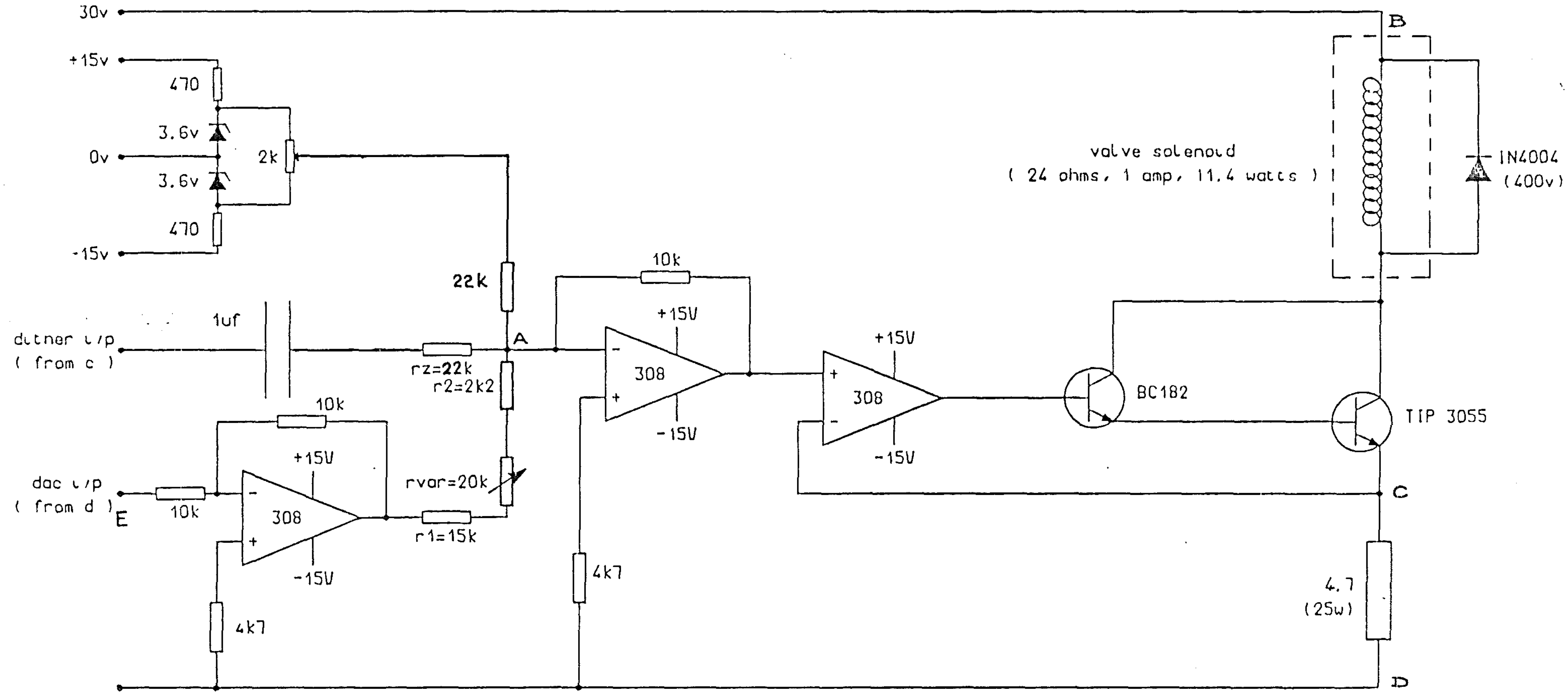
DAC input was simulated by use of a suitable resistor and variable resistors connected across 0.15volts, all tests were performed with a purely resistive load (Thatcher [1983]), including;

(i) linearity - two tests were carried out with different 'range' adjust settings, both show good linearity with gains of 2mA/volt and 0.99mA/volt (Thatcher [1983]), see Figure [3.7.5],

(ii) frequency response - a signal generator with dc offset was connected at the DAC input, and a dual beam oscilloscope was used to measure input and output signals. Two tests were conducted with 5v and 3v

Figure (3.7.3)

SERVO-AMPLIFIER CIRCUIT :



note : corresponds to lut
mk 3.2 servoamplifier

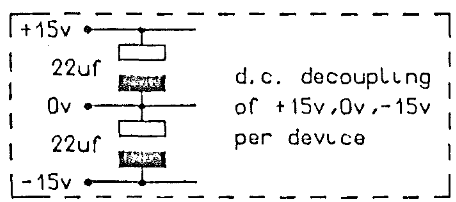


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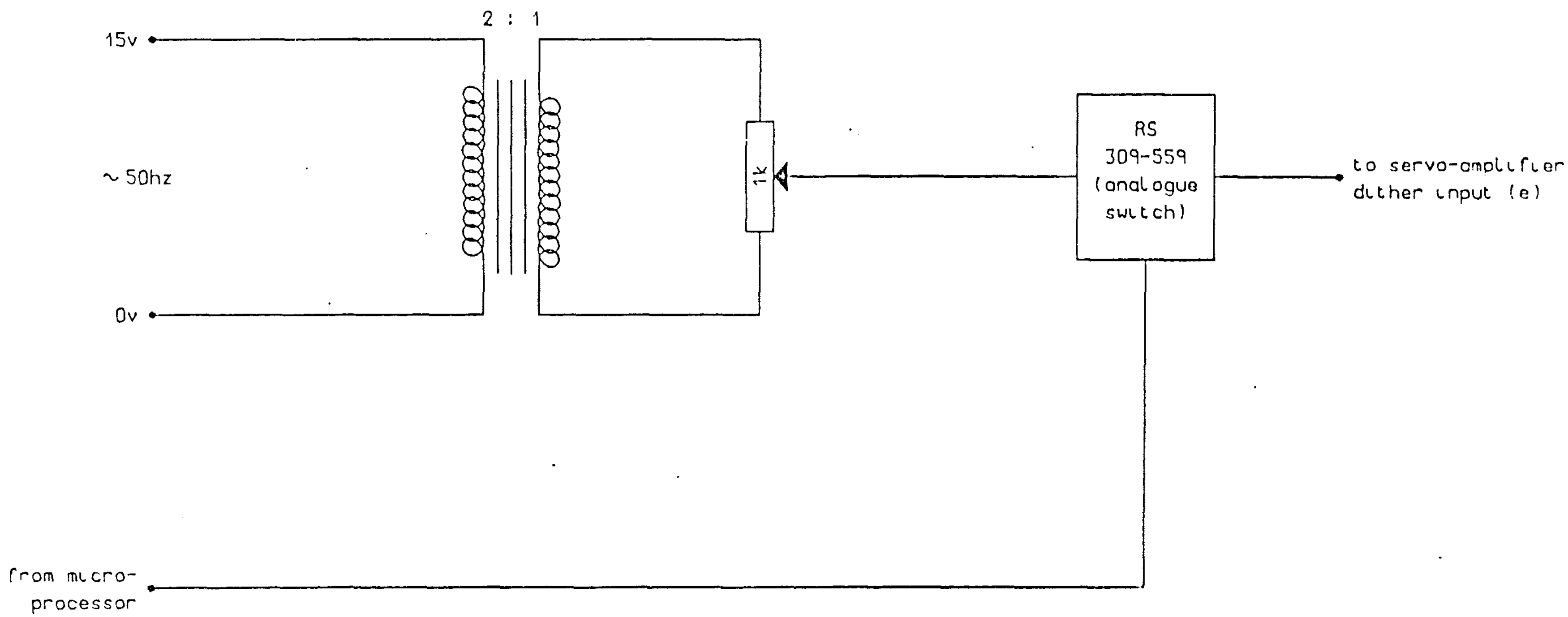
Figure (3.7.4)

DITHER GENERATION CIRCUIT



notes : fixed frequency (50hz)
microprocessor controlled on/off capability
suggest rs 217-797 for transformer
but smaller models maybe available

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dc input levels, in both instances the amplifier gain was found to remain constant up to about 4.5 Khz, having a bandwidth of over 44Khz (Thatcher [1983]), see Figure [3.7.6], and

(iii) temperature stability - a number of temperature stability tests were conducted upon the amplifier; these tests and the results are described in detail by Thatcher T.W. [1983] in an LUT internal report. The MK3.2 amplifier design was found to exhibit acceptable temperature stability characteristics.

3.8. CONCLUSION

In this Chapter the elements of pneumatic drive used in this research study have been described. It should be noted that a number of target systems were necessarily constructed during the course of this study. These included microprocessor based systems for;

(a) pneumo-mechanical 'bang-bang' control strategies,

(b) digitally operated servo-valve systems, and

(c) solenoid operated servo-valve systems.

A significant proportion of time was spent in the design, implementation and evaluation of the various interface electronics hardware for the control strategies investigated. In particlar the development of encoder interface electronics and the proportional servo-valve amplifier involved considerable research effort.

This Chapter considered the types of actuation and feedback system for modules and considered the relevant advantages and shortcomings of each. The prototype module designs used in evaluation of motion control strategies were then considered.

Figure (3.7.5) Mk 3 Servoamplifier : Linearity Check

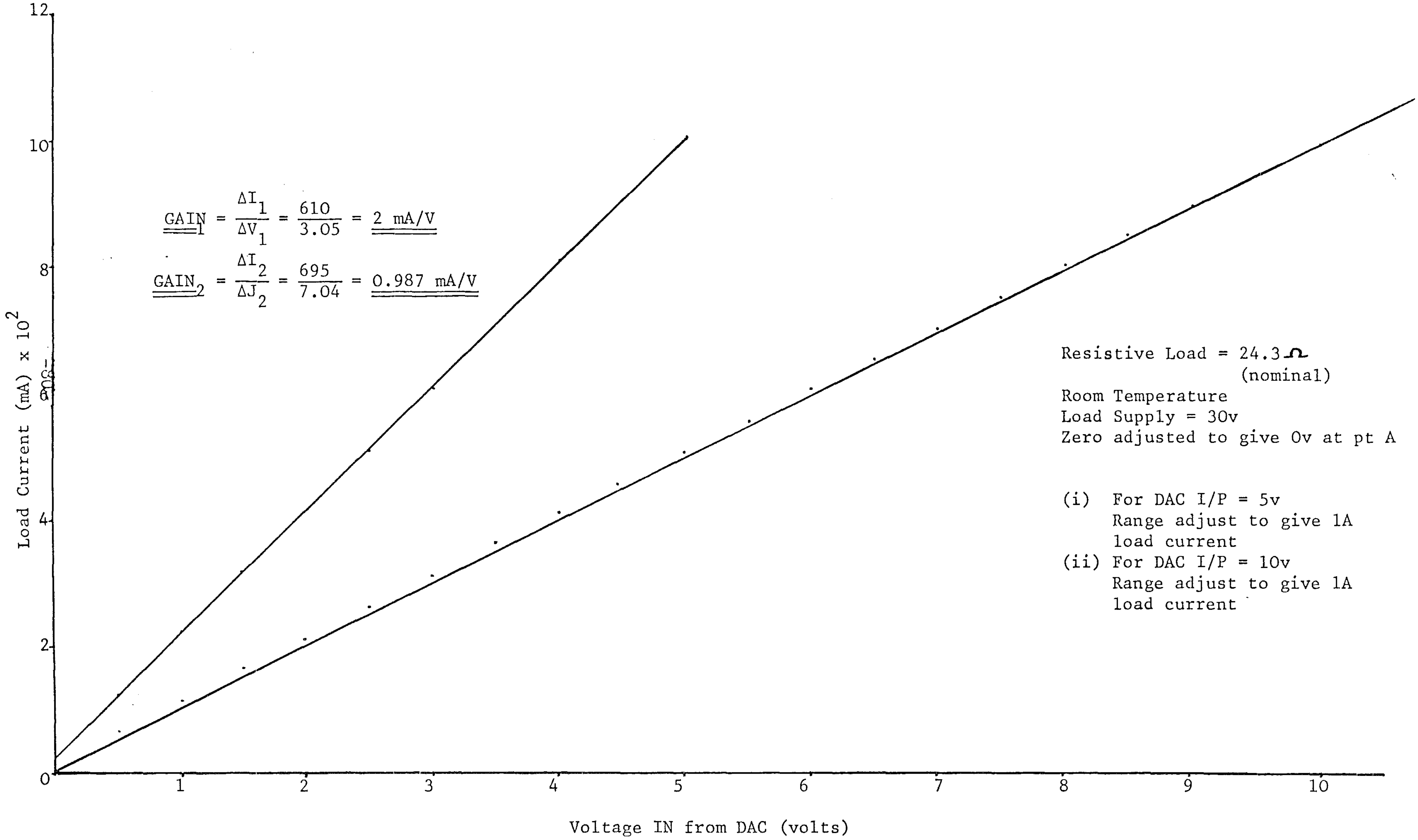
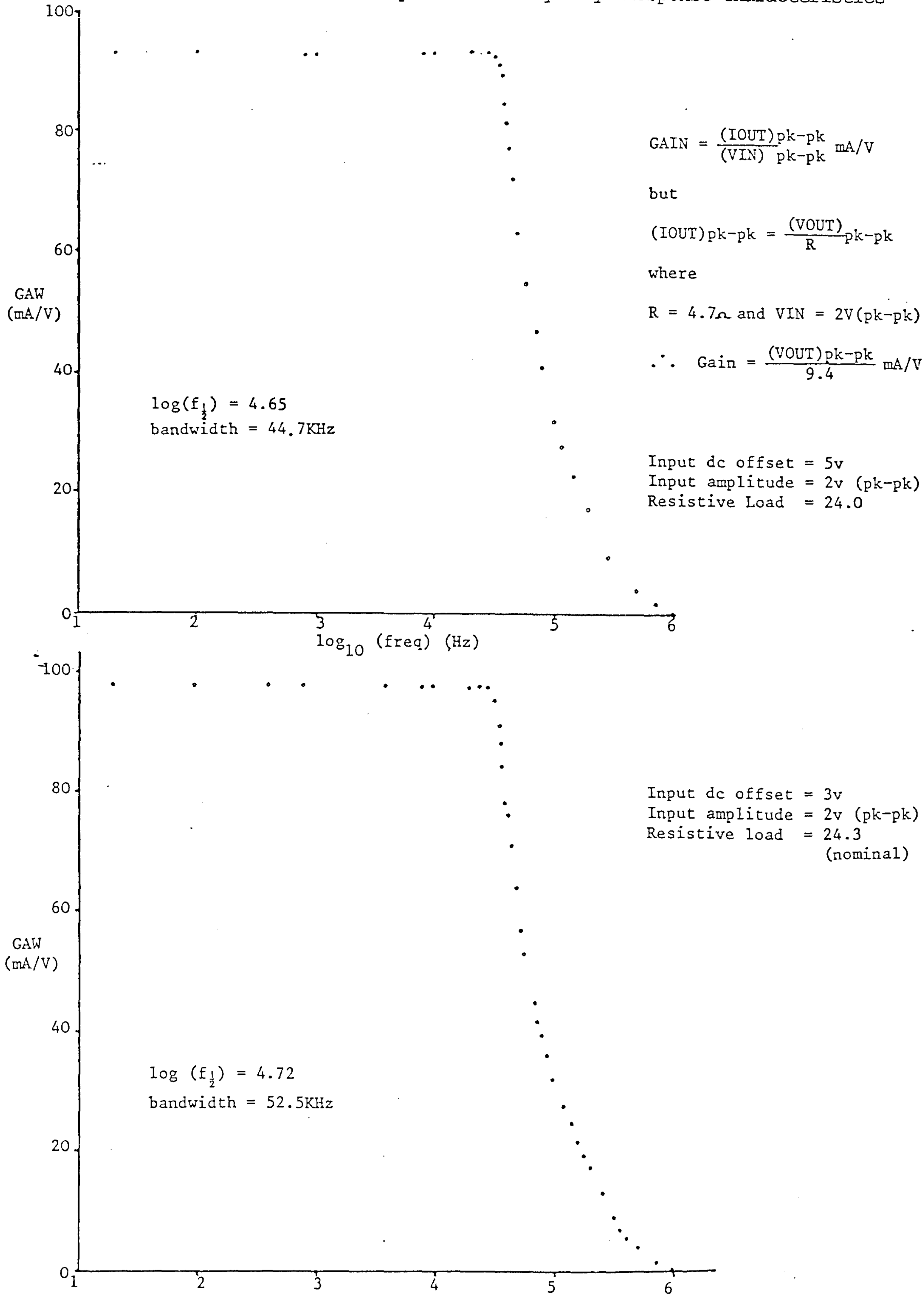


Figure (3.7.6) Mk 3 Servoamplifier : Frequency Response Characteristics



CHAPTER 4

RESUME OF PREVIOUS STUDIES OF PNEUMATIC MOTION CONTROL SYSTEMS

4. INTRODUCTION

When reviewing the literature of previous research studies of 'pneumatic motion control' it has been necessary to consider two broad classifications;

(a) bang-bang positioning systems (where binary or on/off valves are used), and

(b) servo-control positioning systems (where proportional or servo-valves are used).

At the onset of this research study, some attempts at evolving position control strategies for pneumatic drives (which could be utilised with robot modules) concentrated on the use of pneumo-mechanical 'bang-bang' control techniques using 'off the shelf' binary valves. The results of these investigations were documented (Moore [1986]) and some success was achieved. However,

(i) the constraints imposed by the market considerations of the industrial collaborators (Martonair),

(ii) the availability of a prototype 'proportional valve' specially designed by the industrial collaborators,

(iii) the potential to improve further performance criteria, and

(iv) the robustness of any control strategy to accomodate different kinematic and dynamic properties of

modules, ultimately determined that it was necessary to investigate various servo-control positioning strategies which would be suitable using pneumatic actuation for a range of robot modules. This Chapter reviews the literature within the two classifications of drive system providing foundation material for the resulting investigations. The results of the research study into microprocessor based pneumo-mechanical bang-bang control at LUT was beyond the scope of this thesis (Moore [1986]).

4.1. RESUME OF PNEUMO-MECHANICAL BANG-BANG PNEUMATIC DRIVES

As discussed in a previous chapter, the pneumatic drive is the most widely utilised form of actuation for robotic devices. However, in proprietary commercial systems, prior to the commencement of this research, the pneumatic drive was used only to provide mechanical 'end stop' and switchable intermediate 'stop' positioning. However, significant research effort worldwide has centred on the design of pneumatic 'bang-bang' control strategies in an effort to extend the application of pneumatic drives to achieve 'point-to-point' positioning of payloads. This by inference implies that it is necessary to utilise some form of position transducer within the system. In the simplest form this could involve the use of variable location proximity sensors.

Rooks B.W. and Tobias S.A. describe a positioning system that is controlled by a conventional 'sequence interlocked' program controller. Position control is by means of mechanical stops mounted on hexagonal indexing bars, which provide at least six different positions for any one axis. The end stops are designed to detect when close to the end position to allow deceleration of the cylinder or 'ram' to be initiated. A new mechanical stop is selected by indexing the bar to the appropriate position using a small pneumatic actuator through a clutch and ratchet system. The system described allows greater flexibility than 'end stop' systems but does not allow a very large number of programmed positions within the actuator stroke to be achieved: this being limited by mechanical complexity. Although control valving is relatively simple as commercially available binary electro-pneumatic valves can be used, several such valves are necessary. Thus as the number of valves required is large and the mechanical complexity of the system is high, such an approach has severe cost penalties.

Kopp V.Ya and Pogorelou B.V. [1982] describe an analysis technique for the mathematical modelling of a digital pneumatic rotary drive. The use of the model allows parameters to be determined in an effort to optimise the drive performance. The rotary actuator described is similar conceptually to the actuator described by Rooks, in that it provides a number of fixed positions (where the number of vane actuators determines the number of positions). The vane actuators (providing limited rotary motion) are connected sequencially such that the shaft of the preceding actuator rotates the housing of the next actuator (analogous to a binary cylinder). The angle of rotation of the shaft of the next rotary actuator is chosen to give twice the angle of rotation of the preceding shaft. By sequencing the binary control valves to each chamber any descrete combination of shaft positions can be obtained. Dampers provide a deceleration phase operated via mechanical linkages at the extremes of a move. However the cost of such an approach can only be justified in very specialised application areas.

Pietzsch L. et al [1974], Huhne G. and Neugebauer A., [1974], describe a pneumatic drive system for both rotary and linear motion. The drive consists of a pneumatic actuator, a position encoder, a slideway/bearing arrangement and an electro-magnetic brake. The system described here can allow a large number of programmed positions to be selected with a certain level of performance. The control system is of a quasi-continuous cut-off type, that is, to obtain a new 'set-point' position from a 'known' position a cut-off signal for the movement is calculated. The start of the controller braking action is determined by the following control parameters; maximum acceleration, time constant of the brake, inertia of the moving drive components and the inertia of the load. The brake closes with a controlled braking couple and the speed of the actuator will be proportional to the position error deviation. The braking couple is limited by the maximum couple of the brake. The occurance of wear in the system will induce changes in performance characteristics over a period of time, so would changes in slideway friction conditions. The necessary binary control valving is relatively simple, however the brake control system is an additional cost and complexity.

Wauer G. [1980], describes an application of the control strategy which is discussed in some detail in the previous paragraph (Pietzsch et al [1974]). It is used as a drive system for Pfaff Industrial Robots. These include a four degree of freedom cylindrical co-ordinate manipulator and gantry mounted manipulator.

Belfonte G. et al [1981], describe the design, implementation and testing of a dynamic braking system that acts on the rod of the pneumatic cylinder. In the research study described the brake closing was actioned by a sensor actuated by a cam that moves with the cylinder rod. Variations of cylinder velocity, load inertia and sensor positions have been investigated. The brake acts directly on the cylinder rod, the cylinder rod passing through the annular brake, it being actuated by a pneumatic cylinder whose movement parallel to the cylinder rod is translated to clamping motion via the wedge action of conical metal inserts. These conical inserts force four elastic elements (with 'ferodo' brake inserts) to contact the cylinder rod and apply a braking couple. The brake is released by an internal spring when air pressure is released. Such a brake mechanism could find application in a bang-bang control strategy using binary control valving. However, performance would be limited by the consistency of the braking couple.

Gradetskii V.G. and Paroi A.A. [1981] describe special cases for solving the equations of motion of Industrial Robots using pneumatic actuating mechanisms. They propose a methodology for calculating the point at which to initiate braking devices: (i) when an additional volume is automatically connected to the cylinder exhaust chamber; (ii) when at the exit from the exhaust chamber: a brake bushing and needle type restriction are used for exhaust regulation. The methodology proposed in each case was used to implement a computer based solution using numerical methods.

Paroi A.A. [1981], using a similar approach to that used by Gradetskii et al [1981], propose methods for determining the position at which to initiate the braking action when the actuating piston demonstrates varying laws of motion up to and during the braking process (eg variable friction). Practical implementations, based on the results presented, are suggested for designing drive braking units. These drive braking units include exhaust restriction, additional control volumes (cushion) and mechanical braking.

Prokofev V.N. et al [1981] in a general discussion on the use of pneumatic and hydraulic drives for industrial robots, illustrates some of the advantages of a pneumatic drive and references their use in Japanese robots such as the 'Khimeit' and 'SMT-700'. Many types of pneumatic/hydraulic actuators are discussed: the paper describes a linear binary code actuator similar in concept to the rotary actuator detailed by Kopp et al [1982], where the linear actuator is made up of a series of interlinked chambers of varying stroke length, thus with 'n' chambers it is possible to position to 2n-1 descrete positions along the stroke of the actuator. The cost of such actuators is restrictive and they can only be justified in specialised applications.

Krisztinicz P. and Elek I. [1976], describe several types of pneumatic control strategy and present results comparing their performance. The different types of bang-bang control strategy described include; a purely pneumatic system, a pneumatic system with a mechanical brake, and a hydro-pneumatic system. Pneumatic systems only included; (a) positioning by shut-off of a binary directional control valve, triggered by a stop signal produced by a roller limit switch; (b) positioning by shut off of a binary directional control valve, triggered by a stop signal produced by a jet cut-off sensor; (c) positioning by shut off of a binary directional control valve by triggering from the jet cut-off sensor with the additional use of a 'pressure equalisation' valve; (d) positioning by shut off of a binary directional control valve but using two jet cut-off sensors (giving a high speed phase followed by a creep phase) thus further restricting exhaust gases. Pneumatic positioning using a mechanical brake on the piston rod is described, the braking couple being produced by a pneumatic actuator driving clamping collets: again exhaust restriction is used to obtain speed control. Another control strategy described is one combining hydro-pneumatic actuation and binary control valving. For all the tests described a position transducer was used for performance evaluation purposes. Each of the control strategies described provides only limited positions otherwise the strategies become restrictively complex and costly.

Kreinin G.V. [1978] discusses several types of pneumatic drive systems, which include 'end stop' systems with both passive and active dampers (exhaust restriction); pneumatic drives with adjustable stops, produced by a stepped mechanical system (similarly described by Rooks); and drives which use multiple (n) piston stops which allows 2n-1 descrete positions (binary cylinders). Other pneumatic drives described which do not utilise stops, include control strategies using stepped positioning mechanisms with a binary pneumatic control circuit and pneumatic braking; and a stepped pneumatic positioning mechanism with a binary pneumatic control system (where a leadscrew and nut is used for linear motion and harmonic gearing is used for rotary motion). Each of these control strategies is complex mechanically and consequently expensive and only a limited number of positions can be achieved.

The reasons for the preferential use of hydro-pneumatic drives are discussed by Vodopyan P.O. and Oksenenko A.Ya [1978],[1981], who also define areas of application. They provide data concerning new aspects of hydro-pneumatic equipment. A research investigation aimed at producing standardised components for robot drives of all types is discussed.

In two papers Koroljev V.A. et al [1979] and [1982], discuss the problems of achieving saisfactory dynamic responses as determined by the amplitude and 'settling' time for different types of pneumatic robots: the robots using either a mechanical braking system or a pneumatic braking system. This analysis is applied towards the problem of the 'shaft-bush' assembly task.

4.1.1. SUMMARY

General conclusions drawn from this review of research studies into pneumo-mechanical bang-bang pneumatic drive systems are;

(i) the drive systems tend to be relatively complex either in mechanical construction or in the associated pneumatic control valving (both of which imply cost disadvantages);

(ii) many of the drive systems will only allow a limited number of positions which consequently limits the flexibility of the drive;

(iii) the drive systems which use progressive braking techniques have no limitations on the number of positions, however the static and dynamic performance that can be achieved is limited and performance will vary as slideway conditions and payload change;

(iv) the simpler control strategies could find application in specialised applications but are not considered to be generally appropriate as drive systems for flexible robotic elements.

A study of microprocessor based bang-bang control strategies (using pneumo-mechanical control elements) was conducted at LUT (Moore [1986]).

4.2. RESUME OF SERVO-CONTROLLED PNEUMATIC DRIVES

Having considered the use of 'bang-bang' pneumatic drive systems (a resume of previous research studies is given in section 4.1) we shall now consider the findings of various research studies in the use of pneumatic servo-drives.

The term servo-drive or servo-mechanism has been defined in a number of different ways (Burrows [1972]). It has been used to denote a power-amplifying device where the amplifier is error-actuated and it has also been used to describe a feedback control system where the dependant variable is position and/or its derivatives with respect to time (the later definition applies here). The output quantity (position of actuator) is adjusted by the error via the controller and the servo-motor (a control valve). In a digital servo-system the error and the control signal are generated under computer control, the feedback signal being generated by a position transducer which monitors the motion of the actuator/load.

Process control systems (characterised by plants which have long time constants, that is, the output does not change rapidly following a load disturbance; also the input usually remains reasonably constant) make extensive use of 'pneumatic controllers/positioners' using air at a pressure of 3-15psi (Burrows [1972]; Martonair [1982]): this category of positioner is specifically excluded from this literature review as it is functionally different from a servo-drive (servo-drives are associated with applications where the input is continuously varying and the load changes less frequently).

In contrast to hydraulic and electric servo-drives, little attention has been paid, until recently, in the use of pneumatic servo-drives. Foundation studies on pneumatic servo-drives were carried out by Shearer J.L. [1954], [1956] and [1957], who conducted various theoretical and experimental studies involving the use of air at very high pressures. The limited research interest in pneumatic servo-drives since Shearers' foundation work can be attributed to the fact that air is a compressible medium and hence, the associated describing models are more complex than those associated with hydraulic systems. However, 'hot gas servo-mechanisms' have received attention in applications in aircraft and missile systems (Vaughan [1965]). Pneumatic servos have been successfully implemented in the design of high pressure systems (Botting et al [1969]), but for industrial applications safety requires that the supply pressure should be relatively low (below 10 bar). In low pressure servos the inherent disadvantages of using a compressible working medium become more apparent (where the bulk modulus decreases as pressure is reduced). Early implementation studies involved the use of analog control systems which are not particularly suited to facilitate compensation for low drive stiffness and the inherent non-linearities introduced by pneumatic elements. Shearers' work on pneumatic systems also demonstrated the effect of Coulomb friction on drive performance. It was shown that low levels of Coulomb friction were sufficient to cause significant dwell in the response during reversal of the actuator as a result of the time taken for the load pressure to build up (which, in turn, is due to the compressibility of the air).

Burrows C.R. [1968], [1969] and [1969], extends the earlier research work of Shearer, (which was concerned with describing the motion of the load when the piston is operating about the mid-stroke position to account for motion from any initial position). Burrows studies a proportional valve controlled pneumatic servo-mechanism and shows that system stiffness and damping are load position dependant (where, in the mid-stroke position the system is least damped). The system analysed by Burrows comprised a four-way valve controlling the flow of air to a pneumatic cylinder which positions a known load. The valve was operated (or manipulated) by a torque motor using an analog controller which summates the position, velocity, and transient pressure feedback loops. Position dependant drive stiffness has been shown to occur in both 'proportional' and 'on-off' controlled low pressure pneumatic servo-mechanisms (Cutland [1967], Burrows [1968]). The analysis of Burrows has been extended as part of this research study (see

Chapter (7)) yielding a linearised model which has been used as a design aid in evolving the control strategy described within this thesis.

In Burrows' paper [1969], it presents experimental and theoretical transient response and frequency responses results for a low pressure pneumatic servo, comprising of a four-way flapper-nozzle valve (activated by torque motor), pneumatic cylinder and load in the presence of coulomb friction and stiction. He presents a simplified model to describe the system under investigation, results agree with experimental results except for the case where a variable input signal is used, here friction has a significant influence.

Botting L.R. et al [1969] use linearised analysis techniques to design feedback compensating networks for a high pressure pneumatic servo-mechanism using a proportional valve. Digital computation of the complete non-linear equations is used to predict the effect of the following parameters; saturation, inertia load, stiction and coulomb friction, valve laps and leakage. It should be noted that here it is a high pressure servo-mechanism that is analysed using a single stage servo-valve with open centre operation. The equations presented are non-dimensionalised and apply equally to both linear and rotary type actuators. They illustrate that because of high loop gains, required to meet performance specifications, the servo-valve is easily saturated. Also the implications of valve spool lap design are discussed together with the effects of stiction and coulomb friction.

From the analysis and experimentation, the following observations are concluded:

(i) the maximum bandwidth possible is of the order of twice the natural frequency, (but this is only possible using non-linear compensation),

(ii) the stiffness and accuracy of the servo-drive are very sensitive to stiction and leakage (thus they should be minimised), and

(iii) the valve should have its land laps adjusted in conjunction with leakage to give standing pressures between 0.5 and 0.8 of supply pressure (since these values give the best compromise between pressure gain and natural frequency).

Vaughan D.R. [1965] presents a third order model of a hot gas position control system. He shows that due to gas flow saturation certain limitations are imposed on the speed of step response. The results presented indicate that significant improvements in response time can be achieved through the use of non-linear feedback compensation.

Mannetje J.J. [1981] describes a method to control a fluid power servo-system when a compressible medium such as air is used. A high gain pressure feedback loop alters the valve characteristic to a constant pressure source independent of the rate of flow. The control valve described consists of a moving coil in a permanent magnet field, the valve sleeve being rigidly attached to the coil. The pressure feedback loop uses proportional and derivative action only, the valve having a resonant frequency of 190 Hz. The position loop controller described uses simple lead and lag compensation networks. This analysis suggests that this approach when compared to the classical methods gives an improvement in system bandwidth of some twentyfold, higher bandwidth implying higher static gain, better resolution, smaller following error and improved the static and dynamic stiffness.

Nguyen H.T. et al [1984] describe a pneumatic servo-system developed at RWTH Aachen for position control. A simplified theoretical model of the pneumatic system is presented which is used in the design of an appropriate control strategy, suggesting the use of velocity and acceleration feedback loops in addition to the position loops. Earlier foundation studies by Shearer [1954], [1956] and [1957] had suggested similar feedback loop compensation techniques. The positioning system described by Nguyen consists of; a rodless pneumatic cylinder, a single stage servo-valve (operating at 7 bar supply pressure with a maximum flow of 600l/min), a linear incremental encoder with a resolution of 10 microns over a stroke of 650mm, and a microcomputer controller. The controller uses a sixteen bit processor, which runs the control loop every 600 microseconds (interrupt driven), the control algorithm is written in assembler. The encoder is interfaced to the computer via suitable interface electronics, and velocity and acceleration information are obtained by differentiation of the position signal. Performance figures of 0.1mm repeatability and maximum velocity of 1.5m/s for the actuator described were obtained.

Backe W. [1983] describes a pneumatic servo-system that was developed at RWTH Aachen, but in this case two three port proportional servo-valves are used. Performance figures of 0.01mm repeatability, maximum velocity of 2.5m/s and acceleration of 40m/s.s for a 200mm stroke unit are presented. This drive system was marketed by Kienzle, but has since been taken over by the GAS Corporation.

Neuhaus R. [1981] describes a closed loop pneumatic servo-drive which consists of a four port proportional servo-valve controlling a symmetrical actuator and an inertial load. The proportional valve is spring centralised (centre closed) actuated by dual solenoids. Transient response results are presented for various methods of analog compensation technique.

Schwenzer R. [1984] describes a pneumatic servo-system developed at RWTH Aachen using an analog control system with a five port proportional servo-valve operated by two proportional solenoids acting in opposition. The paper also illustrates how 'dither' can be used to reduce hysteresis in the servo-valve, also suggested by Burrows [1972]. The system described here, in terms of system design and performance characteristics is similar to that described by Backe [1983]. A very high resolution feedback linear transducer has been used to obtain the level of static peformance claimed. Schwenzer's results show that as the controller forward path gain is increased drive stiffness is increased, enabling the static positioning performance to be improved.

Drazan P.J. et al, [1976] to [1980] has carried out a range of research studies aimed at using electro-pneumatic drives to actuate multi-axis robots. This work originated at Surrey University with GKN as industrial collaborators. The objective was to design a low cost conventional pedestal mounted robot, by employing pneumatic drives. The system initially consisted of an articulated arm with three degrees of freedom, a base joint (rotating horizontally) and two joints, approach and lift, (rotating in a vertical plane), additional degrees of freedom can be added (eg wrist, gripper etc..). The earlier design used linear actuators and suitable linkages to provide rotary motion about the manipulator joints. Initially a control valve based on a hydraulic proportional servo-valve was used. Changes in manipulator design have now incorporated Kinetrol rotary vane actuators (90 degrees stroke) on each joint and switching of a binary (on-off) valve are used to effectively change it into a proportional servo-valve because of the unavailability of a suitable low cost proportional valve.

A digital control strategy was designed using a linearised model and computer simulations. The final control strategy uses state variable control, state variable data being derived from a position transducer on each joint and numerical differentiation techniques. The digital algorithms ultimately produce a binary logic pattern that is then amplified and used to switch solenoids which in turn actuate on-off pneumatic valves which control the flow of air to and from the actuator chambers. The latest control strategy described uses a position sampling rate of 1ms and the control signal update rate of 20ms. Position data being required more frequently for the algorithms calculating velocity and acceleration information. The approach developed by Drazan is now used commercially in the 'Pendar Placemate' industrial robot which comprises a pneumatically actuated articulated arm with three degrees of freedom with 'point-to-point' positioning capability under computer control.

Mariuzzo C.L. et al [1980] present a non-linear mathematical model to describe the dynamic response of a class of pneumatic actuators. The object of the study was to develop an elaborate system model which would enable the selection of optimised parameters for use in designing pneumatic actuators to meet stringent requirements on speed of response, torque to weight and gas consumption. The rotary actuator modelled consists of twin cylinders with a flapper arrangement on each exhaust port, linear displacement is converted to rotary motion via a linkage mechanism, the flapper being controlled by a push-pull system of solenoid coils which are rigidly attached to each piston.

Mariuzzo et al [1980] describe a simplified non-linear model which was used to conduct analog parametric studies. These studies included both transient and steady state aspects of the positioning servo-system. The effect on the position dead zone, time lag, static gain, response time and torque are considered amongst others. As a result of the analysis, recommendations for the synthesis of this class of servos are presented.

Ivlev I.V. [1983] describes an analog controlled pneumatic servo-drive for an industrial robot arm. The drive consists of a pneumatic cylinder (with teflon for piston seals to reduce friction effects) with a 140mm stroke, a potentiometer position feedback transducer (velocity feedback is obtained by differentiation of the displacement signal) are incorporated together with a servo-valve. Performance figures quoted for this analog servo-system are, 0.5mm repeatability and working frequency of 0.1hz for a payload of 0.3kg. To reduce the air consumption of the servo-valve the supply is disconnected during arm stationary periods with a binary electro-pneumatic valve.

Kraynin G.V. et al [1982] describe how an observer model can be used in an analog control system to compensate a pneumatic servo-drive. An observer model is used to create supplementary feedback loop couplings in the control circuit of the pneumatic servo-drive. It is shown that the introduction of a negative velocity feedback loop, via the observer, can stabilise the dynamic response. All that is necessary is a single displacement transducer signal which is then input into the observer. This provides an alternative to 'direct measurement' of state variables or 'numerical differentiation' of position data.

Eun T. et al, [1982] describe the development of a low cost pneumatic positioning system, which utilises a solenoid operated on-off valve. This analog control scheme utilises an input potentiometer and feedback potentiometer, the two signals are compared and an electronic relay switches the appropriate control solenoid. Methods of compensation investigated by Eun et al [1982] include the use of stabilising tanks, actuator chamber 'coupling' and velocity feedback. Each of these methods are compared in the results presented (each of these stabilisation techniques were investigated in earlier studies by Shearer [1954] and Burrows [1972]).

Kato S. et al [1972] discuss the influence that friction can have on the performance of pneumatic servo-drive. They point out, as many of the earlier references have, that the characteristics of a pneumatic servo-mechanism are susceptible to the prescence of friction owing to the relatively small operational forces exhibited: in particular the effects of viscous friction in the supply piping and the friction of the actuator and load are examined theoretically and experimentally. The experimental data presented in this paper was obtained with a single acting cylinder and a flapper and nozzle valve arrangement.

Hammett G.G. [1984] describes a low cost pneumatic servo-drive. The system uses a pulse width modulated poppet valve which controls the flow of air to the actuator. Feedback signals from the position transducer are processed by the control microprocessor, which generates the control signal to the solenoid operated poppet valve.

4.2.1. SUMMARY

General conclusions that can be drawn from this review of research studies into pneumatic servo-drives are;

(i) in recent years the possibility of using low cost microprocessor based controls, which enable complex compensation techniques to be applied, with a pneumatic servo-drive has stimulated significant research interest;

(ii) the unavailability of proprietary low cost proportional servo-valves has been a significant contributing factor to limiting research success and interest in developing such a drive technology (notable pneumatic servo-drive performance has been achieved in W.German studies (Nguyen [1984], Backe [1983]) using derivatives of hydraulic servo-valves. However, the cost of such valves could limit the acceptability of such drives);

(iii) switching techniques applied to binary electro-pneumatic valves can be used to produce a pneumatic servo-drive (Drazan et al); and

(iv) developments in low cost LSI elements and transducer technology make it technically feasible and commercially attractive to evolve a pneumatic servo-drive.

4.3. RESUME OF MODELLING STUDIES OF PNEUMATIC MOTION CONTROL

A resume of modelling research studies of pneumatic motion control will be described in this section.

4.3.1. MODELLING STUDIES OF PNEUMATIC ACTUATORS

Brown D.E. and Ballard R.L. [1972] developed digital simulation software to obtain the transient performance and response time of a pneumatic actuator and valve combination. The simulation includes the valve flow characteristics. The major difficulty presented in the simulation was found to be in the assessment of suitable values for actuator friction (Ff). The relationship

$Ff = Fc + C1v + C2(P1 - P2)$, where v = velocity and P1 and P2 = chamber pressures, was found to hold reasonably well providing correct values of the coefficients Fc, C1 and C2 could be determined. Factors such as dwell time between strokes, lubricant variability, ambient temperature and load condition all can affect the friction conditions. If the assumptions made about seal friction were accurate the simulation would accurately predict stroking time and pressures in the actuator.

Chitty A. and Lambert T.H. [1976] describe modelling studies on a low pressure pneumatic actuator and load. The results are used to verify the nature of the expansion and compression processes in the actuator and to study the effect of piston leakage and friction. In this research study a glandless labryinth seal was used for the piston seal, which is unrepresentative of most actuators in use, but it allowed a simple representation of friction to be used.

Nazarczuk K. [1979] presents a model of a double acting pneumatic actuator used in simulation studies. The model presented considers the three phases of motion, each described by a set of differential equations. Early experimental results are compared to the simulation results.

Watton J. [1984] studies the open-loop response of servo-valve controlled linear actuators for both extending and retracting motion. In particular, the influence of transmission line dynamics is considered and its effect on the overall system model is evaluated. This analysis considers hydraulic actuation but the synthesis of transmission line dynamics could be modified so that it becomes appropriate in modelling pneumatic drives.

4.3.2. MODELLING STUDIES OF PNEUMATIC SERVO-VALVES

Taft C.K. et al [1981], [1981] describe the development of a proportional electro-pneumatic servo-valve. The valve is a two stage device consisting of a piezoelectric bender element centred between two nozzles. The bender/flapper element is deformed by an electrical input signal, this then causes a change in pressure at the output of the nozzle, which is then amplified to some usable level using cascaded proportional amplifiers. A model of the valves behaviour is discussed and verified with experimental data.

Mayer E.A. [1981] describes a dual solenoid operated valve. Models of the valve are confirmed by experimental data, the valve is designed for use with single acting cylinders primarily for use in temperature or engine control systems.

Araki K. [1971] discusses the modelling and compensation of a force feedback pneumatic servo-valve. The valve design consists of two stages, a flapper-nozzle arrangement at the first stage followed by a spool and sleeve stage. Methods of compensation are compared; these include the use of spring centralising of the spool, and stabilising volumes. Both techniques improve stability and performance, the best results are obtained when used together.

Araki K. [1981] considers the effect of valve configuration on the performance of a pneumatic servo. The spool valve is considered in terms of a mean lap, unevenness of laps and a port width ratio to express the configuration of an underlapped or an overlapped valve. A describing model of a valve controlled cylinder is derived, where load inertia and friction are introduced. Experimental data is presented and compared with simulated results.

Viersma T.J. and Blok P. [1969] suggest methods of stabilisation for servo-systems using both hydraulic and pneumatic actuation. These techniques which relate to two stage valves (flapper/nozzle and spool stages) include chamber bypassing, spring centralising, and various types of pressure feedback which include stabilising volumes.

4.3.3. SUMMARY

For this review of modelling studies of pneumatic motion control several conclusions can be drawn. The design of a representative model of a pneumatic servo-valve controlled actuator and load system (that can be used in the simulation of performance) is extremely complex. However, various aspects of the processes involved have been the subject of research studies from which results can be drawn. The difficulties of modelling friction conditions in a pneumatic drive system is a recurring theme in many of these research studies.

The literature available on the study of pneumatic servo-valves is very limited, and material that is published tends to be very specific to individual valve designs or applications. Research publications in the area of process control valving are more prolific but these were considered to be outside the scope of this literature survey. The shortage of proprietary low cost pneumatic servo-valves is also reflected by the very limited number of research studies in this sphere. A wealth of literature is available describing research studies with hydraulic servo-valves. However, most of this material is not directly of relevance in the design of pneumatic servo-valves (particularly when cost implications are considered).

4.4. CONCLUSIONS

The literature survey of previous research studies in pneumatic motion control provided foundation material on which to build in the research study described in this thesis. Both broad classifications of pneumatic motion control systems have been reviewed: that is; (i) pneumo-mechanical 'bang-bang' control systems (using binary on-off valving where control is not continuous), and (ii) 'servo' control systems (using proportional servo-valves or the equivalent where control is continuous).

In pneumo-mechanical 'bang-bang' control systems in general, the limiting features appear to be the mechanical complexity of the actuator and the associated number of binary electro-pneumatic control valves. A pneumatic drive will only gain wide acceptance if it demonstrates a better relative "cost:performance" ratio than the alternative electric or hydraulic drive technologies. Microprocessor based controls offer the potential to improve the performance of pneumo-mechanical 'bang-bang' drives through the application of software controlled 'switching algorithms' and 'learning strategies'.

The application of pneumatic servo-drives has been severely restricted by the unavailability of proprietary low cost servo-valves, and the technical difficulties that have to be overcome in order to obtain good static and dynamic performance from such a drive. Microprocessor based controls and LSI elements make a technical solution feasible at relatively low cost, by providing compensation, with the use of digital control algorithms.

CHAPTER 5

EXPERIMENTAL EVALUATION SYSTEM

5. INTRODUCTION

The experimental evaluation system used in the design, evaluation and analysis of control strategies is described in this Chapter. Evaluations have been conducted with a range of pneumatic drives operating under different experimental conditions (eg load change). An experimental evaluation system was originally produced to investigate the bang-bang control strategies described in Moore [1986]: additional software and hardware facilities necessary included the facility to automate part of the set-up procedure and the pre-processing of the experimental data.

The experimental evaluation system so configured involves a number of sub-systems, as shown schematically in Figure [5.1]. The sub-systems are;

(i) the 'control system' and its associated elements which comprise the actuator and slideway, a proportional valve, a position transducer, a micro-computer, interface electronics and control algorithms performing realtime control functions (as described in Chapter (8)),

(ii) the 'test bench' and its associated elements, which comprise pneumatic supply components, load test facility, external position transducer and measurement/recording instrumentation, and

(iii) the 'evaluation software' and its associated functional modules which comprise facilities for set-up, test program sequencing and single stepping, diagnostics, control parameter editing, data storage and data analysis.

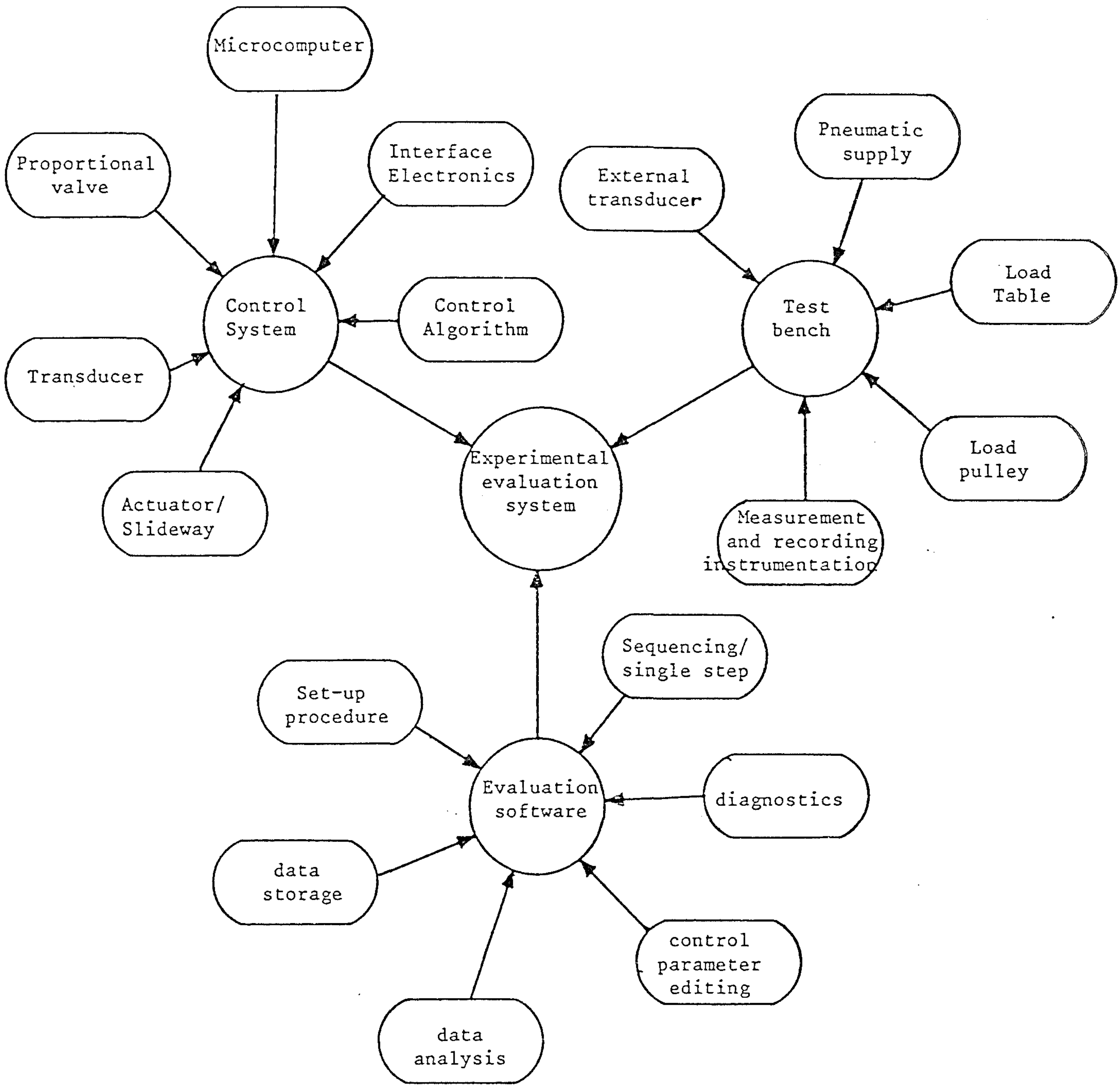


Figure (5.1) Experimental Evaluation System

5.1. THE EXPERIMENTAL TEST BENCH

The 'test bench' was designed to allow a series of experiments to be conducted on a range of different modules. This test bench is required to locate and support the module whilst load testing with both inertial loads and mass loads, is carried out. Instrumentation for measurement and recording of experimental parameters is required and the equipment used comprised of a digital storage oscilloscope, graph plotter, and pressure gauges/transducers. A controlled pneumatic supply was necessary for the proportional control valve and brake mechanism (when used). The external position transducer was used as an independant source for monitoring the performance of a module and can be used to calibrate the integral feedback system used with each module; thereby validating the static performance data derived.

5.1.1. THE TEST TABLE

Linear modules under test were located on the test table which comprised a heavy framed cast iron marking out table. This provided a solid level frame on which to mount the module, via a suitable adaptor plate, and also supported any load test equipment.

5.1.2. PNEUMATIC SUPPLY COMPONENTS

Various components are required to provide pneumatic supply services for the control valves. The pneumatic supply is provided by tapping into the pneumatic ring network which supplies the laboratory. The air is filtered (with a 25 micron screen) and then regulated so that required supply pressure can be obtained with an appropriated amount of lubricant being added to the air. A standard combination unit is used to achieve the necessary filtering, regulation and lubrication.

5.1.3. MEASUREMENT AND RECORDING INSTRUMENTATION

The measurement and recording instrumentation consists of digital storage oscilloscopes. These are used to display position, velocity and acceleration information during dynamic performance testing. The only state variable measured directly is displacement, but velocity and acceleration are derived using digital differentiation techniques. This sampled data, both measured and derived can be output to DAC's available within the prototype target system allowing the state variables to be displayed. If a digital storage scope is used then the transient response data can be displayed in realtime and stored as necessary, from which measurements can be taken.

Once the necessary measurements, such as positioning time (tp), maximum velocity (Vmax) etc, have been measured, the data can then be transmitted to an 'X-Y' plotter under the control of the storage scope so that a permanent record of the transient response can be obtained.

Pressure transducers and associated charge amplifiers were also used to display the pressure changes within the control volumes of the actuator; this facility was utilised in some evaluations.

5.1.4. THE LOAD TABLE

The load table was designed to provide an inertia load test for linear modules, and consists of a table slideway and load carriage. The individual components of the load table are shown in Figure [5.1.4.1].

The load table slideway is mounted onto the surface of the 'test table'

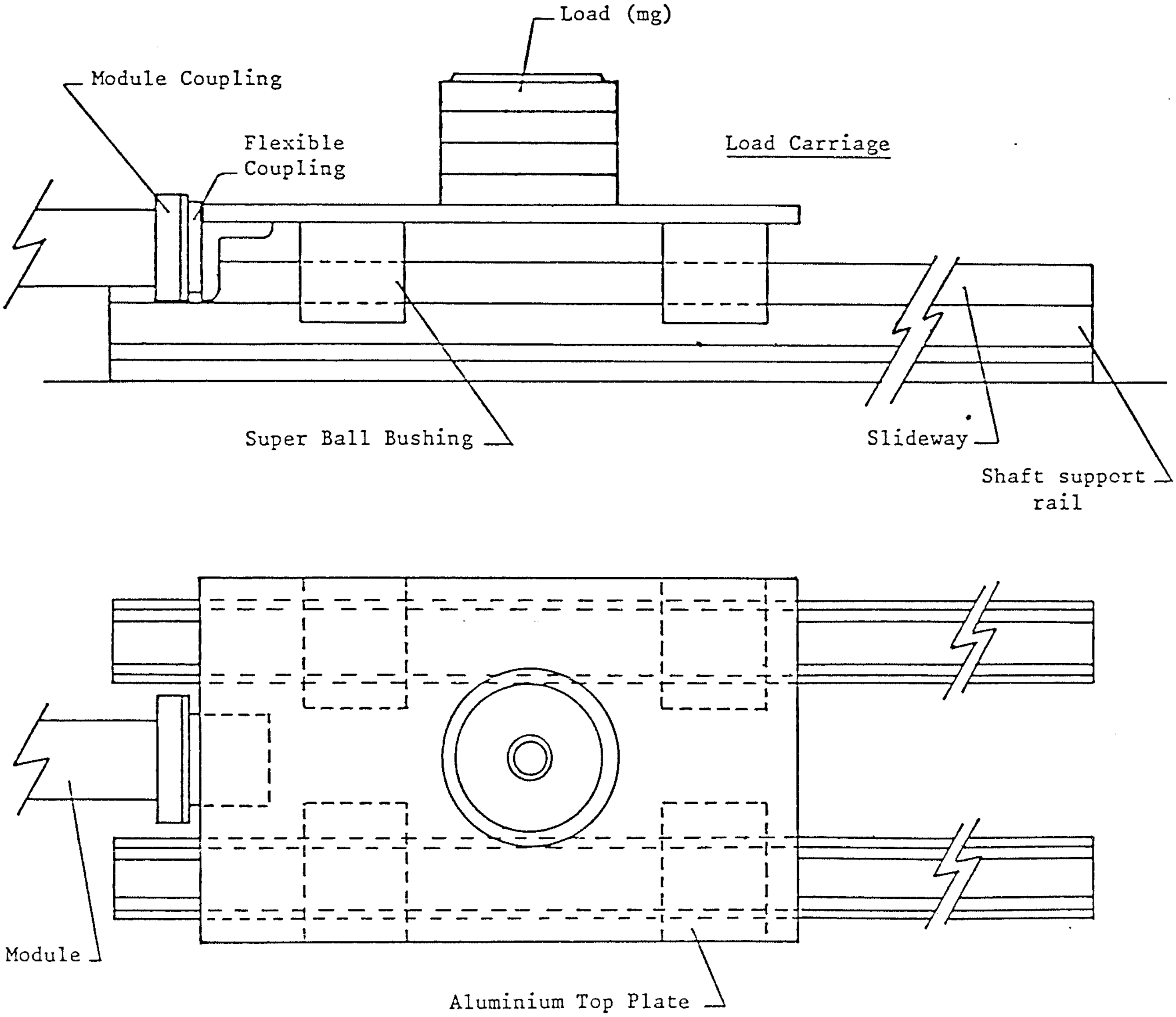


Figure (5.1.4.1) Load Table

using six 6mm caphead screws in each of the two shaft support rails. The table slideway is 1200mm long and is made up of two shaft support rails (4 x 600mm lengths) SR16 and two 1200mm lengths of 1 inch O/Diameter shaft (class L) hardened and precision ground. Each shaft is radially drilled and tapped with twelve M5 x 12mm holes for location on the shaft support rails.

The load carriage consists of an aluminium top plate which is used to locate the ball bushing pillar blocks which carry the load.

The carriage is made from four open super ball bushings and pillar blocks 'SPB.16.OPN', as manufactured by RHP, which locate under the aluminium top plate. The ball bushing operating principle consists of three or more oblong circuits of balls within the bearing, each of which has the balls in one of its straight sides in bearing contact between the inner surface of the ball bushing sleeve and the shaft. As movement between the ball bushing and shaft takes place, ball recirculation is produced: the load being rolled along on the balls in the loaded portion of the circuit (see Figure [5.1.4.2]). The ball bushings are mounted in aluminium pillar blocks and are of the open type to locate over the shaft support rails. This type of bushing will compensate automatically for slight misalignment. Full details and the specification of the bushing is given in the RHP booklet [1983].

The final component of the load carriage is the aluminium adaptor bracket which attaches the load carriage to the module. This consists of the right angled adaptor bracket, a coupling screw and a flexible coupling ring. The flexible coupling ring allows for slight misalignment between the module and load table, see Figure [5.1.4.1].

This design of load table provides a module test facility, from which the dynamic and static performance can be assessed. The super ball bushing and rail arrangement should ensure that the effects of external friction are

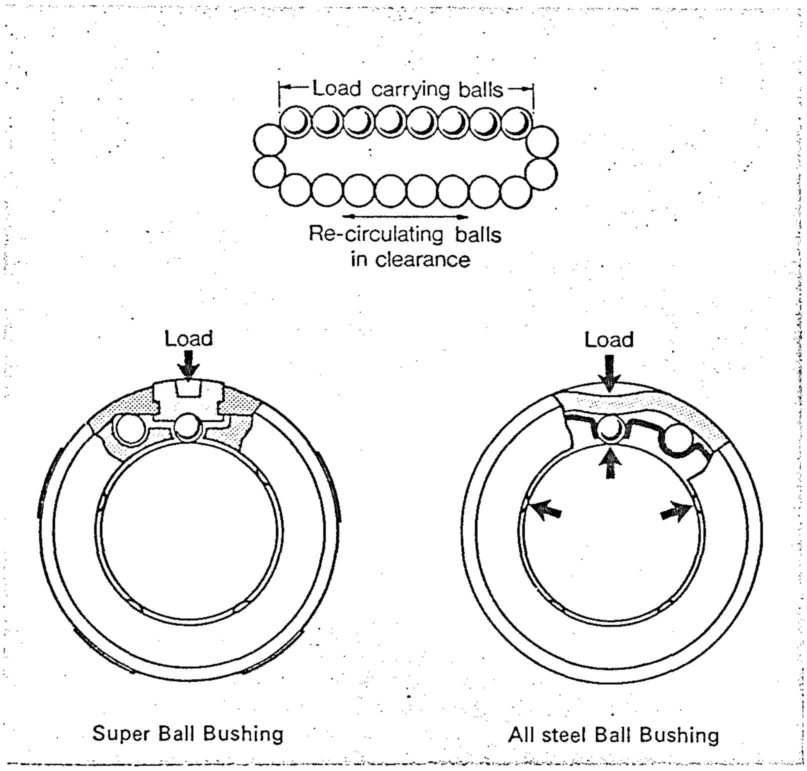


Figure (5.1.4.2) Ball Bushing

consistent and small in magnitude. The design of the carriage coupling arrangement and the bushing pillar blocks allows any slight misalignment between the module and the load table to be accomodated without influencing the performance. The main design criteria, for the load table, was to provide the ability to statically load a range of modules, where the load could be considered as having a purely inertial loading effect, without cantilever loading of the actuator or module slideway.

5.1.5. LOAD PULLEY

The 'load pulley', which is shown in Figure [5.1.5.1], is designed to provide 'axial' loading of the module under test. The test configuration is shown in Figure [8.5.1.1]. This test configuration allows assymetric axial loading of the module. A design criteria of the load test facilities was to allow the performance of position control strategies to be analysed with a variety of static loads, it was not designed to assess the mechanical rigidity and stiffness of the Martonair translation slideways.

The load pulley consists of a nylon cord, an adaptor plate to interface to the module, a load hanger and the pulley block. The pulley block consists of an aluminium U-bracket, for attachment to the test table and two support arms. Onto the support arms are mounted the two pulleys, as shown, which run in ball races. The perimeter of the smaller pulley is arranged to align axially with the module under examination.

5.1.6. EXTERNAL DISPLACEMENT TRANSDUCER

A linear displacement transducer is mounted to the surface of the test table. The transducer is set up so the moving head is offset but has parallel and axial alignment with the load table. It can be used to provide

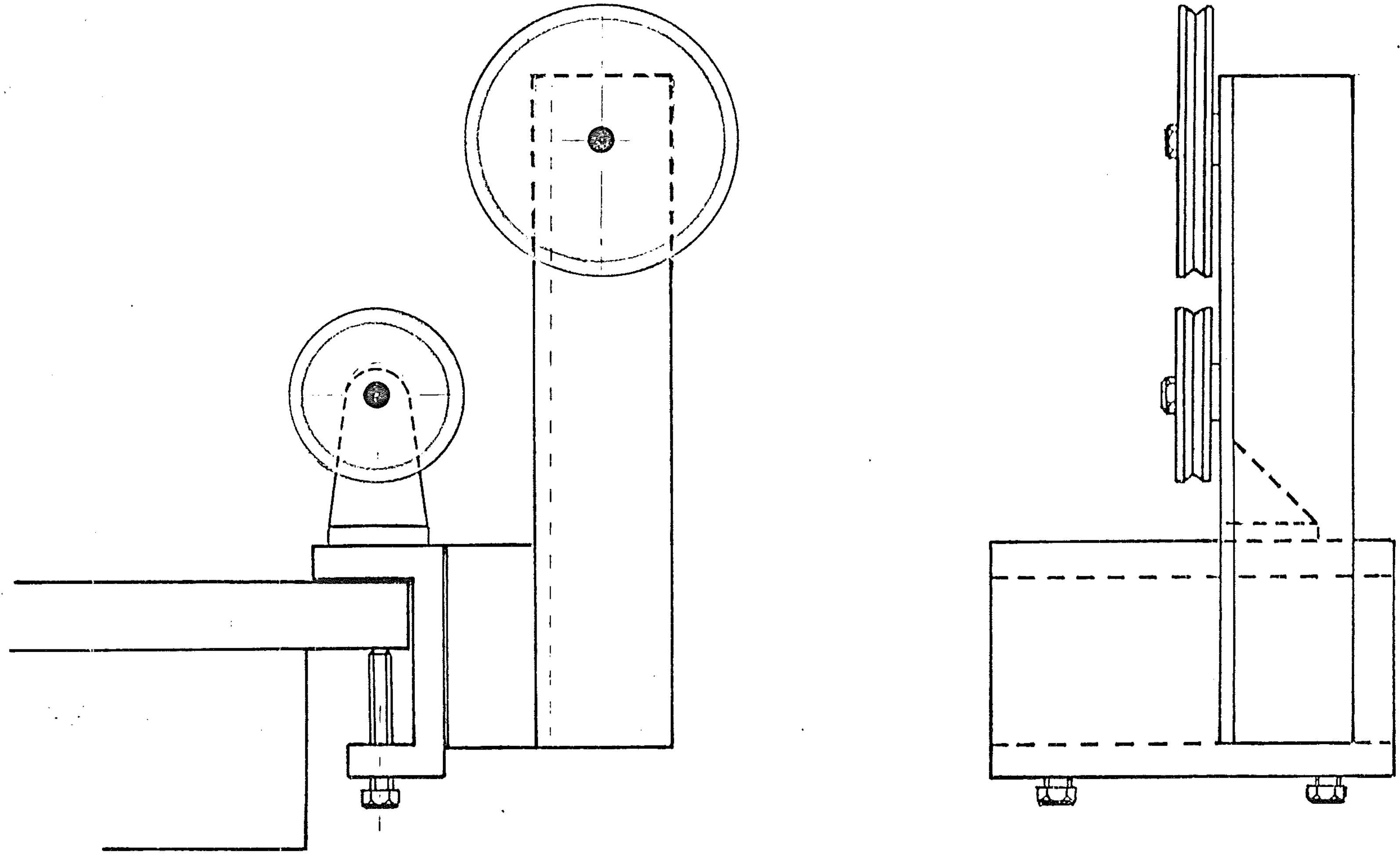


Figure (5.1.5.1) Load Pulley

a means of calibrating the position transducer and transmission system of any module under test. Alternatively it can be used during an evaluation, and having a resolution three times better than that of the conventional feedback system, can be used to provide improved data resolution for realtime control.

The linear transducer selected was a Heidenhain LS513 incremental displacement transducer. The movement of the module is transferred to the transducers' moving head by a coupling arm which directly connects between the module mounting surface and the transducer. The operating principle of the LS linear transducer is shown in Figure [5.1.6.1]. It is analogous to the rotary optical incremental encoder. A precision glass scale, manufactured using the Heidenhain diadur process, is the measuring standard: the LS513 scale has a line grating pitch of 40 microns. Measurement is achieved by photoelectric scanning of the scale via an index plate and an emitter and receiver pair. These receivers generate periodic signals when the scanning/moving head is displaced relative to the scale.

This scanning produces two quadrative trains of output pulses and a reference or marker pulse. The output pulses are in fact sinusoidal in form but are squared by signal conditioning electronics in the scanning head. The square wave quadrative pulse trains so produced can be processed in a similar manner to that used with rotary encoder. Using such an approach a position resolution of 10 microns can be obtained from the LS513 displacement transducer. A cross section through the transducer (Figure [5.1.6.2]) shows the internal details of the device, the scale is cemented to a steel carrier bar and is mounted in an aluminium extrusion which also acts as a guideway for the scanning head. To function correctly the scanning head, when mounted to some external carrier, must align precisely

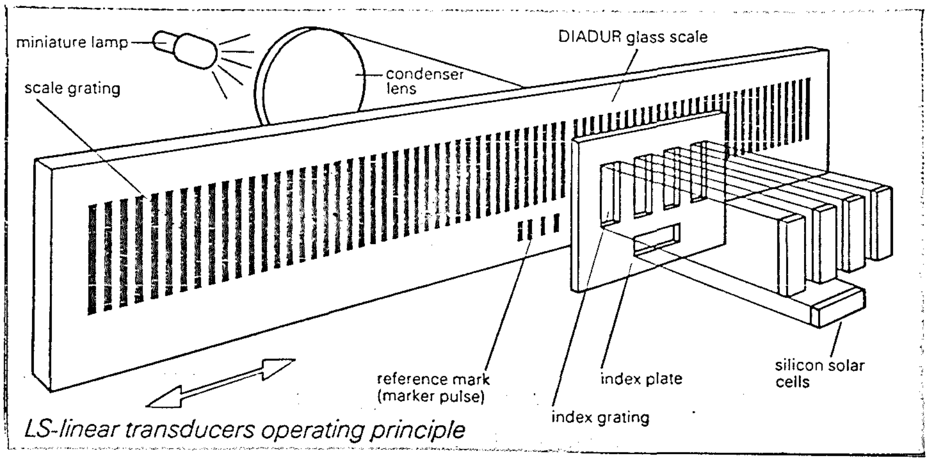


Figure (5.1.6.1) Linear Transducer Principle

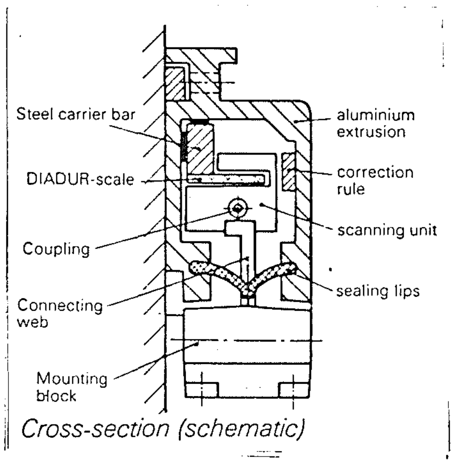


Figure (5.1.6.2) LS Linear Transducer Section with the scale. Full specification, mounting and operational details are given in the Heidenhain catalogue.

5.2. BRITISH STANDARDS BASED EVALUATIONS

It was necessary to devise a standard evaluation procedure which could be used to analyse and compare the performance of modules whilst making design changes in control strategy or control system hardware. In an attempt to control experimental conditions and maintain consistency and reliability of results a set procedure had to be adopted.

A British Standard, BS4151:1967, entitled 'Method of Evaluating Pneumatic Valve Positioners with Input Signal of 3 to 15psi (gauge)', was used as the basis for a test procedure designed to measure quantitatively the static performance of any single degree of freedom positioning system. The use of this procedure coupled with the use of standard measures of dynamic performance (relating to the transient response) has enabled data to be compiled and analysed to achieve the following;

(a) a comparative study of alternative realtime control strategies,

(b) a study of the robustness of a control strategy with respect to the drive system (eg change in actuator, valve, load and pneumatic operational conditions),

(c) performance comparisons to be made with respect to change in control system hardware elements (eg position transducer resolution, valve control resolution, etc), and

(d) assessments to be made with respect to the overall suitability of any particular control strategy for subsequent commercialisation.

BS4151:1967 relates specifically to pneumatic valve positioners, but the measures of performance defined by the standard in terms of 'linearity', 'repeatability' and 'hysteresis' coupled with the methodology defined for assessing such characteristics, is appropriate for the design and assessment of performance for any single degree of freedom positioning system.

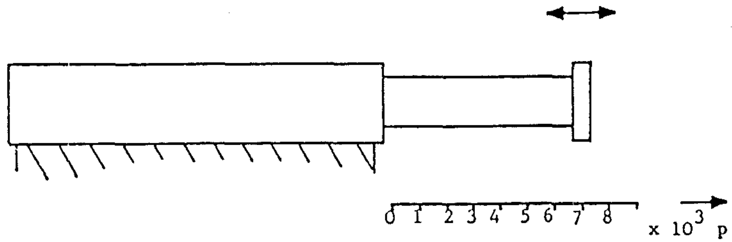
5.2.1. EVALUATION PROCEDURE

The evaluation procedure was designed in two sections; one to quantify the static performance using measures defined in the BS4151 and the other to quantify the dynamic response. The dynamic response test is less well defined in BS4151 thus a software based test procedure was devised: the measures of dynamic performance being chosen from those used in conventional control system transient response analysis.

5.2.1.1. STATIC PERFORMANCE

The static behaviour of the positioning system is assessed with the following test procedure, which is defined in BS4151: the available stroke of the actuator is divided into seven equal lengths, (see Figure [5.2.1.1]), by selecting eight appropriate 'set points'. The two outer set points are turning points and are used as a reference for change in direction during a test. The six inner set points are the active positions from which test data is compiled. The set-point sequence, which is programmed into the prototype microcomputer controller, involves 14 positions: this being necessary so that during one complete sequence each

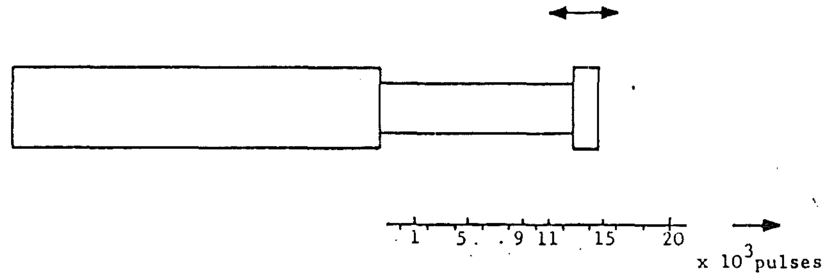
Figure (5.2.1.1) Module Setpoint Sequence



No	Setpoint Sequence (Pulses)	Comment
1	1000	Turning Point
2	2000	Test Point
3	3000	"
4	4000	"
5	5000	"
6	6000	"
7	7000	"
8	8000	Turning Point
9	7000	Test Point
10	6000	"
11	5000	"
12	4000	"
13	3000	"
14	2000	"
	REPEAT	REPEAT OR FINISH

And Evaluation Test Sequence - Static Performance

Figure (5.2.1.2) Module Setpoint Sequence
Evaluation Test Sequence - Dynamic Performance



No	Setpoint Sequence (Pulses)	Comment
1	1000	set-up position
2	20000	test point (+ve)
3	1000	test point (-ve)
4	5000	set-up position
5	15000	test point (+ve)
6	5000	test point (-ve)
7	9000	set-up position
8	11000	test point (+ve)
9	9000	test point (-ve)

active set point is approached in both directions. A typical test program sequence is illustrated in Figure [5.2.1.1].

Each test sequence is programmed so that it is repeated 'N' times (typically N=10) to produce statistically valid data. The evaluation software stores the 'desired position', the 'actual position' achieved and calculates the 'absolute error'. This data is stored for each set point, together with the positioning time, tp (which is evaluated using a specially developed software algorithm). At the conclusion of a test program the mean position for each set point is calculated, together with the position variance (the sum of the squares of absolute error) and the mean positioning time. The calculations are based on the following formula,

Mean position $\bar{x} = \sum x/N$ ---(5.2.1)

Position error variance

$$= \sum (x_d - x)^2 / N \qquad \text{---(5.2.2)}$$

Mean positioning time

$$\bar{tp} = \sum tp/N \qquad \text{---(5.2.3)}$$

where N = number of test repeats,

and x_d = setpoint

Software was also included to provide a summary of each test sequence with associated data for each test being stored in memory so that at the conclusion of a satisfactory test, summary tables can be obtained in hard copy form using a terminal printer. All the test data compiled and stored in memory can be accessed with use of a 'statistical data menu' which is one of the sub-level menus of the evaluation software. If an error occurs during a test sequence (for example due to an unacceptably long positioning time because of some drift problem) then an error flag will be indicated in the test summary tables.

Before an evaluation can take place it is necessary to set the experimental/control conditions and the control parameters associated with a control strategy.

When conducting performance testing the following steps are followed:

(a) down-loading to the target system of the appropriate version of real-time control software,

(b) setting the module 'attitude' and 'load condition',

(c) setting the desired air supply conditions,

(d) setting the correct valve 'dither' (if required),

(e) initialising the displacement transducer interface and 'nulling' the proportional valve,

and(f) programming the evaluation set-point sequence and initialising any measurement instrumentation.

The 'control parameters' required are dependant upon the real-time strategy being employed, ie each control strategy has characteristic loop gains, sampling intervals, etc which must be defined. The design of the control strategy and control system gains must be known so that the appropriate control parameters can be defined. A typical selection of control parameters would include:

(a) sampling interval,

(b) proportional error gain,

(c) saturation levels,

and (d) positioning 'dead-band'.

For an evaluation to be completed successfully 'control parameters' must be selected such that the module under test can always achieve the desired in-position tolerance and it should be able to reach the setpoint within a defined time period. If these criteria are not met then the evaluation becomes invalid as the 'position error variance', or the 'positioning time variance' will overflow. In either case an error flag will be placed in the summary data table.

5.2.1.2. DYNAMIC PERFORMANCE

The dynamic performance or transient response evaluation procedure was designed to compliment the static performance analysis procedure defined in BS4151. Analysis of control systems in the time domain via measurement of the transient response and other state variables (eg velocity) is well defined. It was necessary to devise a test sequence which could be run in 'single-step' mode to allow recordings and measurements to be taken representing a cross section of 'point-to-point' moves within different regions of the actuator stroke: thereby allowing the influence of 'length of stroke' and 'actuator position' on the transient response to be quantified.

The test sequence for a dynamic performance evaluation would, typically, involve six set-points and a program sequence of nine positions, which includes set-up positions. An example of selected set-points and the program test sequence is given in Figure [5.2.1.2] for a module stroke length of 21000 pulses. In a dynamic response test the moves can be categorised in three classes dependant upon the stroke length;

(a) long moves(>= 75% full stroke)

(b) medium moves(>= 25% full stroke & <= 50% full stroke)

(c) short moves(<= 25% full stroke)

The evaluation software associated with dynamic response testing carries out several functions. The system calculates the velocity of the module every sampling interval, stores the maximum velocity attained during any single move displaying the value and calculates and stores any other desired performance indice. During each position control cycle, the evaluation system stores data (associated with a sampling interval) which includes command signal, velocity, acceleration, proportional term, and position error of the module under test. This data can be accessed from the 'statistics' menu and a hard copy taken.

This data can subsequently be used to give the relationship between the control signal and the state variables of the module, assuming that values of velocity and acceleration calculated are representative of the true states. This data can also be used to confirm or otherwise the integrity of a control strategy.

5.2.2. QUANTATIVE PERFORMANCE MEASURES

A range of quantative performance measures have been adopted from BS4151 for the analysis of static performance data and other conventional measures used to analyse the dynamic performance data. The quantative measures used to analyse the static performance data are defined below, these are;

(a) Average departure from linearity

This is defined as the arithmetic mean of the deviation from linearity at each point of measurement and is considered seperately for positive and negative moves. The reference for linearity is defined as the straight line between the points corresponding to the average of the ten position repeats at the innermost set-point and the point corresponding to the average of the ten position repeats at the outermost set-point.

Average departure from linearity is expressed as one value for all positive moves, and one value for all negative moves and has dimensions of millimetres. This measure of linearity is normally obtained by plotting the mean position error against set-point and constructing a 'linearity line'.

$$\text{Linearity} = [\sum(x_l - \bar{x})]/N \qquad ---(5.2.2.1)$$

where x_l is position defined by linearity line

and $\bar{x}$ is mean position achieved

(b) Hysteresis

This is defined as the algebraic difference between the two arithmetic means of the deviations from linearity found for the same set-point when approached from opposite directions, and is expressed in millimetres. This measure of hysteresis is normally obtained from the mean position error against set-point plot, hysteresis is the mean outstroke error minus the mean instroke error.

$$\text{Hysteresis} = \{\sum(\bar{x}_{is} - \bar{x}_{os})\}/N \qquad \text{---(5.2.2.2)}$$

(c) Repeatability

This is defined as the root mean square deviation from set-point for a sequence of positions where the sequence is repeated 'N' times, with values for both positive and negative moves and has dimensions of millimetres.

$$\text{Repeatability} = (\{\sum(x_d - \bar{x})^2\}/N)^{1/2} \qquad \text{---(5.2.2.3)}$$

where x_d is desired position (or set-point)

A profile of repeatability is obtained by plotting the position error standard deviation against set-point.

(d) Accuracy

This is defined as the maximum deviation from the set-point, for a sequence of positions where the sequence is repeated 'N' times. Accuracy has dimensions of millimetres.

(e) Deadband

This is defined as the region about a set-point within which the 'control system' can be defined as 'in-position'.

(f) Step resolution

This is defined as the minimum step change in the set-point to which the system can respond.

The quantative measures used for the analysis of the dynamic performance are defined below, these are

(a) Maximum Velocity

This is defined as the maximum velocity measured during any point to point move. It must be qualified in terms of length of move and load conditions.

(b) Mean Positioning Velocity (MPV)

This is defined as the average value of velocity attained during any given point-to-point move. It must be qualified in terms of length of move and load conditions.

(c) Positioning Time (tp)

This is defined as the total time taken to complete a given point-to-point move. It is used to indicate cycle times and is normally qualified in terms of length of move and the load conditions.

Additional quantitive measures (or indices) of dynamic response (that are conventionally used with relation to transient response) have been used. These indices are described below with reference to Figure [5.2.2.1] which shows a typical transient response. These indices include;

(a) time lag, tl - this is the time required to respond to a step command.

(b) delay time, td - this is the time required to reach 50% of the step command.

(c) rise time, tr - this is the time required to cover a pre-determined percentage of step (typically 10% to 90% for overdamped systems, and 0% to 100% for underdamped systems).

(d) peak time, tpk - this is the time required to reach the peak of the response.

(e) peak overshoot, mp- this is the percentage difference between the peak time response and

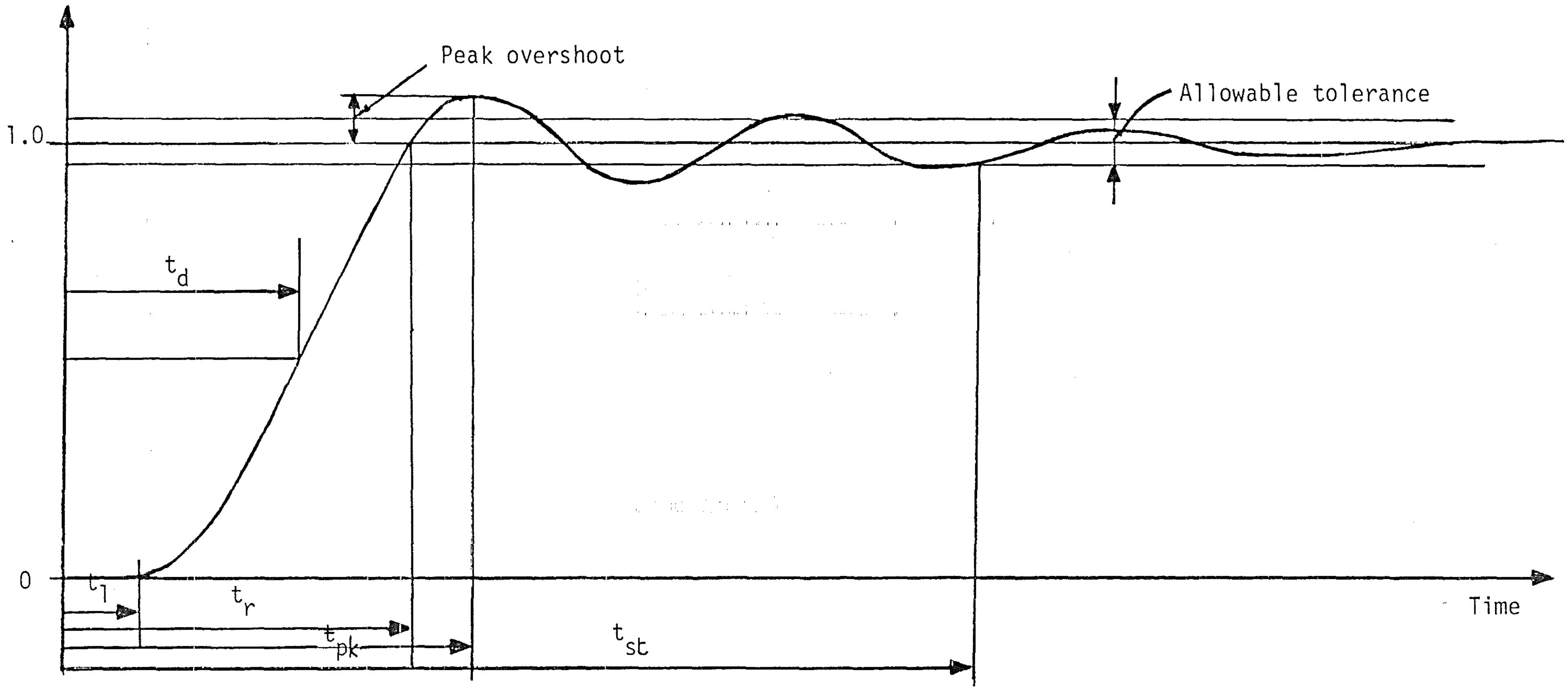


Figure (5.2.2.1) Transient Response Indices the steady state output.

(f) settling time, tst- this is the time required for the response to reach and stay within a specified tolerance (typically 2 or 5% of the step command).

Using such indices the transient response for a system can be defined. Other indices that can be used for analysis of system response include, Integral square error (ISE), Integral absolute error (IAE), Integral time absolute error (ITAE), and Integral time square error (ITSE).

Using any of these indices can simplify the design procedure by reducing the number of variables in response analysis to a straightforward procedure of seeking to minimise the applied index. In the analysis of transient response IAE has been used in addition to the other performance measures.

Any of the performance indices discussed can be used in 'learning' strategies where the indice is the measure of dynamic response. For any quantitive performance measure it is desirable for the indice to demonstrate properties such as sensitivity to parameter variation, selectiveness, and ease of implementation.

Measures of 'constancy' of the system, as defined in BS4151, were not considered to be within the scope of this research study at Loughborough, but this work has been undertaken by Martonair during pre-production trials.

5.3. EVALUATION SOFTWARE

The evaluation software described in this section was produced in linkable software modules to provide a partially automated test environment, which used in conjunction with test hardware, allows a comprehensive evaluation of position control systems to be carried out. The software achieves immediate analysis of some of the test data while other data is stored in memory for post analysis. All the evaluation software modules are written in Texas Instruments Assembler code which are then assembled to produce object code modules; the modules are then linked to produce the complete evaluation package. The modular configuration of the software allows the 'realtime control module' to be changed very easily, thereby facilitating assessment of alternative versions of control strategy. This structure also allows additional features to be incorporated within new modules, without the need for major changes to other stand alone modules. In this way the software development cycle is reduced and documentation is simplified.

The evaluation system was designed to produce statistically valid experimental data, whilst allowing the efficient design and analysis of pneumatic motion control systems.

The evaluation package as indicated is divided into modules which each achieve a specific function or utility. At a lower level the modules can be divided into sub-routines that process individual tasks.

5.3.1. MENU DRIVEN EVALUATION SYSTEM INTERFACE

The evaluation software is driven using a series of screen based menus to simplify the operational procedure for the user. The format of the menu structure is shown in Figure [5.3.1]. The operator enters a character dependant upon the facility required, the facility is executed or the next detailed menu associated with this facility is displayed. Once a facility has been executed it may be necessary for the operator to input more detailed information, for example in the selection of control system gains.

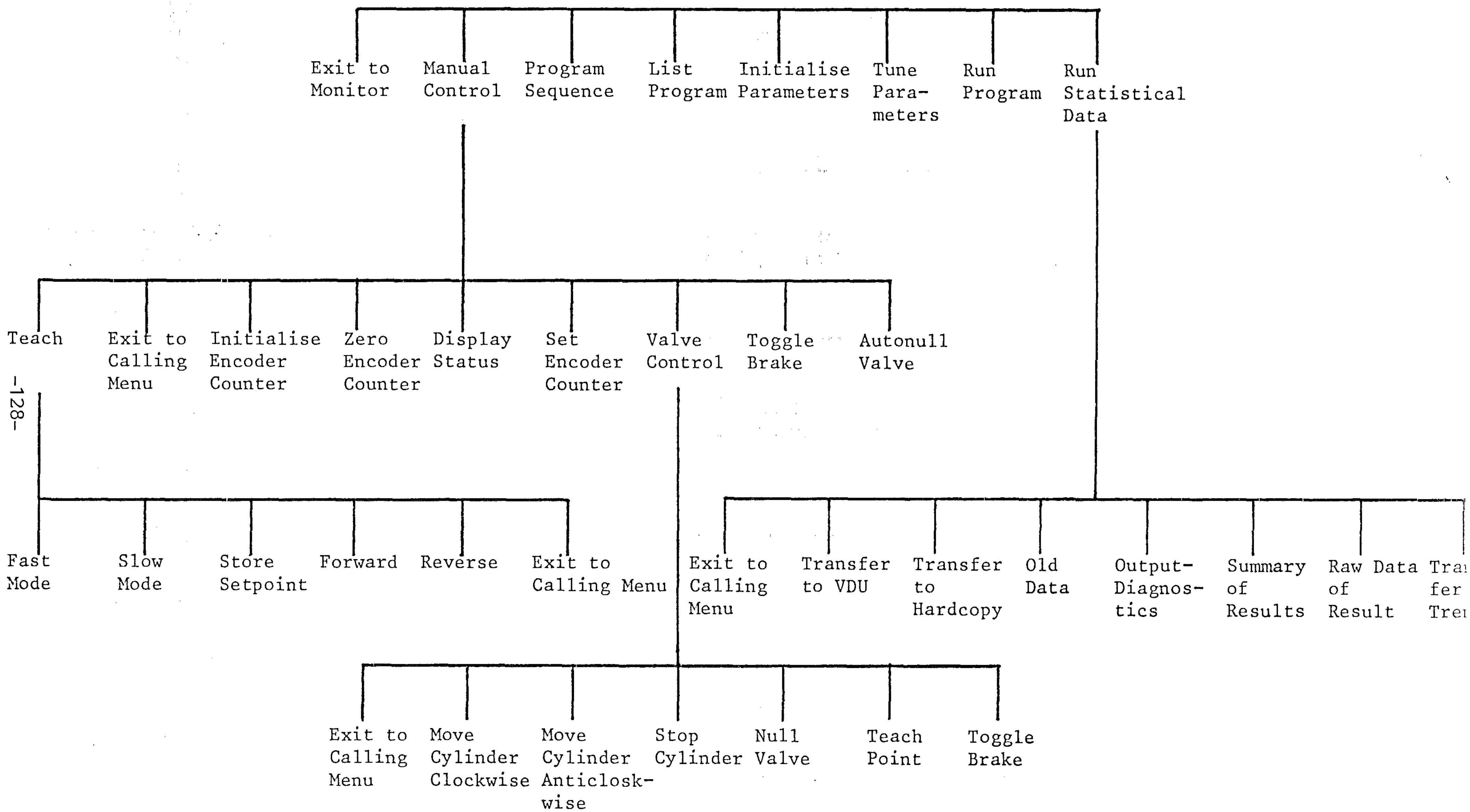


Figure (5.3.1) Menu Structure of Evaluation Software

If no operator input is required the facility when completed returns control to the calling menu.

This menu driven interface allows the operator access to any of the facilities in any sequence. In this way the operator can define the required program 'test sequence', the mode of operation and the number of sequence repeats and the associated control parameters. The operator can then initialise the encoder interface, 'null' the proportional valve, and also drive the module under open-loop conditions or in closed-loop 'teach' mode. Once the evaluation test program has been run, and the data has been compiled, this performance data can be processed under control of the 'statistical data' menu thereby allowing the data to be recorded and output in several formats as required.

In the automatic mode, testing is under control of the 'drive sequencing module' which issues 'desired position' commands for each move (Yd) on a handshake basis to the realtime control module: each new desired position command is transmitted on receipt of an 'in-position' response from the realtime control module. 'In-position' is signalled when the servo-mechanism has entered the controller deadband and maintains zero velocity for a fixed interval. The 'statistics' and 'diagnostics' module is used for the display, storage and limited statistical analysis of data relating to the static and dynamic performance of the drive. During a test run, a number of system variables (such as position error, mean positioning velocity, etc..) are displayed on the VDU while the instantaneous value of other variables (such as position, calculated velocity, controller output command, etc..) can be output as a function of time through digital to analogue converters to a display.

5.3.2. EVALUATION SOFTWARE MODULES

The evaluation software package was designed so that it can be used in a complete RAM memory environment, where the program would be downloaded from disc as required. Alternatively the same modules can be configured to produce an EPROM version. It was such a version of the evaluation software that was used to provide Martonair with an inhouse evaluation facility. The fully linked object code of the RAM version is known as : RAMPRG/OBJ, where the EPROM version is known as : ROMPRG/OBJ. They contain all the same modules but are configured differently to produce the desired ROM/RAM memory split. The software modules are listed in Table [5.3.2].

TABLE (5.3.2)

EVALUATION SOFTWARE MODULES	
:SEQ/SRC	Initialisation
:COMCLC/SRC	Communications Control
:COMIN/SRC	Communications Input
:COMOUT/SRC	Communications Output
:DIAG/SRC	Diagnostics Monitor
:INIT/SRC	Memory Initialisation
:RAM/SRC	Evaluation Variables
:ROM/SRC	Evaluation Constants
:RTC/SRC	Realtime Control
:STAT/SRC	Statistics Package
:SUA/SRC	)
:SUB/SRC	) General Sub-routines
:SUBC/SRC	)

The function of each module is described below:

(a) :SEQ/

This module contains a single sub-routine 'SET-UP' and is the first routine that runs once the evaluation program is executed. The module initialises the control system hardware to a known state, it resets any registers and interrupts vectors, it starts the interval timer and then determines the sense of the brake mechanism on the module.

(b) :COMCLC/

This module contains the data conversion sub-routines 'HEXDEC' and 'DECHEX'. These sub-routines are used by all the other modules which communicate data to or from the operator. The 'HEXDEC' routine takes an hexadecimal input and produces a decimal conversion. The 'DECHEX' routine takes a decimal input and produces a hexadecimal conversion.

(c) :COMIN/

This module contains the operator input communication sub-routines. These include the 'PROGRAM SEQUENCE' routine which is used to set the test program setpoint sequence; the 'PARAMETER TUNING' routine used to set and edit the control parameters associated with a particular control strategy; the 'CONTOUR EDIT' routine which can be used to generate/edit values in the proportional term-error contour; and the 'DECIMAL INPUT' routine which is used to process decimal input by the operator.

(d) :COMOUT/

This module contains the operator output communication sub-routines. Included here is a 'RTC DATA O/P' routine which outputs data in pseudo realtime to the VDU during each point-to-point move. Another sub-routine, 'MOVE SUMMARY', is used to output the error status at the end of a move. The final sub-routine in this module is the 'LIST PROGRAM' routine which lists the current set-point sequence.

(e) :DIAG/

This module contains a single sub-routine which collates the stored diagnostics data, concerning the last move, and outputs this data in the form of a table to the terminal or printer.

(f) :INIT/

This module contains a single sub-routine 'INITIALISE' and associated data which is necessary to initialise RAM memory areas. This ROM to RAM transfer is only necessary once after power up, and then only when working with the evaluation software residing in EPROM memory.

(g) :RAM/

This module contains all the data associated with the evaluation software and all the designated workspaces. The data includes default conditions for all registers, control parameters, and the proportional term error look-up table.

(h) :ROM/

This module contains operator message prompt data strings for the menus and error flags. This section can reside completely in a ROM area of memory.

(i) :RTC/

This module contains the 'REAL TIME CONTROL' sub-routine in which resides the control algorithm. This realtime control module is executed under the control of the interval timer and functions as an interrupt service routine. The control strategies implemented are detailed in Chapter (8).

(j) :STAT/

This module contains the sub-routines associated with storage and the processing of this test data. The module includes a 'DATA STORAGE' routine; a 'STATISTICS CALCULATION' routine; an 'OUTPUT RAW DATA' routine; and a 'DIAGNOSTICS OUTPUT' routine.

(k) :SUA/

This module contains a number of sub-routines for the control of the menu system which are related to one another: these include a 'MENU SPACE' routine; a 'MENU REPLY & SELECTION' routine; and other input control functions.

(l) :SUB/

This module contains a number of sub-routines, they include the following:

(i) the 'SET INTERVAL TIMER' routine,

(ii) the 'BRAKE TOGGLE' routine,

(iii) the 'DAC OUTPUT' routine,

(iv) the 'COUNTER READ' routine,

(v) the 'RUN PROGRAM' routine,

(vi) the 'COUNTER INITIALISE' routine,

(vii) the 'COUNTER ZERO' routine,

(viii) the 'NULL VALVE' routine,

(ix) the 'DISPLAY STATUS' routine,

and (x) a 'MANUAL VALVE CONTROL' routine which is used to control the proportional valve in open-loop mode.

(m) :SUBC/

This module contains additional sub-routines that have been included in the suite of evaluation software. These include:

(i) the 'AUTO VALVE NULL' routine which, with the module somewhere in mid-stroke, will drive the module a set distance forward and then reverse, and from this determine the 'null' of the proportional valve, and

(ii) the 'TEACH' routine which will drive the module under closed-loop control, the speed being determined by the size of set-point step increment, which is set by the operator from the keyboard (fast [f] or slow [s]). Using the 'TEACH' routine the module under test is driven forward or in reverse from the keyboard, and will stop when no input is given. At any point the axis position can be loaded into the set-point program table thereby providing a means of teaching a sequence of set-points.

The evaluation software is fully detailed and flowcharted in Moore [1986].

CHAPTER 6

PNEUMATIC PROPORTIONAL SERVO-VALVES

6. INTRODUCTION

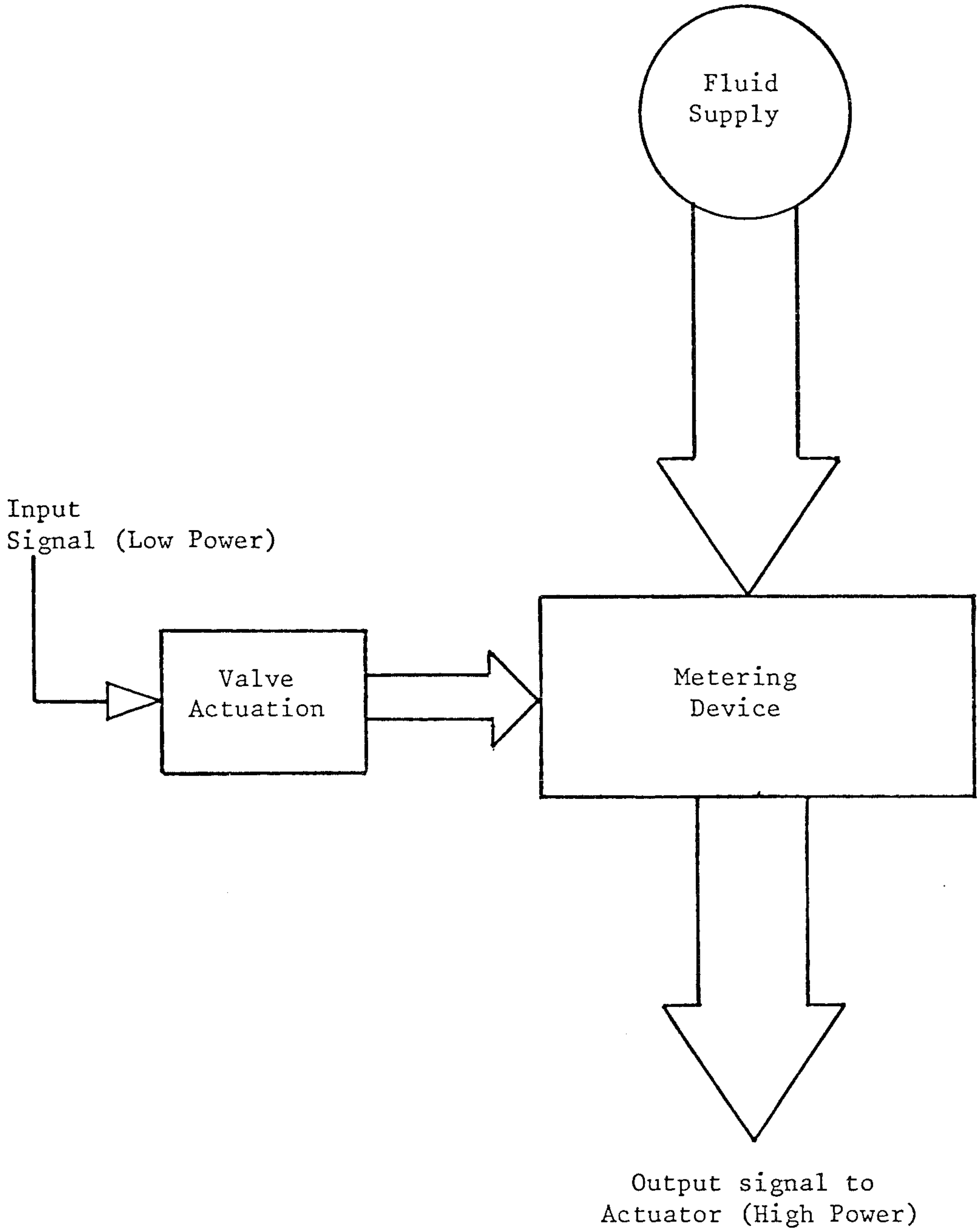
In this chapter the design and use of a 'proportional servo-valve' will be considered. In a fluid power system the proportional servo-valve provides direction and flow control of the working fluid. The design of various types of proportional servo-valve and their characteristics have been assessed by Turnbull [1976]. Control of pneumatic servo-drives is always achieved by means of servo-valve as variable displacement compressors cannot be used.

6.1. TYPES OF PROPORTIONAL SERVO-VALVE

Figure [6.1] illustrates the elements of the proportional valve, which are necessary irrespective of the working medium. The 'input signal' which is normally of low power, is provided by the control system, and is normally an electrical quantity (current or voltage) but it can also take a mechanical form. The 'valve actuator or manipulator' converts the 'input signal' into a mechanical displacement of the 'metering device'. In a conventional valve the electric signal is amplified and converted into a fluid power signal by the 'valve actuator' and 'metering device' The 'valve actuation' can take many forms and these include torque motors, proportional solenoids,and stepper motors.

The metering device controls the flow rate and the direction of the working fluid. It can consist of many different designs, but more fundamentally it can meter the working fluid in one or two directions

Figure (6.1) Proportional Valve Elements



dependant upon the category of the valve (three of five port). Conventionally a five-port valve is necessary to control a double actuating cylinder: a supply port, two exhaust ports, and two control ports (one for each control volume). A three port valve can be used to control a single acting cylinder, or one side of a double acting cylinder.

The 'metering device' design normally determines the valves' characteristics and the classification of the valve. Particularly when a hydraulic working fluid is used, the 'metering device' will often be designed as a multi-stage unit, where at each stage further amplification of the 'output signal' will take place. This is necessary due to the high flow forces that can be present when working with hydraulic fluids at high operating pressures, (Burrows [1972]).

It is possible to classify most types of proportional valve by their basic design, which depends mainly on their operating principle and to a lesser extent, on their geometry (McCloy and Martin [1980]). One of the more conventional classifications that has been adopted is as follows:

(i) Single stage valves, which include;

(a) Spool and sleeve,

(b) Rotary spool,

(c) Split spool,

(d) Sliding plate,

(e) Rotary plate, and

(f) Askania.

(ii) Two stage valves, which include;

(a) Double spool,

(b) Nozzle flapper spool,

(c) Double nozzle flapper spool,

(d) Plate spool, and

(e) Askania spool.

(iii) Three stage valves

In these specialist three stage valves an additional spool stage is normally added to the conventional two-stage valve.

6.2. SPOOL AND SLEEVE VALVES

The most commonly used modulating valve configuration is a spool and sleeve valve. The features of valves of this type are shown in Figure [6.2.1]. Here the spool is displaced relative to the sleeve by either mechanical means, a torque motor, a solenoid working against a spring, or a pair of solenoids in a push and pull mode, (normally spring centralised). The spool-sleeve valve may be used as part of a multi-stage valve.

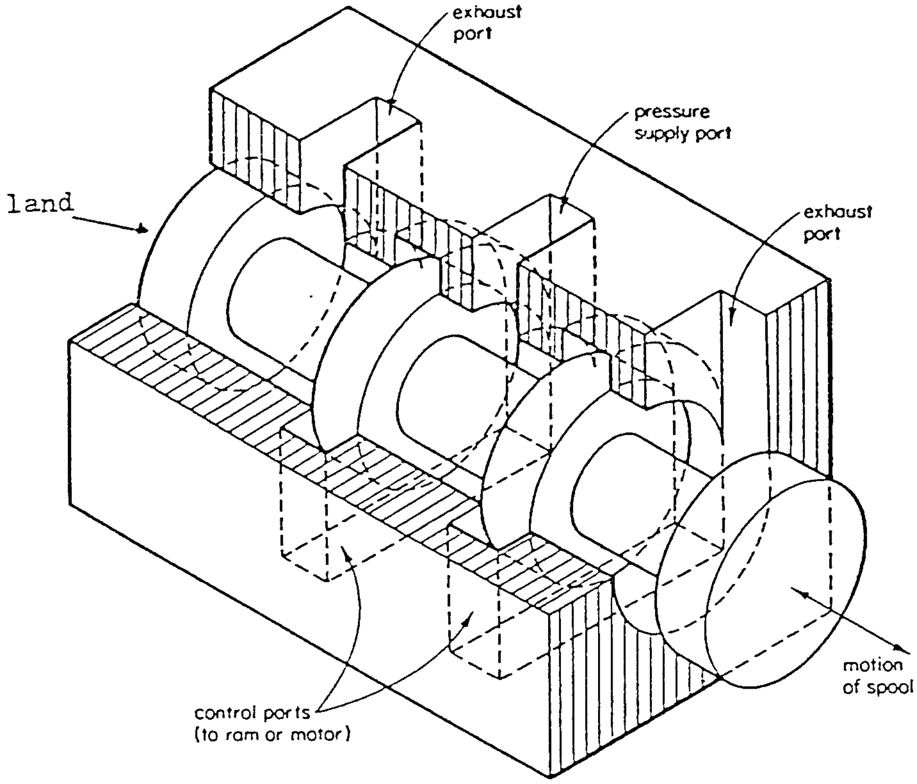
Axial movement of the spool covers and uncovers interconnecting ports. Port covering is provided by lands, or full diameter sections on the spool with intervening waist sections which provide port interconnection through the sleeve.

Spool-sleeve valves can be made with the spool lands either "underlapped", "overlapped", or "critically lapped" with their associated sleeve ports. Critical lapping is also known as a zero-lap condition. The effect of land lap on valve characteristics is discussed by Botting et al [1970], and its ultimate effect on valve design is considered.

6.2.1. SPOOL-SLEEVE SERVO-VALVE CHARACTERISTICS

Spool-sleeve combinations are generally designed to give a linear characteristic such that the flow rate from the valve is directly proportional to the input signal or spool displacement. A typical

Figure (6.2.1) Spool and Sleeve Valve



'spool-sleeve' valve arrangement is shown in Figure [6.2.1].

The valve flow rate 'Q' is a function of the following:

$$Q = f\,(X, Pv, w) \qquad \text{------}(6.2.1)$$

where 'Q' is the valve flow rate

'X' is the valve spool displacement

'Pv' is the pressure drop across the control ports

and 'w' is a parameter relating the area of the port to the valve travel

If the valve is to have a linear flow characteristic, then it is necessary for 'w' to remain constant. This implies rectangular or annular ports are necessary. However, valves with circular ports are easy to manufacture.

It is often necessary to express the characteristics of the servo-valve, and it is convenient to express in terms of valve coefficients, for example;

$$Gq = \left.\frac{\delta Q}{\delta X}\right|_{Pi}$$ - Volumetric flow gain

$$Gp = \left.\frac{\delta P}{\delta X}\right|_{Qi}$$ - Pressure gain or pressure sensitivity

$$Gqp = \left.\frac{\delta Q}{\delta P}\right|_{Xi}$$ - Volumetric flow pressure coefficient

$$C1 = \left.\frac{\delta \dot{M}}{\delta X}\right|_{Pi}$$ - Mass flow gain

$$C2 = \left.\frac{\delta \dot{M}}{\delta P}\right|_{Xi}$$ - Mass flow-pressure coefficient

In pneumatic systems the power medium is compressible as opposed to a hydraulic system where the fluid can often be considered to be incompressible: this accounts for differences in the behaviour of pneumatic and hydraulic systems. For example, when air is introduced into a cylinder when the valve is opened, then sufficient gas must flow into the cylinder chamber to increase the pressure and build up the force to overcome any static friction. In hydraulic systems a very small valve displacement will cause an almost immediate pressure rise in the cylinder.

Hence hydraulic systems have a rapid initial response whilst the pneumatic systems exhibit significant time delays. The pneumatic system will also exhibit limited stiffness, especially to external load disturbances. Pneumatic systems are sensitive to friction as a result of the time taken for pressure build up due to the compressibility. The effect of coulomb friction on pneumatic systems was demonstrated by Shearer [1954]. It was shown that low levels of coulomb friction were sufficient to cause a significant dwell in output during the reversal of the actuator. One possible solution is to use valve dither to overcome coulomb friction acting on the load and valve (Hattori [1980]). This will be discussed in a later section of this chapter.

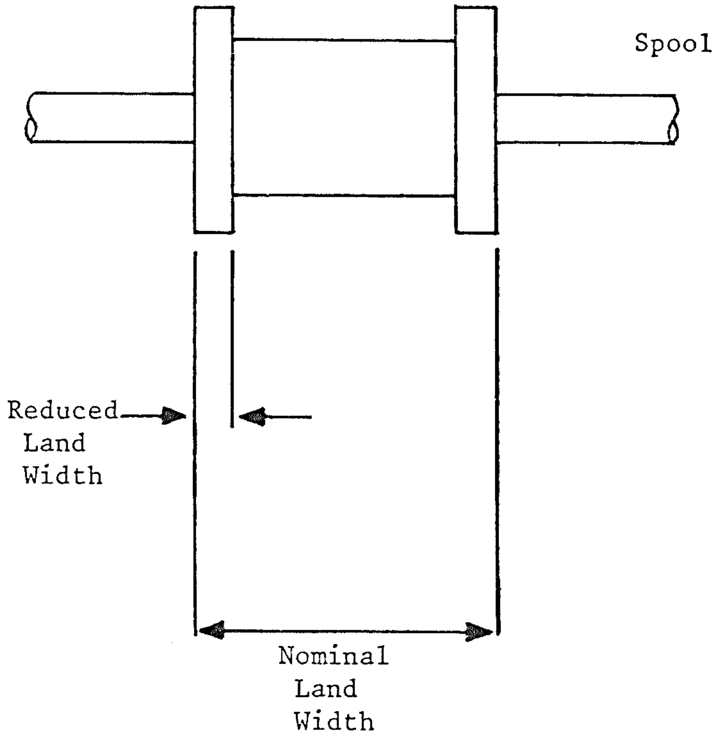
6.2.2. VALVE SPOOL FRICTION

The power and force available to move the valve spool within the sleeve are normally small being limited by the desire to keep size, weight, and cost of the operating mechanism to a minimum. It is therefore essential that frictional forces are kept to a minimum. This is particularly important when using low pressure air as the working medium because the compressible nature of the fluid makes such systems very susceptible to the influence of frictional forces. It is necessary to ensure that there is adequate force available to accelerate the mass of the spool, and to overcome any viscous friction and reaction forces. With a pneumatic servo-valve the reaction forces, if the spool is designed symmetrically, should be quite low, and by definition of a gas medium the natural viscous friction will be quite small. Because of this a spool-sleeve valve for a pneumatic servo operating at low pressures (less than 10 bar) can function successfully using a 'single stage' design. This is obviously desirable as the simplicity of the valve is maintained and the cost kept to a minimum.

The term 'stiction' is normally used to describe the solid friction force which must be overcome before the spool will move and this is usually due to dirt particles, the lack of geometrical perfection, the quality of the surface finish, and the axial alignment of the spool and sleeve. Early investigations on the effects of dirt particles on the performance of the spool valve is discussed by Alcock [1947]. He shows that the valve will operate successfully even when particle size exceeds the radial clearance between the spool and sleeve, as long as no metal burrs or chips are present. Particularly, if dither is used to reduce friction effects it is important to ensure that excessive wear between the spool and sleeve does not take place.

Lack of lubrication in the working fluid can also cause poor characteristics. A simple method of reducing the stiction effects is to use narrow lands. Such an approach was used in the design of the Martonair proportional servo-valve used for the investigations described in this thesis. Eynon [1960] suggests a method for land width design, a typical spool land is shown in Figure [6.2.2.1].

Figure (6.2.2.1) Spool Land Design



6.3. PROPORTIONAL SERVO-VALVE ACTUATION

Single stage proportional servo-valve actuation can be produced in several ways, which include mechanical linkage, electro-magnetic force from an analog signal, and electro-magnetic force from a digital command.

Mechanical linkage is used in proportional valves used in 'pneumatic positioners', where the spool position is adjusted dependant upon the location of a lever feedback mechanism.

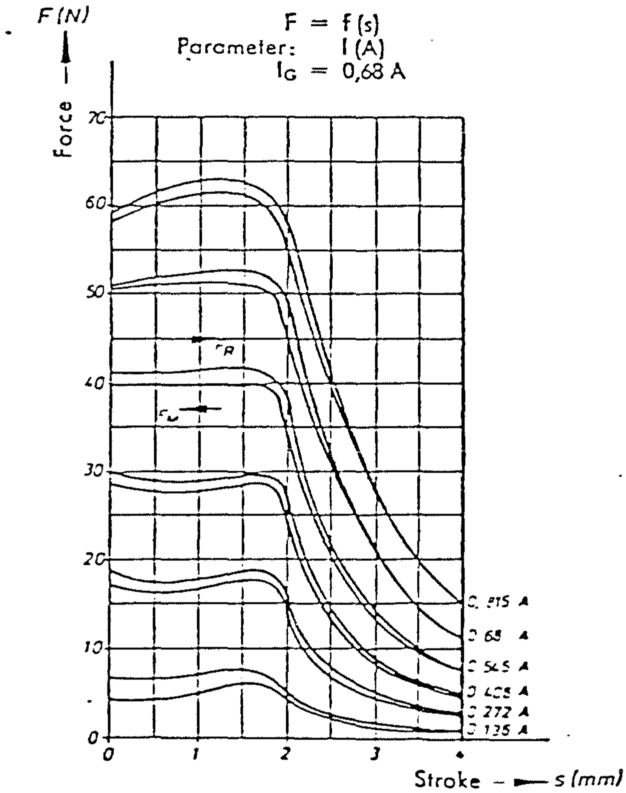
Electro-magnetic operation of a proportional valve can take one of several forms, a torque motor where the torque produced by the motor is proportional to the current applied, a proportional solenoid where the force produced is proportional to the current applied (analogues to a linear torque motor), and a conventional motor providing either torque or force (such as a stepper motor).

6.3.1. PROPORTIONAL SOLENOID

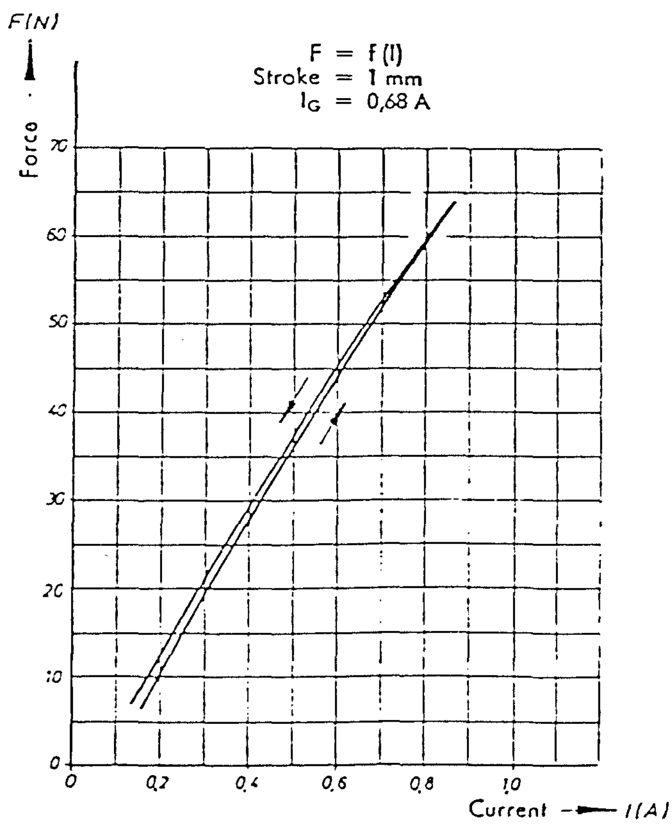
During this research study a proportional valve, using solenoid actuation of the spool, has evolved and is now marketed by Martonair (SPGB 18913). This solenoid is manufactured by the Emessem solenoid company, and is a dc wet pin regulating solenoid (Type GRFY 035). An oil immersed (wet) solenoid (designed for regulating control valves at up to 350 bar), will provide a force in proportion to the current applied.

The solenoid is flange mounted to the valve body. The solenoid is single acting and operates in a push mode against a return spring in the valve. The solenoid operating characteristics are shown in Figures [6.3.1.1] (a), (b), and (c) where the force/stroke graph, force/current graph, and the force/time graph are depicted. The solenoid provides 4mm of stroke, but

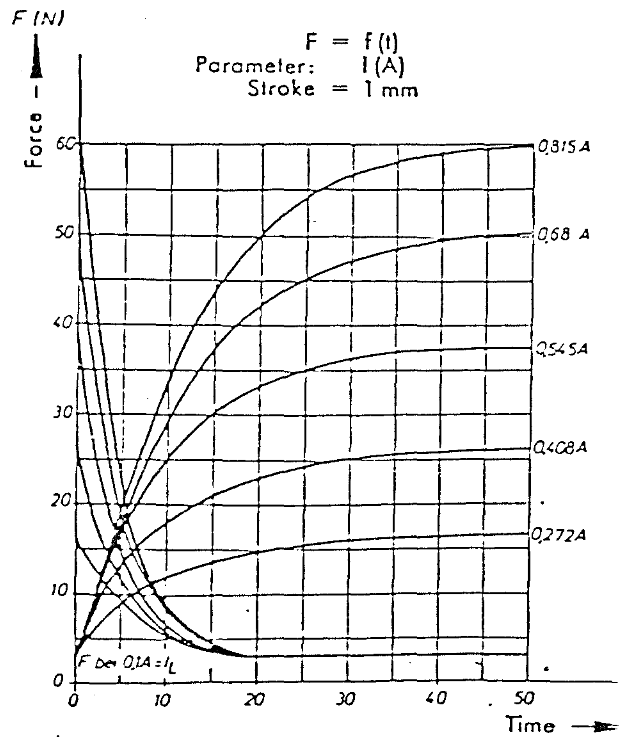
Figure (6.3.1.1) Proportional Solenoid Characteristics



(a)
Force/Stroke Graph



(b)
Force/Current Graph



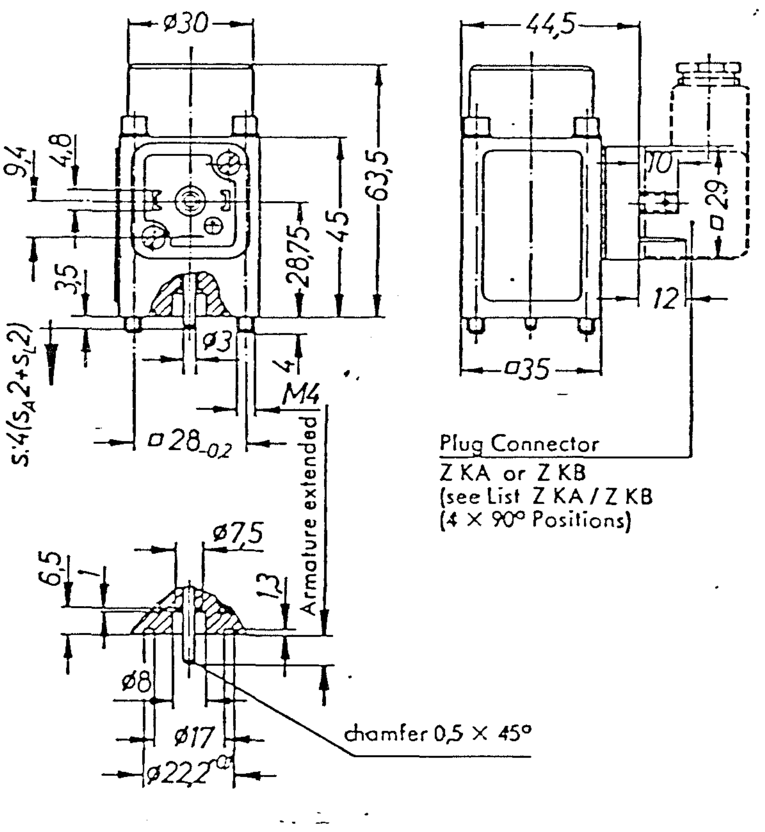
(c)
Force/Time Graph

only 2mm of the availale stroke provides a proportional force/current relationship. The available force becomes insignificant beyond the initial 2mm. The limited proportional stroke of the solenoid imposes a limitation on the valve operational characteristics. It imposes a saturation on the fully open port size of the valve (which limits the flow capacity of the valve). When the solenoid is fitted on the valve body,if it is not adjusted so that the valve 'null' occurs in the middle of the proportional range of the solenoid then an asymmetry in valve saturation levels will occur. This phenomena is illustrated in the transient responses discussed subsequently. The solenoid is rated to operate at up to 0.68 amps at between 24v and 30v dc, and has a specified hysteresis of 3% force and 3.5% current. The dimensions of the MSM solenoid are given in Figure [6.3.1.2]. The bandwidth of the proportional solenoid operated servo-valve with blocked load ports is about 60 hz for a constant amplitude input signal of 100 mA(p-p) as shown in Figure [6.3.1.3].

6.3.2. DIGITAL LINEAR STEPPER MOTOR

The proportional servo-valve can be used with an alternative form of actuation and during this research project a proportional valve using a digital linear actuator was constructed and tested. This digital linear actuator consists of a stepper motor which was modified to incorporate an internally threaded rotor fitted with a leadscrew shaft. When energised with a standard four phase sequence, bi-directional control of the shaft is obtained, with linear increments of 0.001 or 0.0005 inches depending on the translator mode used (half or full step actuation). Microstepping could be used to improve the resolution of spool positioning but this would include significant cost penalties.

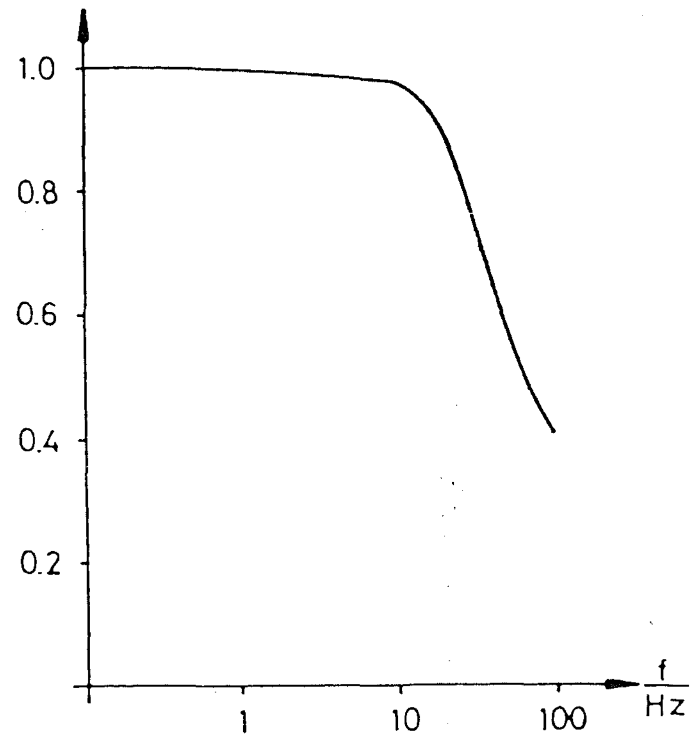
Figure (6.3.1.2) Proportional Solenoid



Type G RF Y 035 F 20 A 01

Dimension Table

Figure (6.3.1.3) Proportional Solenoid Actuated Valve Bandwidth



There is a wide range of manufacturers from which to select an appropriate proportional solenoid, but this does not appear to be the case for digital linear actuators (Ekere [1984]). A large number of manufacturers produce variants of the stepper motor, but it would have been necessary to develop a suitable rotary to linear transmission to make use of a standard stepper motor. It was decided to purchase a commercially available digital linear actuator which could be used directly when suitably mounted on the valve body.

The actuator selected for this study is manufactured by Airpax (USA), and distributed in the U.K. by Mclennan. From the 92000 series, a L92411 digital linear actuator was chosen. The actuator is shown in Plate [6.3.2]. A diagram of the valve and digital actuator is shown in Figure [6.3.2.1]. Martonair have also produced prototype variants of the digitally operated valve using the smaller L92121 and K92211 actuators, which can be driven directly from a proprietary I.C. driver. This has cost advantages but imposes limitations on the flexibility of the interface. To maximise flexibility when evolving motion control strategies a decision was made to purchase the more powerful L92411 actuator and to interface to the device using the Mclennan EM162 translator which enables higher step rates by employing drive pulse ramping techniques (using the Mclennan EM211 'pulse buffer module') and an L/2R drive configuration. Another advantage was that the translator allowed the motor to be controlled in half step mode, which was important in that it allowed an improvement in linear resolution (this being particularly important when close to the valve null).

The two methods used to interface to the digital linear actuators are shown in Figure [6.3.2.2], where mode (2) is used for the L94211 actuator. When used in full step mode the digital actuator gives a resolution of

PLATE (6.3.2) SERVO-VALVE ASSEMBLY - DIGITAL OPERATION

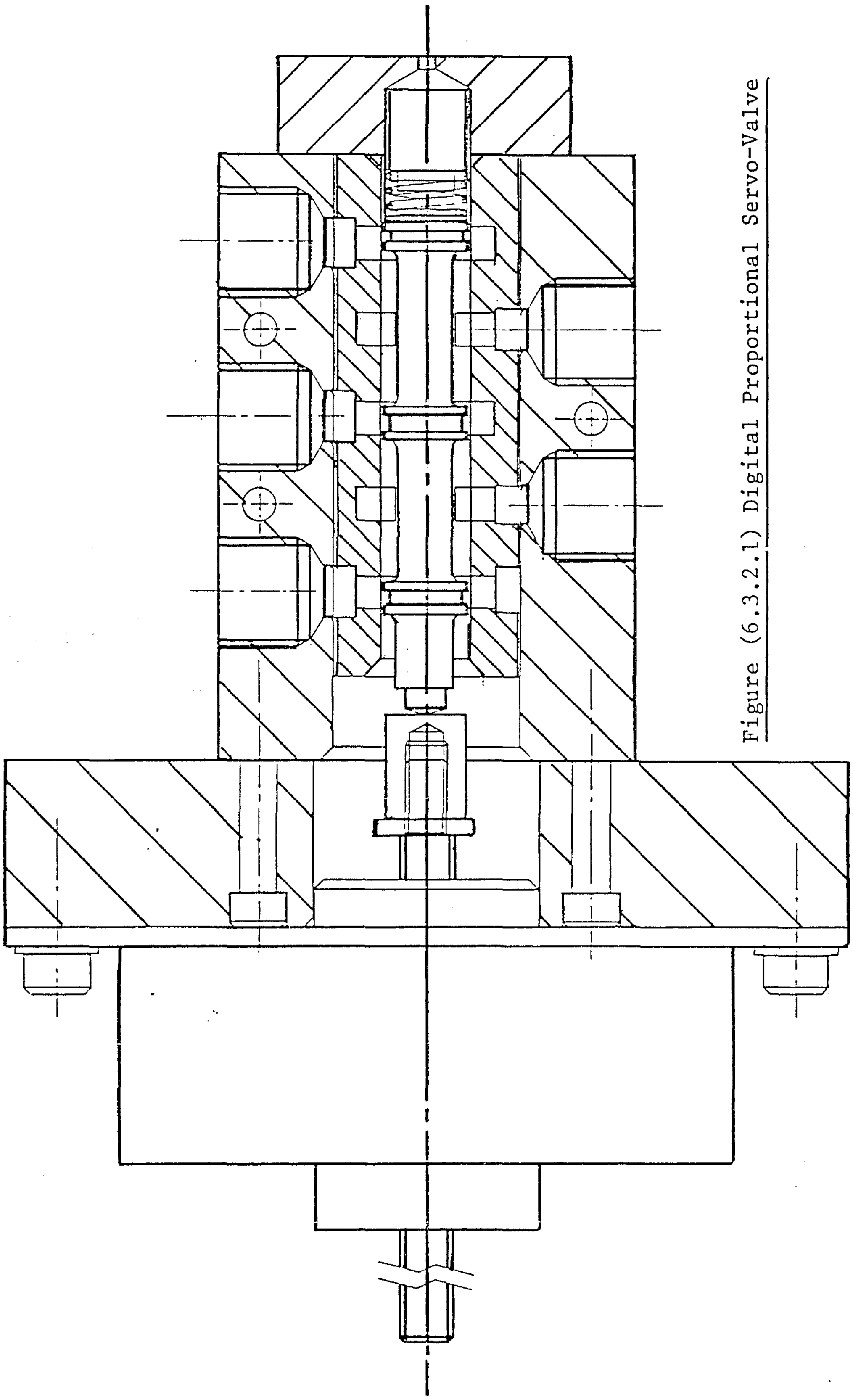
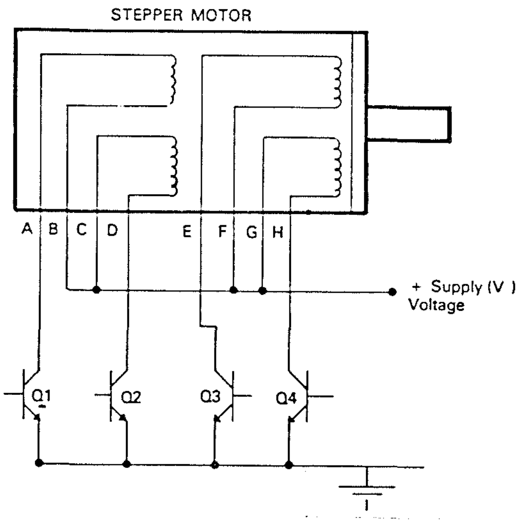


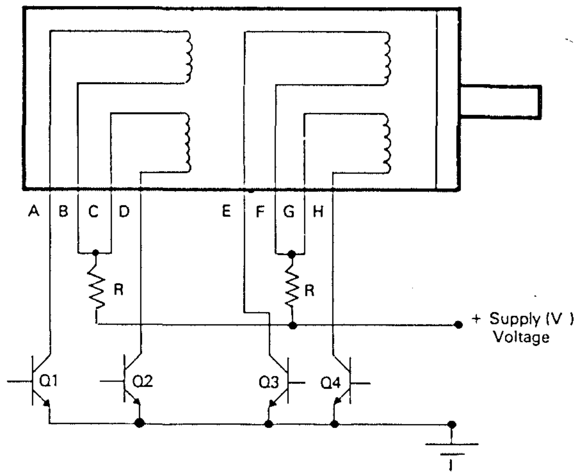
Figure (6.3.2.1) Digital Proportional Servo-Valve

Figure (6.3.2.2) Methods of Interfacing the Digital Linear Actuator

Mode 1



Mode 2



approximately 6 bits within the 2mm working range. This is improved to 7 bit resolution when using half-step mode. The digital actuator has reduced resolution when compared with the proportional solenoid, but it does enable the full flow characteristics of the valve to be achieved as sufficient stroke is available to fully displace the spool.

6.4. USE OF DITHER IN SERVO-VALVE ACTUATION

It has been found, by experimentation, that sluggish jerking motion within a position control system can be caused by non-linear friction forces within the positioning system (Hattori M.[1980]). When using fluidic actuators, such as proportional valves and cylinders, valve spool stiction and seal friction can be a major factor in producing non-linear friction characteristics. Other factors that also influence friction characteristics include the slideway bearings and the external load. With dc/ac servo motors the same effect is produced by varying torque components with changes in angular position (Hattori M [1980]).

The use of a relatively high frequency dither component superimposed upon the system input command signal, helps to significantly reduce the effects of non-linear friction forces upon the system. In this way the position control system is made more sensitive to changes in input commands, operating response speed is improved and hysteresis is substantially reduced in the valve.

The value of amplitude and frequency of the dither component is critical to its effectiveness upon the position control system. The dither source used in this research study is a sinusoidal waveform, the amplitude (a) necessarily produces a force component greater than is required to overcome the friction present within the overall positioning system. The frequency of the dither is necessarily greater than the natural frequency of the positioning system. In this research study dither was only used with the solenoid operated valve via the analog interface.

The value of the dither source used is in the region of 20 to 100 hz with 0.9 volts (peak to peak) amplitude. These values were determined by a variety of experimental tests carried out when using the servo-valve. The reduction in the level of hysteresis present was chosen as a measure of the effectiveness of the dither input amplitude and frequency setting. The reduction in hysteresis levels present within the valve indicates the degree to which stiction has been overcome. The optimal level of dither amplitude and frequency required will vary as the control system parameters are altered (eg valve actuator type, slideway type, load, etc).

Disadvantages of using dither include; (i) air consumption is increased, (ii) wear in the valve can result, and (iii) ambient noise levels are increased.

6.5. COMMERCIALLY AVAILABLE PNEUMATIC PROPORTIONAL SERVO-VALVES

Exploitation of pneumatic motion control systems has been restricted by a number of factors, as previously discussed. Of the limited attention such control systems have received much of the work involved the use of bang-bang motion control techniques (reviewed in Chapter (4)). The early studies on pneumatic proportional valve controlled servo-systems utilised either hydraulic system proportional valves or special purpose designed valves such as the plate valve (Shearer [1954]).

This trend has been a function of the limited commercial availability of suitable proportional servo-valves. Of the valves now available, selection is limited by valve design and characteristics, valve actuation and suitable interface electronics, cost, and application expertise. A brief review of the commercially available proportional valves known to the author is presented in Moore [1986].

6.5.1. SUMMARY OF COMMERCIAL PROPORTIONAL SERVO-VALVES SURVEY

A limited number of proportional servo-valves are now available that are suitable for use with pneumatic actuators (eg Bosch, Herion, Dynamic, PTM, SMC, etc..). Almost certainly a number of the hydraulic valve manufacturers worldwide must have or are considering producing pneumatic proportional valves. Of the proportional valves known to the author, the majority are not ideally suited for motion control systems, because of the valve configurations. At RWTH Aachen (WG) development of a fast response two stage pneumatic proportional valve (3 and 5 port) has proceeded (Backe et al).

During this research project, the industrial collaborating company, Martonair Ltd, developed a pneumatic proportional control valve, which was made available for use in the ensuing research study. This proportional valve is now available in commercial form from Martonair (No.SPGB 18913), as an industrial component. This valve, its characteristics and interface components are described in section (6.6). The development of this valve has been an integral part of this research study and various prototypes have been used and tested within the various motion control investigations carried out at LUT.

6.6. MARTONAIR PROPORTIONAL SERVO-VALVE

The Martonair proportional servo-valve was developed from a proportional valve used in a 'Pneumatic Positioner' device marketed by Martonair for use in process control applications, where the positioner moves relative to a pressure control signal.

6.6.1. PROPORTIONAL SERVO-VALVE DESCRIPTION

The 'proportional valve' SPGB 18913 is shown in Plates [6.6.1] and [6.6.2], where Plate [6.6.1] shows the complete valve assembly with solenoid operation, and Plate [6.6.2] shows the individual valve components. The valve in its conventional form is operated by a proportional solenoid, but alternative methods can be used. A section through the valve is shown in Figure [6.6.1.1].

This proportional valve is a 'single stage' device and consists of a glandless spool and sleeve design. The valve has five ports, two control ports, two exhaust ports, and a supply port. The valve is designed to work with dry or lubricated air with a maximum supply pressure of 10 bar, but this is more typically 6 bar. In the 'null' condition the valve is 'mid-centre closed', to reduce quiescent leakage, and the three spool 'lands' cover the 'supply' port and the two 'exhaust' ports. The valve lands are zero lapped in the standard version, flow control being obtained to either control port of the valve by displacement of the spool relative to the valve sleeve. In this way the supply orifice to either control volume can be adjusted, with the flow capacity being proportional to the spool displacement from 'null'.

The valve consists of the following components, which are shown in Figure [6.6.1.1]:

(i) The 'spool' which is a stainless steel cylindrical component with three control lands which are ground to very close diametrical tolerances. The valve relies on close tolerances between the spool and sleeve

PLATE (6.6.1) SERVO-VALVE ASSEMBLY - SOLENOID OPERATED

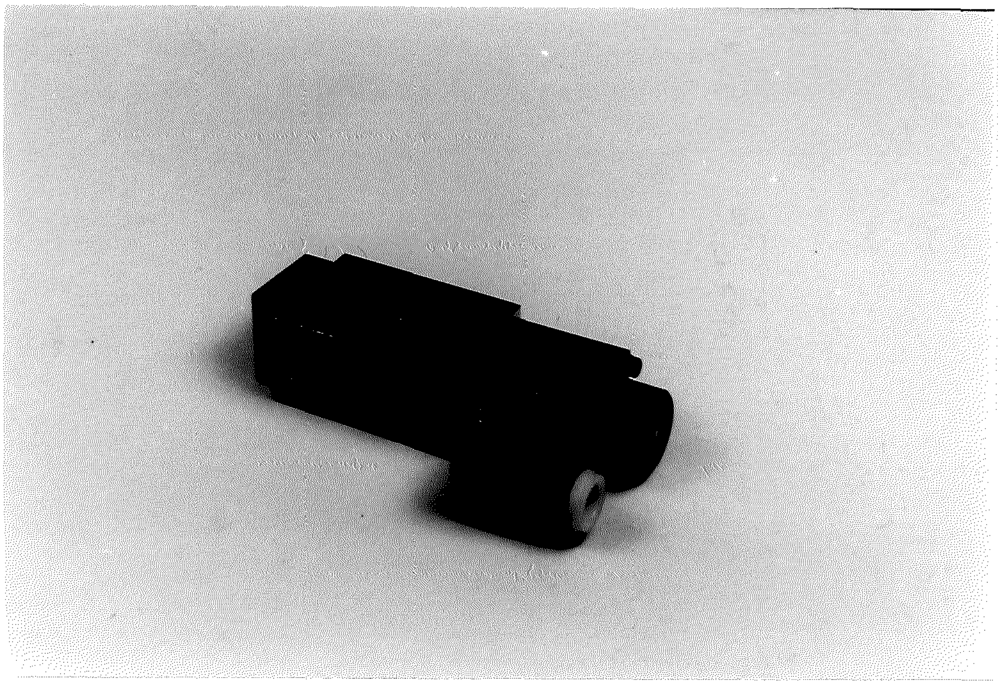


PLATE (6.6.2) SERVO-VALVE COMPONENTS

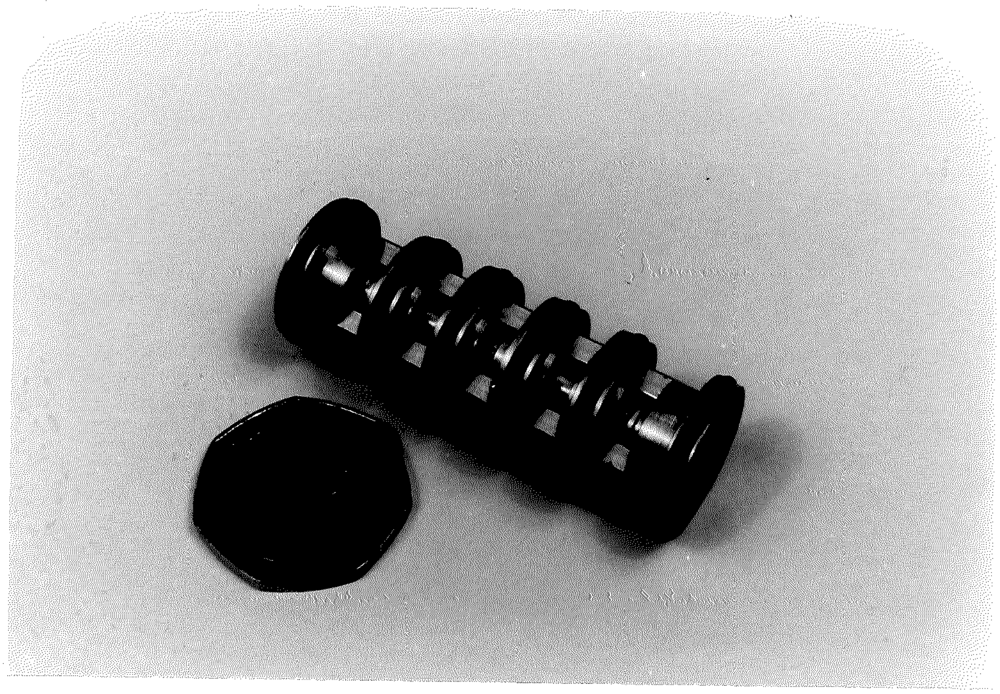
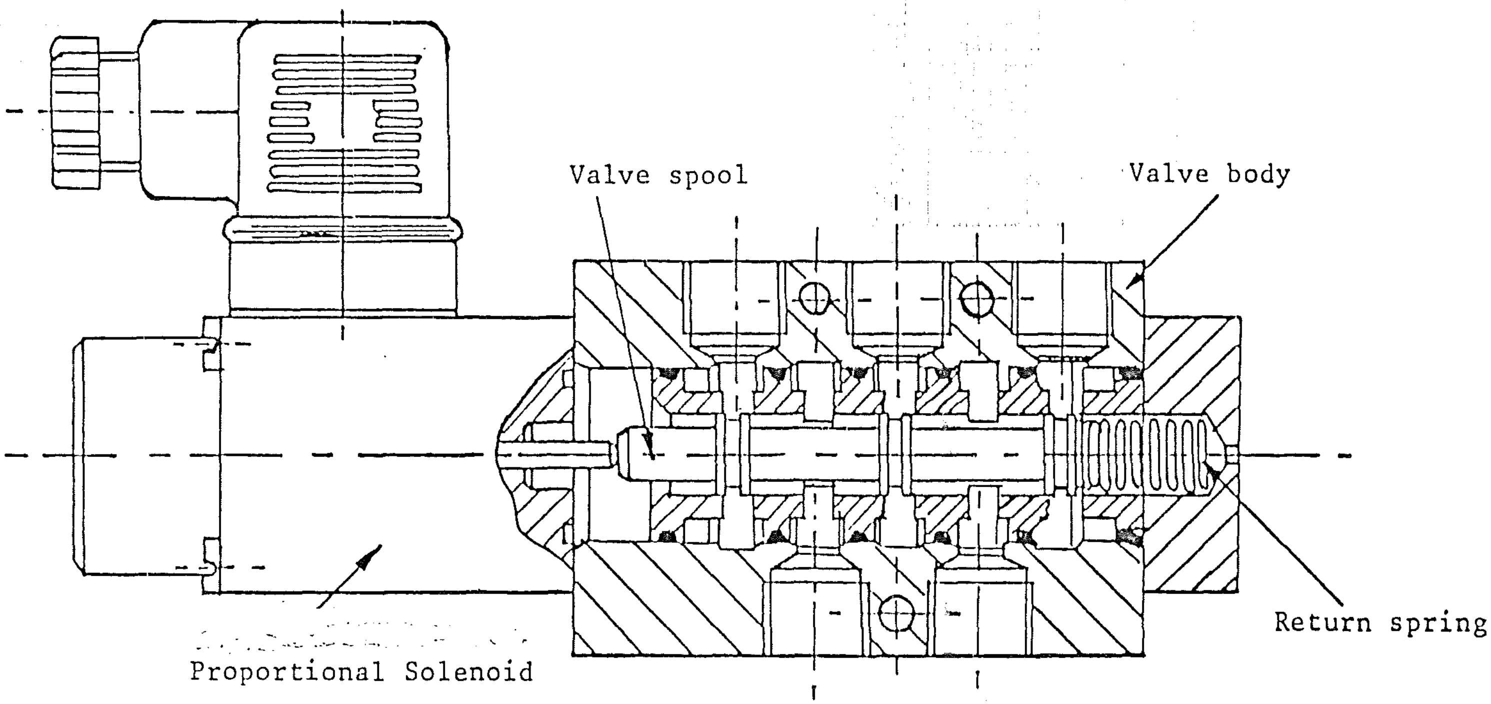


Figure (6.6.1.1) Mechanical Arrangement of the Proportional Valve



(12-14microns) to maintain the 'standing pressure' between the control volumes and minimise valve leakage. The control lands have recesses to reduce stiction between the spool and sleeve, by reducing the annular contact area. The spool is designed with a location for the centralising spring at one extreme and an endcap at the other extreme for the spool actuator. The spool has a hollow centre to reduce its mass. The spool (M/P22947) is shown in Plate [6.6.2].

(ii) The 'sleeve' which is a brass cylindrical component, contains the control ports of the valve, these control ports consist of two milled slots on opposite sides of the sleeve. The port size is adjusted by either altering the depth of the slot (which is very limited) or changing the width of the slot. The bore of the 'sleeve' is honed to obtain a high quality surface finish and a close diametric tolerance to match the spool. The outside of the 'sleeve' is recessed on an even pitch with 'o' ring location grooves. The 'o' rings provide a seal between the 'sleeve' and the valve housing. The 'sleeve' (M/P22946) is shown in Plate [6.6.2]. (The dimensions of a typical spool and sleeve are shown in Figure [6.6.1.2]).

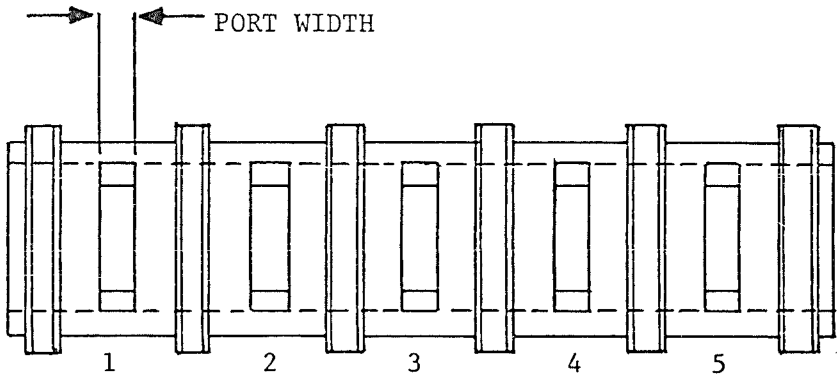
(iii) The 'valve housing' is a machined aluminium block that provides location for the 'sleeve' in its bore, location for the spool actuator, and the tapped holes for the pneumatic fittings which are positioned over the ports in the 'sleeve'. It also provides location for the end-plate. The valve housing is shown in Plate [6.6.2].

(iv) The 'return spring' is a helical compression spring, against which the spool actuator works. If a solenoid actuator is used, it is necessary to select the return spring so that the solenoid works in its proportional force range.

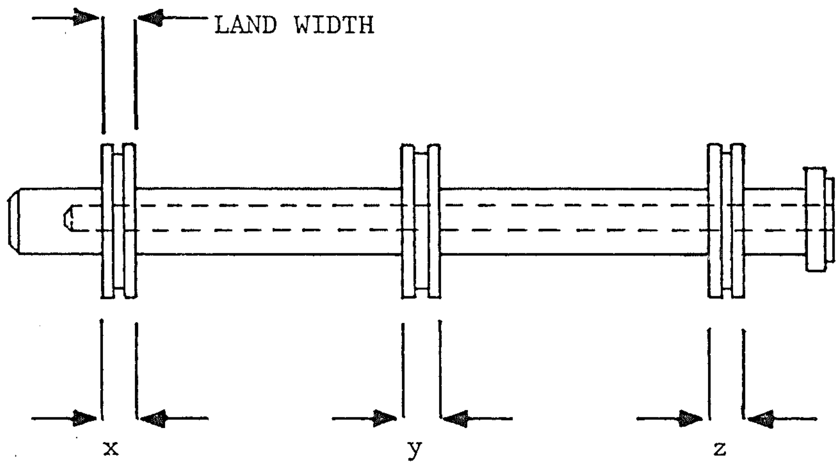
(v) The 'end plate' is used to retain the return spring in position. It

Figure (6.6.1.2) Proportional Valve - Spool/Sleeve Detail

SLEEVE



SPOOL



PORT WIDTH (mm)

Port Number	1	2	3	4	5
Size	4.040	4.020	4.008	4.009	4.062

LAND WIDTH (mm)

Land	x	y	z
Size	3.897	4.007	3.900

is shown in section in Figure [6.6.1.1]. The end plate also includes a pressure relief vent. This allows any back pressure from the exhaust port or leakage across the spool lands to escape to atmosphere.

6.6.2. PROPORTIONAL SERVO-VALVE PRESSURE AND FLOW CHARACTERISTICS

The operating characteristics of the proportional valve are conventionally represented by the 'pressure-displacement' and the 'flow-displacement' profiles. The pressure-displacement data will show 'pressure sensitivity' of the valve and the saturated 'standing pressure' of the valve. The flow-displacement data will show the maximum flow capacity for a given supply pressure, the linearity of the flow for each control port, and the valve flow gain. The manufacturers quote steady state valve co-efficients of;

(i) $G_p = 8 \times 10^9\ Nm^{-2}\ m^{-1}$, and

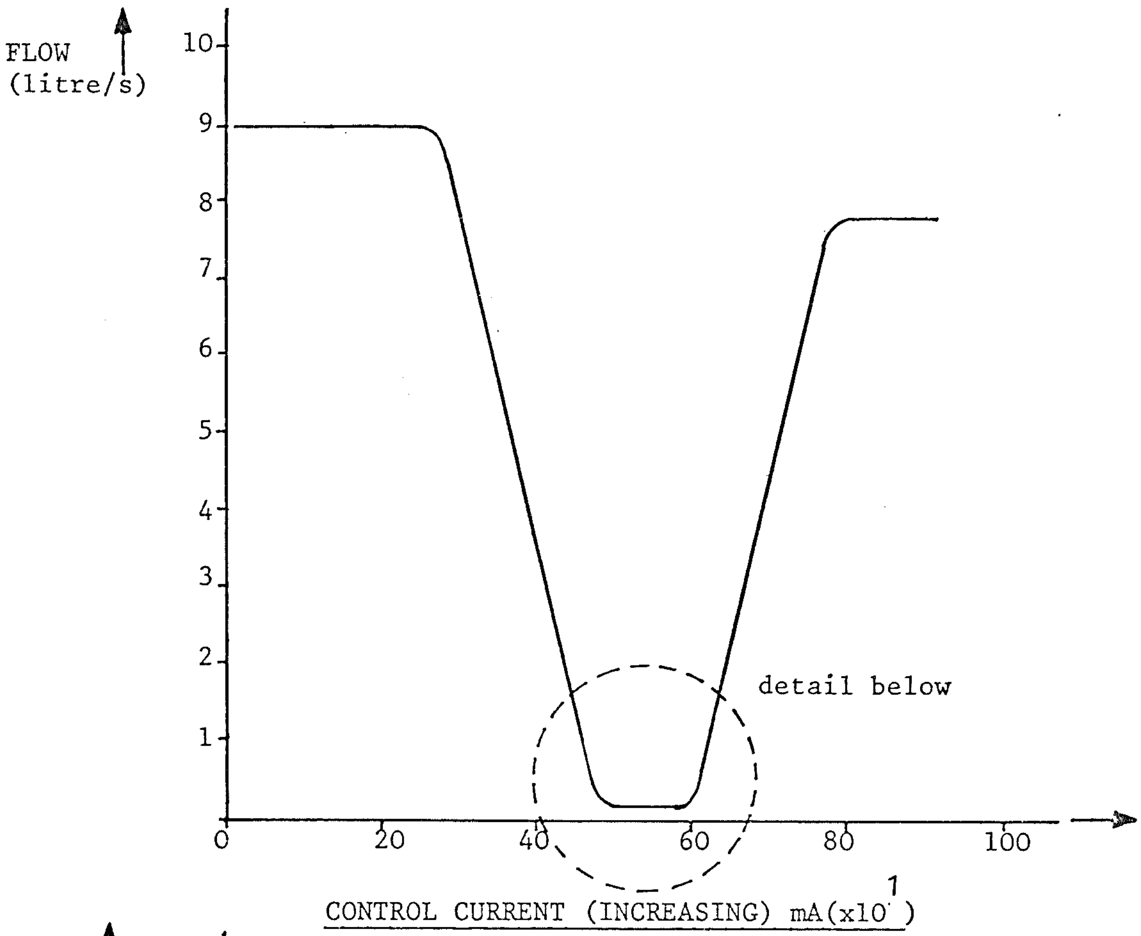
(ii) $G_q = 2.0\ m^3\ s^{-1}\ m^{-1}$

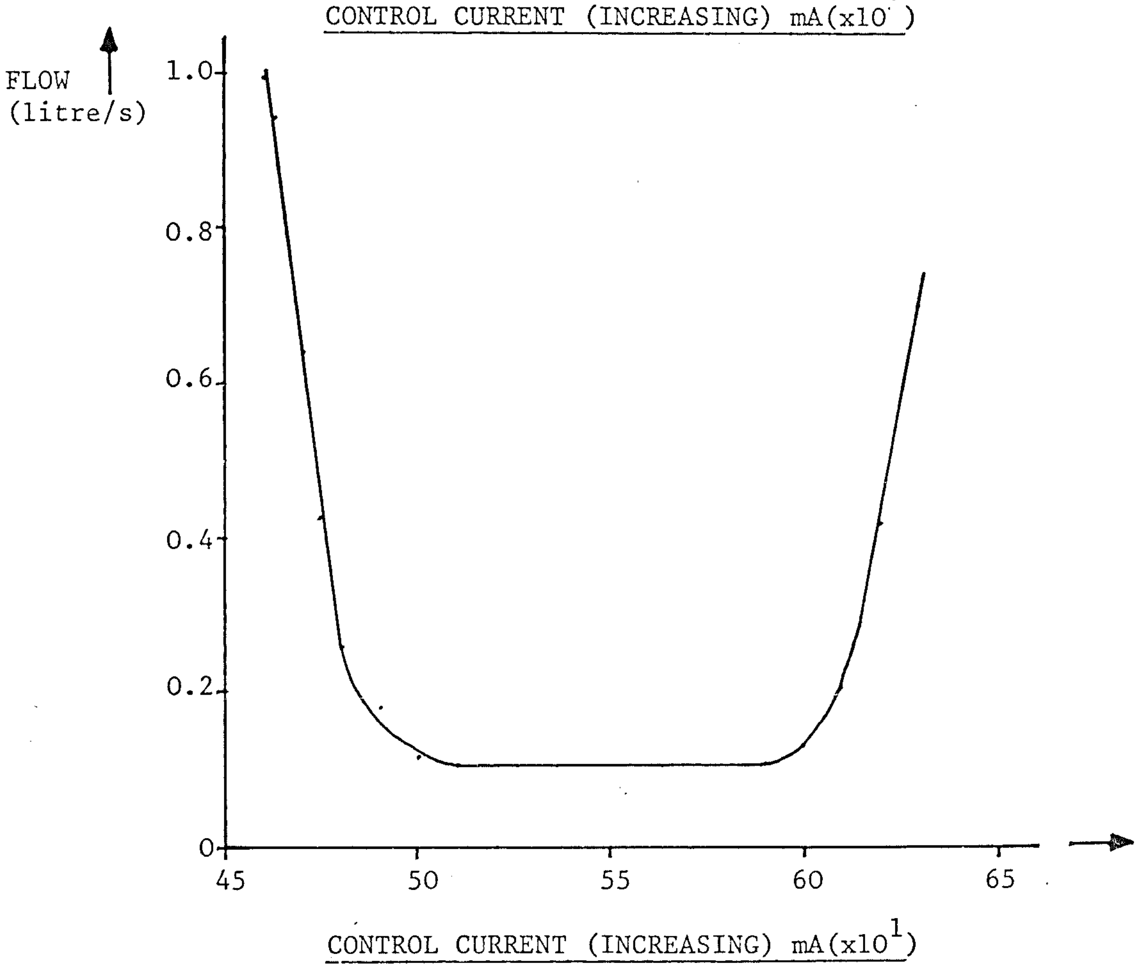
The flow/current characteristics for the proportional valve SPGB 18913 are shown in Figure [6.6.2.1]. These characteristics were obtained using the standard proportional valve with solenoid operation. The supply current does not include a dither component, investigation of dither is considered in section (6.6.4).

A flow/current test was carried out by the valve manufacturers, using a 6 bar supply pressure and an increasing supply current. A variable dc power supply was used to power the valve solenoid. The Martonair flow test rig was used to establish flow/current characteristics using a turbine meter No.3 and a Krohne (0.1 to 1.1 litre/sec) variale area meter. The low range flow readings were checked using a GEC Elliot 'Rotameter' flow kit. Exhaust ports of the valve were blocked.

The characteristics show that the flow/current relationship in the

Figure (6.6.2.1) SPBG 18913 - Proportional Valve Flow/Current Characteristics





operating range is linear. The valve leakage in the 'null' condition is constant at approximately 0.1litre/second. Using a glandless spool and sleeve valve design there will always be a quiescent leakage associated with it. The graph also shows that the maximum flow saturation level is not identical for each control port.

The flow and pressure characteristics for the proportional servo-valve, (with the solenoid removed), were determined by manually displacing the spool in known increments. A depth micrometer was used directly for spool displacement, pressure transducers P1 and P2 measure the control chamber pressures and a flowmeter measures the air flow to the valve supply port. The valve is connected to a symmetrical actuator locked in mid-stroke. The experimental procedure used is described below;

(i) the valve was adjusted to achieve the null, where pressure Pa=Pb, the reading of the micrometer being noted as reference,

(ii) the spool is then displaced by a known increment,

(iii) the differential chamber pressure is recorded, (Pa-Pb), after reaching a steady state, and the flow meter reading is also noted,

(iv) steps (ii) and (iii) are repeated, until a saturated chamber pressure is reached, and

(v) the valve spool is then reset to the 'null' position and the test procedure is repeated for the reverse spool displacement.

The flow rate is calculated using the following formula;

$$Qv = Cc.fx. ([Ps + P']/P')^{1/2}$$

where Qv = Flow rate (m^3/min)

Ps = Inlet/supply gauge pressure (kN/m^2)

P' = Atmospheric pressure (kN/m^2)

Cc = Flowmeter constant (0.049)

fx = Flowmeter reading

The test results are tabulated for flow and differential pressure in Tables [6.6.1] and [6.6.2] respectively. Figure [6.6.2.2] shows the graph of 'flow-spool displacement'. This shows that the valve flow characteristic is symmetrical and the valve is zero lapped.

The 'differential pressure-spool displacement' graph is shown in Figure [6.6.2.3]. This shows that the pressure characteristic is not symmetrical. The shape of a valve pressure characteristic will be influenced by the manufacturing precision, and each valve checked would vary slightly dependent upon the spool-sleeve combination. The graph also shows that the 'standing pressure' will fully switch from one control port to the other within 0.2mm of displacement about the valve null.

6.6.3. OPTIMISATION OF PROPORTIONAL 'SERVO-VALVE' DITHER

The prototype servo-valves supplied by Martonair exhibited considerable hysteresis when a control signal was applied. This hysteresis makes the valve of this type very non-linear and difficult to control, particularly as the hysteresis represents a significant proportion of the operating characteristic. A small percentage of the valve hysteresis is due to electromagnetic hysteresis in solenoid (approx 3%), but the main proportion being due to stiction of the mechanical components in the valve (and cylinder if connected).

As discussed previously, a conventional way of reducing hysteresis in hydraulic proportional valves is to apply a 'dither' component superimposed upon the control signal. The 'dither' component may also influence the

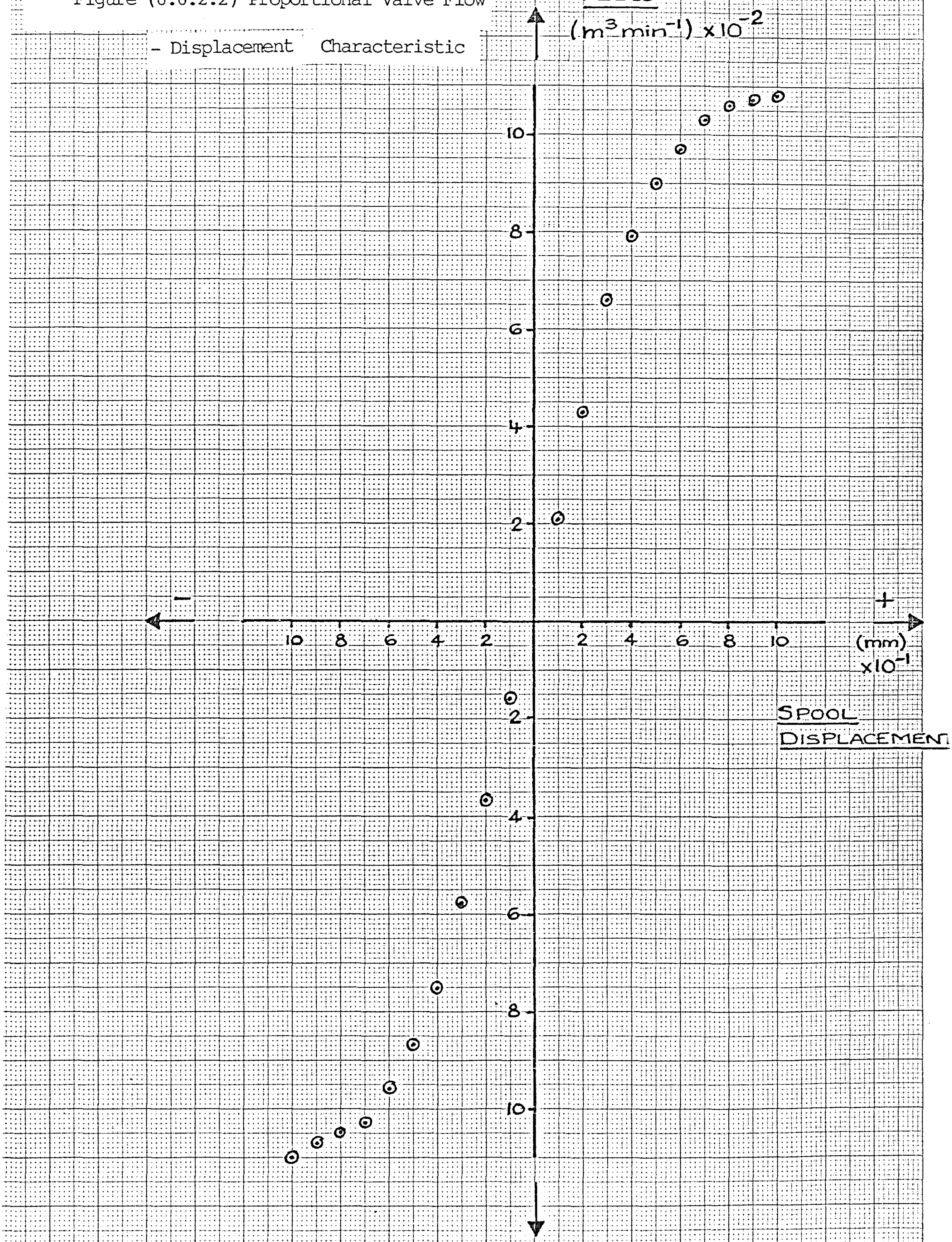
Spool Displacement (mm)	Differential Pressure $(P_1 - P_2)$ (kN/m^2)	
	+ve Displacement	-ve displacement
0.01	103	83
0.02	207	165
0.03	276	200
0.04	324	227
0.05	379	248
0.06	407	248
0.07	421	262
0.08	427	276
0.09	434	324
0.10	434	359
0.11	441	386
0.12	448	421
0.13	448	434
0.14	455	434
0.15	469	441
0.16	469	448
0.17	476	448
0.18	476	462
0.19	483	469
0.20	483	483

Table (6.6.1) Pressure Characteristic

Spool Displacement (mm)	Flow Rate $m^3 min^{-1}$ $(x10^{-2})$	
	+ve Displacement	-ve displacement
0	0	0
0.1	2.1	1.6
0.2	4.3	3.7
0.3	6.6	5.8
0.4	7.9	7.5
0.5	9.0	8.7
0.6	9.7	9.6
0.7	10.3	10.3
0.8	10.6	10.5
0.9	10.7	10.7
1.0	10.8	11.0

Table (6.6.2) Flow Characteristic

Figure (6.6.2.2) Proportional Valve Flow - Displacement Characteristic



* Manual Spool Displacement using depth micrometer

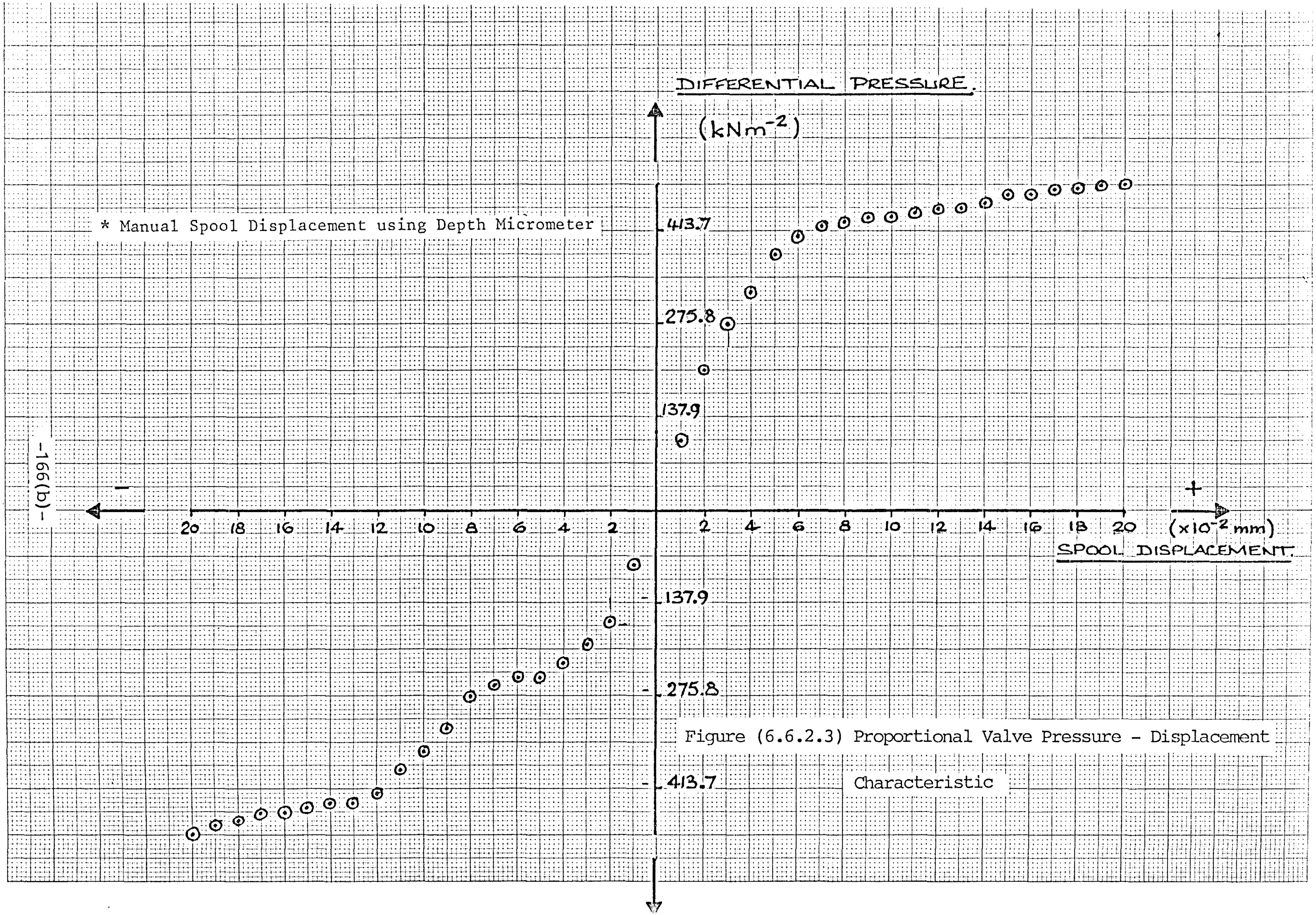


Figure (6.6.2.3) Proportional Valve Pressure - Displacement Characteristic level of friction present in the pneumatic actuator itself, possibly influencing seal friction, etc.. The use of a 'dither' component improved the linearity and consistency of the valve characteristics and hence allow improved performance in a servo control system using the valve. It would have been possible to use a PWM amplifier design to drive the solenoid operated valve as an alternative to using a linear amplifier and a dither source, however this was not investigated in this study.

A series of tests were carried out to assess the optimum level of dither amplitude and frequency in order to minimise valve hysteresis (Thatcher [1984]). A schematic of the test equipment is shown in Figure [6.6.3.1].

The test results are shown in Figure [6.6.3.2] and Figure [6.6.3.3]. They show that minimum hysteresis in the valve occurs with a dither frequency of about 20hz, and also that hysteresis can be reduced to a minimum value by increasing the amplitude of the dither. This, intuitively is correct, as the dither amplitude necessarily produces a force component greater than that required to overcome friction present in the system. These results show that a very significant improvement in the valve characteristic results with the inclusion of an optimised dither component. Where no dither was used the valve shows a hysteresis of 340mA, but with dither of 20hz and 100mA amplitude the hysteresis is reduced to only 28mA.

As a result of the tests, it was decided to standardise on the use of a dither component in any evaluation work using the solenoid operated proportional servo-valve. To simplify the design of the dither generation circuit, mains supply derived 50hz dither frequency is used and a suitable amplitude can be selected by manually adjusting a "gain potentiometer".

Figure (6.6.3.1) Hysteresis - Dither Experimental Arrangement

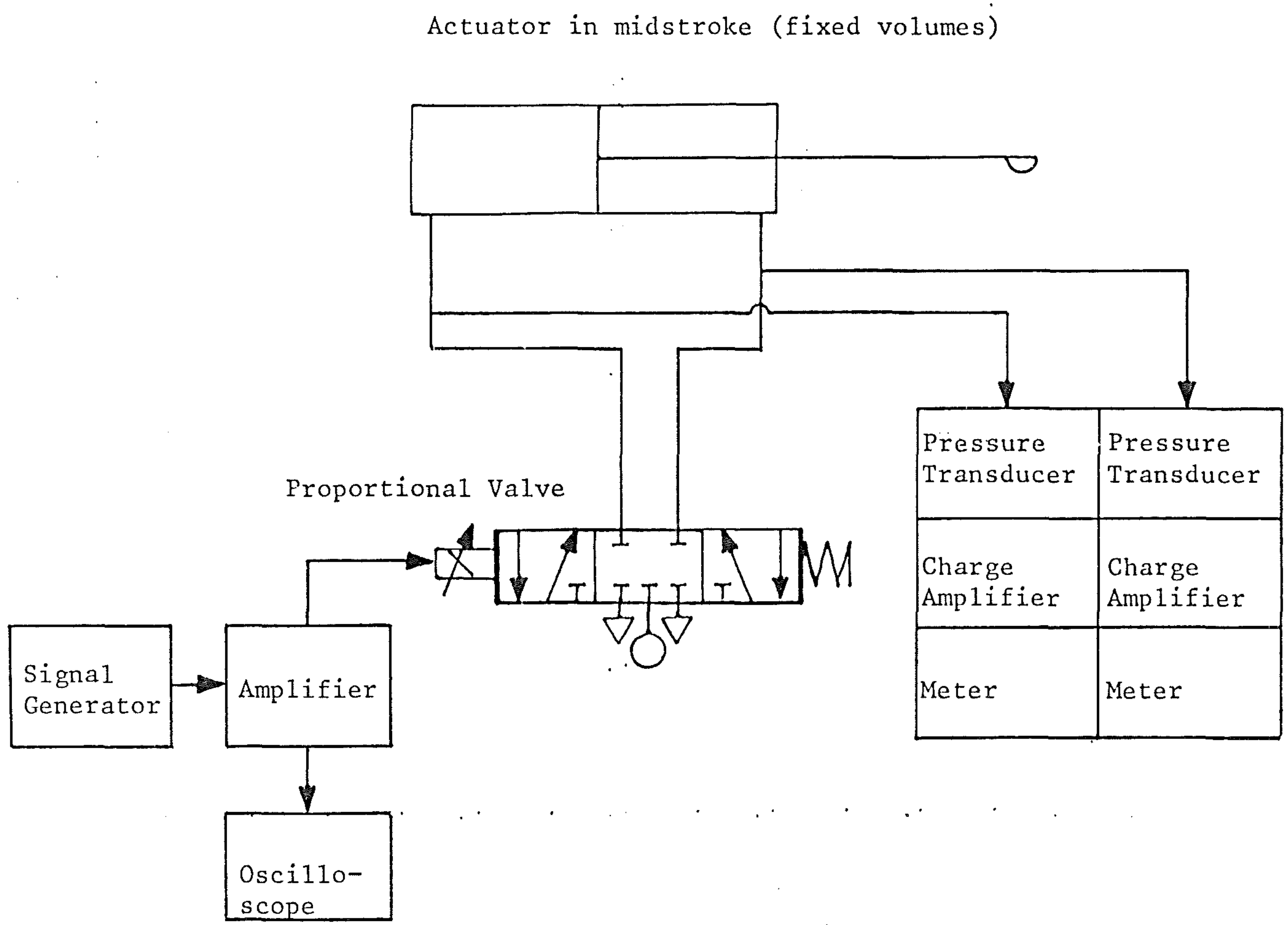


Figure (6.6.3.2) Hysteresis Characteristics of the Solenoid Operated Proportional Valve as a Function of Dither Frequency and Amplitude

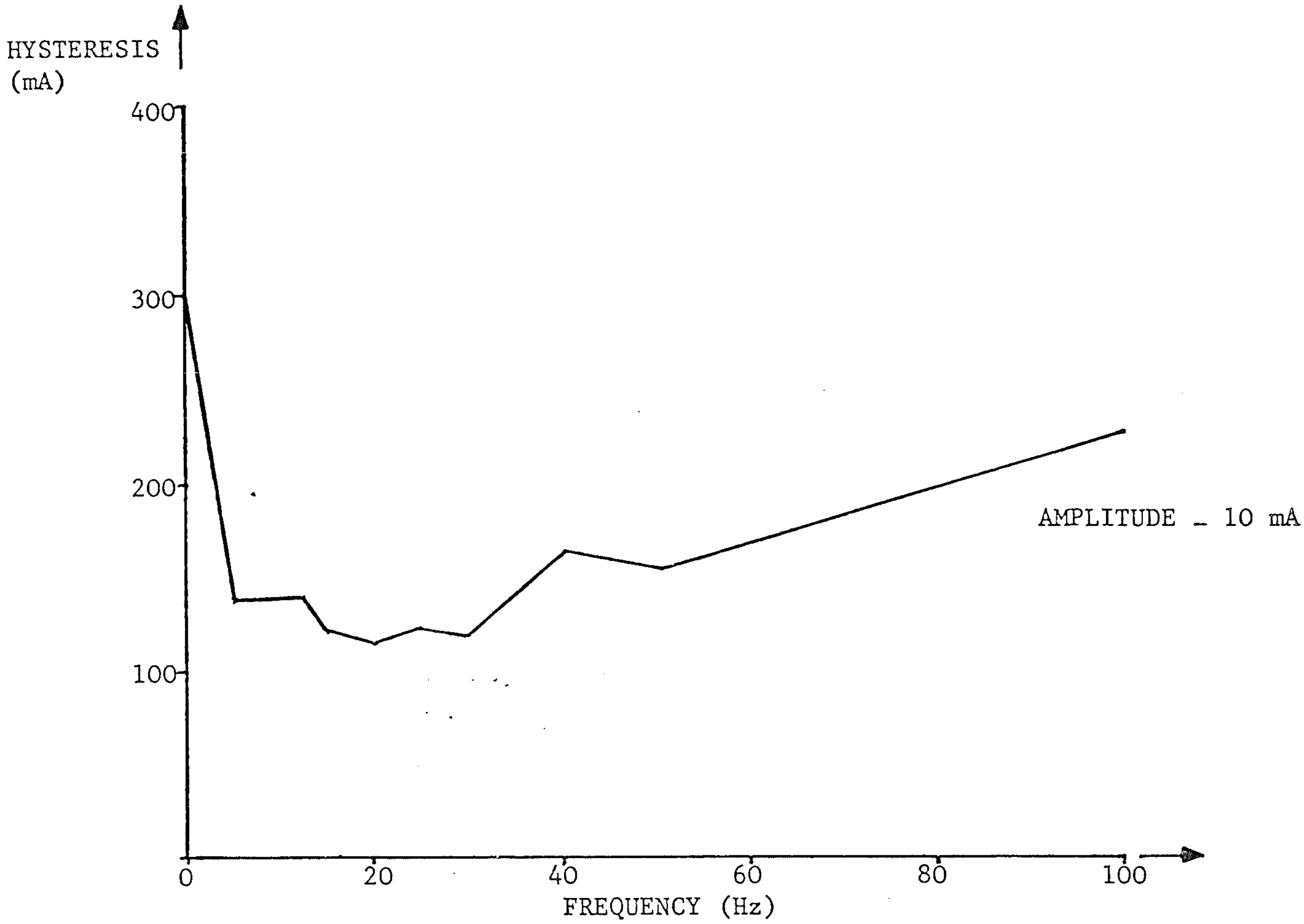
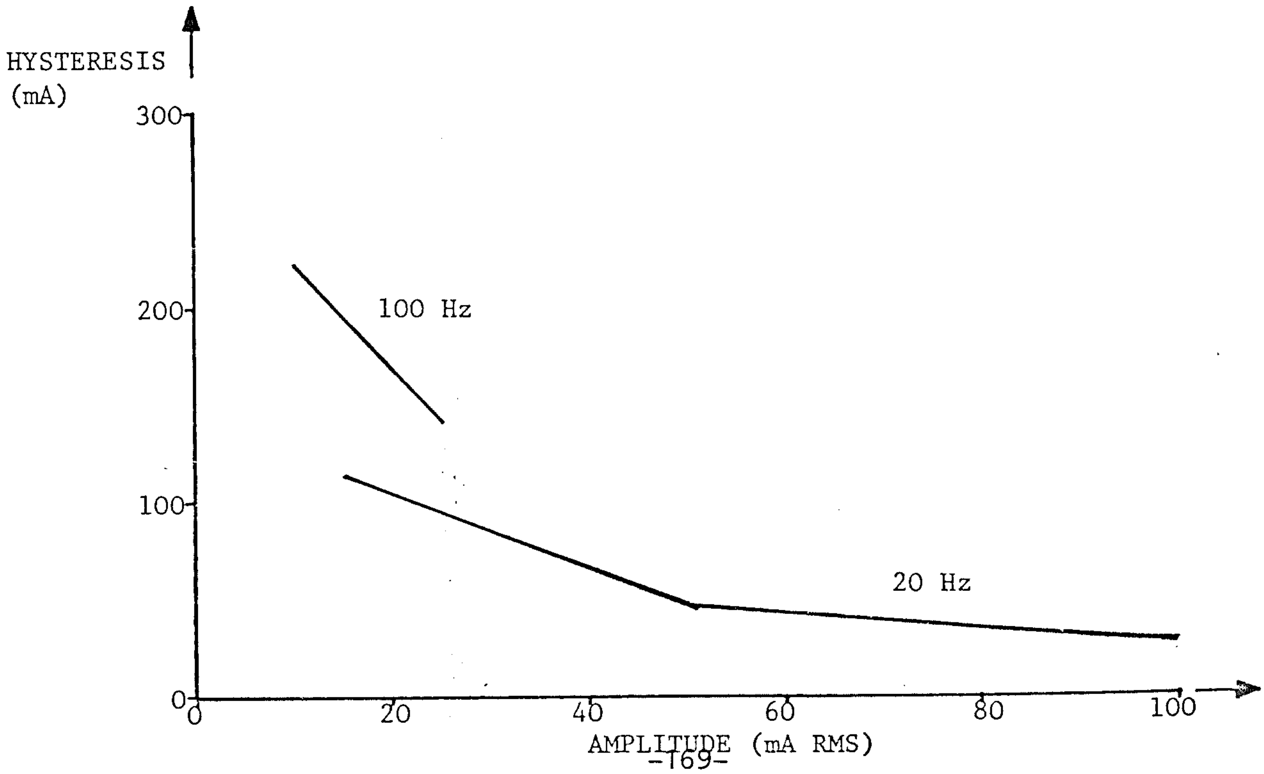


Figure (6.6.3.3)



CHAPTER 7

A LINEARISED MODEL FOR THE DESIGN OF PNEUMATIC SERVOS

7. ANALYSIS OF A PNEUMATIC ACTUATOR, PROPORTIONAL VALVE, AND LOAD SYSTEM

In this chapter a model is presented for a pneumatic actuator, servo-valve and load system. The objective in formulating a simplified model is to represent the dynamic behaviour of the physical system involved. The system model can then be utilised in a number of ways which include;

(i) to formulate an understanding of the open-loop physical system,

(ii) used in the synthesis of closed loop control strategy designs,

(iii) used in the analysis of closed loop control strategies to investigate the generalised effect of changes in feedback loop design, and

(iv) as a design tool to investigate the specific effect of changes in system parameters on performance (eg feedback loop gain selection; to simplify any system design and tuning procedure involved with a range of robot modules and actuators).

However, by definition before some or all of these objectives can be achieved it is necessary to verify any model formulated.

Linearisation techniques have been successfully applied in the analysis of electric and hydraulic drive systems, however the validity of such an approach needs to be considered when applied to a highly non-linear pneumatic drive system. However, linearised analysis remains the most attractive choice to the control system designer without embarking on very extensive component orientated modelling studies (which were outside the terms of reference of this research study).

Pneumatic actuators have found limited industrial use as primary drive elements for servo applications (in contrast to hydraulic and electric actuators) as a result of the difficulties encountered when using compressed air to transmit controlled power. Air, as a working fluid exhibits very low stiffness and little inherent damping at the operating pressures commonly used in manufacturing plants, resulting in servo-control features which are typified by a low natural resonant frequency and hence sluggish response times. Additionally, complex non-linearities are an inherent feature of a valve controlled linear actuator and its associated load system.

These features lead to complex describing models, when compared with counterpart models for electric and hydraulic drives and until the advent of low cost LSI devices it was generally impractical to achieve a level of compensation which would give adequate static and dynamic performance characteristics for general industrial use. The reader should be aware that the use of high cost control system elements used to achieve compensation would largely negate the cost effective use of pneumatic servos possibly leading to equipment of similar or greater cost than electric or hydraulic servos with inferior performance characteristics. Thus prior to the widespread use of microelectronics, pneumatic motion control received comparatively little attention within research communities in the western world.

The original stimulus to the study of pneumatic servomechanisms can be attributed to foundation research by Shearer, [1954], [1956], [1957], who analysed an open-loop system consisting of a plate valve controlling a cylinder using air at 55 bar. However, though Shearers' fundamental equations have been successfully applied to high pressure pneumatic systems, for industrial applications safety requirements dictate that the supply and operating pressure should be low. In low pressure servos the inherent disadvantages of using a compressible working medium become more apparent.

The linearised model presented here is derived from an earlier model presented by Burrows, [1968], [1969], [1969].

7.1. NOMENCLATURE

C_1	= flow gain or flow sensitivity	Valve coefficients
C_2	= flow-pressure coefficient	
m	= mass	
A	= ram area	
$(Y-Y_m)$	= displacement from the mid-stroke position	
V_i	= volume either side of the piston in mid-stroke position	
P_q	= the quiescent pressure	
R	= gas constant	
T_s	= supply temperature	
τ	= time constant = $V_i/\gamma R T_s C_2$	
γ	= ratio of specific heats	
K_p	= position-loop gain	
K_v	= velocity-loop gain	
K_a	= acceleration-loop gain	

The bar notation is used to denote changes from the initial conditions. Subscripts 'a' and 'b' refer to the ends 'a' and 'b' of the cylinder, and subscript 'i' refers to the initial value.

7.2. SIMPLIFYING ASSUMPTIONS

It is very important to realise that the following mathematical analysis presented in this chapter necessarily includes a number of simplifying assumptions. The assumptions on which the analysis is based include;

(i) the supply pressure and temperature are constant. The supply pressure is also assumed to be low (less than 10 bar typically for industrial applications),

(ii) a reversible adiabatic process is assumed,

(iii) the piston moves only small distances from its initial condition, and the pressures developed on either side of the piston vary by small amounts from an initial steady state value (small perturbation analysis theory),

(iv) the valve coefficients are constants and do not vary with spool displacement (an ideal valve),

(v) a symmetric actuator is used, and

(vi) only an inertial load is applied so that friction effects (and the non-linearities introduced therein) are negligible.

One of the most severe problems when modelling pneumatic systems is in describing the friction forces which are non-linear, a function of more than one variable, and can vary from stroke to stroke of the same actuator. (Chitty et al [1976]).

7.3. OPEN-LOOP ANALYSIS OF THE PNEUMATIC ACTUATOR, SERVO-VALVE, AND LOAD SYSTEM

The open-loop transfer function is an expression relating the load displacement to the valve spool displacement; the system is shown in Figure [7.3.1]. A representation of the open-loop transfer function G(s) is shown in Figure [7.3.2].

Linearised Mass Flow Rate equations (Shearer [1956]) are;

$$\underline{\dot{M}}_a = \frac{P_{ai}\,\underline{\dot{V}}_a}{RT_s} + \frac{V_{ai}\,\underline{\dot{P}}_a}{\gamma RT_s} \qquad - \quad (7.3.1)$$

and similarly

$$\underline{\dot{M}}_b = \frac{P_{bi}\,\underline{\dot{V}}_b}{RT_s} + \frac{V_{bi}\,\underline{\dot{P}}_b}{\gamma RT_s} \qquad - \quad (7.3.2)$$

where γ is the ratio of specific heats.

Linearising the valve pressure-flow characteristic to give

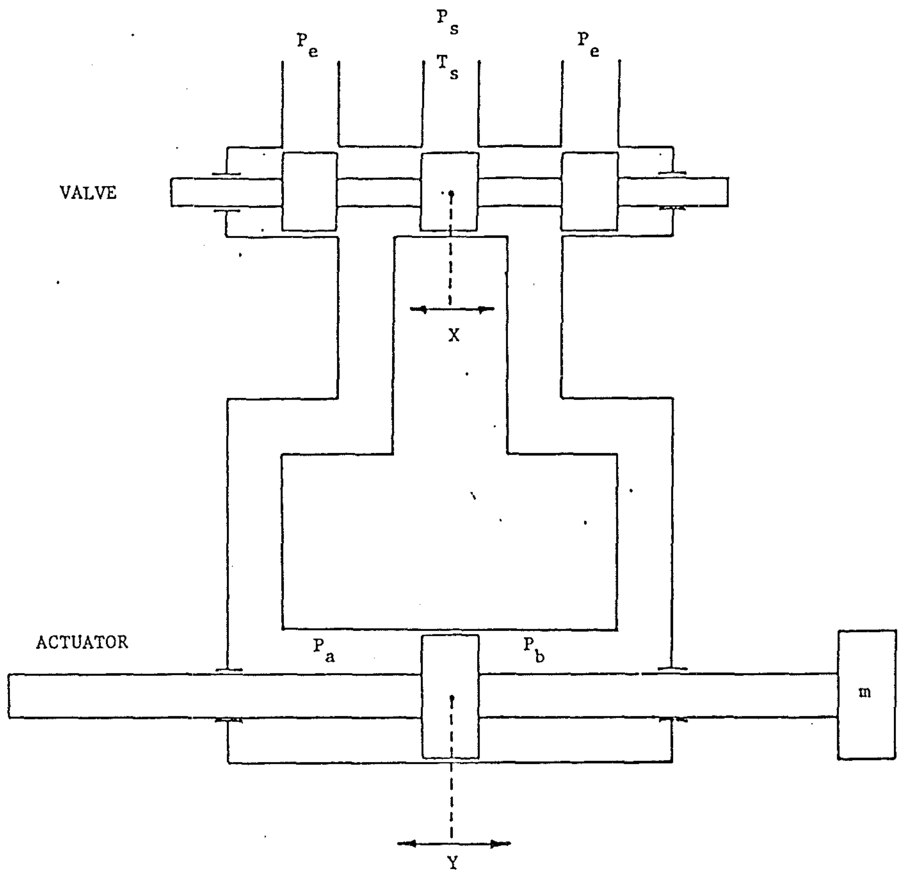
$$\underline{\dot{M}}_a = C_1\,\underline{X} - C_2\,\underline{P}_a \qquad - \quad (7.3.3.)$$

and

$$\underline{\dot{M}}_b = -C_1\,\underline{X} - C_2\,\underline{P}_b \qquad - \quad (7.3.4.)$$

where the mass flow rate through the valve is a function of spool displacement and cylinder pressure.

For compressible fluids the valve characteristic is conventionally a plot of 'mass flow rate' versus pressure, and a plot of 'mass flow rate' versus spool displacement. (Volumetric flow rate is normally used for incompressible fluids).



where: X - spool displacement
Y - actuator displacement

Figure (7.3.1) Open Loop System

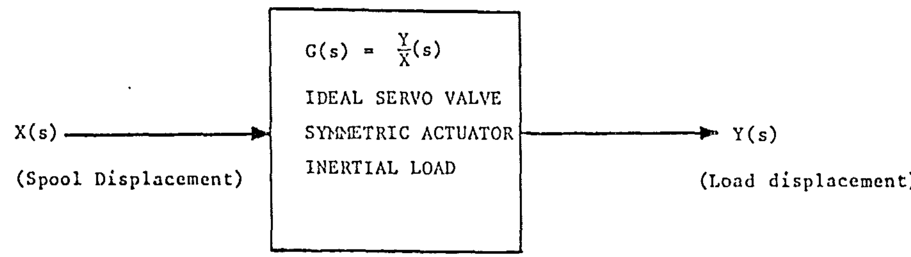


Figure (7.3.2) Open Loop System Representation

The valve coefficients are;

$$C_1 = \left.\frac{\partial \dot{M}_a}{\partial X}\right|_{P_{ai}} = \left.\frac{\partial \dot{M}_b}{\partial X}\right|_{P_{bi}} \qquad \text{Flow Gain or Flow Sensitivity}$$

and

$$C_2 = \left.-\frac{\partial \dot{M}_a}{\partial P_a}\right|_{X_i} = \left.-\frac{\partial \dot{M}_b}{\partial Pb}\right|_{X_i} \qquad \text{Flow-Pressure Coefficient}$$

The load motion equation, assuming friction effects are negligible and a rigid coupling between the actuator and load, for zero external forces is;

$$(P_a - P_b)A = m\,\underline{\ddot{Y}} \qquad - \quad (7.3.5.)$$

Let the volume on either side of the piston, when in mid-stroke position (i.e. Y = Ym), be V_i;

then $$V_{ai} = V_i + A\ [Y - Y_m] \qquad - \quad (7.3.6.)$$

and $$V_{bi} = V_i - A\ [Y - Y_m] \qquad - \quad (7.3.7.)$$

Volume changes from the steady state

$$\underline{V}_a = A\underline{Y} \qquad - \quad (7.3.8.)$$

and

$$\underline{V}_b = -A\underline{Y} \qquad - \quad (7.3.9.)$$

Assuming equal ram areas and absence of an external load;

$$P_{ai} = P_{bi} = P_q \quad \text{the quiescent pressure}$$

An open-loop system linearised analysis has been presented by Burrows [1969] which accounts for displacement from the midstroke.

Reducing equations (7.3.1) to (7.3.4) and (7.3.6.) to (7.3.9.) to get

$$\frac{2PqAs\underline{Y}}{RT_s} = 2C_1 \underline{X} - \left(C_2 + \frac{V_i s}{\gamma RT_s} - \frac{(A \; [Y-Y_m]/(\gamma RT_s)^2 \; s^2}{V_i s/(\gamma RT_s) + C_2}\right) \underline{\Delta P} \quad - \quad (7.3.10.)$$

The Laplace operator has replaced the Newtonian dot notation.

Similarly we can substitute as follows;

$$\underline{u} = s\underline{Y} \quad \text{the actual velocity change}$$

and

$$\underline{\Delta P} = \underline{P}_a - \underline{P}_b$$

also let

$$\tau = \frac{V_i}{\gamma RT_s C_2} \quad \text{the Pneumatic Time Constant} \qquad - \quad (7.3.11.)$$

then

$$\frac{2P_q A\underline{u}}{RT_s} = 2C_1\underline{X} - C_2 \left[1 + \tau s - \frac{\left[\frac{A\tau \; [Y-Y_m]}{V_i}\right]^2 s^2}{(1 + \tau s)} \right] \underline{\Delta P}$$

or

$$\frac{2P_q \underline{Au}}{RT_s} = 2C_1\underline{X} - C_2\left[1 + \tau s - \frac{N^2 s^2}{(1+\tau s)}\right]\underline{\Delta P}$$

where $N = \frac{A\tau}{V_i}[Y-Y_m]$ – (7.3.12.)

Thus

$$C_2\left[\frac{(1+\tau s)^2 - N^2 s^2}{(1+\tau s)}\right]\underline{\Delta P} = 2C_1\underline{X} - \frac{2P_q \underline{Au}}{RT_s}$$

or

$$\underline{\Delta P} = \left[(2C_1\underline{X} - \frac{2P_q \underline{Au})}{RT_s}\right] \cdot \frac{(1+\tau s)}{C_2\,[1 + 2\tau s + (\tau^2 - N^2)\,s^2\,]}$$

– (7.3.13.)

so

$$\underline{\Delta P} = \frac{\frac{2C_1}{C_2}\left[\underline{X} - \frac{\underline{Au}}{L}\right](1+\tau s)}{[1 + 2\tau s + (\tau^2 - N^2)\,s^2]}$$ – (7.3.14.)

where

$$L = \frac{RT_s C_1}{P_q}$$ – (7.3.15.)

Equation (7.3.14.) may be re-written in the form

$$\underline{\Delta P} = \frac{\frac{2C_1}{C_2}\left[\underline{X} - \frac{\underline{Au}}{L}\right](1+\tau s)}{(\tau^2 - N^2)\left(s^2 + \frac{2\tau s}{(\tau^2 - N^2)} + \frac{1}{(\tau^2 - N^2)}\right)}$$

For the case of real and distinct poles may be written as

$$\frac{\Delta P}{} = \frac{K' \left[\frac{X}{} - \frac{\frac{Au}{}}{L}\right](1+\tau s)}{(s+a)(s+b)} \qquad - (7.3.16.)$$

where

$$K' = \frac{2C_1}{C_2(\tau^2 - N^2)} \qquad - (7.3.17.)$$

The poles are identical to the roots of the characteristic equation

$$s^2 + \frac{2\tau s}{(\tau^2 - N^2)} + \frac{1}{(\tau^2 - N^2)} = 0 \qquad - (7.3.18.)$$

hence

$$s = \frac{1}{(\tau^2 - N^2)}\left[-\tau \pm N\right]$$

which gives

$$a = \frac{-\tau + N}{(\tau^2 - N^2)} \quad \text{and} \quad b = \frac{-\tau - N}{(\tau^2 - N^2)}$$

Substituting into equation (7.3.16.)

$$\frac{\Delta P}{} = \frac{K' \left[\frac{X}{} - \frac{\frac{Au}{}}{L}\right](1+\tau s)}{\left(s + \frac{\tau + N}{(\tau^2 - N^2)}\right)\left(s + \frac{\tau - N}{(\tau^2 - N^2)}\right)} \qquad - (7.3.19.)$$

rearranging and resubstituting for K'

$$\frac{\Delta P}{} = \frac{\frac{2C_1}{C_2}\left[\frac{X}{} - \frac{\frac{Au}{}}{L}\right](1+\tau s)}{(1+\tau_1 s)(1+\tau_2 s)} \qquad - (7.3.20.)$$

Where $$\tau_1 = \frac{(\tau^2 - N^2)}{(\tau + N)} \qquad - (7.3.21.)$$

and $$\tau_2 = \frac{(\tau^2 - N^2)}{(\tau - N)} \qquad - (7.3.22.)$$

The expression for pressure change given in (7.3.20) is a second order relationship between $\underline{P}$, $\underline{X}$ and $\underline{u}$. However at the mid-stroke position $Y = Y_m$, for which

$$N = \frac{A\tau (Y-Y_m)}{V_i} = 0, \text{ giving } \tau_1 = \tau_2 = \tau$$

This reduces equation (7.3.20.) to a first order relationship, where

$$\Delta\underline{P} = \frac{\frac{2C_1}{C_2}\left[\underline{X} - \frac{A\underline{u}}{L}\right]}{(1 + \tau s)}$$

The analysis can be extended as follows;

From equation (7.3.5.)

$$\Delta\underline{PA} = m\underline{\ddot{Y}} = ms^2\,\underline{Y} \qquad \text{(using Laplace operator)}$$

also as $\underline{u} = s\underline{Y}$

Then substituting in equation (7.3.20.) we obtain

$$ms^2\underline{Y} = \Delta\underline{PA} = \frac{\frac{2AC_1}{C_2}\left[\underline{X} - \frac{As\underline{Y}}{L}\right]}{(1 + \tau_1 s)(1 + \tau_2 s)} (1 + \tau s)$$

expanding to give

$$\frac{\underline{Y}}{\underline{X}} = \frac{\frac{2AC_1}{m\tau_1\tau_2 C_2}(1 + \tau s)}{s^4 + \frac{(\tau_1 + \tau_2)}{\tau_1\tau_2}s^3 + \frac{m + \frac{2C_1 A^2}{C_2 L}}{m\tau_1\tau_2}s^2 + \frac{\frac{2C_1A^2}{C_2 L}}{m\tau_1\tau_2}s} \qquad - (7.3.23.)$$

This is the open-loop transfer function for the linearised model relating actuator/load displacement to the spool displacement.

The open loop transfer function can further be expressed as follows;

$$G(s) = \frac{Y}{X}(s) = \frac{K(1+\tau s)}{s^4 + b_1 s^3 + b_2 s^2 + b_3 s} \quad - (7.3.24.)$$

where

$$K = \frac{2AC_1}{\tau_1 \tau_2 m C_2}$$

The coefficients of the transfer function can be evaluated as;

$$K = \frac{2AC_1C_2(\gamma RT_s)^2}{m\left[V_i^2 - A^2(Y-Y_m)^2\right]} \quad - (7.3.25.)$$

alternatively

$$K = \frac{2C_1}{\frac{mC_2\tau^2}{A}\left[1 - \left[\frac{A(Y-Y_m)}{Vi}\right]^2\right]}$$

Similarly

$$b_1 = \frac{2V_i \gamma RT_s C_2}{\left[V_i^2 - A^2(Y-Y_m)^2\right]} \quad - (7.3.26.)$$

alternatively

$$b_1 = \frac{2}{\tau\left(1 - \left[\frac{A(Y-Y_m)}{V_i}\right]^2\right)}$$

Similarly

$$b_2 = \frac{m\,(\gamma RT_s C_2)^2 + \left[\dfrac{2A^2\,C_1\,V_i}{C_2}\right]\left[\dfrac{(\gamma RT_s\,C_2)^2}{RT_s\,C_2}\right]\cdot\dfrac{P_q}{RT_s\,C_1}}{m\left[V_i^{\,2} - A^2\,(Y-Y_m)^{\,2}\right]} \qquad - (7.3.27.)$$

alternatively

$$b_2 = \frac{\dfrac{mC_2}{A} + \dfrac{2AP_q\tau}{RT_s}}{\dfrac{mC_2\tau^2}{A}\left[1 - \left[\dfrac{A\,(Y-Y_m)}{V_i}\right]^2\right]}$$

Similarly

$$b_3 = \frac{2\,P_q\,A^2}{C_2 mRT_s\,\tau^2\left[1-\left[\dfrac{A\,(Y-Y_m)}{V_i}\right]^2\right]} \qquad - (7.3.28.)$$

alternatively

$$b_3 = \frac{(2\,P_q\,A)/RT_s}{\dfrac{C_2 m\,\tau^2}{A}\left[1 - \left[\dfrac{A(Y-Ym)}{V_i}\right]^2\right]}$$

An alternative form of the open-loop transfer function of equation (7.3.20.) is:

$$\underline{\Delta P} = \frac{Gp\left[\underline{X} - \dfrac{Au}{Gq}\right](1+\tau s)}{(1+\tau_1\,s)\ \ (1+\tau_2 s)} \qquad - (7.3.29.)$$

By identity with (7.3.20.) yields:

$$Gp = 2\,\frac{C_1}{C_2} \qquad \text{Valve pressure gain} \qquad - (7.3.30.)$$

$$Gq = L = \frac{RT_s C_1}{P_q} \qquad \text{Valve flow gain} \qquad - (7.3.31.)$$

Equation (7.3.29.), an alternative expression which can be useful if Gp and Gq are readily available.

Now combining equations (7.3.29.) and (7.3.5.) to develop:

$$G(s) = \frac{Y}{X}(s) = \frac{\dfrac{Gp\ A}{m\tau_1\tau_2} \cdot (1+\tau s)}{s^4 + \left[\dfrac{\tau_1+\tau_2}{\tau_1\tau_2}\right]s^3 + \left[\dfrac{m + \dfrac{Gp}{Gq}A^2\tau}{m\tau_1\tau_2}\right]s^2 + \dfrac{GpA^2}{\left[Gqm\tau_1\tau_2\right]}s} \qquad -(7.3.32.)$$

Equation (7.3.32.) is an alternative form of the open loop transfer function.

The pneumatic time constant τ is defined in terms of C_2, however:

$$Gp = \frac{2C_1}{C_2} \quad \text{and} \quad Gq = \frac{RT_sC_1}{P_q}$$

thus

$$C_1 = \frac{P_q\ Gq}{RT_s} \qquad - (7.3.33.)$$

then substituting in equation (7.3.30.)

$$Gp = \frac{2\ P_q\ Gq}{RT_s\ C_2}$$

hence

$$C_2 = \frac{2\ P_q\ Gq}{RT_s\ Gp} \qquad - (7.3.34.)$$

Thus substituting in equation (7.3.11.)

$$\tau = \frac{V_i \; Gp}{2\gamma \; P_q Gq} \qquad - \; (7.3.35)$$

From equations (7.3.21.) and (7.3.22.), τ_1 and τ_2 can be evaluated.

$$\tau_1 = \frac{Gp}{2\gamma P_q Gq}\left[\frac{V_i^{\,2} - A^2\,[Y-Y_m]^2}{V_i + A\,[Y-Y_m]}\right] \qquad - \; (7.3.36.)$$

and

$$\tau_2 = \frac{Gp}{2\gamma P_q Gq}\left[\frac{V_i^{\,2} - A^2\,[Y-Y_m]^2}{V_i - A\,[Y-Y_m]}\right] \qquad - \; (7.3.37.)$$

The coefficients of the open loop transfer function, equation (7.3.32.), can be evaluated from equation (7.3.24.) where

$$G\,(s) = \frac{\underline{Y}\,(s)}{\underline{X}} = \frac{K\,(\tau s + 1)}{s^4 + b_1\, s^3 + b_2\, s^2 + b_3\, s} \qquad - \; (7.3.24.)$$

$$K = \frac{4A\,(\gamma\, P_q\; Gq)^2}{Gpm\; V_i^{\,2}\left[1 - \left[\dfrac{A(Y-Y_m)}{V_i}\right]^2\right]} \qquad - \; (7.3.38.)$$

Similarly

$$b_1 = \frac{4\,\gamma\; P_q Gq}{GpV_i\left[1 - \left[\dfrac{A(Y-Y_m)}{V_i}\right]^2\right]} \qquad - \; (7.3.39.)$$

Similarly

$$b_2 = \frac{4m\,(\gamma\; P_q Gq)^2 + 2\gamma\; P_q\; A^2\; V_i\; Gp^2}{mV_i^{\,2}\; Gp^2\left[1 - \left[\dfrac{A\,(Y-Y_m)}{V_i}\right]^2\right]} \qquad - \; (7.3.40.)$$

Similarly

$$b_3 = \frac{4\ (\gamma\ P_q\ A)^2\ Gq}{mGp\ V_i^{\ 2}\left[1 - \left[\dfrac{A\ (Y-Y_m)}{V_i}\right]^2\right]} \qquad - \ (7.3.41.)$$

7.4. APPLICATION OF THE LINEARISED MODEL

The open-loop model described in equation (7.3.32) will now be further analysed with a view to designing closed-loop control strategies and associated compensation methods. The advantage of deriving a linearised model despite the gross simplifying assumptions embodied therein, is to predict system behaviour using well established analysis techniques.

Thus far the analysis presents a model originally derived by Burrows [1969] which accounts for system response about any initial position in the stroke. However, the analysis is extended to incorporate an acceleration feedback loop in place of the quasi-transient pressure feedback. The concept of using a combination of velocity and acceleration has been postulated by a number of researchers where implementation was expected to utilise analog control architectures.

Velocity feedback is commonly used with commercial electric and hydraulic servos and has a similar effect on stability to that of increasing viscous damping acting on a load; unlike viscous damping however, it does not dissipate energy. Acceleration feedback can further stabilise the response when used in conjunction with velocity feedback. In the digital control system implemented in this research study minor loop compensation is achieved with velocity and 'regenerative' (or positive feedback) acceleration, as this was found to be most effective in stabilising the system. The use of 'regenerative' acceleration and 'negative' velocity feedback is a feature quite unique to this system in the compensation of pneumatic servo-drives as far as the author is aware. The technique has been advocated for compensation of hydraulic systems by Burrows [1972].

An early study of digital electro-pneumatic control systems is described by Drazan [1978] and it is shown that velocity and acceleration feedback can be used to improve system stability. This study is illustrated by root loci from the characteristic equation, where the lead term introduced modifies the system behaviour (both velocity and acceleration are required for inherent closed loop stability).

The effect of a velocity and acceleration feedback combination on the root locus for an electro-pneumatic system has also been demonstrated by McCloy and Martin [1980], where the ratio of the feedback coefficients is shown to determine the conditional stability of the system, clearly indicating that the feedback loops need to be carefully tuned.

7.4.1. EVALUATION OF THE COEFFICIENTS OF THE OPEN-LOOP TRANSFER FUNCTION

The coefficients of the linearised open-loop transfer function (equation (7.3.32)) can be evaluated for the drive system elements under examination (modules and proportional valves). To illustrate this we will consider the particular use of the Martonair M60110/600 module. This module has a 25mm diameter piston, a maximum stroke length of 600mm and has an intrinsic mass (for the moving element) of 3.5kg.

For this analysis we will assume that the air supply is at a pressure of 6.8 bar and temperature of 293K. On making these assumptions the following parameters can be calculated:-

$$m = 3.5\ kg$$

$$A = (\pi D^2)/4 = 4.91 \times 10^{-4}\ m^2$$

$$V_i = A(S/2) = 1.47 \times 10^{-4}\ m^2$$

$$P_q = 6.87 \times 10^5\ Nm^{-2}$$

Subsequently these parameters can be used in evaluating the corresponding coefficients of the open-loop transfer function provided that the necessary coefficients associated with the proportional valve (Gp and Gq) can be determined. Representative values for these valve coefficients are not easily assimilated; it is necessary, in an attempt to simplify analysis, to assume steady state values here, even though it is likely that these coefficients will vary under transient conditions. Furthermore, inspection of the steady state pressure and flow characteristics for a proportional valve shows that the coefficients are non-linear. Despite these problems estimates of the valve coefficients were made, but it should be made clear to the reader that significant doubt must be associated with the choice made.

However, using valve coefficients of

$$\left.\begin{aligned} G_p &= 8 \times 10^{9}\ \mathrm{Nm}^{-2}\,\mathrm{m}^{-1} \\ \text{and } G_q &= 2\mathrm{m}^{3}\,\mathrm{s}^{-1}\,\mathrm{m}^{-1} \end{aligned}\right\}*$$

(* as quoted by the valve manufacturer)

the coefficients τ, K, b1, b2 and b3 can be evaluated using equations [7.3.35] and [7.3.38] to [7.3.41] respectively. But inspection of the above equations shows that the magnitudes of the coefficients are significantly influenced by the values for Gp and Gq.

On evaluating coefficients (assuming midstroke operation) the values obtained for the Martonair M60110/600 module and proportional valve were as follows;

$$\tau = 0.304$$

$$K = 12.1 \times 10^{6}$$

$b_1 = 6.6$

$b_2 = 10.8$

and $b_3 = 2974.7$

Correspondingly an alternative set of coefficients can be obtained using valve coefficients of

$$\left.\begin{aligned} Gp &= 8.82 \times 10^{8} \ Nm^{-2} m^{-1} \\ \text{and } Gq &= 3.33 \ m^{3} s^{-1} m^{-1} \end{aligned}\right\} **$$

(** these being obtained from valve characteristics presented in Chapter (6)). The coefficients relating to the alternative valve coefficients were found to be;

$\tau = 2.01 \times 10^{-2}$

$K = 3.04 \times 10^{8}$

$b_1 = 99.3$

$b_2 = 2463$

and $b_3 = 4.48 \times 10^{4}$

Comparison of these two sets of open-loop coefficients serves to illustrate the importance of evaluating values for Gp and Gq. Investigation of the open-loop system behaviour shows that the system would be inherently unstable (ie open-loop poles in the right hand half of the s-plane) when using the valve coefficients as quoted by the manufacturers. However, the alternative valve coefficients give a better interpretation of the open-loop system stability (this is illustrated in the following section).

7.4.2. CLOSED-LOOP SYSTEM ANALYSIS

Knowing the open-loop transfer function, a range of classical analysis techniques are available to anticipate in either the time or frequency domain the effectiveness of any control strategies (Kuo [1975]). Here the root locus technique is used to study closed-loop system stability and behaviour with extensive use being made of a computer aided control system design software package from UMIST known as SVLIN. When using this package the user defines parameters of the transfer function and a corresponding root locus is automatically calculated through the use of the Newton-Raphson technique.

The root locus technique was selected as it is a well established graphical technique for gaining an insight into the effect of changing parameters of the closed-loop transfer function. The SVLIN package simplifies the process of calculating roots (or pole positions) and the subsequent plotting of the root locus once the transfer function is formulated. However, it must be emphasised that the merit of any information derived depends directly on the validity of the transfer function representing the system model. To illustrate the use of the technique we will, however, consider two alternative closed-loop control strategies.

7.4.2.1. PROPORTIONAL ERROR CONTROL

Proportional error control is a commonly used control strategy where only the position loop is closed. Proportional control is now considered in the context of a pneumatic drive system (see Figure [7.4.1]) where the closed-loop transfer function can be shown to be:

$$W(s) = \frac{Kp.G(s)}{1+G(s).Kp.Kf}$$

where $Kf = Ky$

If we now consider the characteristic equation, which determines stability, we have:

$$1 + \frac{KpK\{Ky\tau s + Ky\}}{s^4 + b_1 s^3 + b_2 s^2 + b_3 s} = 0 \qquad \text{---(7.4.1)}$$

If we now consider the system behaviour (about the midstroke position) then we use equation (7.4.1) to determine the locus of the roots of this equation for values of the proportional coefficient (Kp) between zero and infinity.

When anticipating motion control through the use of a microprocessor based controller a knowledge of the nature of the proportional coefficient (Kp) is necessary so that we can interpret information contained within the root locus of the system. The proportional coefficient constitutes two seperate elements, viz:

(1) the hardware dependant gain of the system (Kp"), and

(2) the software programmable gain (Kp'), where

$$Kp = Kp'.Kp'' \qquad \text{---(7.4.2)}$$

Figure (7.4.1) Proportional Error Control

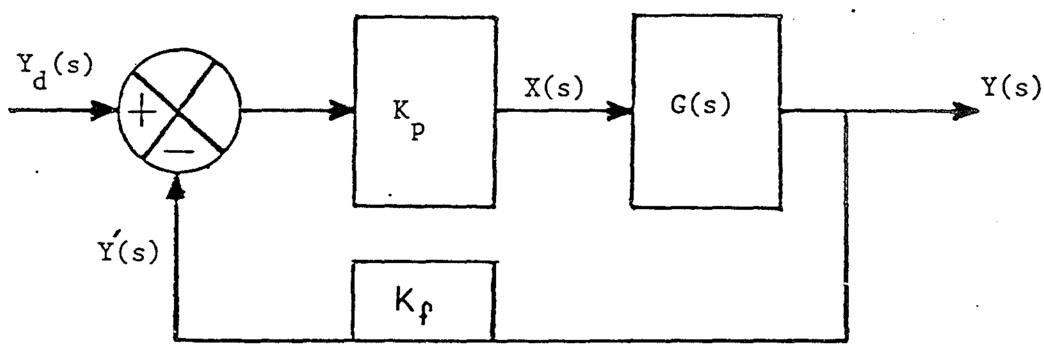
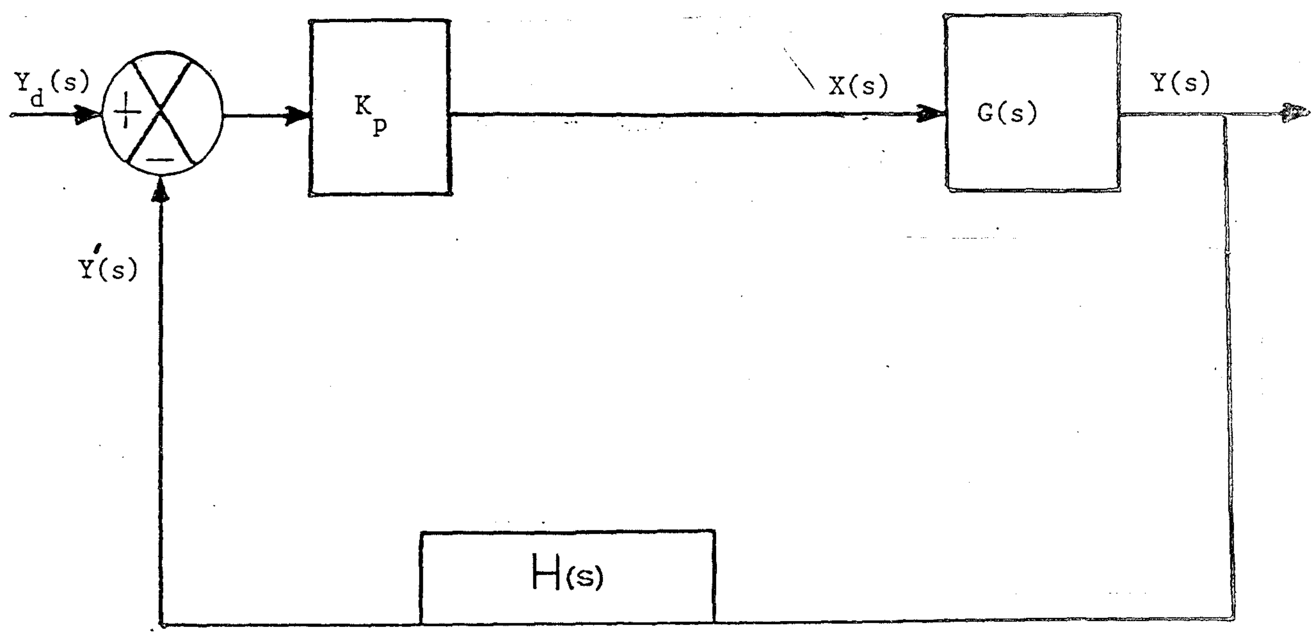


Figure (7.4.2) Proportional Error Control
with Minor Loop Compensation

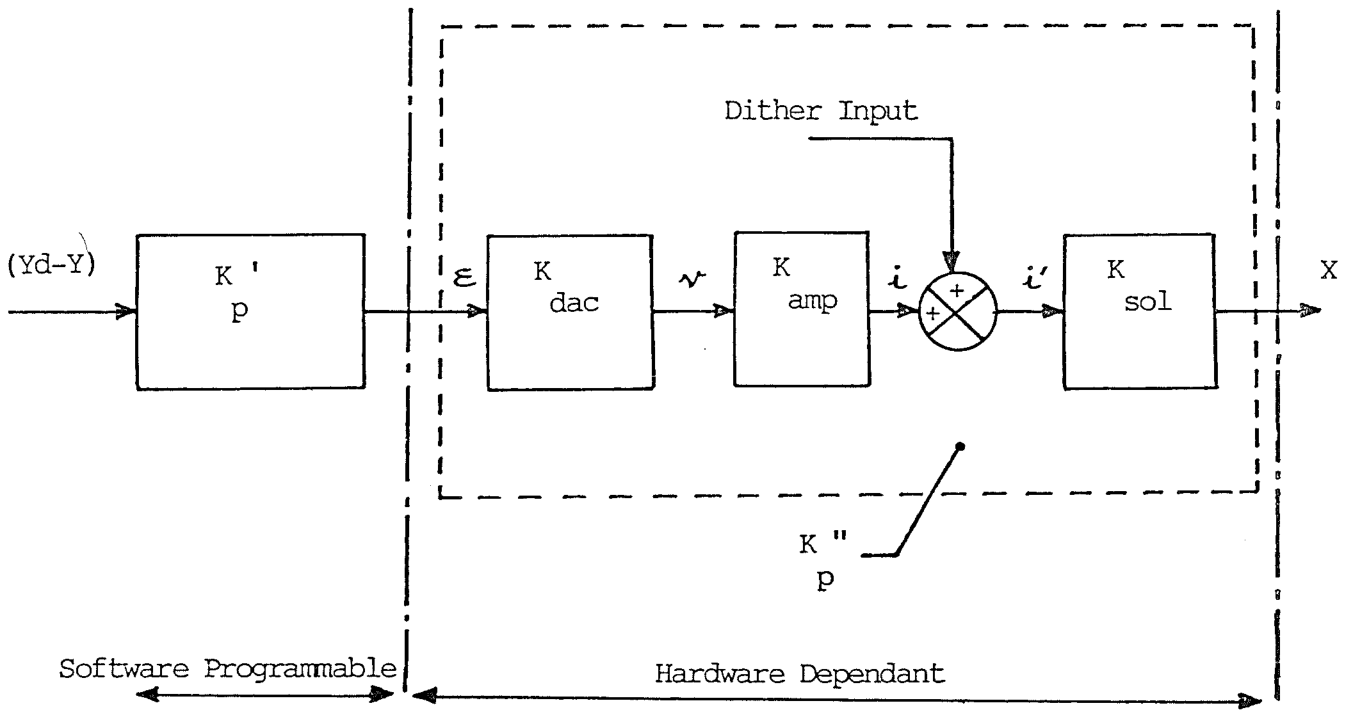


The elements that constitute the hardware dependant gain are illustrated in Figure [7.4.3], and consist of coefficients associated with the DAC, linear amplifier, and the proportional solenoid. If an idealised situation is considered each component of the open-loop system will saturate for the same input demand. For any digital input in the range of 0 to 1024, this should result in a displacement of the valve spool (X) in the range 0 to 2mm. However, it should be remembered that physical saturation within the real system can occur for different input demands (non-ideal component matching), as well as saturation effects occurring in software due to overflow conditions in calculations etc..

Having considered this background information we will now interpret the root loci obtained. Root locus (1) corresponds to the situation where the valve coefficients are chosen as being those quoted by the manufacturers, and predicts an inherently unstable system. This is clearly unsatisfactory as variation of the proportional gain can be shown to produce a spectrum of stability conditions in the real system (see Figure [9.2.1]). As mentioned earlier, alternative estimated values for the valve coefficients can be obtained, and from these values new coefficients of the characteristic equation can be derived.

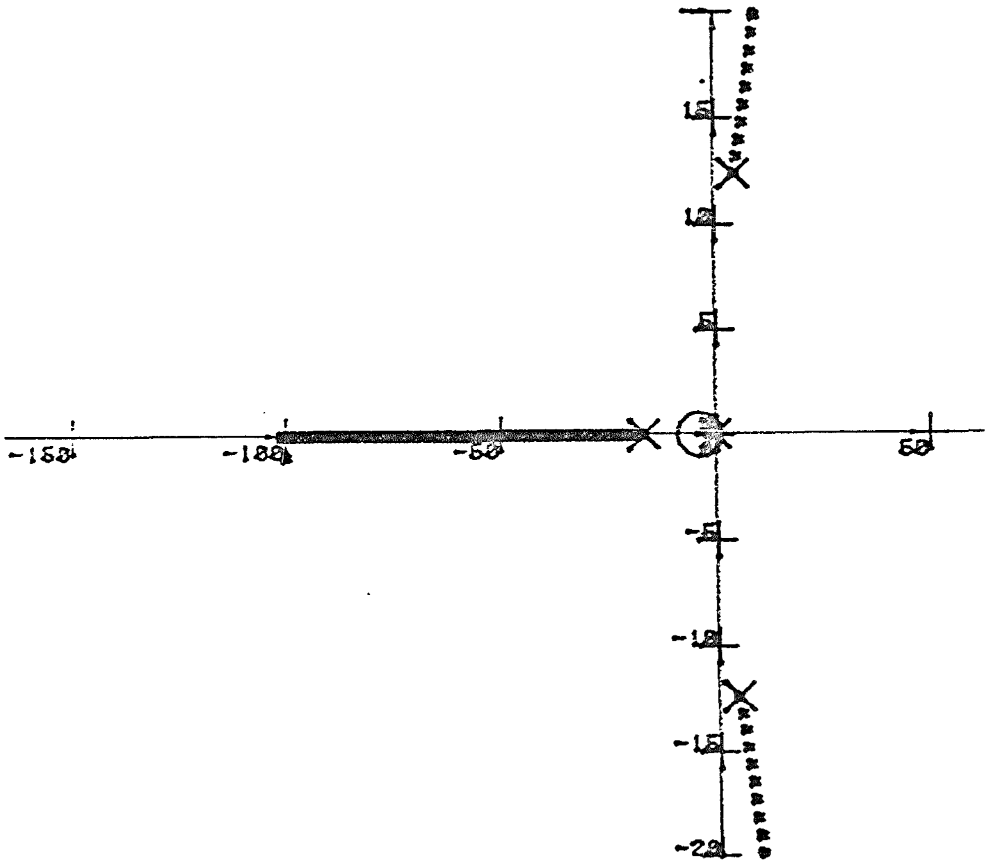
Root locus (2) corresponds to the alternative characteristic equation (embodying the modified coefficients), where the system is shown to be conditionally stable; the stability being dependant on the chosen value of proportional gain (the root locus is duplicated to show the position of the roots for various values of gain). Again however, it should be emphasised that any predictions with respect to system stability will only be valid within the limitations of the model.

As we have seen, it is difficult to determine values for system



where $K_p'' = 2 \times 10^{-6}$ m/increment

Figure (7.4.3) Elements of the Open Loop Pneumatic Drive

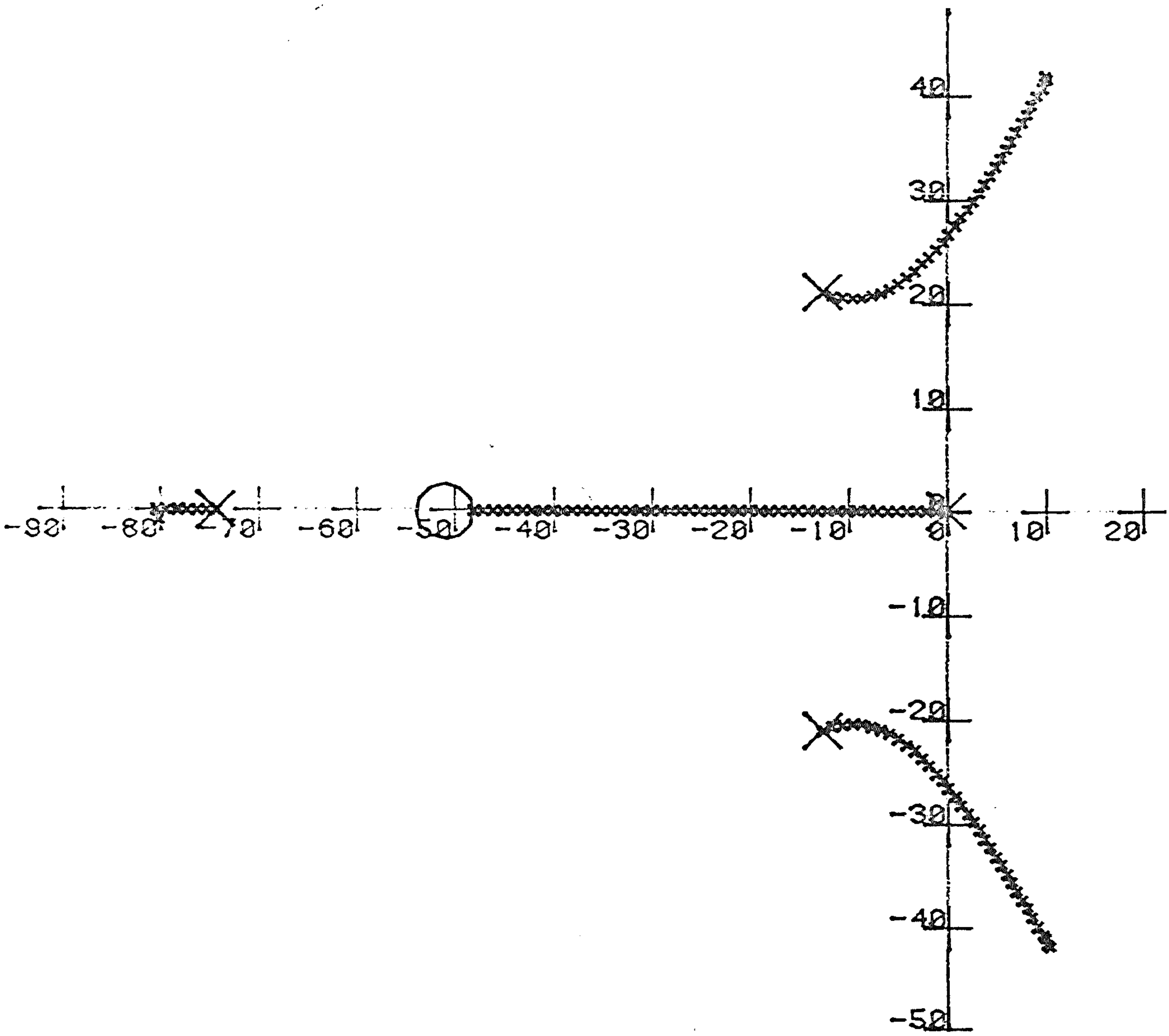


Root Locus (1)
Proportional Control
inherently unstable

$$G_p = 8 \times 10^{9}\ \mathrm{Nm}^{-2}\ \text{per m}$$

$$G_q = 2\ \mathrm{m}^{3}\,\mathrm{s}^{-1}\ \text{per m}$$

Zero		Poles	
-3.30	0.00	0.00	0.00
		5.00	12.40
		5.00	-12.40
		-16.70	0.00

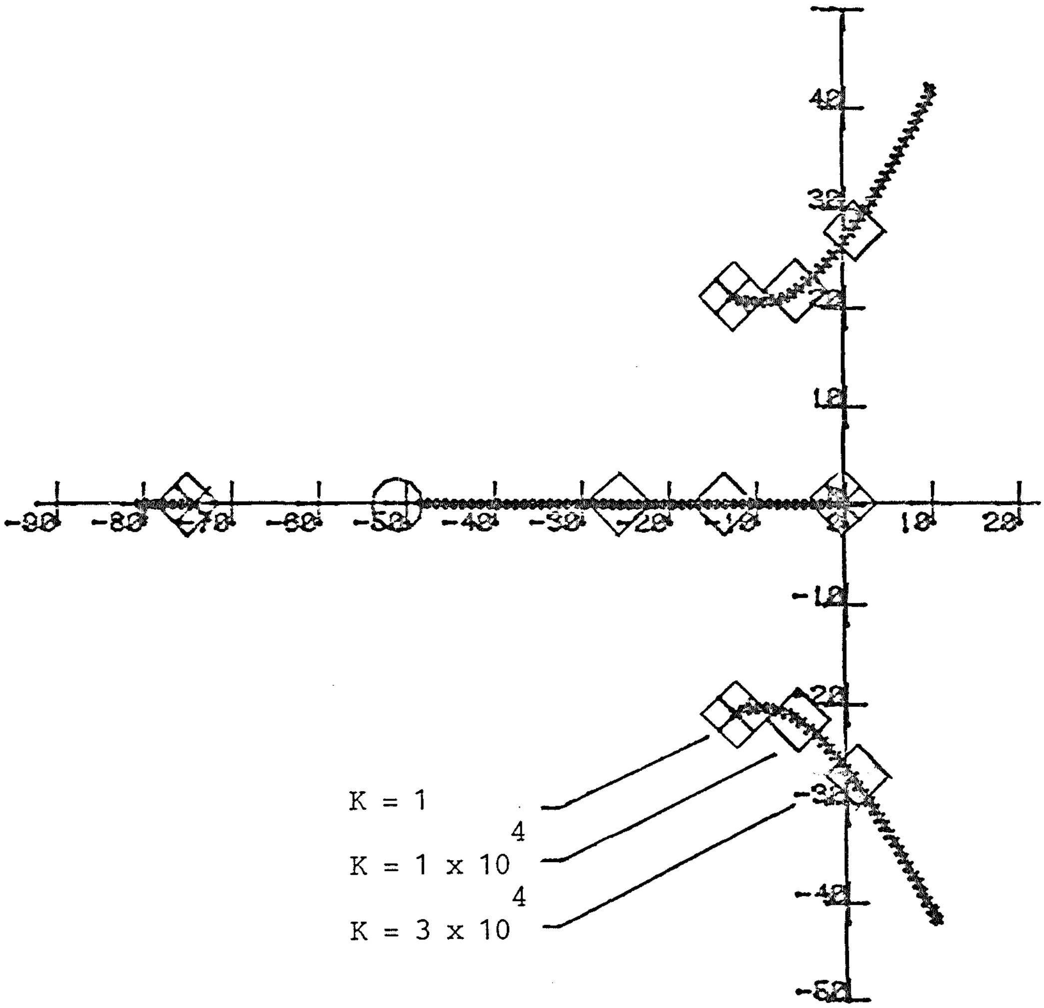


Root Locus (2).a.
Proportional Control

$$G_p = 8.82 \times 10^{8}\ \mathrm{Nm}^{-2}\ \text{per m}$$

$$G_q = 3.33\ \mathrm{m}^{3}\,\mathrm{s}^{-1}\ \text{per m}$$

Zero		Poles	
-50.00	0.00	0.00	0.00
		-12.53	21.09
		-12.53	-21.09
		-74.23	0.00



Root locus (2).b.
Proportional Control

coefficients which enable the linearised model to closely represent the real system behaviour. However, the modified coefficients at least indicate a generalised stability and behaviour spectrum of a type that can be identified on the real system (Figure [9.2.1] illustrates such a range of stability conditions).

The root loci thus far would indicate that with proportional control an increase in drive stiffness will quickly lead to unacceptable damping and eventually instability. In fact, in general, manufacturing drive systems for machines such as robots will not be required to demonstrate overshoot (ie they should not be underdamped). Inspection of equation (7.3.24) and equation (7.4.1) shows that with proportional control the coefficients of the characteristic equation will vary with load position. If the proportional gain is a constant then drive stiffness will vary as a function of load position. Despite the uncertainties associated with identifying suitable values for coefficients the form of equation (7.4.1) serves to illustrate the difficulties embodied in achieving good dynamic response characteristics for pneumatic servos, particularly with variation in load mass.

7.4.2.2. MINOR LOOP COMPENSATION

Figure [7.4.2] illustrates conceptually how velocity and acceleration minor loops can be introduced to compensate for unacceptable dynamics. Here we will consider the case were Ka=0 (ie only a velocity minor loop is introduced) for which the closed loop transfer function becomes:

$$W(s) = \frac{Kp.G(s)}{1+ \quad Kp.G(s).H(s)}$$

where

$H(s) = \{Ky + Kv.s\}$

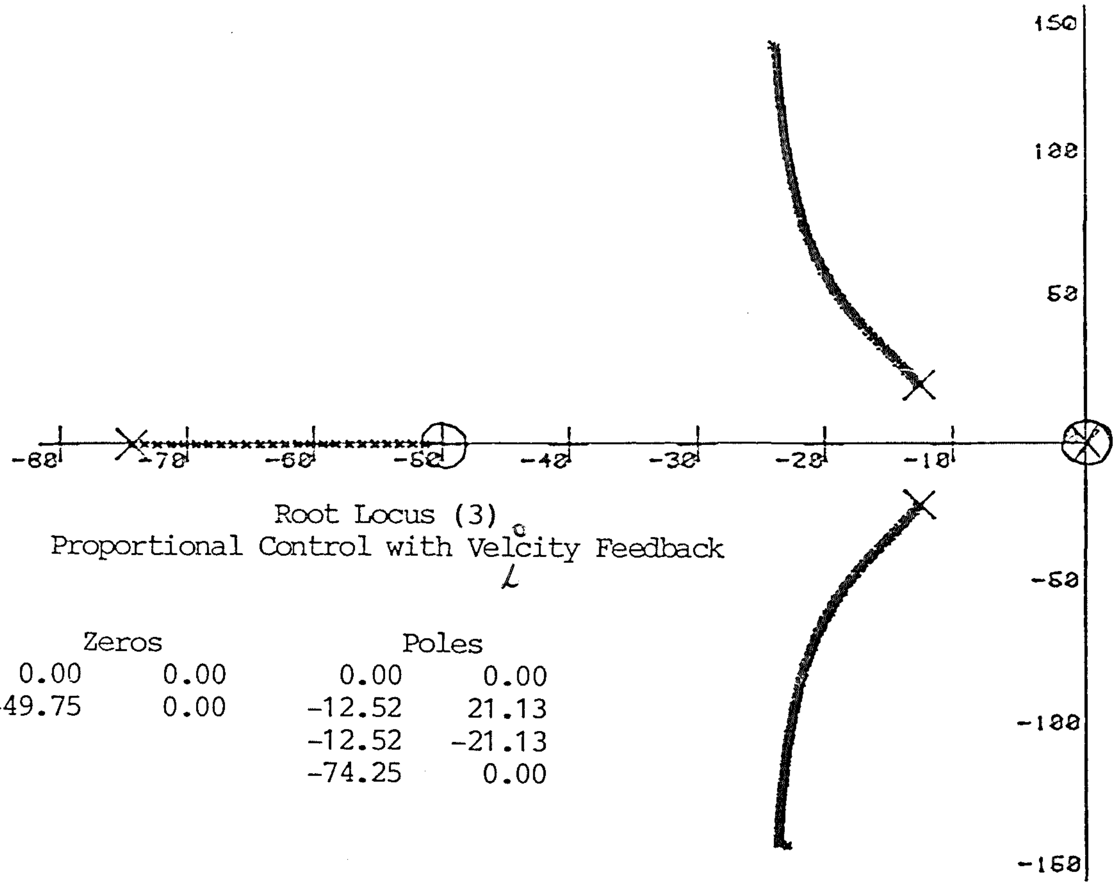
If we now consider the characteristic equation we have

$$1 + \frac{KpK\{\tau Kvs^{2} + (Kv + \tau Ky)s + Ky\}}{s^{4} + b_{1}s^{3} + b_{2}s^{2} + b_{3}s^{1}} = 0 \qquad \text{---(7.4.3)}$$

from which root locus (3) can be plotted. Again we will consider the case of a digital implementation. Furthermore we will assume that position of the load is measured and input to the controller in sampled data form (ie a new digital word representing position is sampled by the controller every ts seconds). We will also assume that the actual velocity of the load is not measured directly, thereby reducing the hardware cost of the drive, but is determined in software by evaluating the change in the position word in time ts (ie differentiating the position response). We will assume a position feedback resolution of 0.0285mm/inc for the M60110/600 module, and a controller sampling interval (ts) of 2 milliseconds.

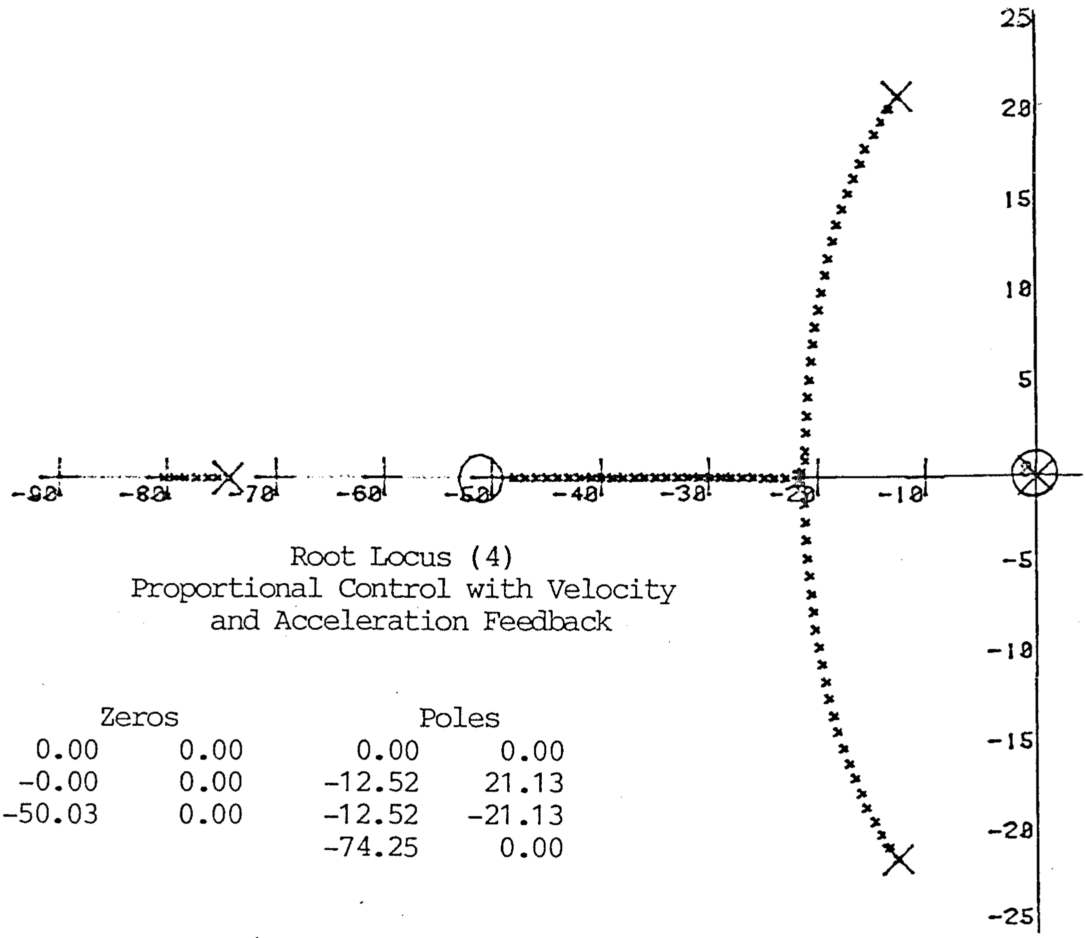
Root locus (3) relates to the case where a controller is used in which a velocity feedback gain takes a value Kv =100 set in software, and the coefficients of the characteristic equation are calculated using the modified values of Gp and Gq. The velocity feedback loop results in a modified characteristic equation, and predicts an improved system stability.

However, the system behaviour can be additionally compensated when a controller is used where Ka>0 (ie an additional acceleration minor loop is introduced). Thus for this particular pneumatic drive this leads to the characteristic equation:



Root Locus (3)
Proportional Control with Velcity Feedback

Zeros		Poles	
0.00	0.00	0.00	0.00
-49.75	0.00	-12.52	21.13
		-12.52	-21.13
		-74.25	0.00



Root Locus (4)
Proportional Control with Velocity
and Acceleration Feedback

Zeros		Poles	
0.00	0.00	0.00	0.00
-0.00	0.00	-12.52	21.13
-50.03	0.00	-12.52	-21.13
		-74.25	0.00

$$1 + \frac{KpK \{Ka\tau s^3 + (Ka+Kv\tau)s^2 + (Kv+Ky\tau)s + Ky\}}{s^4 + b_1 s^3 + b_2 s^2 + b_3 s^1} = 0 \qquad ---(7.4.4)$$

from which root locus (4) can be plotted. The velocity and acceleration minor loops result in an inherently stable system behaviour. For the practical implementation used in this study, acceleration information concerning the load was also derived indirectly. Thus the difference between two velocity words in consecutive sampling intervals could be derived in software to obtain an acceleration word. Constant feedback loop gains of Kv =50 and Kv =100 were used where it can be seen that two zeros and one of the poles are almost coincident at the origin. The effect of changing the minor loop gains was not investigated due to the uncertainty of choice of coefficients embodied within the characteristic equation.

7.5. IMPLICATIONS WITH RESPECT TO THE ANALYSIS AND DESIGN OF PNEUMATIC SERVOS

The overall objective was to formulate a system model which could be used for:

(i) control system design purposes in general terms, to make generalised conclusions concerning the gross effects of feedback design, and

(ii) selecting controller gains for different robot modules and actuators to simplify tuning etc..

Comparison between practical and theoretical studies have shown that the model can be used to make general conclusions concerning the effect of gross changes in parameters (eg area of cylinder, stroke length, mass, etc..) and feedback design. However, this work has shown that it is impossible to make more specific use of the model (ie selecting gains to simplify tuning etc..), this being emphasised by the practical problems of establishing even the order of magnitude of some of the values of system parameters.

Assuming that it is acceptable to describe the functioning of a pneumatic servo by a small perturbation linearised model it has been shown that the use of minor loop compensation, in the form of velocity and acceleration feedback, can be used to stabilise a pneumatic servo based on the use of a conventional proportional position loop. The theoretical analysis has been verified in broad terms within Chapters (8) and (9) where the influence of minor loop compensation on system stability is demonstrated.

It is apparent from the linearised model that the midstroke position should correspond to a situation where there is least damping (Burrows [1969]) and that the drive stiffness varies as a function of load position. When minor loop compensation in the form of velocity and acceleration feedback is used, the feedback loop gains can be chosen to alter the coefficients of the linearised closed loop transfer function, W(s), and hence improve the stiffness and damping of the system.

From the analysis, having established it has severe limitations, we can still use the linearised model to advantage to identify generalised controller designs. For example, in a digital controller it is possible to implement 'gain scheduling' by selecting loop gains as a function of load position thereby improving the dynamic performance. Similarly, if the proportional gain is increased the drive stiffness is increased and as such improves system accuracy. An alternative additional control technique which could be utilised would be to select loop gains as a function of change in the load mass (m). In robotic type applications the load mass is normally 'known' and this knowledge of the system can be incorporated within the controller thereby facilitating an appropriate choice of loop gains.

7.6. CONCLUSIONS

A linearised model for a pneumatic valve controlled actuator and load system has been formulated in this Chapter. The model has enabled an understanding of the physical system to be gained, providing a mechanism for understanding the inter-relationships of component elements within the system and a means of assessing the effect of gross changes in coefficients relating to the components. The linear model is used to approximate the behaviour of the physical system; however, many simplifying assumptions are embodied within the formulation of the model and additionally, linearisation techniques are only valid for small perturbations about initial conditions. The model can be used to illustrate the generalised effect of changes in controller feedback design, and also alternative controller designs can be summised from the nature of the model.

However, the linearised model cannot be used more specifically in controller design (eg selecting controller gains to simplify tuning for the range of robot modules and actuators likely to be encountered) as it cannot adequately represent the highly non-linear system. As such the linear model has only partially achieved the objectives that stimulated the original modelling studies. We could have considered developing a more detailed component orientated model or a non-linear analysis of the system but even these routes would not have necessarily provided an adequate system model. For those reasons the ensuing research study was largely formulated around testing new control system design hypotheses via practical implementations and evaluations.

The linearised model offers advantages of simplicity and can be analysed by a range of well proven techniques. However, the model has only limited validity because the linearisation is applied to a highly non-linear system. It is important that a reader wishing to extend the use of this model should proceed with caution and be fully aware of the models' limitations. Reasons why the linearised model is not closely representative of the pneumatic servo-drive can be attributed to several factors which include:

(i) to apply the model it is necessary to establish representative system parameters from the real system and this can be difficult,

(ii) the significance of component saturation effects (and the non-linearities introduced) is difficult to determine,

(iii) friction effects (and the non-linearities introduced) are ignored within the model formulation, and the possible effect of changes in friction forces in seals, slideways, etc., is difficult to satisfactorily establish,

(iv) saturation type non-linearities which are likely to occur and result because of: limits and overflow in calculations in software, electronic interface component saturation (eg linear amplifier), and saturation of mechanical system components (eg solenoid, proportional valve, etc..). Saturation can occur simultaneously in a number of components (where we have ideal component matching), but is more likely to occur in a single component,

(v) the proportional valve demonstrates asymmetry in its characteristics which cannot be accounted for within the model. Factors influencing valve asymmetry include: the matching of the active range of the amplifier to the solenoid: the assembly of the solenoid to the valve body (so that mid-range solenoid displacement results in valve null), and the use of asymmetric valve spool actuation, and

(vi) the model is not sufficiently robust to account for the diverse range of operating conditions likely to be encountered within a family of robot modules, namely variations in actuator size, payload type, friction forces, and move types.

To briefly summarise, a linear model has been formulated that has allowed an understanding of the physical system to be gained. The model can be used to illustrate the generalised effect of changes in feedback loop design, and the nature of the model serves to aid the design of new controllers. However, the model cannot be used more specifically in controller design (eg gain selection, etc..) as it does adequately represent the non-linear system behaviour.

CHAPTER 8

MOTION CONTROL STRATEGIES FOR PNEUMATIC SERVO-DRIVES

8. INTRODUCTION

In this Chapter control strategies, which achieve motion control of pneumatic servo-drives are described. An experimental based evolution of control strategy is necessary without an adequate model of the pneumatic drive existing. In this research study microprocessor based digital control was utilised. The availability of relatively low cost LSI elements coupled with developments in actuation and sensor technology make the use of microprocessor-based control systems an attractive solution. In particular, where a pneumatic drive is to be used (which demonstrates significant compressibility, little natural damping and non-linearities) then complex compensation techniques are required if performance characteristics are to be obtained to satisfy the requirements of robots and other servo-controlled manipulators.

In evolving realtime control strategies, it has been necessary to study the characteristics of the elements of the pneumatic drives constructed.

Here, the open loop response of a proportional valve controlled actuator is considered, with reference to various step inputs. Subsequently the results of a study of 'proportional' closed loop control is described. The design of more sophisticated control strategies employing compensation techniques is then described, these include the use of minor loop compensation, non-linear control system gains and outer loop 'front end' control schemes.

The control system software was designed such that the realtime control algorithm executes within the interrupt service routine corresponding to the interrupt level generated by the interval timer. The control algorithms execute at a fixed sampling rate determined by the programming of the TMS 9901 interval timer.

The background program is used to service tasks such as the processing of test data, process communication with the VDU, and update the command signal.

The evolution of control strategies was necessarily achieved using a number of module designs: the design features of these modules are described in Chapter (3).

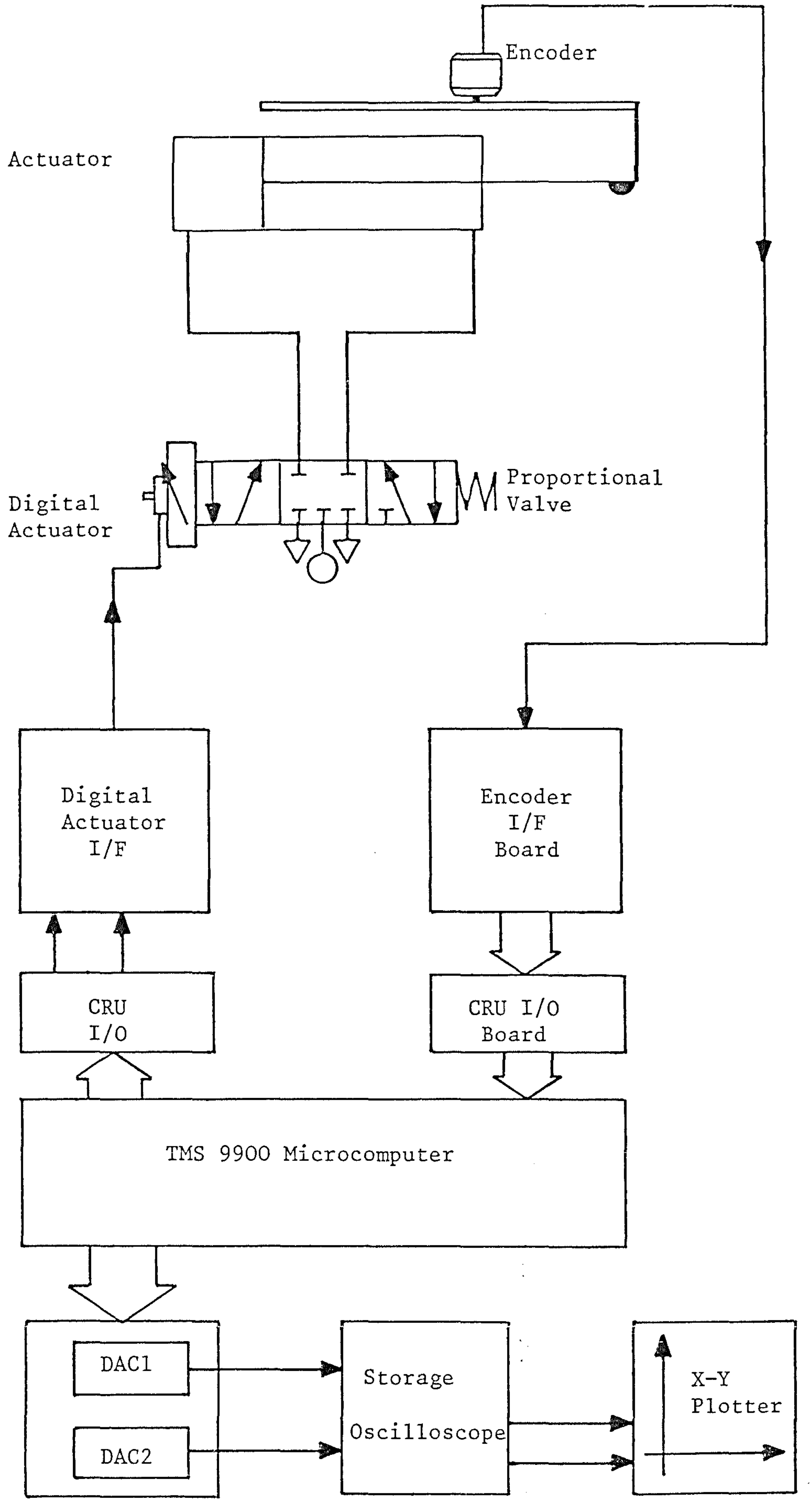
8.1. OPEN LOOP RESPONSE TESTING

The objective of open loop response testing was to identify actuator and valve characteristics. Associated with each pneumatic actuator will be a time lag, due to the compressible nature of the drive medium, the response time associated with the operation of the proportional valve, the issue of the demand, and the influence of the load and slideway friction. For a given supply pressure, the open loop response of the drive to a maximum step input to the valve, can be used to determine the maximum attainable velocity and acceleration characteristics of the drive for a given load condition: the open loop transient response thus being a quantitive measure of dynamic performance which can be compared with the transient responses when employing various forms of closed loop control.

8.1.1. OPEN LOOP RESPONSE TEST FACILITIES

Open loop responses of pneumatic actuators controlled with a proportional valve were investigated. Software facilities provided a simple test

Figure (8.1.1) Open Loop Test Hardware



environment from which 'step' input commands can be selected together with a sample data rate. The module under test is driven in open loop mode from one extreme of stroke: the open loop response characteristics of the module for a variety of valve 'step' displacements being recorded and made available for 'realtime' display. The test hardware is shown schematically in Figure [8.1.1]. The position and velocity data is output via two twelve-bit DAC's on the RT1-1241 board, which resides in the microprocessor target system, to the recording instrumentation.

When using a digitally operated proportional valve command signal updating (from the microprocessor to valve) introduces a significant time delay, which is a function of the command size. The stepper motor can only respond to control pulses at a limited frequency without losing synchronisation. The pulse buffer module ensures that the command pulse train has a controlled acceleration and deceleration rate to ensure synchronisation is maintained.

8.1.2. OPEN-LOOP RESPONSE EVALUATION

Rotary module No.3 was selected for evaluation as this incorporates a symmetrical vane actuator and rolling element slideway bearings. As such the effect of valve asymmetry can be demonstrated with reference to the transient response.

To obtain the step responses, the module is positioned at one extreme of stroke with the valve 'nulled'. A spool displacement is then demanded and the position and velocity response of the module are monitored. The procedure is repeated for a range of step changes and directions as required.

Figure [8.1.2.1] through [8.1.2.4] show the position-time and the

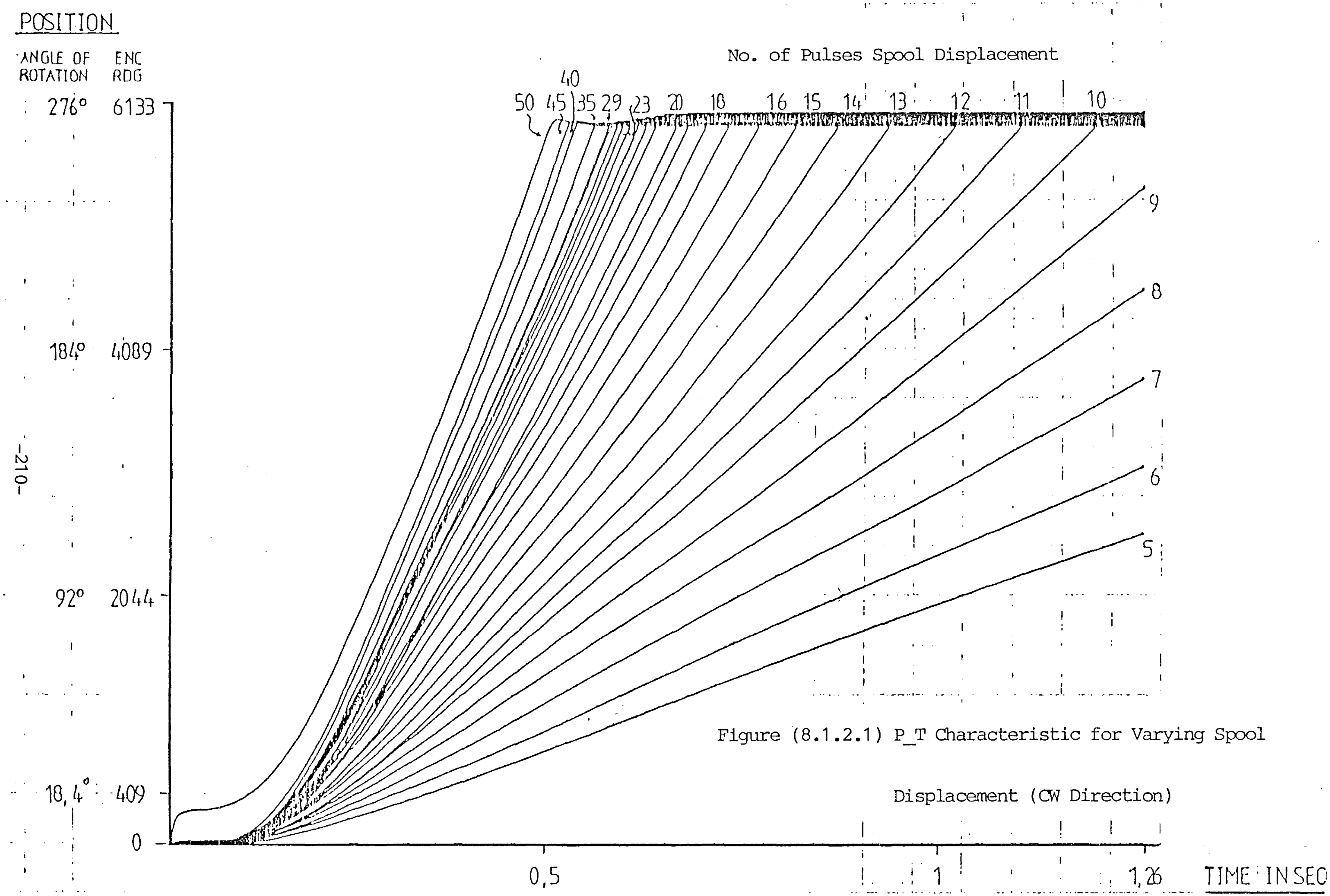


Figure (8.1.2.1) P_T Characteristic for Varying Spool Displacement (CW Direction)

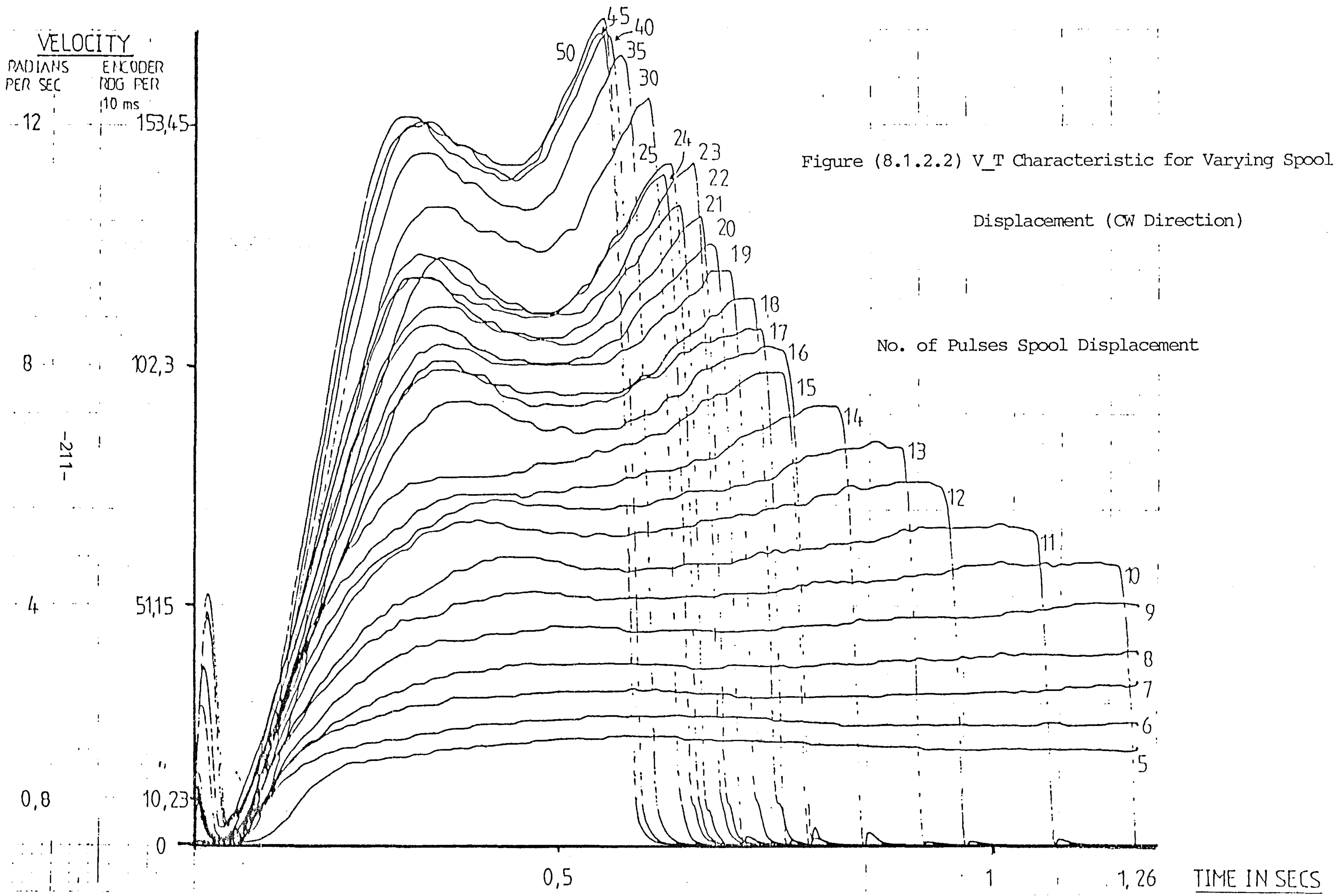


Figure (8.1.2.2) V_T Characteristic for Varying Spool Displacement (CW Direction)

No. of Pulses Spool Displacement

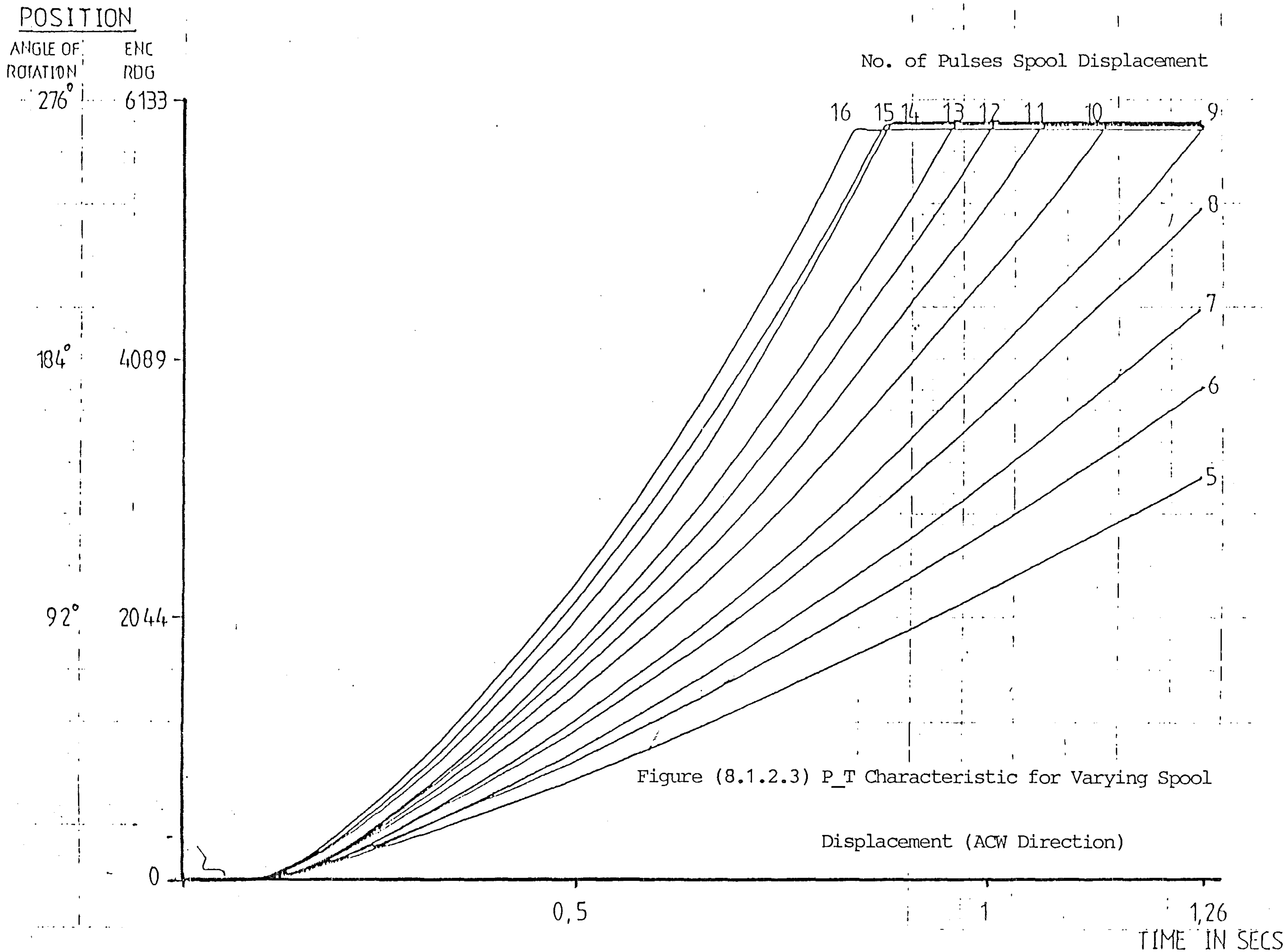


Figure (8.1.2.3) P_T Characteristic for Varying Spool Displacement (ACW Direction)

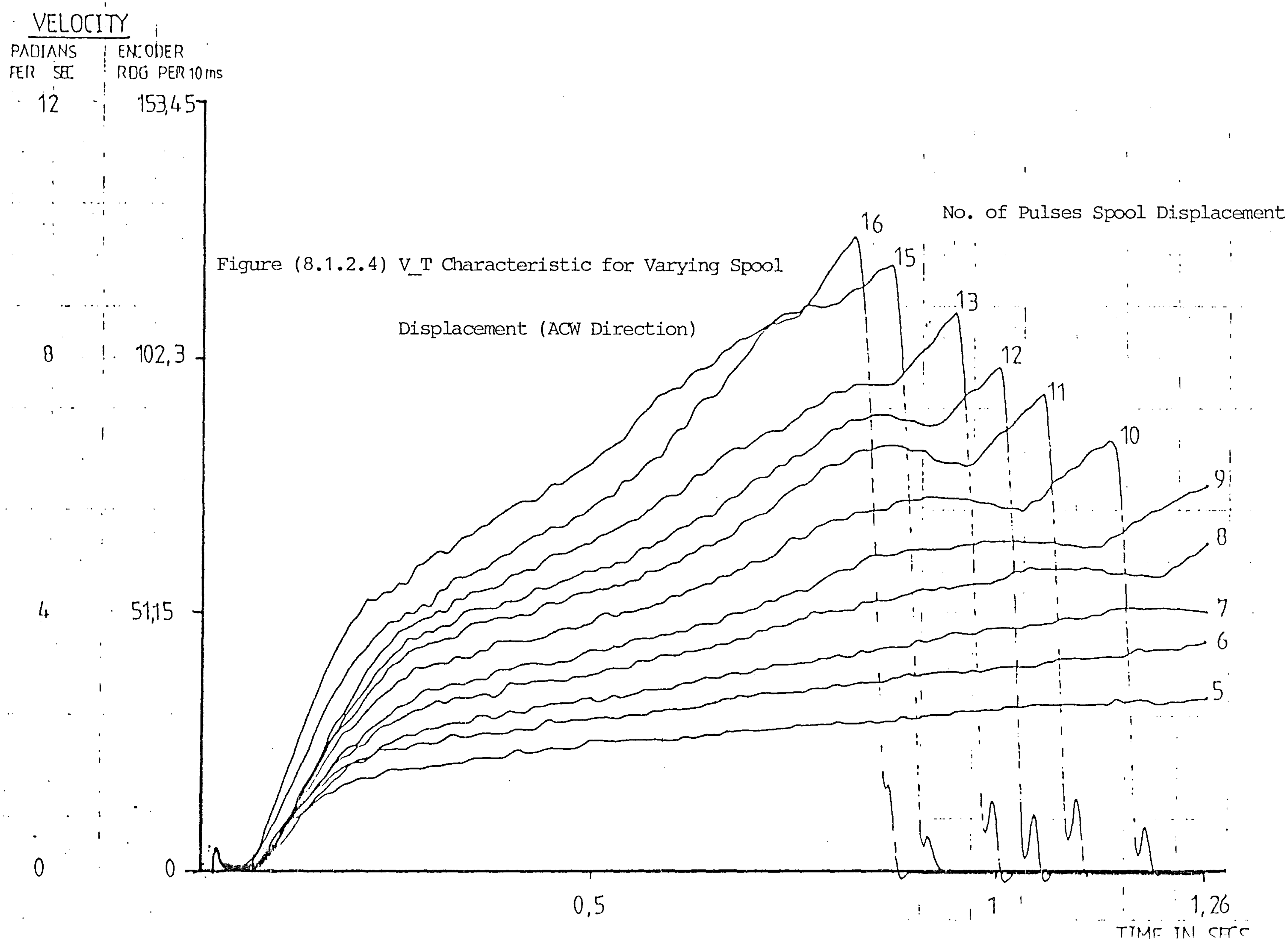


Figure (8.1.2.4) V_T Characteristic for Varying Spool Displacement (ACW Direction) velocity-time characteristics for both clockwise and anti-clockwise motion of the module for varying step commands of pulses (1 pulse = 0.025mm). The results show that the response is consistently faster for anti-clockwise motion. The asymmetry in the valve characteristics can be attributed to the valve spool sleeve arrangement and manufacturing tolerances, and could also be influenced by the asymmetric nature of the spool actuation (direct drive in one direction and spring return). The results also show that the speed of response of the module increases as the step command is increased. The time lag before the module responds to a command is attributable in part to the time response of the proportional valve, but mainly is a function of the pneumatic delay which in turn is a function of the pressure gain of the valve, the size of the control volumes and the friction characteristics of the module.

8.2. PROPORTIONAL CONTROL

Proportional error control is the most commonly used form of closed loop control strategy, where the command signal is directly proportional to the position error. Such a control strategy based on analog control system components is commonly used for the control of electric drives. With proportional control the position error (Yd-Y) is calculated and multiplied by the proportional gain (Kp) to produce the proportional term,

$$Cp = Kp(Yd-Y) \qquad \text{---(8.2.1)}$$

Proportional control is shown in Figure [8.2.1].

The relationship between the command signal and the position error is determined by the proportional gain. The higher the magnitude of gain the sooner that command signal 'saturation' is reached. With the pneumatic servo studied here, 'saturation' is determined by the physical properties of the proportional valve: saturation being a function of the port geometry and the available spool movement. When using the digitally operated valve, the command signal 'saturation' is determined by the control port geometry as sufficient valve spool displacement stroke is available to fully open the ports. In the case of the solenoid operated valve the available proportional stroke is limited to 2mm, and as such this imposes a saturation condition. Details of the valve spool 'land' and sleeve 'port' geometry is given in Chapter (6) where the proportional valve is discussed in detail. Stability assessment for proportional control using the Root Locus technique has been demonstrated in Chapter (7).

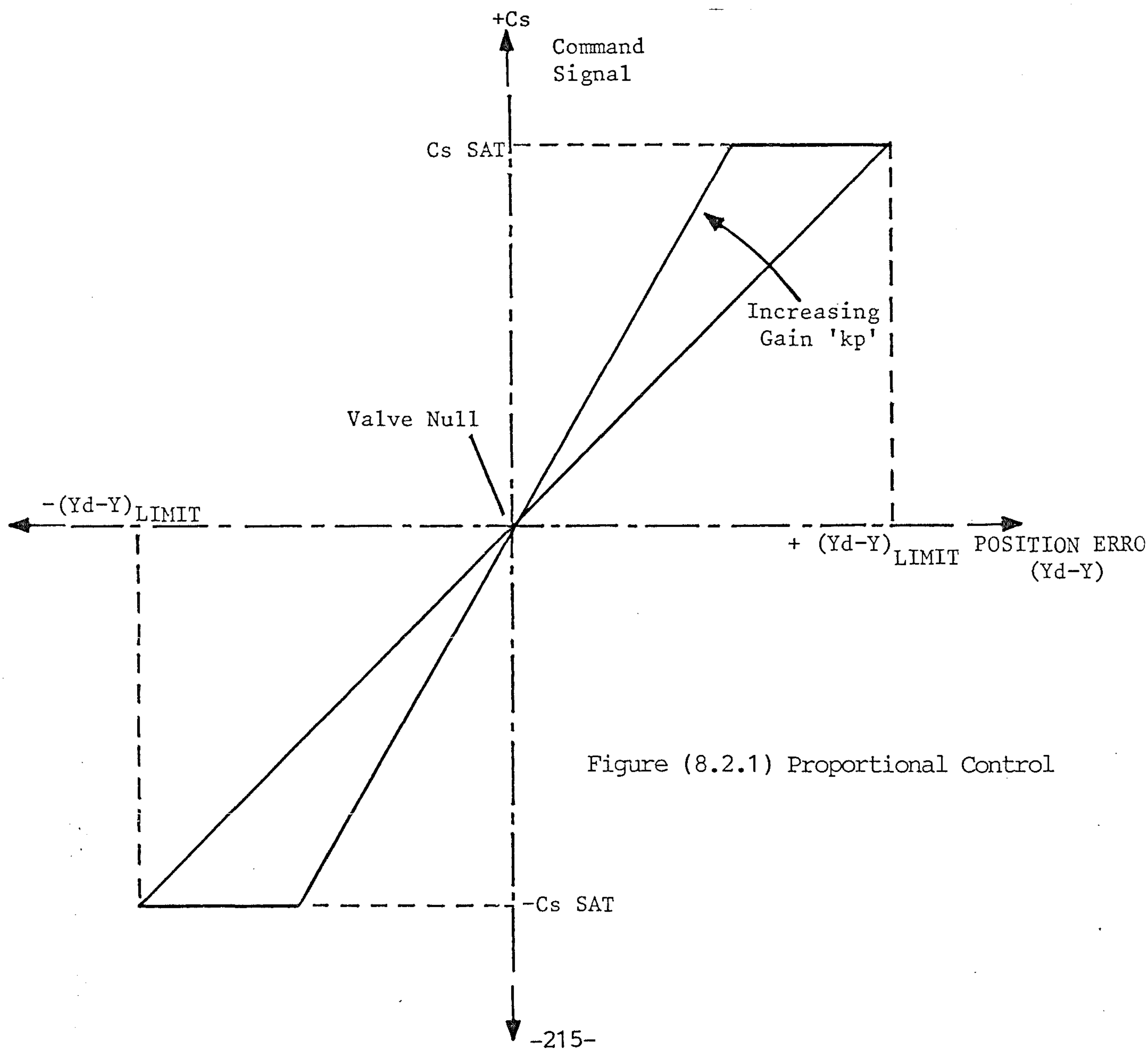


Figure (8.2.1) Proportional Control

8.2.1. PROPORTIONAL CONTROL STRATEGIES

Proportional control strategies were implemented to allow this conventional form of closed loop control to be assessed when used with pneumatic actuation. A version of 'proportional control' is shown in flow chart form in Figure [8.2.1.1]. The control algorithm is described by the equation:

$$Cs = Cp + Cnull \qquad \text{---(8.2.1.1)}$$

where $Cp = Kp(Yd-Y)$ ---(8.2.1.2)

and Cnull = valve offset for null

The proportional control strategies implemented assumed symmetry of the valve actuator and load, which often is not the case. However, the addition of bi-direction proportional gains is easily implemented digitally if so desired, see Figure [8.2.1.2], where gains Kpn and Kpp can take different values. Bi-directional gains are essential when the module is loaded asymmetrically such as when working in a vertical orientation/attitude. In this example, appropriate selection of gains Kpn and Kpp allows cycle times to be minimised for load condition and still maintain stability.

A 'brake control' algorithm is incorporated in the control scheme to control the operation of the 'lock mechanism' when the module has achieved 'in-position'. The alternative to using a brake/lock mechanism is to maintain realtime control continuously and hence compensate for any drift in position that occurs.

In-position is defined as a tolerance zone about the setpoint, within which the drive is defined as 'in-position'. This zone is known as the position 'deadband' and it normally takes a symmetrical value either side of the setpoint. Realtime control is active within the 'deadband' until the brake is applied, at which point 'null' is applied to the valve (an offset command resulting in balanced flow conditions). When the 'deadband' is set at its smallest level it defines the incremental resolution or threshold

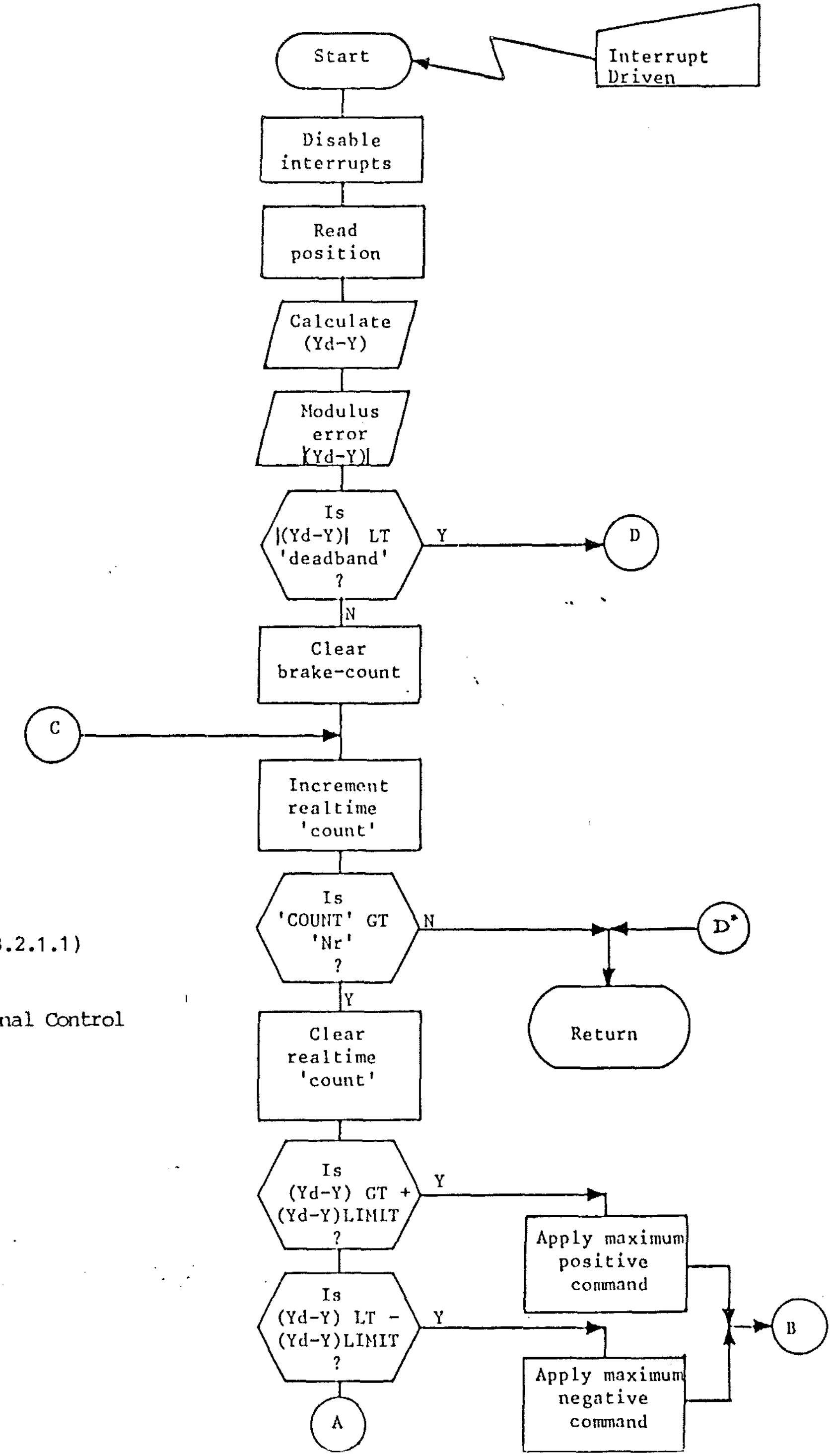


Figure (8.2.1.1)

Proportional Control

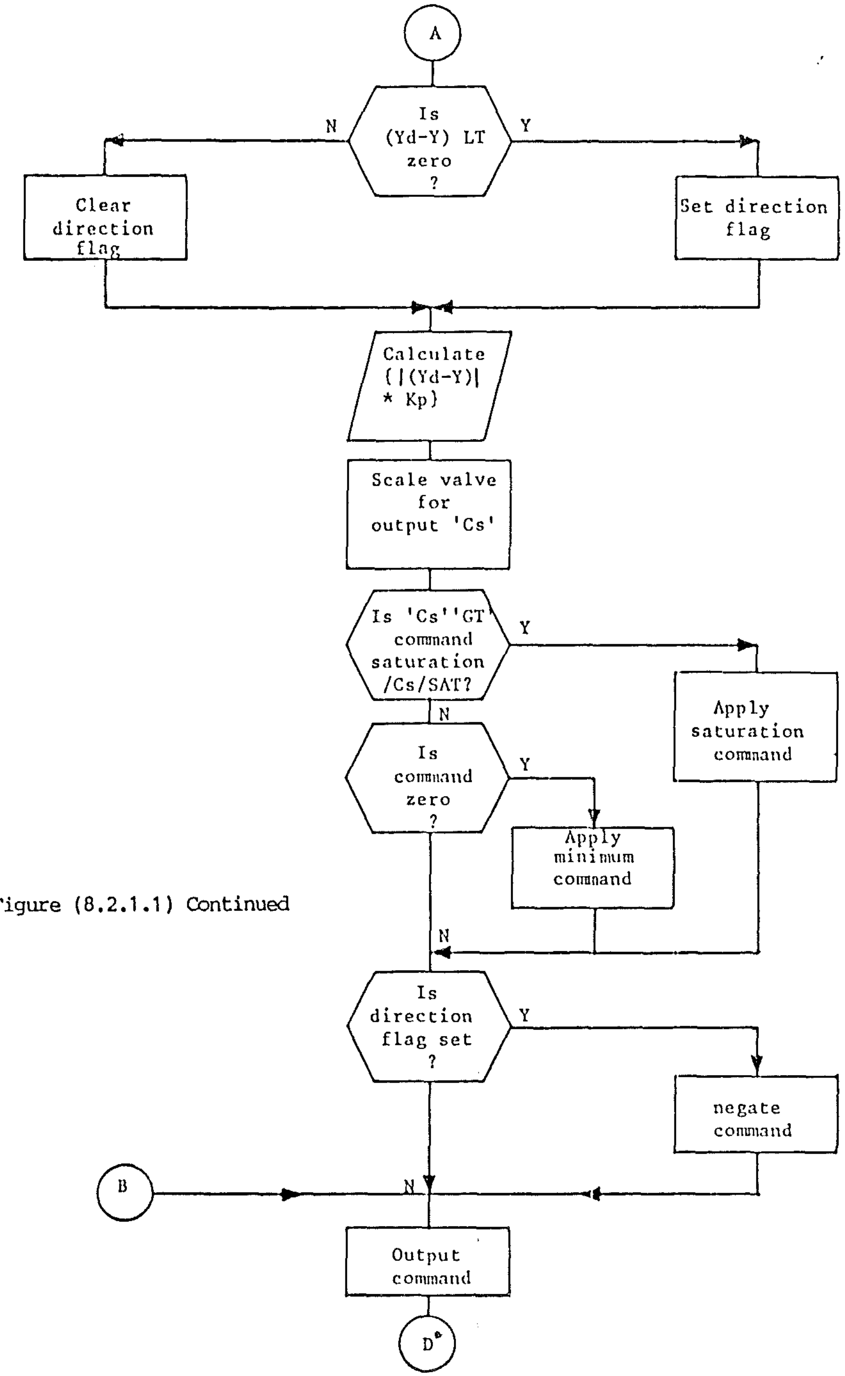


Figure (8.2.1.1) Continued

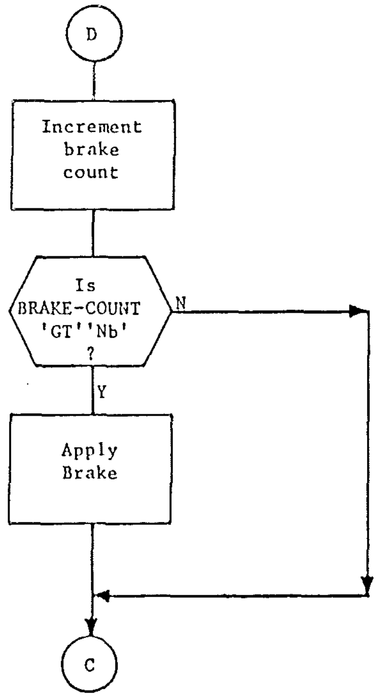


Figure (8.2.1.1) Continued

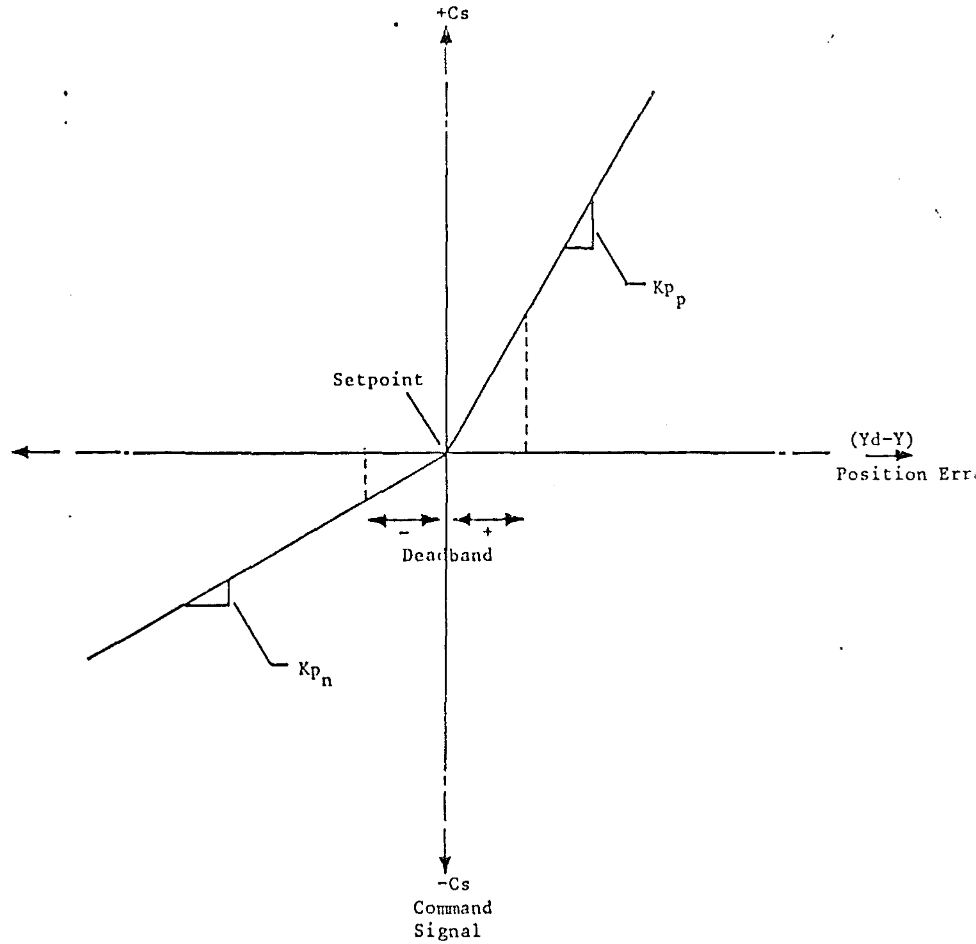


Figure (8.2.1.2) Bi-directional Gains

deadband of the servo-drive, and any step change in the setpoint smaller than the deadband would not be followed by the drive system. The incremental resolution of the servo determines the accuracy that can be achieved in 'teach' mode, and is significantly influenced by friction characteristics of the slideway and the stiffness of the servo.

The command signal is re-calculated every 'Nr' times through the interrupt service routine: in this way it is possible to select a very short sampling interval (ts) and the 'calculation rate' can be set as an increment of the sampling interval.

The algorithm incorporates saturation limits which are necessarily applied at two stages. For position errors greater than a certain value, /(Yd-Y)/limit, then the control system imposes the command saturation, Csat. By imposing such limits any overflow condition during the calculation of the command is prevented.

The algorithm sets the direction flag, according to the sense of the position error, calculates {Kp./(Yd-Y)/} and then checks for command saturation. The command signal Cs is then scaled before being output. This scaling of the command signal can cause problems, particularly where the output device has poor resolution, which is the case when using the digitally operated valve. In this case, the scaling creates a command signal 'dead-zone' when {Kp./(Yd-Y)/}<Sf, the scaling factor. This problem is overcome by introducing a minimum command, Cmin, when the scaling factor produces a zero command term.

8.2.2. PERFORMANCE EVALUATION - PROPORTIONAL CONTROL

The experimental conditions relating to the evaluation are detailed in Table [8.2.2.1]: the LUT linear module No.2 and the digitally operated

Table (8.2.2.1) Experimental Conditions

CONTROL CONDITIONS	COMMENT
Module	LUT Linear Module No 2
Module Attitude	Vertical
Load Condition	Slideway approximately 16 kg
Position Resolution	0.032 mm/pulse
Proportional Valve	Digital Actuation
Dither Frequency	N/A
Brake Mechanism	Yes
Supply Pressure	5.5 bar
Control Strategy	Proportional Control
Version	Prop Stepcon Ver 1.0
Control Parameters	PGAIN = 2 Deband = 10 (.22 mm) Spltim = 64 (ts = 0.683 ms) Clcnum = 5 (tc = 3.413 ms) Maxsat = +60 Minsat = -60 Mincom = 1 CNTRNG = 8000 Inpnum = 1

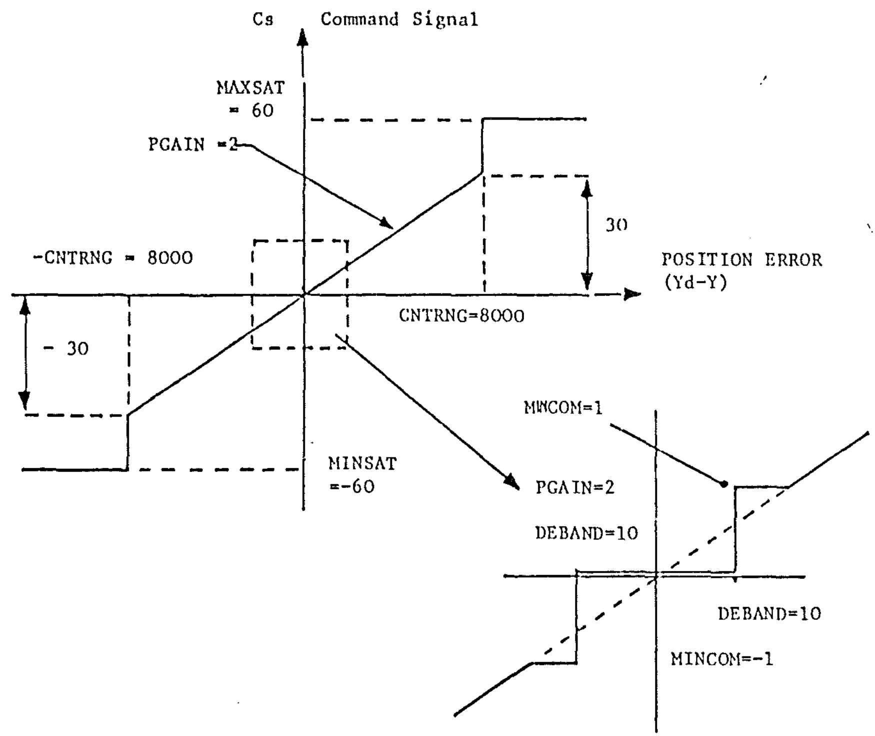


Figure (8.2.2.1) Proportional Control

valve were used. The control parameters used are listed in Table [8.2.2.1]. A standard static performance evaluation was carried out with set point changes of 64.5mm (2000 pulses): this corresponding to 13% of full stroke. Sample transient responses were recorded for a number of move lengths.

The proportional control strategy used is illustrated in Figure [8.2.2.1]: as can be seen saturation limits are included in the control.

The command signal Cs is given by the following

for /(Yd-Y)/ >= CNTRNG and either (Yd-Y) > 0, Cs=MAXSAT

or (Yd-Y) < 0, Cs=MINSAT

for CNTRNG > /(Yd-Y)/ then Cs = Kp.(Yd-Y)

where Kp = PGAIN/512

for 512.MINCOM/PGAIN >=/(Yd-Y)/ > DEBAND and (Yd-Y) > 0

therefore Cs = MINCOM

or (Yd-Y) < 0 therefore Cs = -MINCOM

and for DEBAND >= /(Yd-Y)/, Cs = 0

8.2.3. EVALUATION RESULTS - PROPORTIONAL CONTROL

The evaluation results for static performance testing are summarised in Table [8.2.3.1] and Figure [8.2.3.1], with the results of the dynamic performance tests being presented in Table [8.2.3.2]. Observation of this data and Figure [8.2.3.1] show that certain positions in the stroke exhibit relatively poor static performance. This indicates that the sliding friction characteristics of the module change as a function of position. Thus when using only proportional error control with a low loop gain, the stiffness of the drive system is low and hence the system is very sensitive to changes in load condition.

This evaluation has shown that it is possible with proportional error

Figure (8.2.3.1) Proportional Control

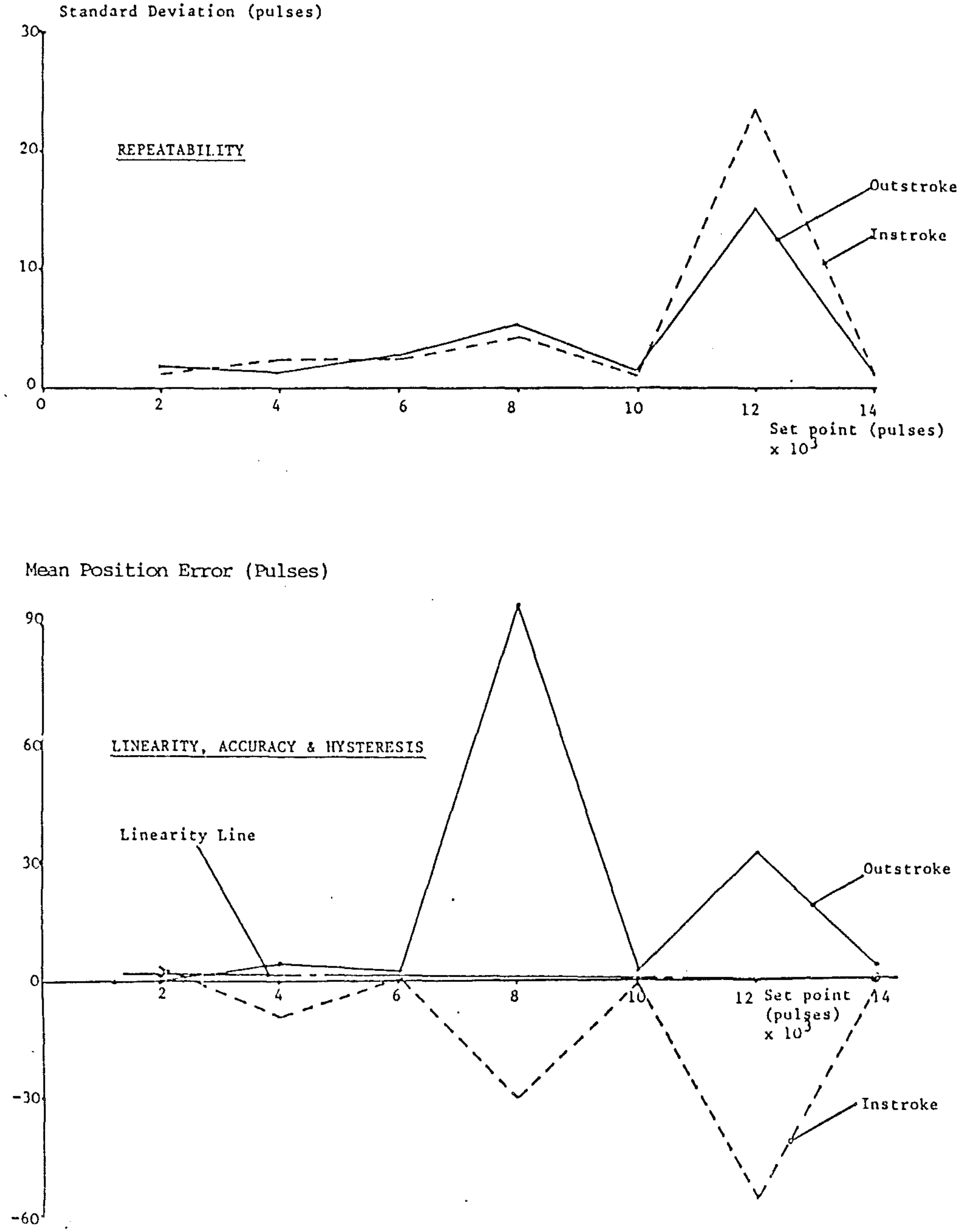


Table (8.2.3.1) Results Summary - Static Performance

Parameter	Positive /OUT	Negative /IN
Linearity (mm)	0.62	0.50
Repeatability (mm)	0.13	0.163
Hysteresis (mm)	1.12	
Accuracy (mm)	2.90	

Table (8.2.3.2) Results Summary - Dynamic Performance

MOVE	MAXIMUM VELOCITY (mms^{-1})	MEAN POSITIONING VELOCITY (mms^{-1})
Small move out (TOP)	224.2	40.1
Small move in (TOP)	131.0	34.3
Small move out (BOTTOM)	157.6	84.3
Small move in (BOTTOM)	214.7	84.0
Medium move OUT	335.8	113.5
Medium move IN	343.7	71.5
Long move OUT	439.0	151.0
Long move IN	581.7	127.0

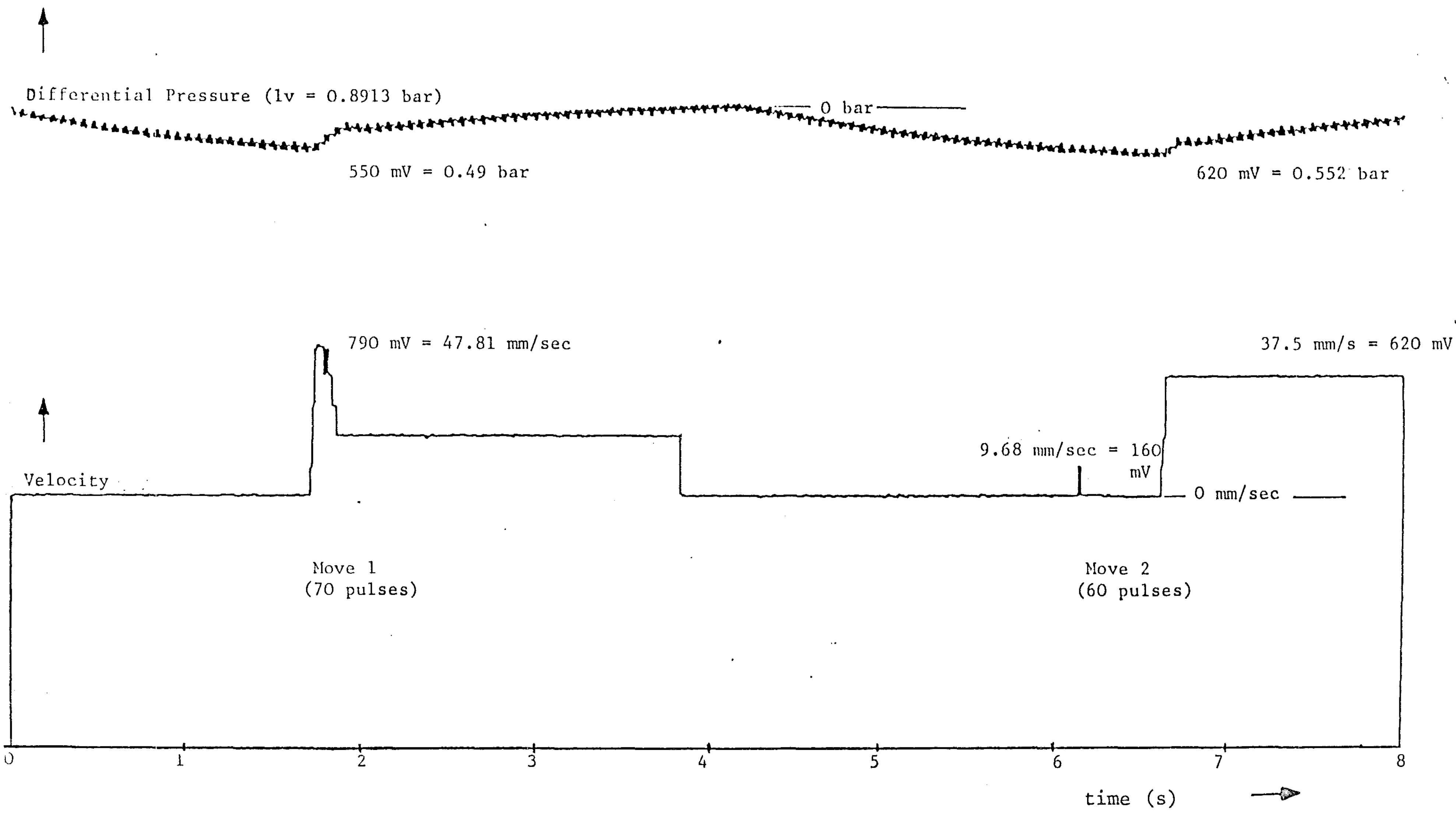
control to obtain a mean repeatability of the order of 0.17mm and accuracy of 2.9mm. This level of performance can only be achieved with the use of non-linear gains in the form of saturation limits and offsets on a module that exhibits inherently high damping. Maximum velocities of 0.5m/s were obtained for long moves, with mean positioning velocities of approximately one third the maximum velocity. It should be noted that the control parameters had been tuned to give a marginally overdamped response and that by increasing the proportional gain the response rapidly becomes unstable.

The problem of overcoming friction, particularly where short moves are encountered, highlights the limitations of proportional control. The module under test exhibits more inherent mechanical damping than other modules evaluated, but even here the forward path gain of the system $Kf=K'Kp$ must be relatively low to obtain a stable response with little overshoot: consequently the stiffness of the drive is low and hence the system is sensitive to friction conditions particularly with short moves.

The results of a position resolution test illustrated this phenomena quite clearly. A set-point sequence was selected so that a reduction in the move length was programmed with each successive command. The data from this test showed that for moves below 70 pulses (2.24mm) the static performance deteriorates dramatically.

A sample velocity characteristic is shown in Figure [8.2.3.2] which illustrates the effect of changes in differential pressure across the actuator. The differential pressure initially rises and then decays, with a small time lag the piston velocity characteristic follows this pattern. The piston movement accentuates the drop in differential pressure to such an extent that the differential pressure is no longer sufficient to overcome friction and the actuator stops. The actuator remains stationary

Figure (8.2.3.2) Response Characteristic
- differential pressure: time
- velocity: time



while the differential pressure builds up and then jerks forward again; the process is then repeated which in consequence results in poor dynamic response and considerable overshooting of the set-point with the velocity peaking at approximately 35mm/s.

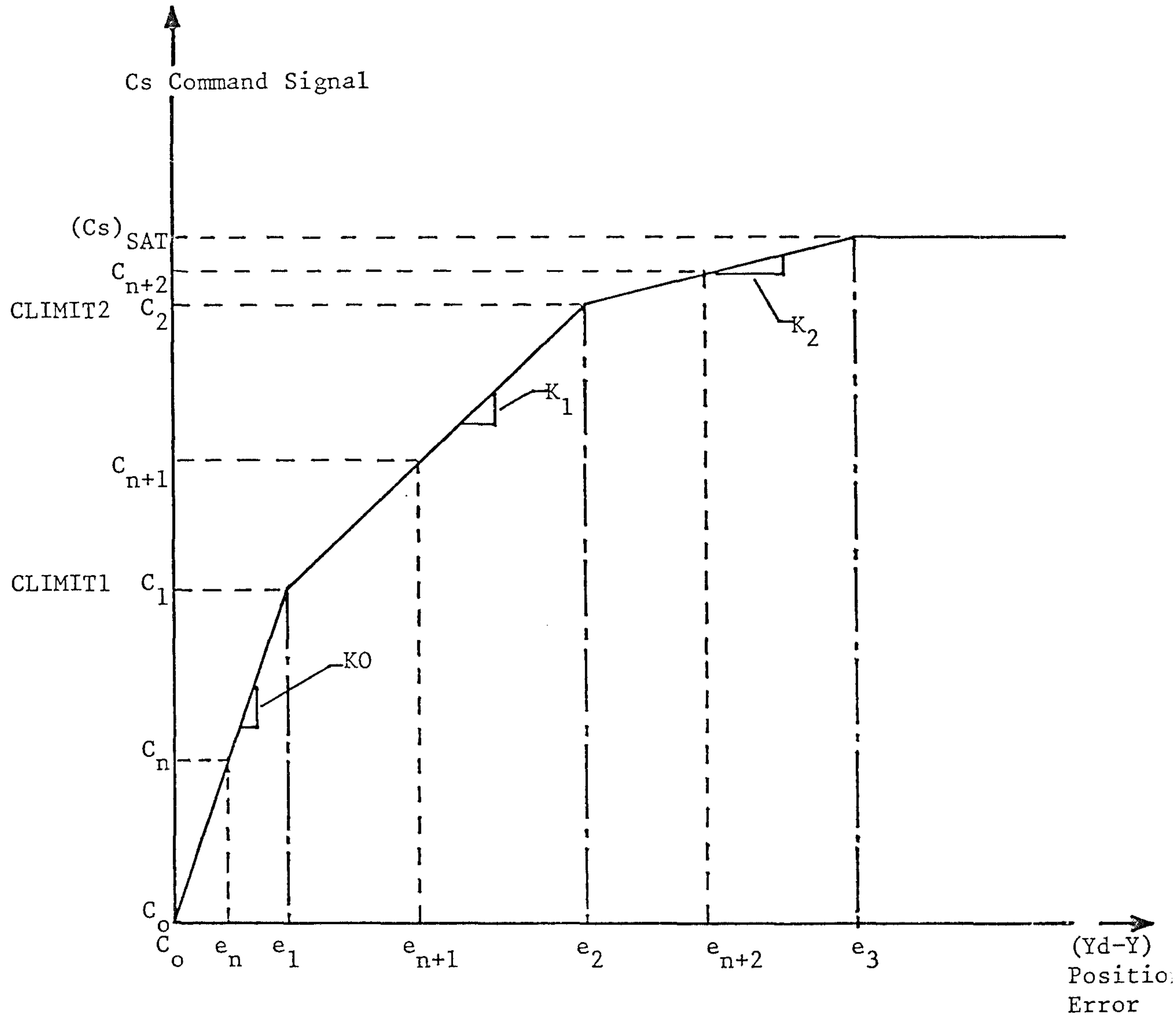
When using the digitally operated proportional valve, hysteresis in the spool and sleeve is less of a problem (ignoring backlash) but with no dither component being used its beneficial effects on reducing friction in the actuator and slideway are lost.

Proportional error control with constant loop gain when applied to modules exhibiting low levels of mechanical damping (eg M35 arm) results in an unstable response (when a high loop gain is used) or large absolute position errors (when a low loop gain is used), typically several millimetres. This effect is illustrated clearly in the next Chapter where the dynamic response of a module under proportional error is analysed. The use of non-linear proportional error gain results in an improved dynamic response and static performance.

8.3. PROPORTIONAL CONTROL - NON-LINEAR GAINS

Conventional proportional control can be modified to incorporate non-linear or multiple proportional gains. The control strategy is shown in Figure [8.3.1]. The use of multiple proportional gains allows the 'proportional gain' to increase as the error reduces: thereby applying a chosen non-linear loop gain profile. The stiffness of the drive can be increased as the setpoint is approached, thereby allowing the 'deadband' to be reduced. The lower proportional gains maintain stability of response in a long point-to-point move, but allow high gains close to the setpoint improving static performance.

Figure (8.3.1) Non-linear Gain Proportional Control



Where; $K_o > K_1 > K_2$ (9.3.1)

Example;

$$C_n = K_o \times (e_n - e_o) + C_o \tag{9.3.2}$$

$$C_{n+1} = K_1 \times (e_{n+1} - e_1) + C_1 \tag{9.3.3}$$

$$C_{n+2} = K_2 \times (e_{n+2} - e_2) + C_2 \tag{9.3.4}$$

e_1 = control zone limit (1)

e_2 = control zone limit (2)

e_3 = control zone limit (3)

The proportional gains K0, K1, and K2 have the relationship as shown in equation (8.3.1). The command signals for position errors e(n), e(n+1), and e(n+2) are calculated as shown in equations (8.3.2) through to (8.3.4) respectively. It is also necessary to define the associated control zone limits e1, e2 and e3. Once these limits are known the control strategy switching points C1 and C2 can be calculated, where;

$$C1 = (K0.e1) \qquad ---(8.3.5)$$

and $$C2 = (K1.[e2-e1]) + C1 \qquad ---(8.3.6)$$

The multiple gain proportional control strategy is shown in flow chart form in Figure [8.3.2]. The beneficial influence of a non-linear proportional gain on static and dynamic performance is evaluated in the next chapter.

8.4. MINOR LOOP COMPENSATION

Control strategies using minor loop compensation were designed with reference to the linearised model which describes the pneumatic actuator, valve, and load system, (Weston et al [1984]), and the effect of such compensation on stability is illustrated in Chapter (7). The model is an extension of an analysis presented by Burrows [1969], (which accounts for variation in system transfer function with load position), where the pressure feedback loop is replaced by an acceleration loop.

The model assumes an ideal proportional valve, a symmetrical actuator and pure inertial load and as such only approximates to the physical system. In recent years a number of researchers (Drazan et al [1976], Neuhaus [1981], Schwenzerger [1984], etc.) have investigated the use of computer controls in an attempt to compensate for the problems caused by the compressibility of the working fluid which results in low stiffness, little

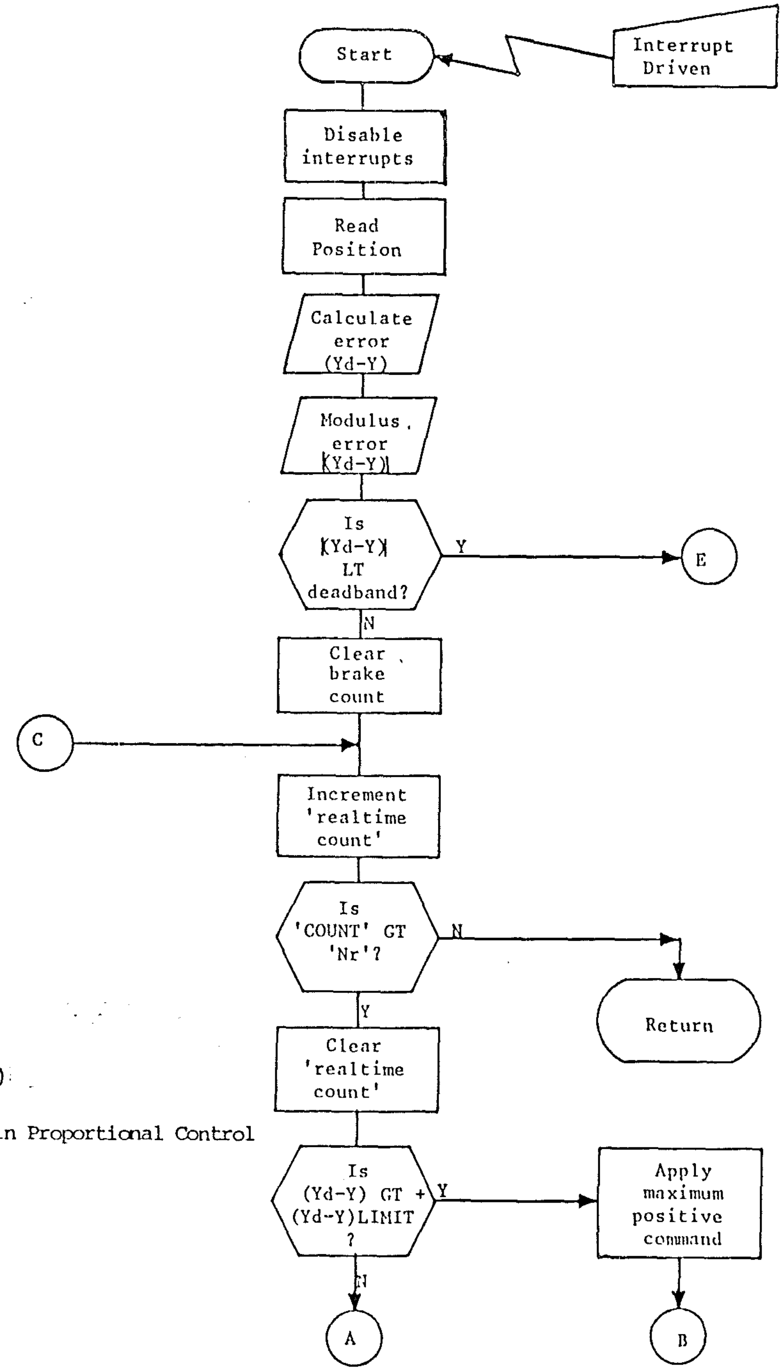


Figure (8.3.2)

Non-linear Gain Proportional Control

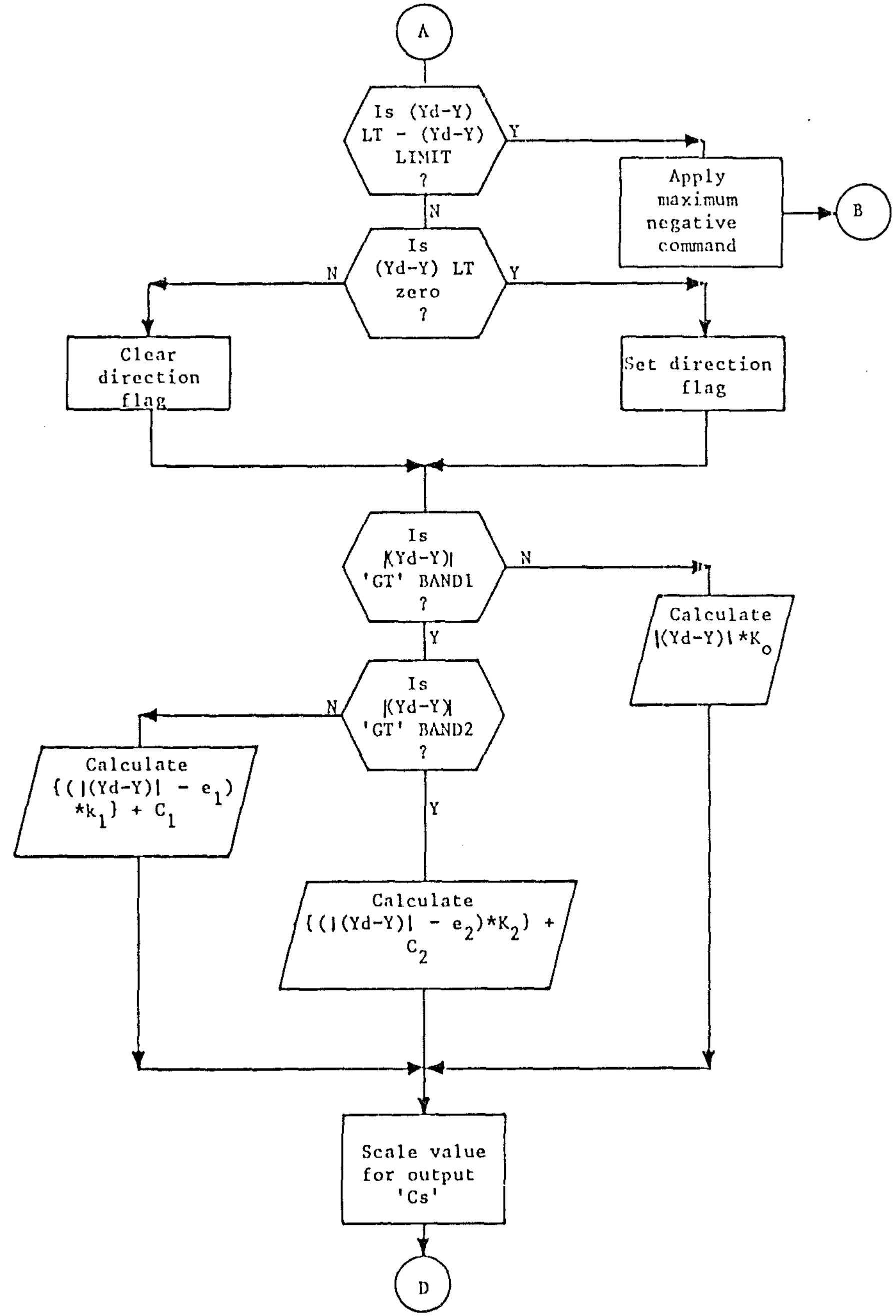


Figure (8.3.2) Continued

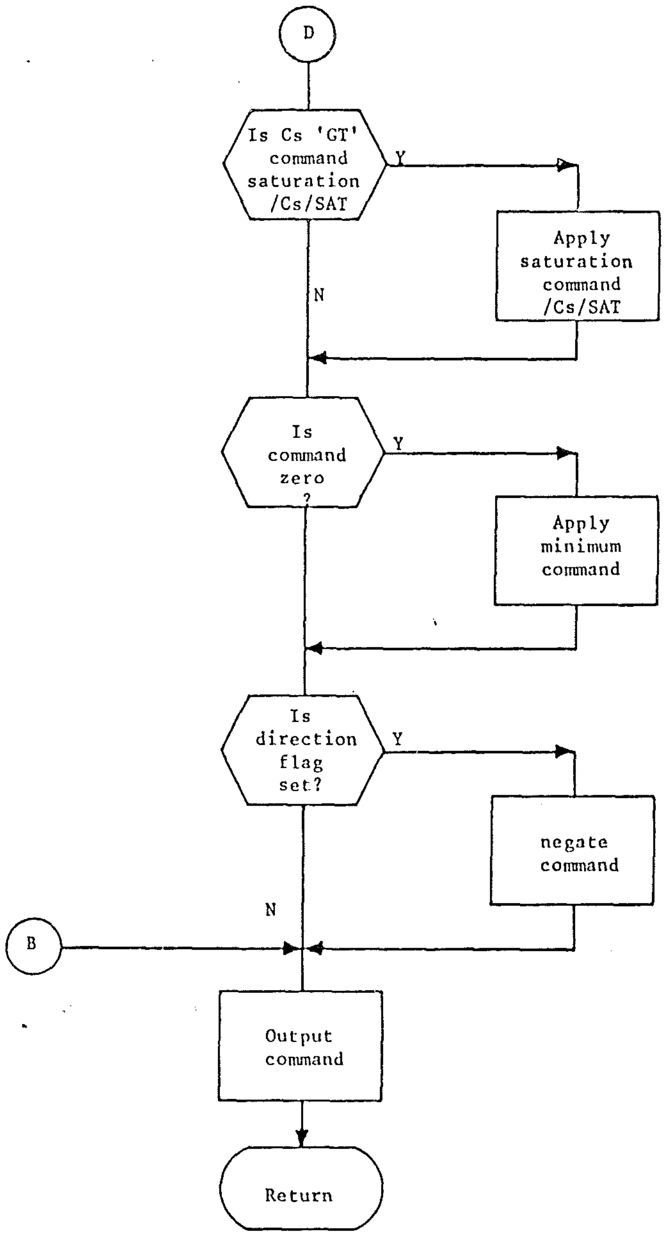


Figure (8.3.2) Continued

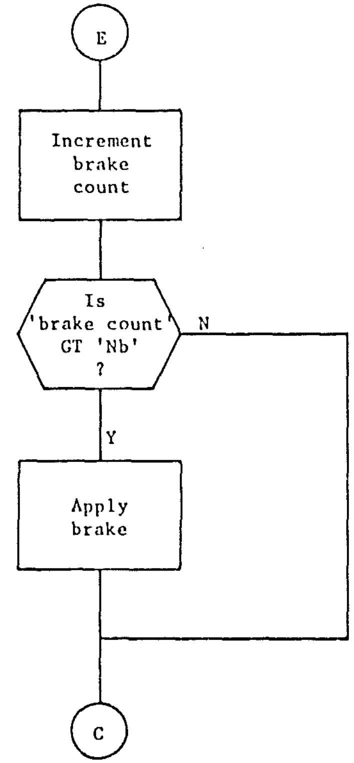


Figure (8.3.2) Continued

natural damping and complex non-linearities.

With reference to the linearised model it can be shown that the addition of a velocity and acceleration minor loop feedback terms allows adjustment of the coefficients of the characteristic equation to be made, with appropriate selection of the feedback loop gains.

The family of control strategies to be described are known as MILCOM, which is derived from the name 'MInor Loop COMpensation'. The use of velocity and acceleration feedback is only an attractive compensation technique, with respect to a commercial system, if the feedback data can be derived from a single transducer

Methods available for state variable measurement include;

(a) additional velocity and acceleration transducers
for direct measurement, and

(b) digital differentiation of the displacement
transducer signal using software algorithms.

The best quality of state variable measurement would be obtained using direct measurement, but this is unattractive due to the cost implications of additional transducers and associated interfaces. The most attractive approach is to derive the feedback data directly from the displacement transducer signal in software.

8.4.1. DERIVATION OF VELOCITY AND ACCELERATION DATA

The optical encoder converts the load displacement into two quadrature pulse trains, from which the encoder interface then interrogates the incoming pulses to produce a sixteen-bit position word. The counter buffer can be accessed at any time by the processor to provide displacement data. The displacement resolution is a function of the encoder grating line density and the feedback transmission ratio. The displacement resolution defines the maximum length of stroke that can be accomodated with the present position feedback system.

The problem is to design a software algorithm to derive velocity and acceleration of an acceptable resolution from the sampled displacement data: techniques for deriving velocity and acceleration information are described below;

(a) DIRECT DIFFERENTIATION OF STATE VARIABLES

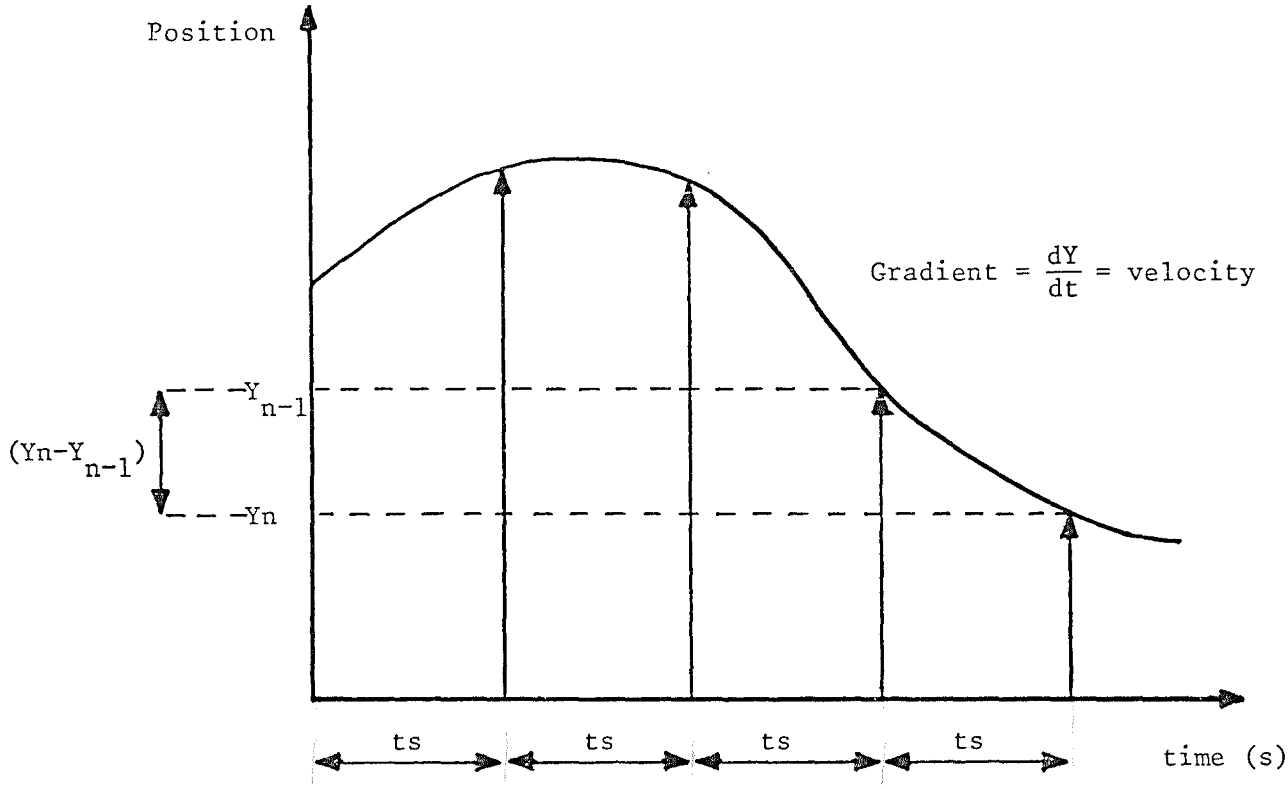
Velocity and acceleration data can be derived from the changes in displacement and velocity , respectively, over a constant period, namely the sampling interval. Figure [8.4.1.1] shows that the slope of the position-time characteristic for any sampling interval (ts) represents the average velocity over this interval.

The average acceleration is calculated from the slope of the velocity-time characteristic for any sampling interval (ts).

The sampling interval (ts) determines the resolution in measuring the velocity and acceleration. Using such an approach, a time delay is introduced: the velocity and acceleration information being delayed by ts/2 seconds. For a discrete digital system to approximate to a continuous system it is desirable to have the sampling interval as small as possible, but this implies that at low velocities the resolution on velocity measurement will be poor, with a high noise content and the acceleration data will be a factor worse. At the other extreme a large sampling interval will give good resolution, but with high module velocities the delay in the measurement of the velocity and acceleration becomes increasingly significant.

Using direct digital differentiation it is necessary to select a sampling

Figure (8.4.1.1) Velocity Derivation



interval to give a compromise between resolution and sample data rate requirements.

The velocity being calculated as follows;

$$V_n = dY/dt = [Y_n - (Y_{n-1})]/t \qquad \text{---(8.4.1.1)}$$

and acceleration;

$$A_n = d(dY/dt)/dt = [V_n - (V_{n-1})]/t \qquad \text{---(8.4.1.2)}$$

where

Y_n = position after n samples

Y_{n-1} = position after n-1 samples

V_n = velocity after n samples

V_{n-1} = velocity after n-1 samples

A_n = acceleration after n samples

t = sampling interval

(b) DIFFERENTIATION OF STATE VARIABLES WITH Nth ORDER HOLD

In this method 'm' consecutive positions are stored together with 'm' consecutive velocities in tables which have been calculated by;

$$V_n = [Y_n - (Y_{n-m})]/t$$

and the acceleration is calculated by;

$$A_n = [V_n - (V_{n-m})]/t$$

As new position and velocity values become available the two tables are updated on the basis of the oldest data is removed and the new data forces the other data to shuffle up one location.

For example, store 'm' positions over consecutive sampling intervals, for m=5, we have

SAMPLING TIME	t	2t	3t	4t	5t	6t	7t
POSITION	Y1	Y2	Y3	Y4	Y5	Y6	Y7
VELOCITY	V1	V2	V3	V4	V5	V6	V7

The velocity and acceleration are derived as follows;

VELOCITY	ACCELERATION
.	.
.	.
.	.
V5=[Y5-Y0]/mt	A5=[V5-V0]/mt
V6=[Y6-Y1]/mt	A6=[V6-V1]/mt
V7=[Y7-Y2]/mt	A7=[V7-V2]/mt
.	.
.	.
.	.
Vn=[Yn-(Yn-m)]/mt	An=[Vn-V(n-m)]/mt

In a system where rapid dynamic changes are present it is necessary to sample data very rapidly, and the approach described here will yield a better data resolution. The velocity and acceleration data is averaged over an extended period, which should result in less noise content in the data.

However, there is no ideal digital differentiation technique to derive state variable data, a compromise between sampling rate and resolution has to be made.

8.4.2. MINOR LOOP COMPENSATION WITH CONSTANT GAINS (MILCOM VERSION 1.0)

A control strategy incorporating minor loop compensation was implemented; the control scheme is illustrated schematically in Figure [8.4.2.1]. The proportional error gain 'Kp', the velocity feedback gain 'Kv', and the acceleration feedback gain 'Ka' take constant values.

The control calculation is shown in flow chart form in Figure [8.4.2.2]. The command signal within the control range is described by equation (8.4.2.1).

$$Cs = Kp(Yd-Y) - KvV - KaA + Cnull \qquad ---(8.4.2.1)$$

The strategy includes a 'control range' which defines the zone about the setpoint in which the minor loop compensation is active. Beyond these limits the command signal is set to the appropriate saturation limit. The control range is used to prevent an arithmetic overflow occurring. Overflow conditioning is a necessary feature to incorporate in any digital motion control systems.

The relationship between control system gains is shown in Figure [8.4.2.3]. As can be seen, control is active within the deadband until the brake is applied. The realtime control calculation operates every 'n' sampling intervals. The command signal will only reach zero when the setpoint is reached and the corresponding velocity and acceleration data measured is zero.

8.4.2.1. SUMMARY

The minor loop compensation control strategy has been illustrated here in its most basic stage of evolution. No formal standard performance evaluations were carried out with this implementation. Problems associated

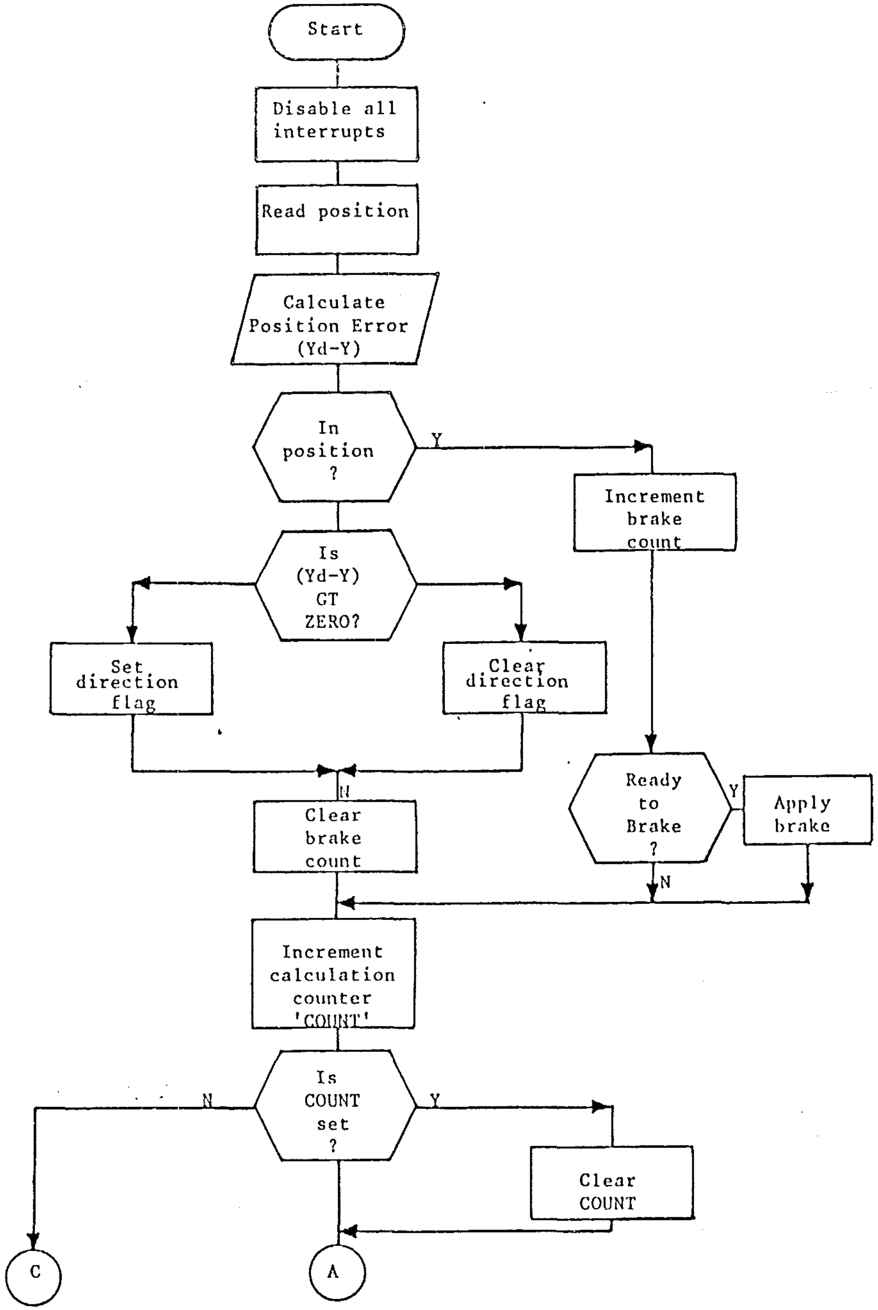


Figure (8.4.2.2) RTC - Milcom Version 1.0

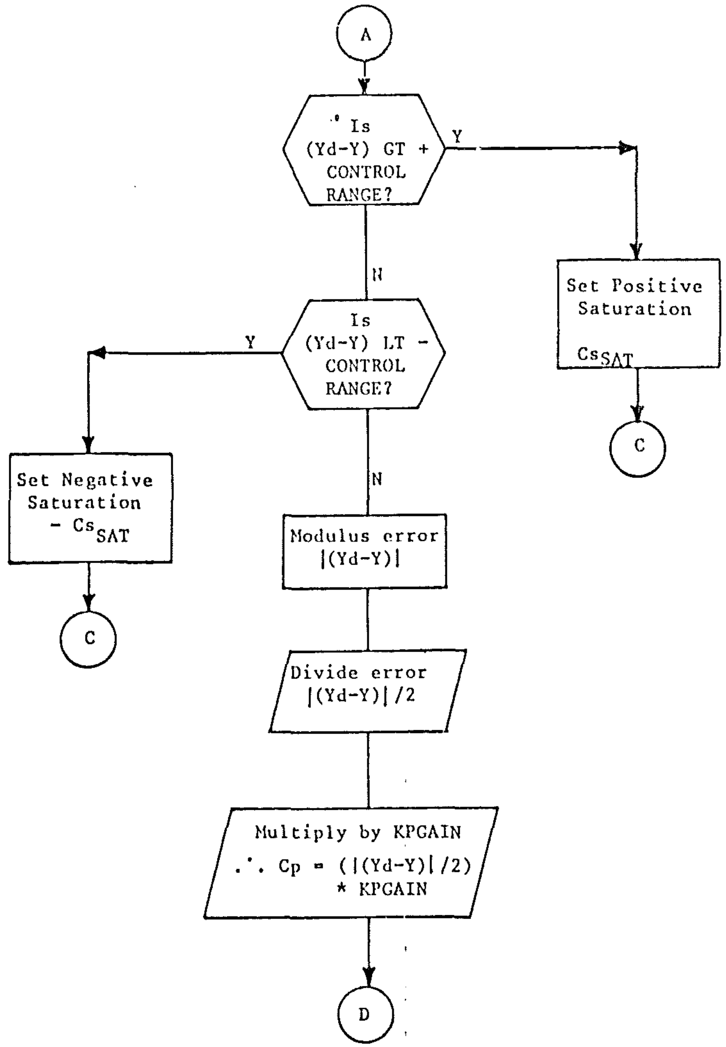


Figure (8.4.2.2) RTC - Milcom Version 1.0

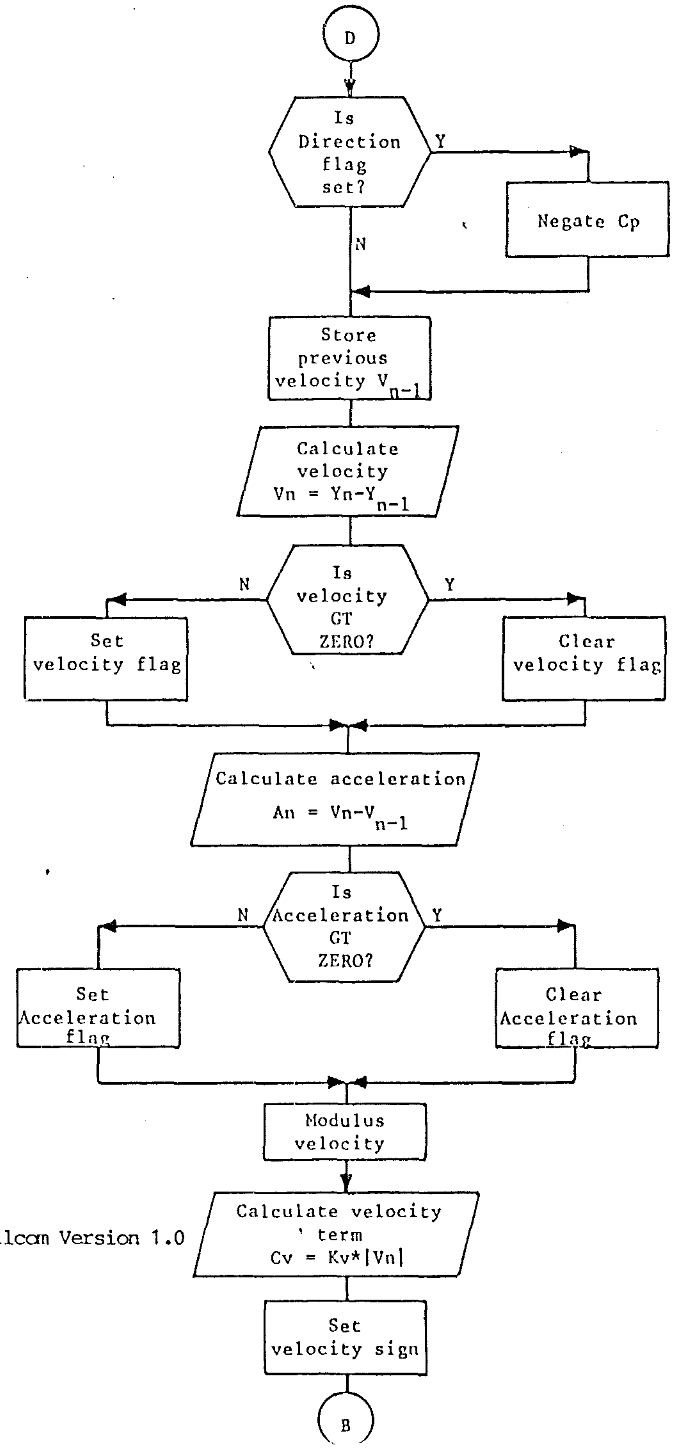


Figure (8.4.2.2) RTC - Milcom Version 1.0

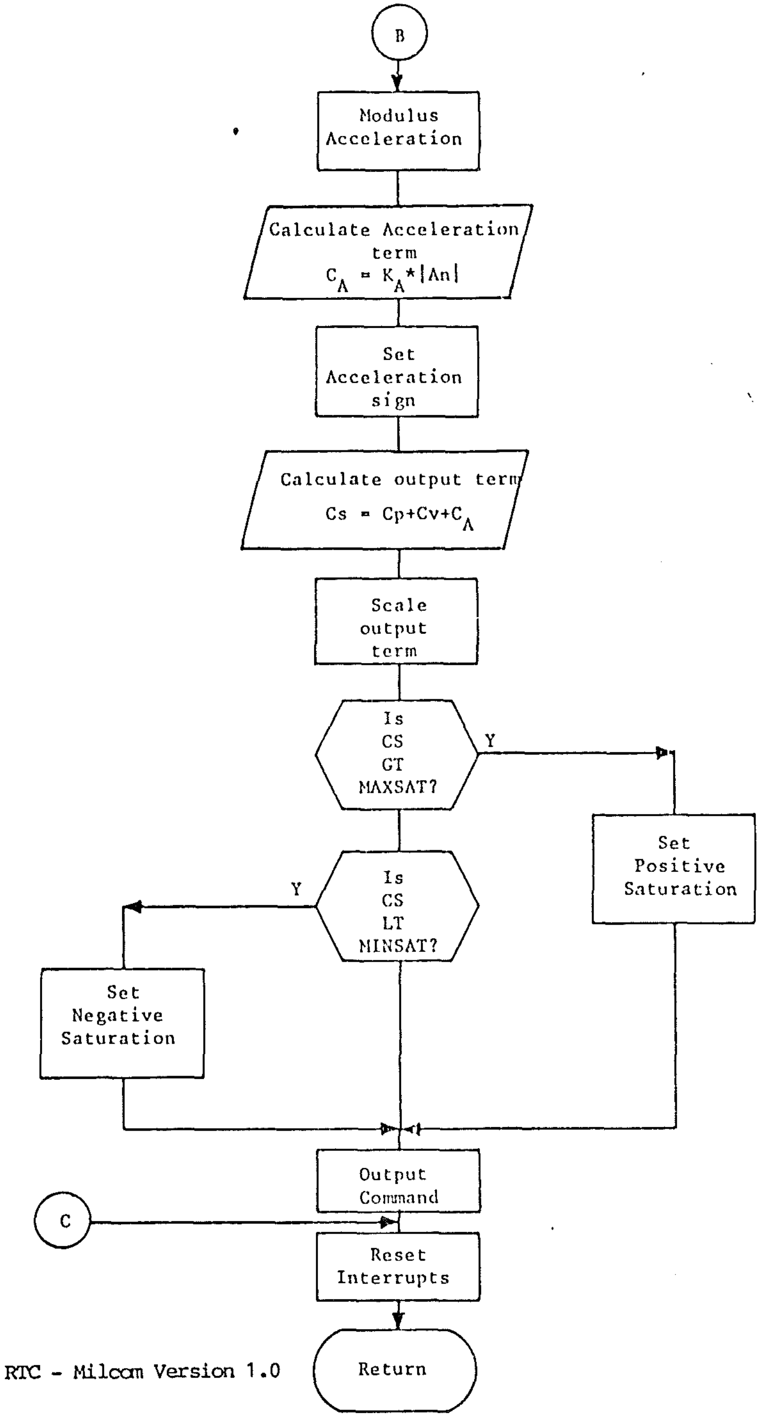


Figure (8.4.2.2) RTC - Milcom Version 1.0

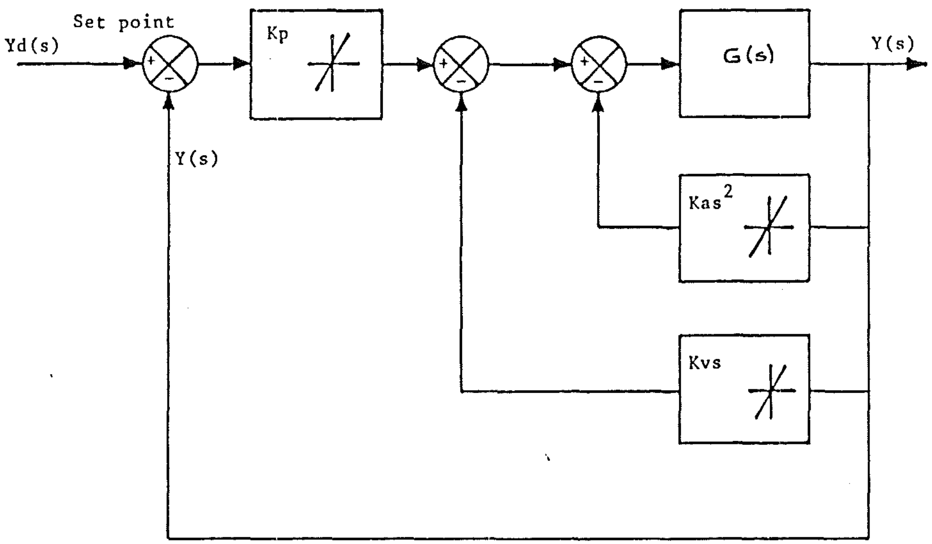


Figure (8.4.2.1) Minor Loop Compensation - Constant Gains

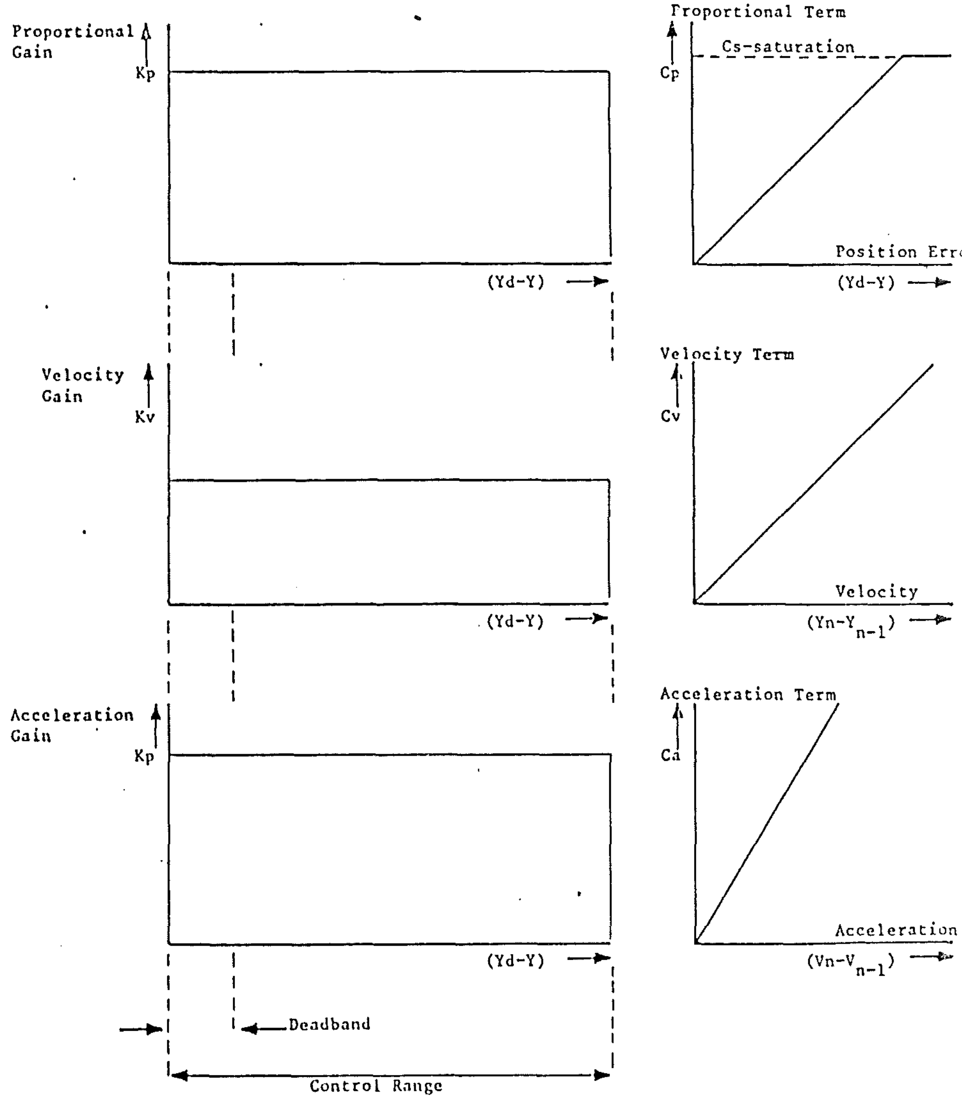


Figure (8.4.2.3) Minor Loop Compensation - Version 1.0

with the use of the digitally operated proportional valve with respect to; (a) speed of response, and (b) limited spool displacement resolution, prevented the control strategy being fully evaluated. However, during informal trials an increase in system damping due to the minor feedback loops was noticeable. The subsequent implementation of the control strategy, where the solenoid operated valve was used, includes additional control features which are illustrated in the next section.

8.5. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR (MILCOM VERSION 2.0)

The design of a control strategy with minor loop compensation incorporating a proportional term contour is now described. The inclusion of minor loop feedback enabled the level of damping to be significantly increased by appropriate selection of minor loop gains Kv and Ka.

With the level of damping relatively controllable, it was possible to improve the response time and increase the 'static stiffness' of the servo by increasing the forward path gain. This was achieved by increasing the proportional gain 'Kp'. In many practical situations sliding friction will be a dominant factor in determining the deadband of the servo. In such a case if the sliding friction force is F, when the system is stationary at the extremity of the deadband then;

$$KpK'(Yd-Y) = F \qquad \text{---(8.5.1)}$$

where KpK' = forward path gain of the system

and (Yd-Y) = the deadband about the setpoint (Yd) assuming symmetry

Clearly, as the gain Kp is increased the deadband will be reduced, consequently improving the static performance of the system. However, there is a limit on the magnitude of Kp as determined by the resulting stability of the system.

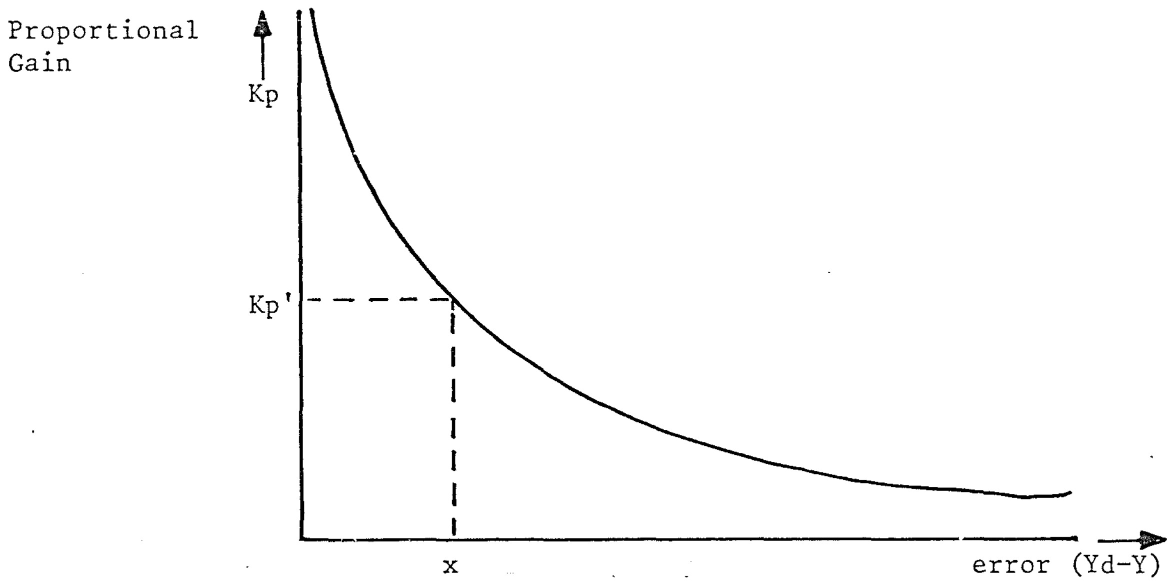
When high controller gains are encountered the command signal will saturate rapidly and the command signal will alternate between opposing saturation levels. If the command signal saturates very close to the setpoint instability can result. For these reasons it is desirable to have a proportional gain 'Kp' that is non-linear, so that 'Kp' can be chosen as a function of error (Yd-Y) so that high static stiffness can be introduced for small errors without introducing instability of larger error values.

A desirable characteristic for the proportional gain 'Kp' is shown in Figure [8.5.1], where it takes a non-linear relationship as a function of the position error. To implement such a control characteristic it is necessary to have a function relating the gain to the position error. This function could be parabolic or hyperbolic in form, each would provide a desirable characteristic, but both include a squared term. These functions are difficult to implement using integer arithmetic and would significantly increase the processing time. The proportional gain characteristic shown in Figure [8.5.1] would produce a proportional term of the form shown in Figure [8.5.2].

The non-linear characteristic of the gain can be pre-calculated and stored as a contour/look-up table in memory. It is then only necessary to look up the appropriate gain and multiply by the error (Yd-Y) to produce the proportional term.

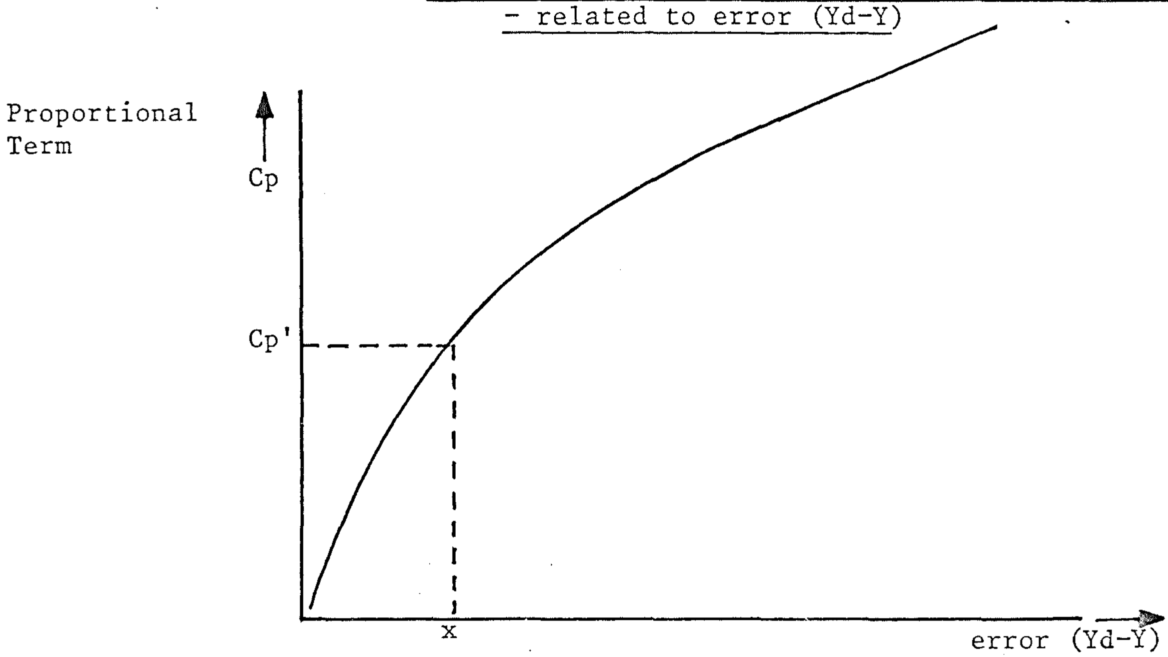
In implementing such a strategy it was felt that the process could be further simplified by pre-calculating the proportional term 'Cp' and storing this directly for reference as a contour in memory. This has further benefits in regard to quantisation of signal levels. In the

Figure (8.5.1) Desirable Proportional Gain Characteristic - non linear function related to position error (Yd-Y)



where $Cp' = Kp' x (Yd-Y)_x$

Figure (8.5.2) Proportional Term associated with a non-linear gain - related to error (Yd-Y)



algorithm calculation it is then a simple case of looking up the appropriate proportional term value for a particular position error. This value can then be scaled and used in the command signal calculation. By scaling the 'Cp' contour it is possible to match exactly the physical characteristics of any module to achieve a desired performance. The effect of scaling the contour can be seen in Figure [8.5.3]. The parameter used to select the contour scaling factor is 'MNGCNT' where the contour scaling factor (C.S.F.) is calculated as follows;

$$C.S.F. = 16 / \text{'MNGCNT'} \qquad ---(8.5.2)$$

from which the scaled proportional term is derived;

$$Cp' = Cp.(16 / \text{'MNGCNT'})$$

The software design for processing the proportional term and scaling is shown in flow chart form in Figure [8.5.4]. The proportional term contour can be 'mapped' over any desired length of the stroke by selecting the appropriate value of the parameter 'CNTRNG'. When an error generates an index value greater than 4064 the index is reset to its maximum value, and the proportional term saturates.

The command signal within the control range can be described by;

$$Cs = Kp_{nl} (Yd-Y) - KvV - KaA + Cos + Cnull \qquad ---(8.5.3)$$

where

Kp_{nl} = non-linear proportional gain

Cs = command signal

$Cnull$ = offset to achieve valve 'null'

Cos = valve offset from null to reduce system hysteresis

This control scheme is shown schematically in Figure [8.5.5].

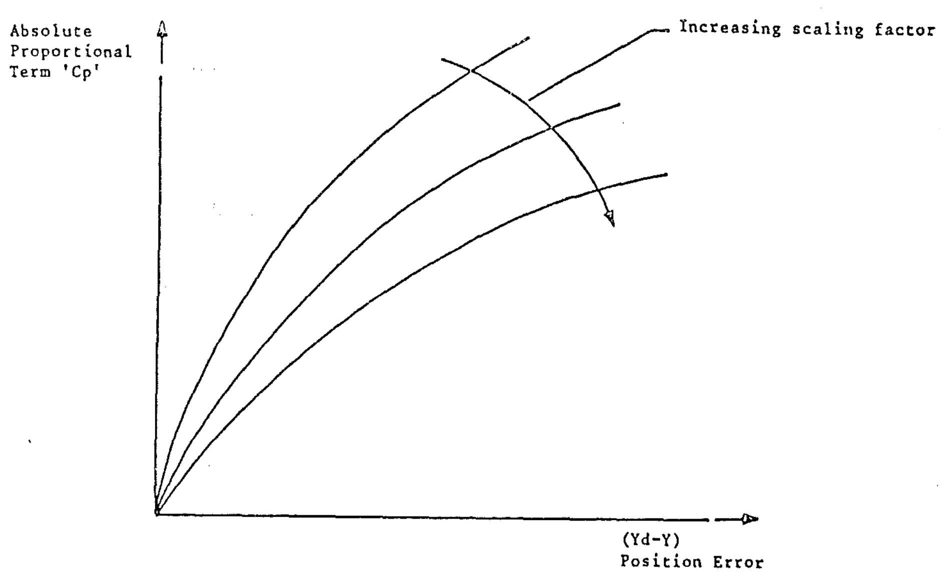


Figure (8.5.3) Effects of Contour Scaling

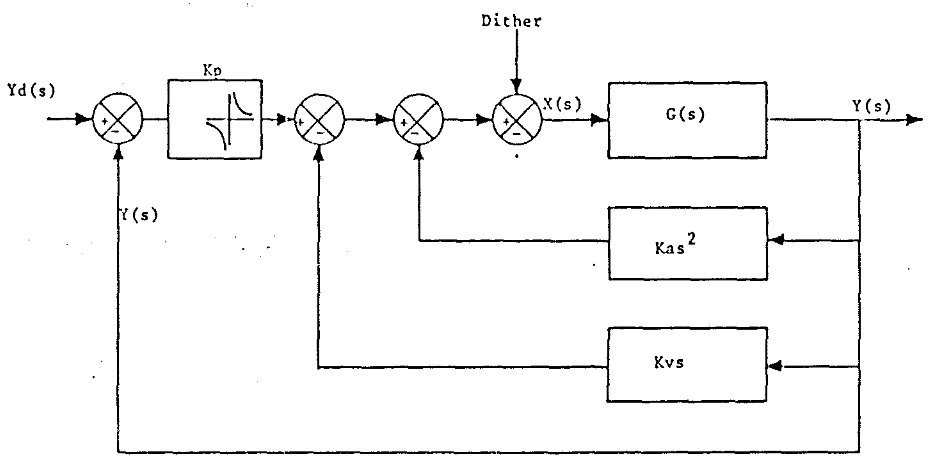
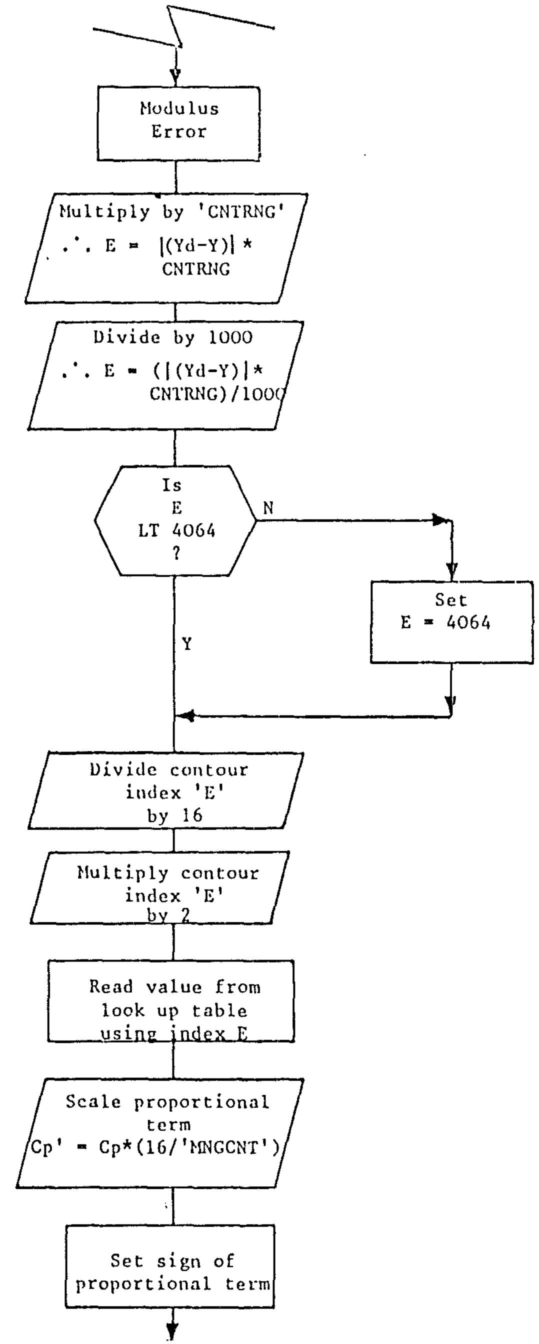


Figure (8.5.5) Minor Loop Compensation - Version 2.0

Figure (8.5.4) Proportional Term Contour Selection Process



The offset 'Cnull' is necessary to produce valve 'null' when using the solenoid operated proportional valve, it relates to the mid-point of the DAC operating range. The solenoid operated valve was used for all subsequent work in the research study, as the solenoid was attractive from the commercial point of view (lower unit cost) as well as technically it provided a faster response time and greater spool displacement resolution.

The use of dither overcomes many of the hysteresis problems associated with the solenoid operated valve, and also reduces other mechanical hysteresis in the drive. The control system has an associated operating 'dead zone', which is significantly reduced with the inclusion of the dither component. This dead zone can be attributed to friction in the actuator/slideway and valve, and electromagnetic hysteresis in the solenoid. With such a dead zone it is necessary to generate a minimum command value from 'null' before any response of the load is registered. In a digital system it is possible to include discontinuities in the control to accomodate these conditions. Inclusion of the valve offset 'Cos', (see Figure [8.5.6]), where Cos is a constant for any actuator and load condition, allows compensation to be made for this dead zone in the operating characteristic, linearising the response. To overcome asymmetry this valve constant is set using the parameters 'OFFPOS' and 'OFFNEG' respectively for opposing directions of motion.

8.5.1. PERFORMANCE EVALUATION - MINOR LOOP COMPENSATION (MILCOM VERSION 2.0)

Two standard performance evaluations were carried out on linear modules which exhibit differing pneumo-mechanical characteristics to determine the effectiveness of this control strategy. The modules under evaluation

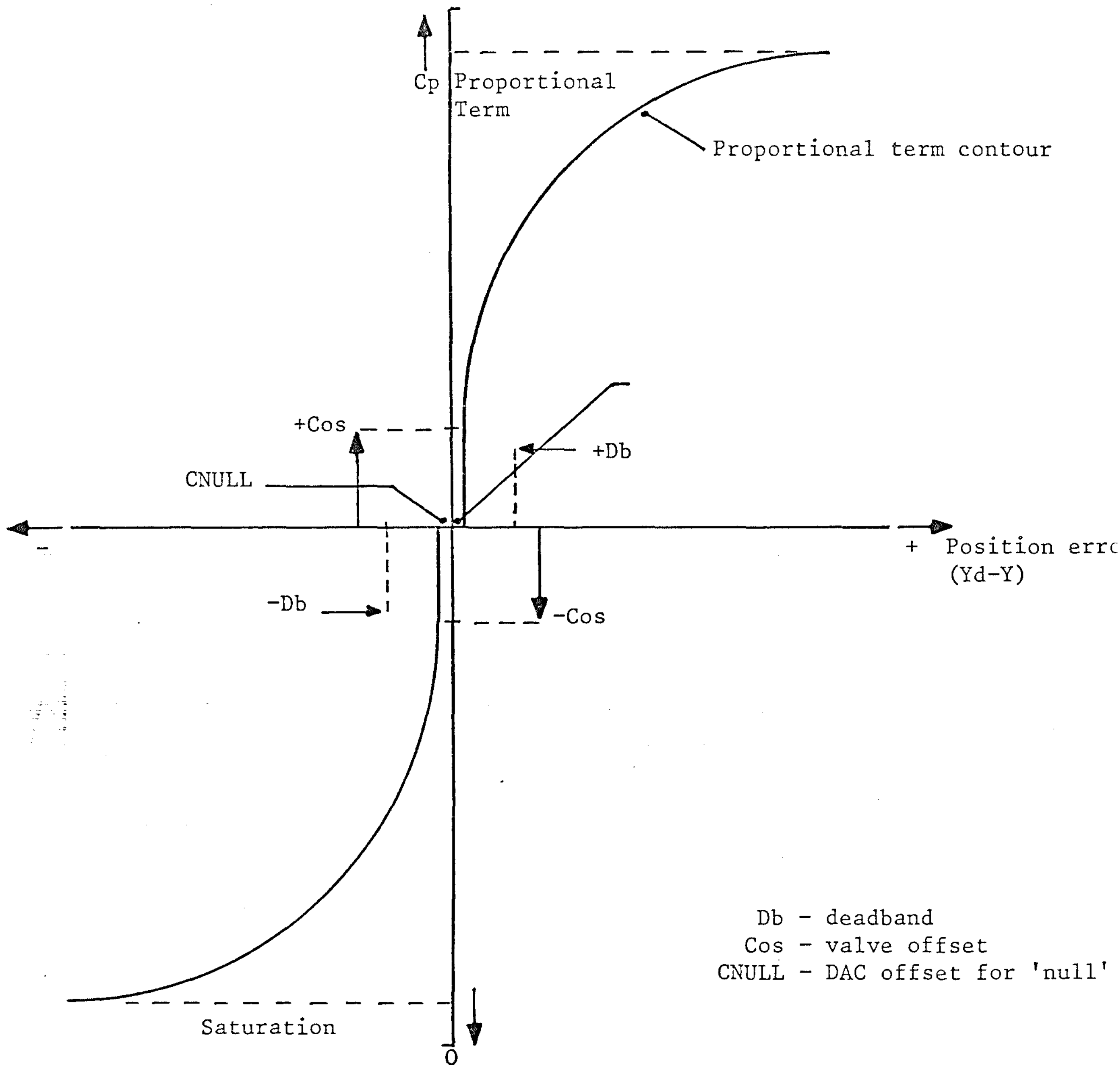


Figure (8.5.6) Minor Loop Compensation (Version 2.0)
- proportional term relationship about null (including control law non-linearities)

demonstrate a large variation in slideway friction and load characteristics and additionally use different size actuators.

(a) MINOR LOOP COMPENSATION EVALUATION (1)

The experimental conditions for this evaluation are detailed in Table [8.5.1.1] together with a list of associated control parameters. A standard static performance evaluation was conducted using the M35 linear module with a move length of 50mm: this corresponds to twelve percent of full stroke.

The control strategy used was Milcom version 2.0, which consists of a non-linear proportional error term (stored in the form of a look-up table in memory) and a linear velocity and acceleration term. The control range was defined as approximately eight percent of full stroke, outside this region software saturation levels are imposed. The dynamic response evaluation consisted of a series of transient responses for short moves in different regions of the stroke.

The objective of this evaluation was to assess the performance of such a control strategy with a module which demonstrates low inertia and low friction.

(b) MINOR LOOP COMPENSATION EVALUATION (2)

The experimental conditions for this evaluation are detailed in Table [8.5.1.2] with an associated list of control parameters. A standard static performance evaluation was conducted using the LUT linear module No.2 with a move length of 66.8mm: this corresponds to thirteen percent of full stroke. The control strategy used was Milcom version 2.0, as used in the previous evaluation. The objective of this evaluation was to assess the performance of such a control strategy with a linear module demonstrating high inertia and friction.

Table (8.5.1.1) Experimental Conditions - Milcom Ev.(1)

CONTROL CONDITION	DESCRIPTION
Module Module Attitude Load Condition Position Resolution Proportional Valve Dither Frequency Brake Mechanism Supply Pressure	M35 Arm Horizontal Slideway - 3 kg 0.025 mm/pulse Solenoid Actuated (10 bit DAC) 50 hz Yes 5.5 bar
Control Strategy Version	Milcom Ver 2.0
Control Parameters	KVGAIN = 4 KAGAIN = 24 Clenum = 3 (1.6 ms) Deband = 6 (±0.15 mm) Spltim = 50 (≃ 0. 5 ms) Maxsat = 380 Minsat = -380 Inpnum = 40 Offpos = +9 Offneg = -9 Mngent = 184 Cntrng = 3200

Table (8.5.1.2) Experimental Conditions - Milcom Ev.(2)

CONTROL CONDITION	DESCRIPTION
Module Module Attitude Load Condition Position Resolution Proportional Valve Dither Frequency Brake Mechanism Supply Pressure	LUT Linear module No 2 Vertical Slideway - 16 kg 0.033 mm/pulse Solenoid Actuated 50 hz Yes 5.5 bar
Control Strategy Version	Milcom Ver 2.0
Control Parameters	KVGAIN = 9 KAGAIN = 35 Clenum = 5 (≃2.6 ms) Deband = 6 (±0.2 mm) Spltim = 50 (0.5 ms) Maxsat = +500 Minsat = -500 Inpnum = 40 Offpos = +22 Offneg = -22 Mngent = 144 Cntrng = 3500

8.5.2. EVALUATION RESULTS

(a) MILCOM EVALUATION (1)

The results for a standard static performance test are summarised in Table [8.5.2.1] and Figure [8.5.2.1]. A summary of dynamic response is given in Table [8.5.2.2]. The results of static performance show that the levels of linearity, accuracy and hysteresis achieved are significantly better than those achieved in any previous evaluations on any linear module; however the repeatability has not shown the same improvement. A study of the static test data showed that it was only the isolated result that produced a wide position variance. From these results and observations at the time of the test it became apparent that periodically, once the setpoint was achieved, before the brake mechanism could actuate and lock the slideway, the module was drifting out of position. This position drift was caused by a variation in the steady state conditions probably influenced by leakage in the pnematic supply and the mechanical operation of the wrap spring lock mechanism.

In this version of the control strategy realtime control is maintained until the module achieves in-position; the lock mechanism is then actuated and realtime control ceases until a new setpoint is demanded. In subsequent versions of the control strategy a modified brake control algorithm is used where realtime control is maintained until the braking action is complete, or alternatively realtime control can be maintained continuously and use of the brake mechanism is optional.

Some informal tests were carried out with no brake mechanism, and these tests showed that no contribution to positioning accuracy and repeatability is made by the brake, this being a highly desirable feature as brake

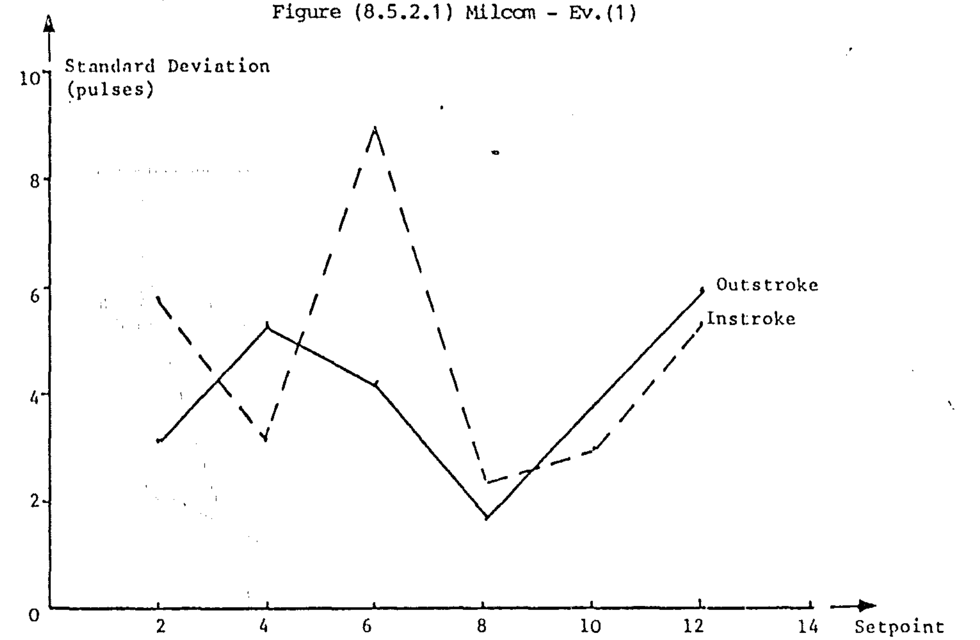
Table (8.5.2.1) Results Summary - Static Performance

PARAMETER	OUT	IN
Linearity (mm)	0.023	0.044
Repeatability (mm)	0.101	0.120
Hysteresis (mm)	0.046	
Accuracy (mm)	0.15	

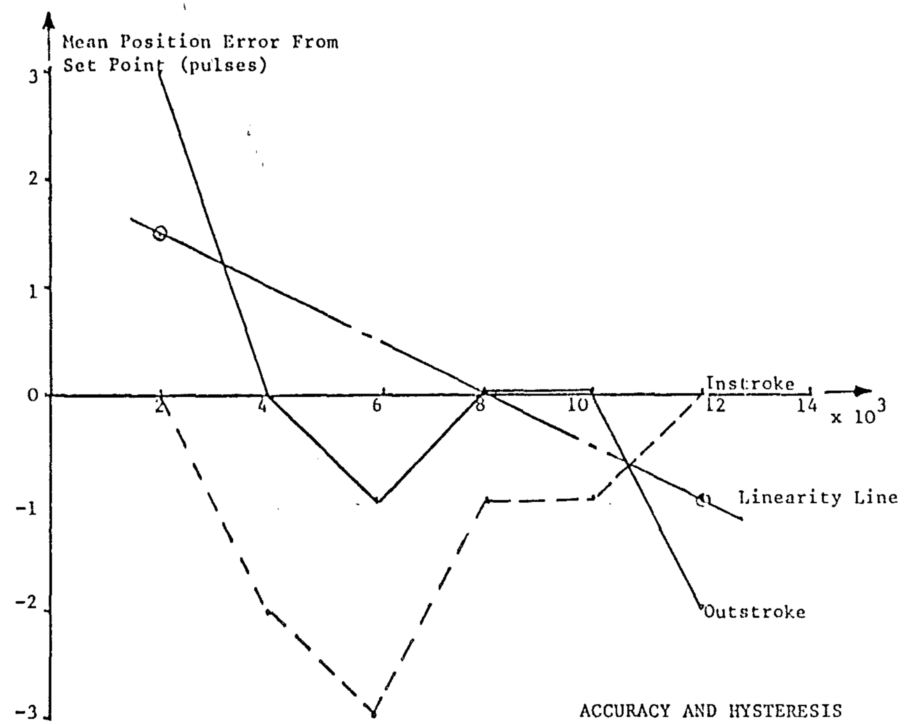
Table (8.5.2.2) Results Summary - Dynamic Performance

MOVE	DESCRIPTION	PEAK VELOCITY (mm/s)	MEAN POSITIONING VELOCITY (mm/s)	POSITIONING TIME (ms)
1	Small move out (TOP)	281	59.2	844.5
2	Small move in (TOP)	328	47.1	1061
3	Small move out (BOTTOM)	203	59.6	838.5
4	Small move in (BOTTOM)	266	61.8	808.5
5	Small move OUT	266	56.3	888
6	Small move IN	312	57.8	865

Figure (8.5.2.1) Milcom - Ev.(1)



REPEATABILITY



ACCURACY AND HYSTERESIS

characteristics are unpredictable and change as wear takes place and operating conditions vary. The grab action of the wrap spring brake was identified on occasions to disturb the module and this would have influenced the positon variance. It was concluded that this design of brake mechanism, with its inherent backlash, is not suitable. A conventional disc brake arrangment is offered as an optional lock mechanism on the commercial family of linear modules.

The static performance evaluation was repeated some days later under identical test conditions to confirm that the results could be repeated.

The dynamic performance figures obtained for short moves (50mm) were encouraging with maximum velocities of 300mm/s, and positioning times (tp) of less than one second being achieved: it should be remembered that positioning time is an important criteria of the dynamic response as in industrial applications short cycle times can be identified with increased manufacturing efficiencies.

To summarise, the results of the first evaluation using a control strategy incorporating minor loop compensation were most encouraging, where good accuracy and a stable response to a step demand were achieved.

(b) MILCOM EVALUATION (2)

The data derived from a standard static performance test is summarised in Table [8.5.2.3] and Figure [8.5.2.3]. A summary of dynamic response is given in Table [8.5.2.4].

The static performance achieved was not as good as that achieved in the previous evaluation, and as would be expected, the deficiencies of the brake control algorithm were also exhibited here, this influencing the static performance achieved. As with the previous evaluation positioning could be successfully achieved without the use of the brake mechanism.

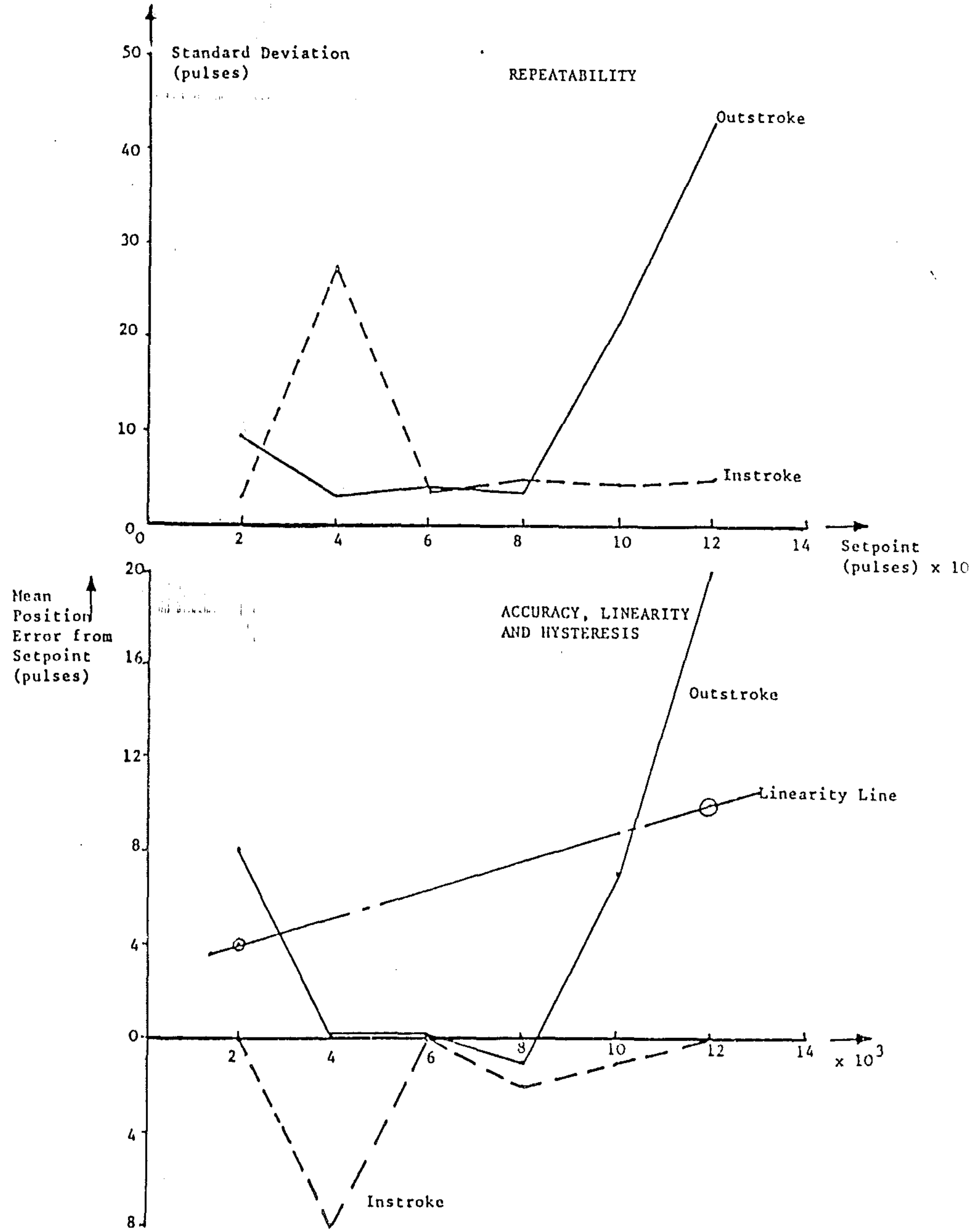
Table (8.5.2.3) Results Summary - Static Performance

	OUT	IN
Linearity (mm)	0.211	0.284
Repeatability (mm)	0.494	0.268
Hysteresis (mm)	0.25	
Accuracy (mm)	0.66	

Table (8.5.2.4) Results Summary - Dynamic Performance

DESCRIPTION	MAXIMUM VELOCITY (mm/s)	MEAN POSITIONING VELOCITY (mm/s)	POSITIONING TIME 'tp' (s)
Small move out (TOP)	376	30	2.227
Small move in (TOP)	188	24.6	2.714
Small move out (BOTTOM)	163	31.6	2.115
Small move in (MID)	288	28.6	2.331

Figure (8.5.2.3) Milcom - Ev.(2)



The dynamic performance results showed that positioning times in excess of two seconds were necessary. The increased inertia of the linear module under examination here will obviously influence the dynamic response, but the industry partners advised that the speed of response would need to be improved for commercial exploitation to be assured. From the transient response it was seen that the time delay before the module responds to the command signal accounts for about one quarter of the total positioning time. This lag in the response is most significant in short moves.

8.5.3. SUMMARY

These performance evaluations have illustrated that minor loop compensation can be used to significantly improve the static and dynamic characteristics of a pneumatic drive under varying load and friction conditions.

8.6. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR

The control strategy described in the previous section (Milcom version 2.0) was the subject of a number of design modifications to enhance both the static and the dynamic performance characteristics. These design changes are illustrated in section (8.6) and as appropriate reference is made to evaluations.

8.6.1. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR (MILCOM VERSION 2.10)

Milcom version 2.10 was designed such that the proportional term contour can be 'mapped' over any desired 'control range' by simply selecting the control parameter 'CNTRNG', in pulse counts. To improve the speed of response (particularly for long moves), involved a modification of the control strategy outside the 'control range'. The command signal is set to the positive or negative saturation level, so that outside the 'control range' the command signal takes a constant value determined by the control parameters 'MAXSAT' and 'MINSAT' respectively.

These modifications are shown in flow chart form in Figure [8.6.1.1]. The relationship between the control system gains is shown in Figure [8.6.1.2]. The proportional term contour used here is as previously described.

The command signal within the 'control range' can be described by;

$$Cs = Kp_{nl}\ (Yd-Y) - KvV - KaA + Cos + Cnull \quad ---(8.6.1.1)$$

Outside the 'control range' the command signal is

$$Cs = Csat \quad ---(8.6.1.2)$$

Other than the modifications discussed Milcom version 2.10 is the same as

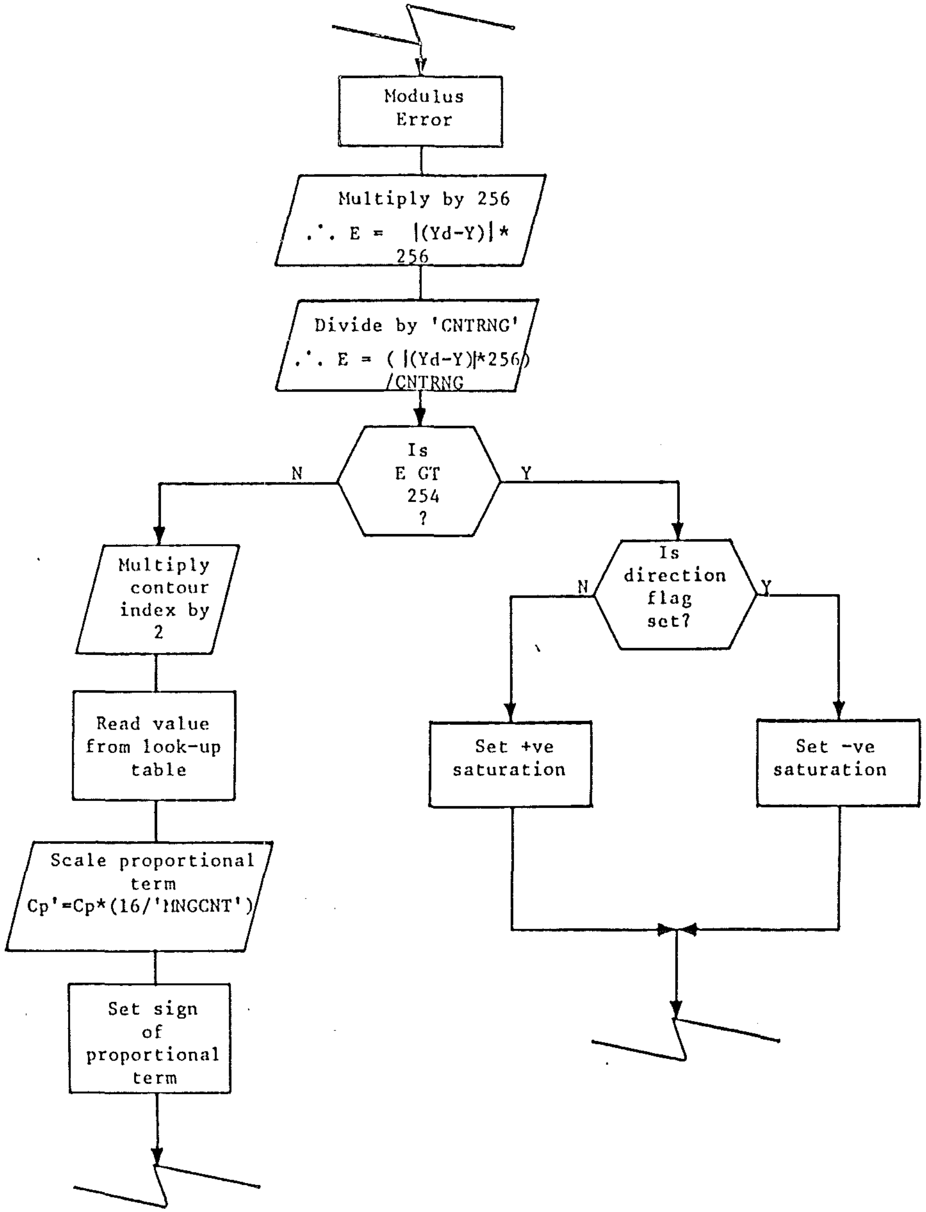


Figure (8.6.1.1) Proportion Term Contour Selection Process - Milcom Version 2.10

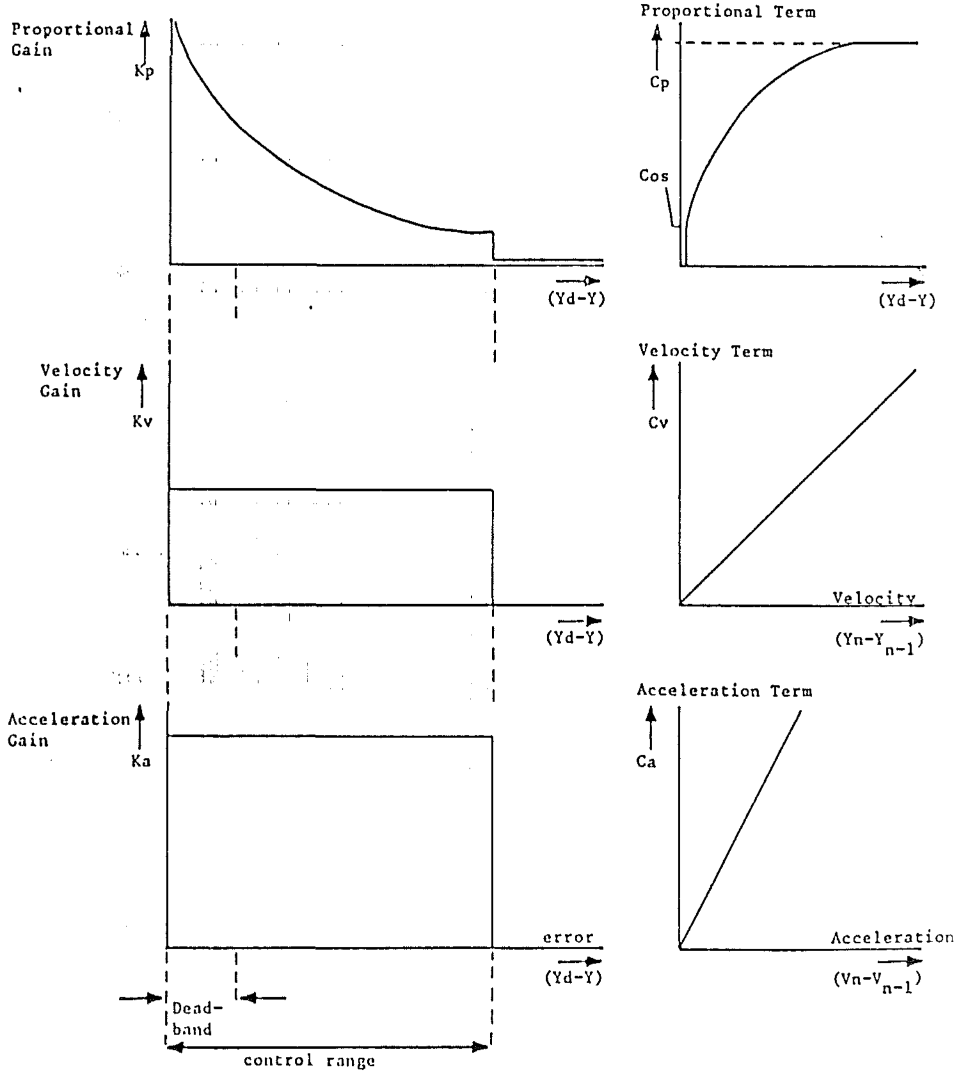


Figure (8.6.1.2) Minor Loop Compensation Version 2.10 - Control Gains version 2.0.

8.6.2. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR (MILCOM VERSION 2.20)

Problems were encountered using a step function to saturation level in the control scheme when outside the control range. When a small step function was used the response was unacceptably slow. If a large step function was used then an acceptable speed of response could be obtained, but significant overshoot would result.

To obtain better control of the dynamic response, the step function, which operates outside the control range, was replaced by a proportional error control term. With an appropriate selection of the proportional gain 'Kp', being set using the control parameter 'KGAIN',

where, $Kp = 1/KGAIN$ ---(8.6.2.1)

and the constant 'Cc', the dynamic response was significantly improved for the complete range of moves. The controller constant 'Cc' was introduced to include the valve offset 'Cos'. This constant is set using the control parameter 'PROPST', and can be calculated approximately from the relationship

$$Cc = PROPST = [(CNTRNG) / KGAIN] + Cos \quad ---(8.6.2.2)$$

Figure [8.6.2.1] shows the software modifications for the proportional term calculation outside the control range.

With the introduction of the proportional control zone there are now three distinct control schemes in operation, dependent upon the position error. The control zones can be represented by the following equations;

for $/(Yd-Y)/ < Ecr$ (within the control range)

$$Cs = Kp_{nl}\ (Yd-Y) - KvV - KaA + Cos + Cnull \quad ---(8.6.2.3)$$

for $Ecr <= /(Yd-Y)/ < Es$

$$Cs = Kp(Yd-Y) + Cc + Cnull \qquad ---(8.6.2.4)$$

for $/(Yd-Y)/ >= Es$

$$Cs = Csat \qquad ---(8.6.2.5)$$

These relationships are illustrated by Figure [8.6.2.2].

8.6.3. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR (MILCOM VERSION 2.30)

The minor loop compensation control strategy, as described in the previous section, was achieving target performance levels acceptable for commercial positioning systems with respect to its static characteristics. The one characteristic of the transient response that was still undesirable was the low amplitude oscillation close to setpoint at certain positions in the stroke.

The first design change to overcome this problem was to implement a new algorithm for the derivation of the state variables. The method of direct derivation of velocity and acceleration was replaced by the nth order hold approach, which involves the storage of position and velocity data over 'm' consecutive sampling periods. The velocity and acceleration can be re-calculated every sampling interval, but the values calculated are the average over the previous 'm' sampling intervals.

This approach did allow improved resolution to be obtained on velocity and acceleration measurement, but the time lag introduced did appear to have a detrimental effect on stability. This effect was not fully quantified due to time constraints in the research study.

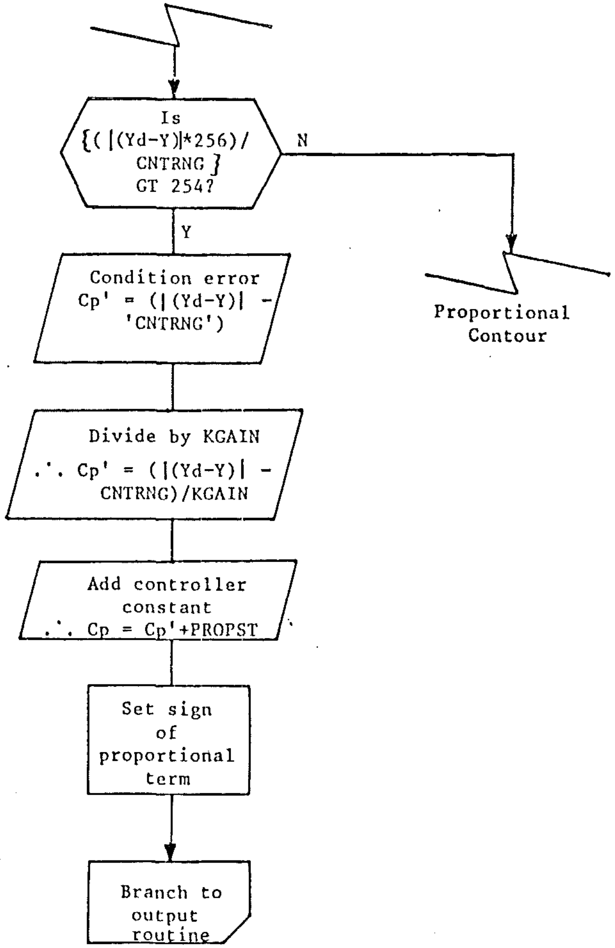


Figure (8.6.2.1) Proportional term calculation - Milcom version 2.20

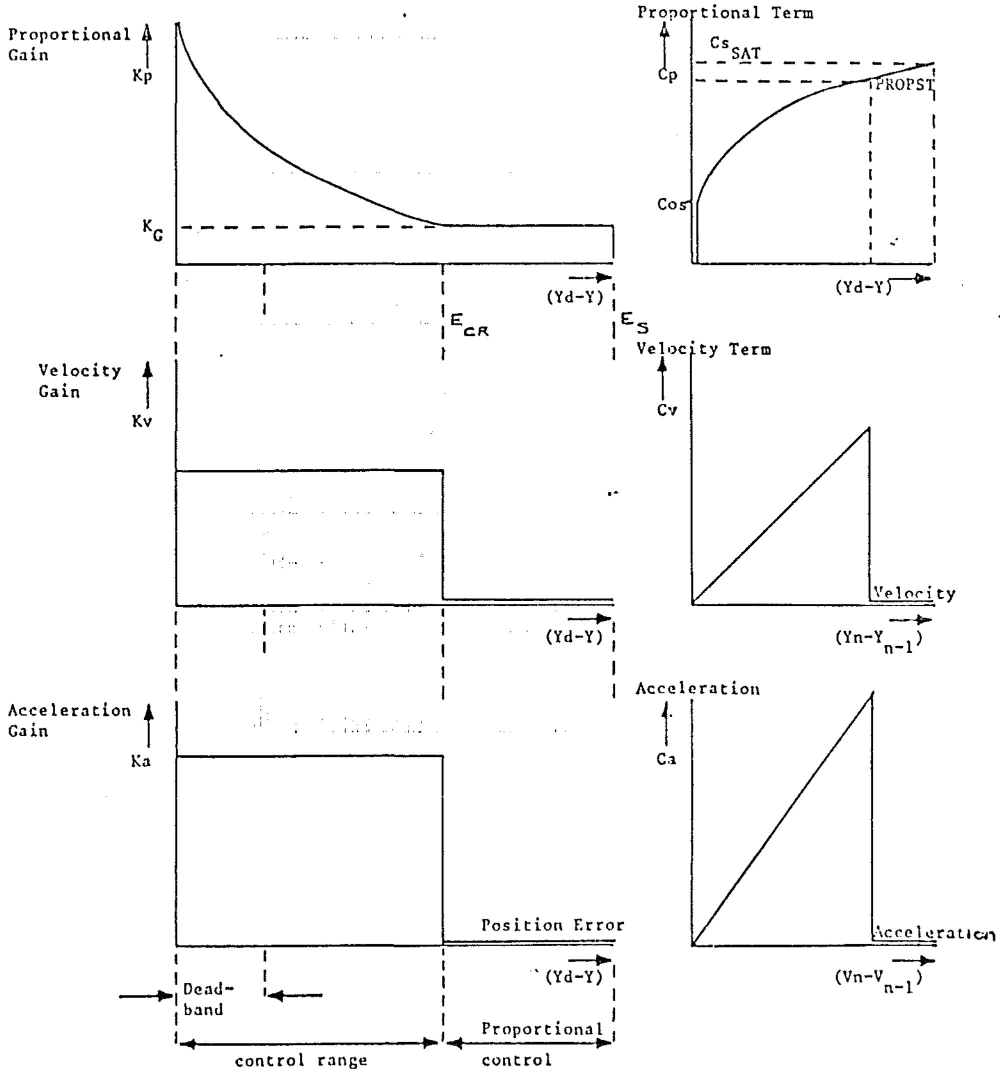


Figure (8.6.2.2) Minor Loop Compensation - Version 2.20 - Control Gains

8.6.4. MINOR LOOP COMPENSATION WITH PROPORTIONAL TERM CONTOUR (MILCOM VERSION 2.40)

The problem of reducing the low amplitude oscillation at the setpoint (also resulting in a reduced settling time) was achieved by implementing subtle changes to the minor loop compensation strategy (version 2.20). An additional control zone was introduced as part of the existing control range (see Figure [8.6.4.1]). The final section of the proportional term contour (the last 'm' points in the table closest to 'null') are replaced by a linear proportional term. The contour at this point will approximate to a tangent with a gradient 'K1' and so for each discrete error value in this region a proportional term can be calculated.

This additional control region overcomes the problem of discontinuities in the look-up table. These discontinuities are a result of the quantising of the contour profile and the length of stroke to which the look-up table has been 'mapped'. The larger the length of stroke over which 'mapping' occurs, the larger the resulting discrete incremental changes will be. These discrete changes obtained from the contour will only be significant when close to the null, and the problem disappears when this additional control zone is introduced.

Another change to the control scheme that can be implemented is to stop the control within the 'deadband'. This means that when the 'deadband' is entered, the controller is immediately set at null. This change is designed to reduce the settling period. The effect of these changes on the proportional term is illustrated in Figure [8.6.4.2]. The deadband now defines a region about the setpoint within which the controller is inactive, an analogy to programmable compliance can be drawn. The software modifications necessary are shown in flowchart form in Figure [8.6.4.3]. It

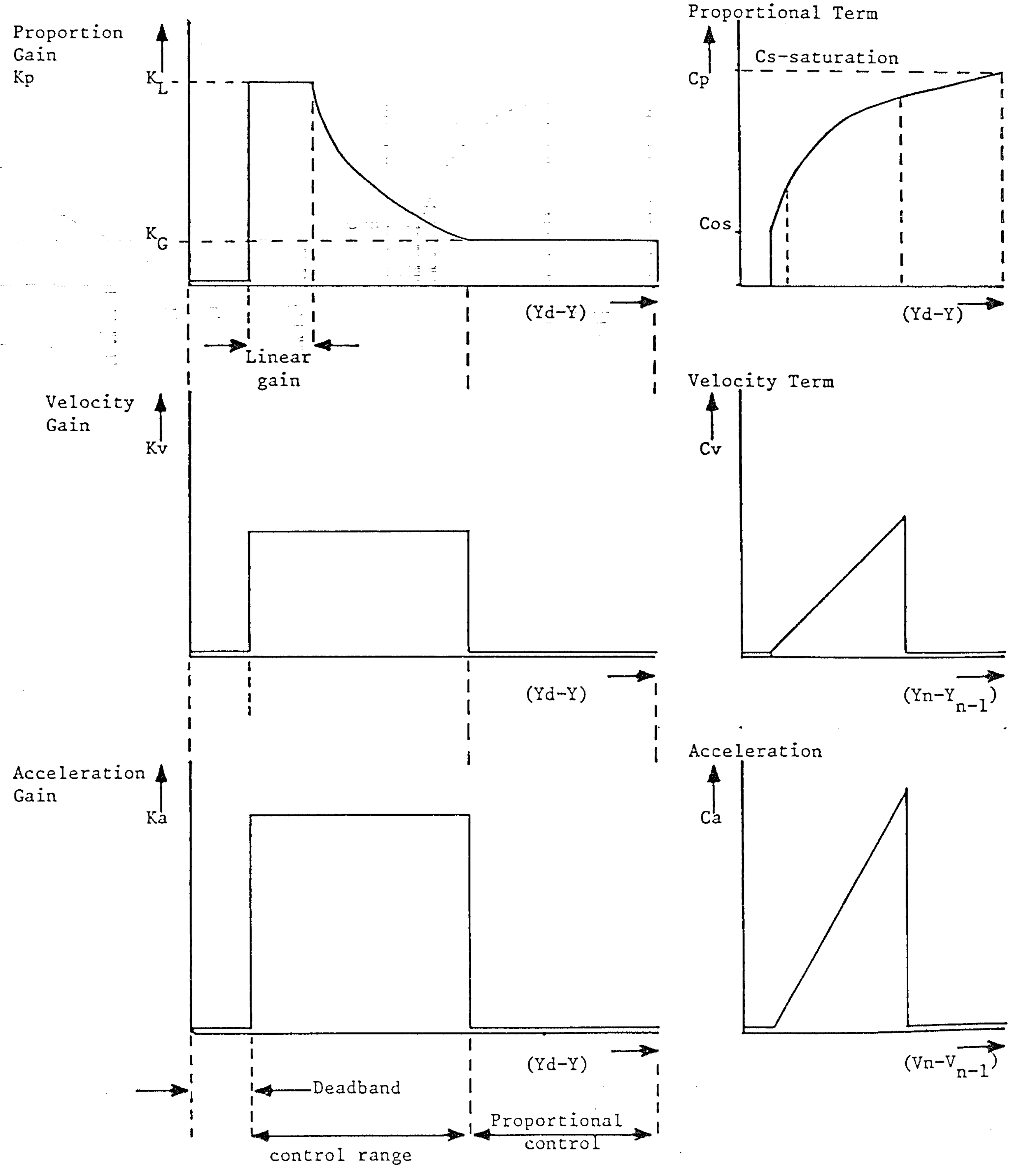


Figure (8.6.4.1) Minor Loop Compensation - Version 2.40 - Control Gains

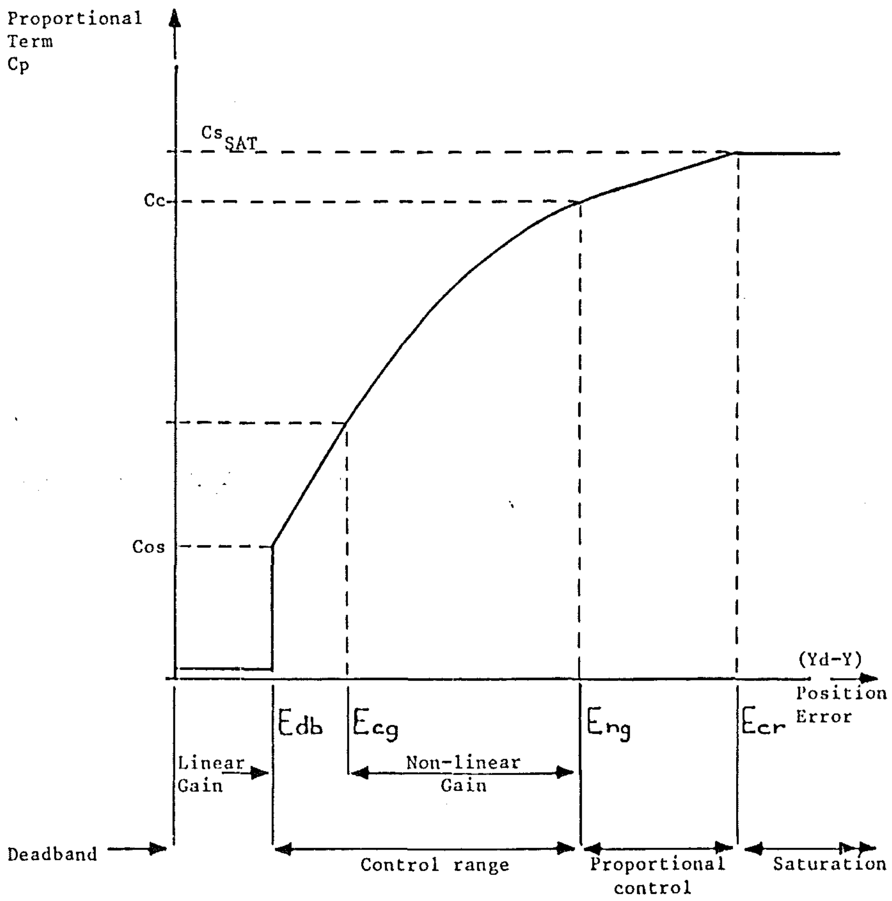


Figure (8.6.4.2) Minor Loop Compensation - Version 2-40 - Proportional Term Control Zones

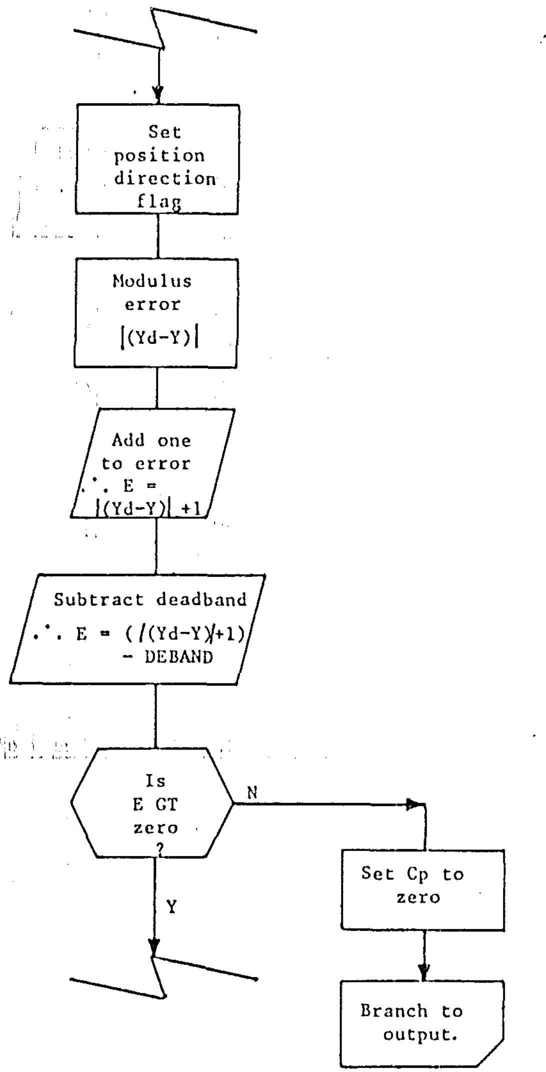


Figure (8.6.4.3) Proportional Term Calculation - Milcom version 2.40

can be seen that to prevent the servo from stopping at the edge of the deadband it was necessary to increment the error before the proportional term was calculated.

The command signal for each control zone can be represented by the following equations;

For /(Yd-Y)/ < Edb (within the deadband) then,

$$Cs = Cnull \qquad ---(8.6.4.1)$$

For Edb <= /(Yd-Y)/ < Ecg then,

$$Cs = Kp_{l} (Yd-Y) - KvV - KaA + Cos + Cnull \qquad ---(8.6.4.2)$$

For Ecg <= /(Yd-Y)/ < Eng then,

$$Cs = Kp_{nl} (Yd-Y) - KvV - KaA + Cos + Cnull \qquad ---(8.6.4.3)$$

For Eng <= /(Yd-Y)/ < Ecr then,

$$Cs = Kp(Yd-Y) + Cnull + Cc \qquad ---(8.6.4.4)$$

and for Ecr <= /(Yd-Y)/ then

$$Cs = Csat \qquad ---(8.6.4.5)$$

This control strategy achieved target performance levels necessary for a commercial pneumatic positioning system with respect to both static and dynamic performance criteria, as illustrated in the following section.

8.6.5. PERFORMANCE EVALUATION - MINOR LOOP COMPENSATION (MILCOM VERSION 2.4)

Two standard performance evaluations were conducted using the minor loop compensation control strategy (Milcom version 2.4) to quantify the performance level.

(a) MINOR LOOP COMPENSATION EVALUATIONS (4) AND (5)

The experimental conditions for these evaluations are detailed in Table

[8.6.5.1] with a list of associated control parameters for each evaluation. Using the M35 arm a standard static performance test was conducted in each evaluation together with a series of dynamic response tests. The realtime control software used for both evaluations was Milcom version 2.40. This strategy includes a proportional control region and a 'control range' in which the proportional term can take either a linear or non-linear relationship, the velocity and acceleration minor loops have constant gains.

The objective of these evaluations was to determine the level of performance that could be achieved using the M35 linear module: this module demonstrated similar mechanical characteristics to the projected production modules. Two parallel evaluations were conducted necessarily in an attempt to quantify the effectiveness of the non-linear proportional term when compared to a linear proportional term. Milcom evaluation (4) was conducted using a linear proportional term within the control range where the gain 'Kp' has a constant value

for 4 < /(Yd-Y)/ <= 255 then Kp = 1

Outside the control range the proportional term has a gain of Kp=1/100.

Milcom evaluation (5) was conducted using a non-linear proportional term within the 'control range' where the non-linear contour values are scaled by a factor of 1.6, the last fifteen points of the contour having a gain of Kp=1. Outside the control range the proportional term has a gain of Kp=1/80.

The M35 linear module has the following loop gains associated with the control system used in this research study, where;

Proportional error loop gain = Kp.400 volts/m

Velocity loop gain = $Kv.2X10^{5}$ $volts/ms^{-1}$

Acceleration loop gain = $Ka.1X10^{8}$ $volts/ms^{-2}$

8.6.5.1. EVALUATION RESULTS - MILCOM EVALUATION (4)

The results of a standard static performance test are summarised in Table [8.6.5.1.1] and Figure [8.6.5.1.1]. The transient response is shown in Figure [8.6.5.1.2] with a summary of dynamic response given in Tables [8.6.5.1.2](a) and (b).

The static performance evaluation results show that very good performance was achieved for all the performance criteria. The quantitive measures of linearity, accuracy, repeatability and hysteresis show levels of performance necessary in a commercial motion control system. These static performance results are better than any achieved in previous evaluations with a linear module.

The dynamic performance results show that for long moves velocities in excess of 1000mm/s are achieved, with mean position velocities in excess of 300mm/s. The positioning time 'tp' for all categories of move was found to be less than one second. The transient responses show that the settling phase, though reduced, still has a significant influence on the positioning time and that the response lag is also significant for short moves: this is why the positioning times cannot be directly correlated to length of move.

8.6.5.2. EVALUATION RESULTS - MILCOM EVALUATION (5)

The test data for a standard static performance test are summarised in Table [8.6.5.2.1] and Figure [8.6.5.2.1]. A typical transient response is shown in Figure [8.6.5.2.2] with a summary of dynamic performance given in Tables [8.6.5.2.2](a) and (b).

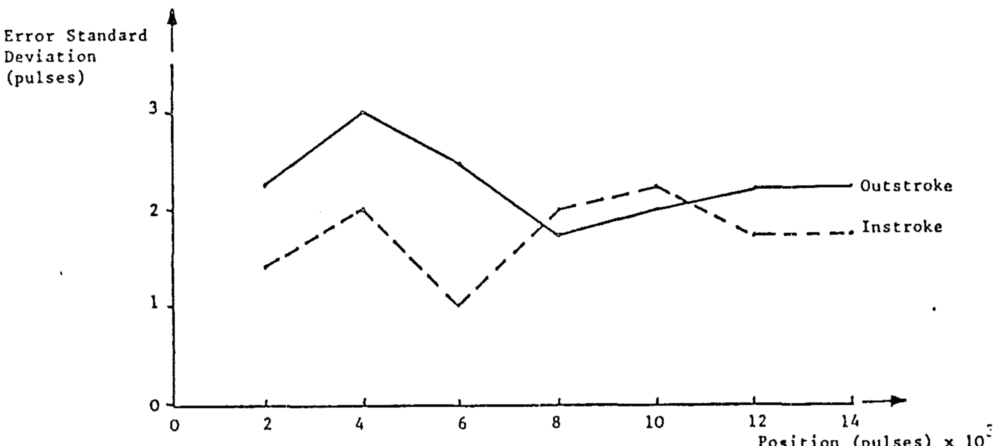
Table (8.6.5.1) Experimental Conditions - Milcom Ev.(4) and Ev.(5)

CONTROL CONDITION	DESCRIPTION
Module Module Attitude Load Condition Position Resolution Proportional Valve Dither Frequency Brake Mechanism Supply Pressure	M35 arm Horizontal Slideway - 3 Kg 0.025 mm/pulse Solenoid Actuated 50 hz NOT USED 5.5 bar
Control Strategy Version	Milcon Ver 2.40
Control Parameters	
<u>Milcom Ev(4)</u> Kvgain = 50 Kagain = 100 Deband = ±4 (0.1 mm) Spltum = 200 (2.13 ms) Maxsat = +60 Minsat = -60 Inpnum = 60 (128 ms) Offpos = +12 Offneg = -12 Mngcnt = XXX (dont care) Cntrng = 255 (6.37 mm) Propst = 40 Kgain = 100 Lstcnt = 255 (6.37 mm) Klgain = 16	<u>Milcom Ev(5)</u> Kvgain = 50 Kagain = 100 Deband = ±4 (0.1 mm) Spltum = 200 (2.13 ms) Maxsat = +60 Minsat = -60 Inpnum = 60 (128 ms) Offpos = +12 Offneg = -12 Mngcnt = 10 Cntrng = 500 Propst = 50 Kgain = 80 Lstcnt = 15 Klgain = 16

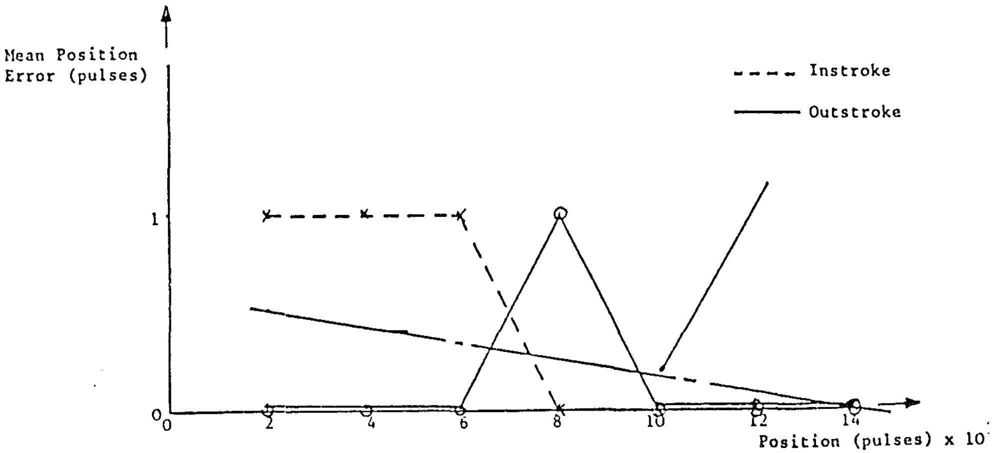
Table (8.6.5.1.1) Static Performance Summary

PARAMETER	OUTSTROKE (mm)	INSTROKE (mm)
Linearity	0.008	0.008
Repeatability	0.057	0.043
Hysteresis	0.014	
Accuracy	0.1	

Figure (8.6.5.1.1) Milcam - Ev.(4)



REPEATABILITY



ACCURACY, LINEARITY AND HYSTERESIS

Table (8.6.5.1.2) (a) Dynamic Performance Summary - STD Test

PARAMETER	MOVE 50 mm OUTSTROKE	MOVE 50 mm INSTROKE
Mean Positioning Time (ms)	850	840
Time Standard Deviation (ms)	50	42
Mean Positioning Velocity (mm/s)	58.8	59.5

Table (8.6.5.1.2) (b) Dynamic Response Summary

DIR	MOVE (mm).	POSITIONING TIME (ms) tp	VELOCITY MAX (mms^{-1}) Vmax	MEAN POS VEL (mms^{-1}) MPV
+ OUT/S	LONG 275	937	892	293
- IN/S	LONG 275	824	1020	334
+ OUT/S	MED 137.5	940	633	146
- IN/S	MED 137.5	710	692	194

MOVE +50mm
(2000 PULSES).
t_p = 595 ms.
V_{max} = 363 mms^{-1}
(31 p/ts).

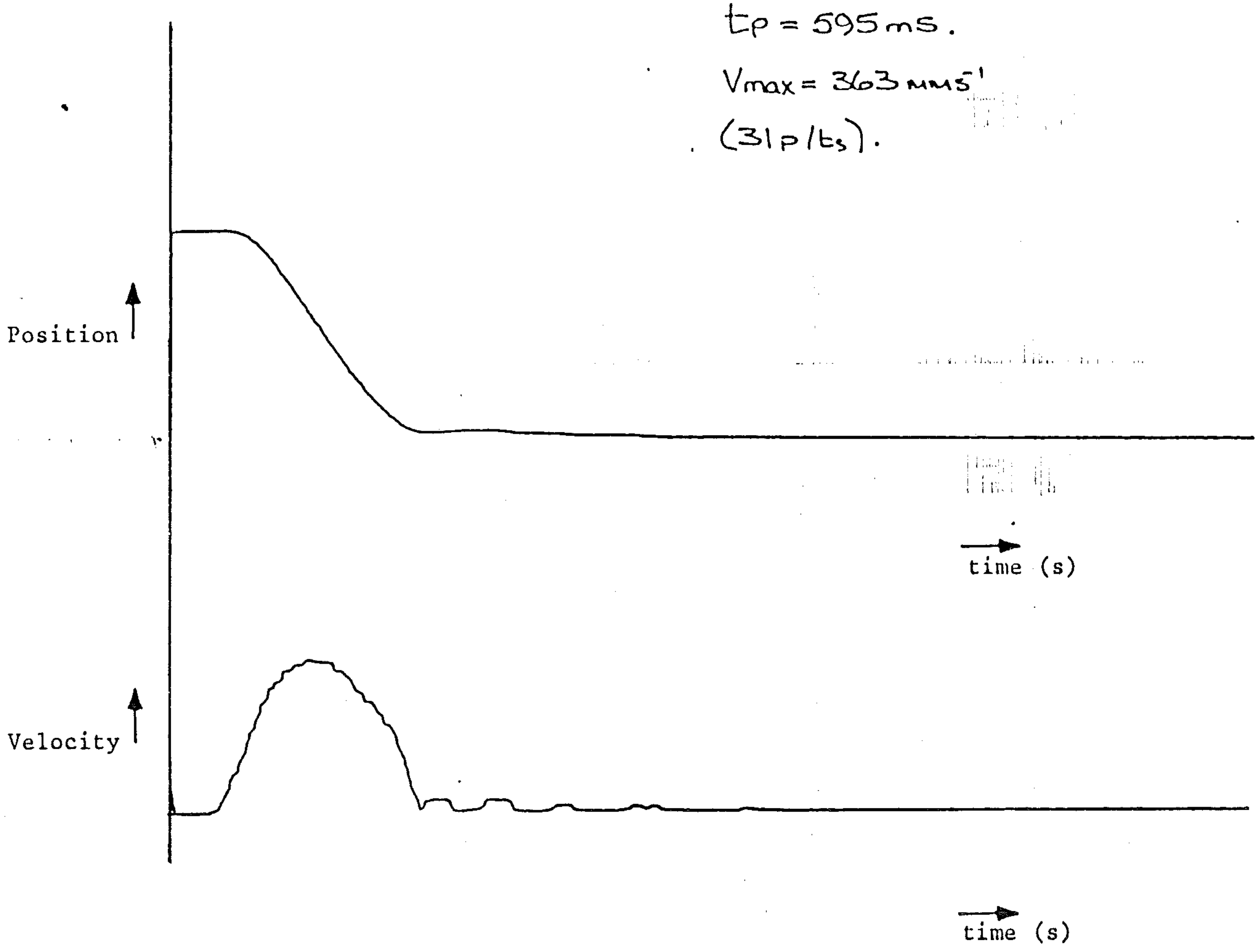


Figure (8.6.5.1.2) Dynamic Response

The static performance evaluation shows that as with evaluation (4) very good results were obtained for all categories of performance measure, with no substantial differences to those of evaluation (4). The notable improvement is that relating to mean positioning times (tp) of the 50mm moves in the standard test. For the full test sequence, tp=723ms for outstroke moves and 696ms for instroke moves; these compare with tp=850ms and 840ms respectively for evaluation (4) where a linear proportional term was used. A reduction in mean positioning time of this order is quite significant in a series of short moves, and can be attributed to the higher average forward path gain when using a non-linear proportional term. The beneficial influence of a non-linear position loop gain is demonstrated in a further study of dynamic response in the next Chapter. It should be noted that the positioning times quoted in any evaluation includes the 'in-position' time, which in this case was 128ms. It is appropriate to say that the static performance demonstrated by the M35 linear module in this evaluation would be acceptable for many industrial motion control applications.

The dynamic performance test results show that for long moves velocities in excess of 1200mm/s are achieved, with mean positioning velocities in excess of 300mm/s. Particularly with short moves, the use of non-linear position loop gains has resulted in a faster dynamic response. The use of a non-linear velocity loop gain can further enhance the dynamic response, but techniques to reduce the 'response lag' in short moves and the 'settling phase' result in a more significant improvement.

8.6.5.3. SUMMARY

The evolution of a motion control strategy for pneumatic servo-drives has

Milcom eu(5)

Table (8.6.5.2.1) Static Performance Summary

PARAMETER	OUTSTROKE (mm)	INSTROKE (mm)
Linearity	0.013	0.012
Repeatability	0.046	0.040
Hysteresis	0.021	
Accuracy	0.10	

Figure (8.6.5.2.1) Milcom - Ev.(5)

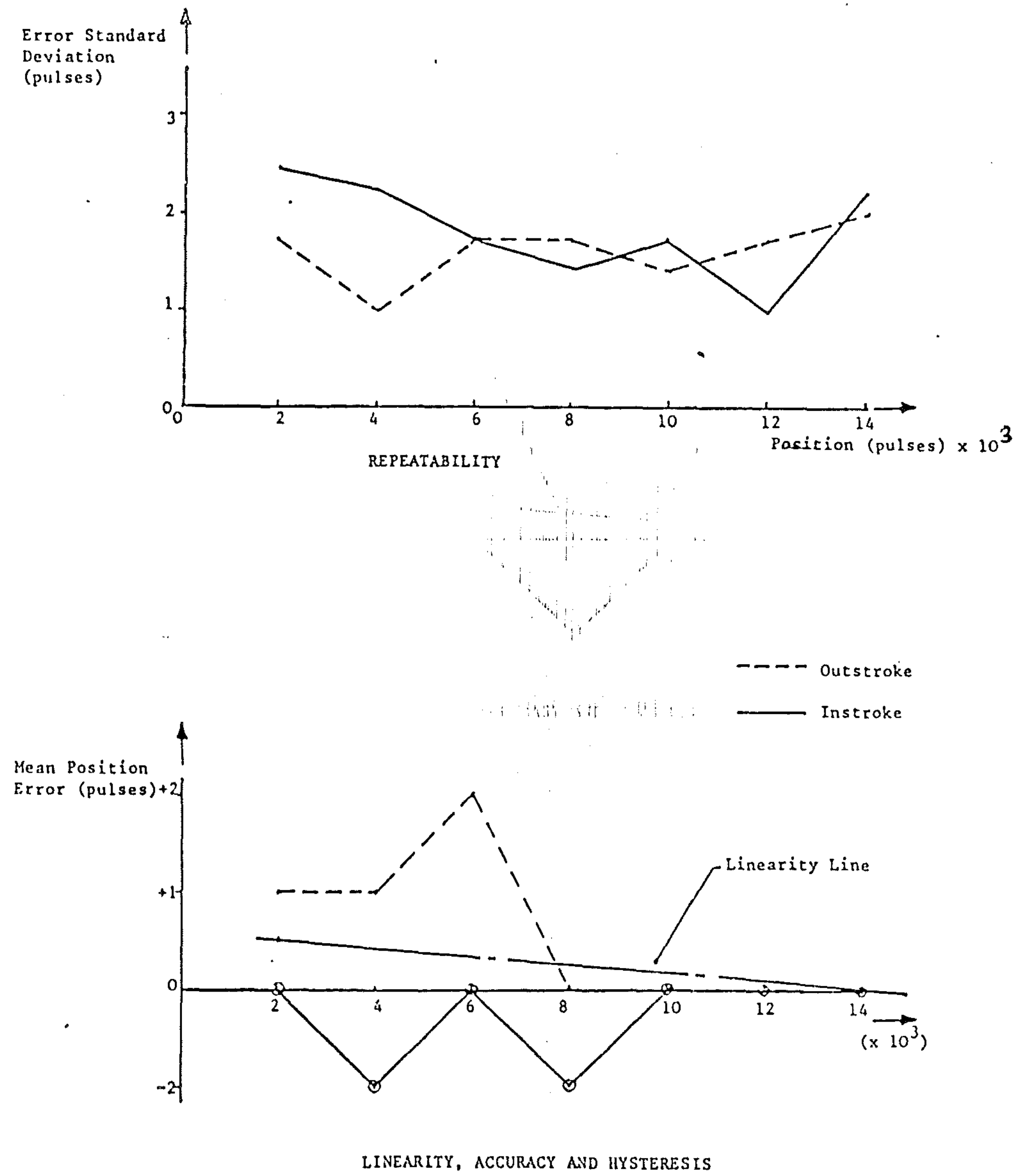


Table (8.6.5.2.2) (a) Dynamic Performance Summary

PARAMETER	MOVE 50 mm OUTSTROKE	MOVE 50 mm INSTROKE
Mean Positioning Time (ms)	723	696
Time Standard Deviation (ms)	46	46
Mean Positioning Velocity (mm/s)	69.2	71.8

Table (8.6.5.2.2) (b) Dynamic Response Summary

DIR	MOVE (mm)	POSITIONING TIME (ms) tp	VELOCITY MAX (mms^{-1}) Vmax	MEAN POS VEL (mms^{-1}) MPV
+ O/S	LONG 275	963	1043	285
- I/S	LONG 275	802	1230	343
+ O/S	MED 137.5	974	727	141
- I/S	MED 137.5	722	879	190

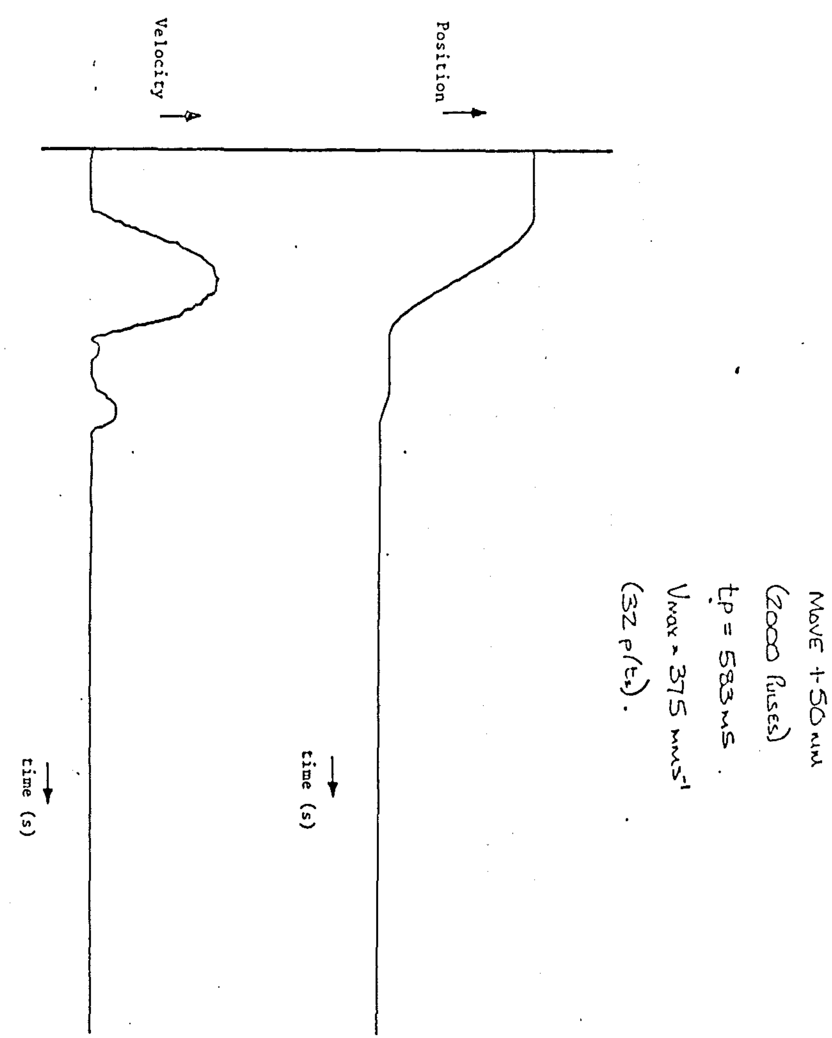


Figure (8.6.5.2.2) Dynamic Response

been described. The use of minor loop compensation techniques and non-linear feedback loop gains in a multi-strategic control system has resulted in a robust control scheme demonstrating significanlty improved static and dynamic performance. This control strategy will now be described under a series of load evaluations. However, the implementation of the control strategy has been modified to include bi-directional gains and global saturation limits (Milcom version 2.50). These changes allow valve, actuator and load asymmetry to be accommodated by the control strategy.

The control strategy (Milcom version 2.5) will be described in some detail in the following section as this implementation (with the addition of a 'front-end' control scheme) is used in the commercial motion control system sold by Martonair Ltd.

8.7. A CONTROL STRATEGY FOR COMMERCIAL PNEUMATIC MOTION CONTROL (MILCOM VERSION 2.50)

The control strategy is designed as a multi-strategic control scheme incorporating minor loop compensation and non-linear gains which can be represented schematically by Figure [8.7.1]. Milcom version 2.50 was implemented to incorporate the most successful features of the early control strategies using minor loop compensation and non-linear feedback loop gains, for use in the prototype single axis controllers (see Chapter (10)). Certain additional features, until now omitted to reduce complexity, were now necessarily incorporated within the control strategy.

It was necessary to introduce bi-directional controller gains to accomodate asymmetry in the actuator, valve and load conditions. Earlier versions of Milcom allowed the controller to fully saturate the command signal to the valve when outside the control range (saturation limits could

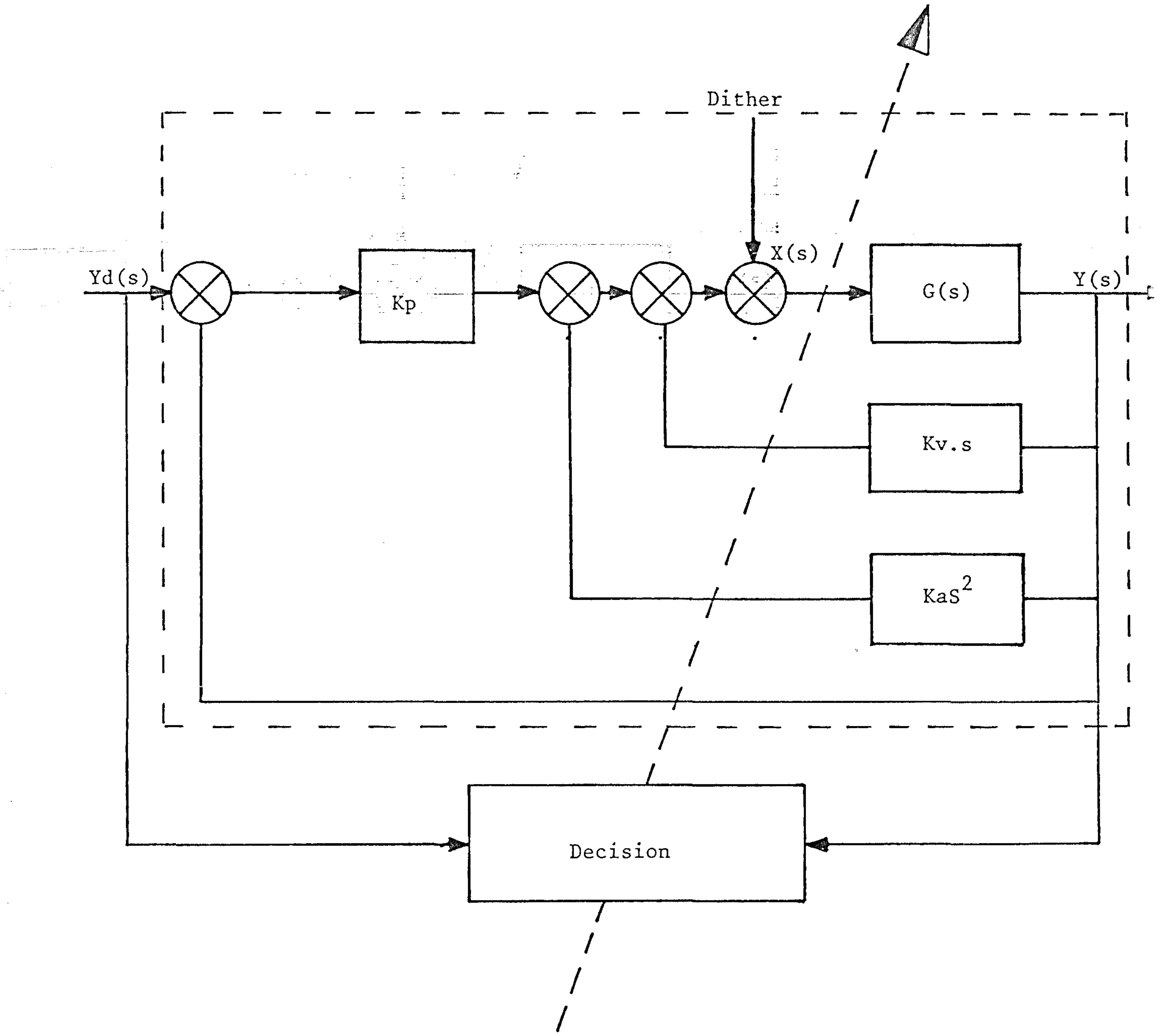


Figure (8.7.1) Multi-strategic Control

Algorithm Scheduling

be set within the control range). Here parameters are introduced to impose global saturation limits on the controller. These software limits are set using the control parameters 'SATPOS' and 'SATNEG' for each direction. Milcom version 2.50 is described in Figure [8.7.2] and [8.7.5] and in section (8.7.1) presented in Appendix (8).

8.7.2. LOAD TESTING OF A LINEAR MODULE (M60110/600)

WITH MINOR LOOP COMPENSATION CONTROL STRATEGY

Motion control strategies using minor loop compensation techniques have been evaluated with a range of prototype modules. In this section a series of performance evaluations are presented with variable load systems: the tests being carried out using the Martonair M/60110/600 linear module, which is one of a range of handling modules marketed commercially with an LUT designed motion control system. It should be noted that prototype Martonair linear units became available subsequent to the Milcom control strategy evaluations in this research study.

8.7.3. EXPERIMENTAL CONDITIONS

The experimental conditions are described for a series of performance evaluations, which were conducted to analyse the effect of variable load systems.

The realtime control software used for all the load system evaluations was Milcom version 2.50. Real time control is maintained continuously negating the requirement for the lock mechanism.

(a) LOAD EVALUATION (1),(2)and(3)

The experimental conditions for load evaluation (1),(2)and(3) are detailed in Table [8.7.3.1] with an associated list of control parameters. A representation of the load test configuration is shown in Figure [8.7.3.1], the M/60110/600 linear module is mounted horizontally on the test table with the load acting in the vertical plane via the pulley system. This configuration ensures that a purely axial load is applied to

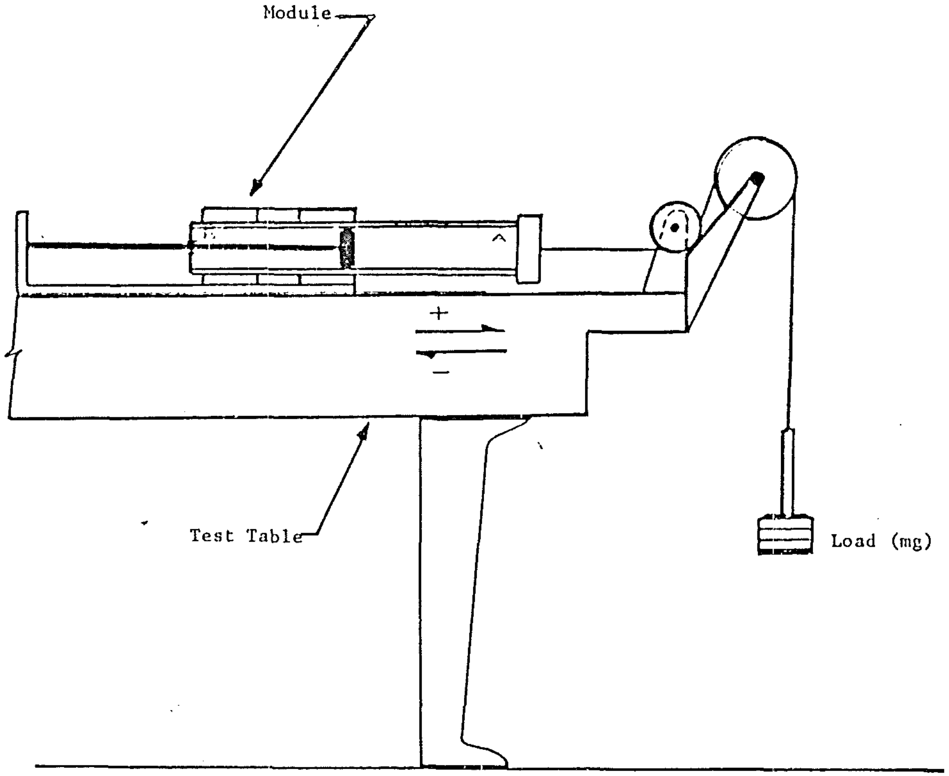


Figure (8.7.3.1) Load Test Configuration

Table (8.7.3.1) Experimental Conditions

Load Evaluations (1), (2) and (3)

CONTROL CONDITION	DESCRIPTION	
Module Module Attitude Load Condition Position Resolution Proportional Valve Dither Frequency Brake Mechanism Supply Pressure	M/60110/600 Linear Unit Horizontal Slideway 3.5Kg + Load 0.028 mm/pulse Solenoid Actuated 50 hz Not used 5.5 bar	
Control Strategy Version	Milcom Ver 2.50	
Control Parameters	Kvgain = 70 Kagain = 150 Deband = 5 (±0.14 mm) Spltum = 200 (2.15 ms) Maxpos = +60 Maxneg = -60 Inpnum = 60 Offpos = +17 Offneg = -17 Mngctn = 8 Mngctp = 8 Cntrng = 600 (~16.7 mm)	Propst = 30 Kgnpos = 60 Kgnneg = 40 Lstcnt = 5 Klgpos = 5 Klgneg = 5 Satpos = 650 Satneg = 375

Table (8.7.3.2) Load Evaluation

TEST	STATIC LOAD (Kg)
Load Evaluation (1)	13
Load Evaluation (2)	5
Load Evaluation (3)	0

the module: thus additional static loading can be incorporated with the intrinsic mass of the slideway (3.5kg), the additional static loading used for load evaluations (1),(2)and(3) is listed in Table [8.7.3.2].

Standard static performance evaluations were conducted using a move length of 84mm(3000 pulses) and the dynamic responses associated with these evaluations were also analysed. Other dynamic response testing included the monitoring of the transient response over various lengths of move in each direction, to determine the effect of load.

The module under test, the M/60110/600 linear unit, has a 600mm stroke length. As can be seen from Figure [8.7.3.1] the actuator body moves with respect to the piston (the reverse of a conventional pneumatic cylinder). Hence to provide positive motion (outstroke) air is applied to chamber (A) with negatve motion induced by supplying air to chamber (B). An actuator piston area differential is present biasing motion in the negative direction. In each evaluation the 'null' of the valve was set in the loaded condition.

(b) LOAD EVALUATION (4) - INERTIAL LOAD

The experimental conditions for load evaluation (4) are detailed in Table [8.7.3.3] with an associated list of control parameters. The control parameters were re-selected for this evaluation to accomodate the changed load system configuration.

A representation of the load test configuration is shown in Figure [8.7.3.2]: the M/60110/600 linear module is mounted horizontally on the test table with the load carriage mounted directly onto the module via a flexible coupling. This configuration was chosen to determine the effect of an inertial load of 6.5kg on performance, the intrinsic mass of the module slideway (3.5kg) plus the mass of the load carriage of 3kg.

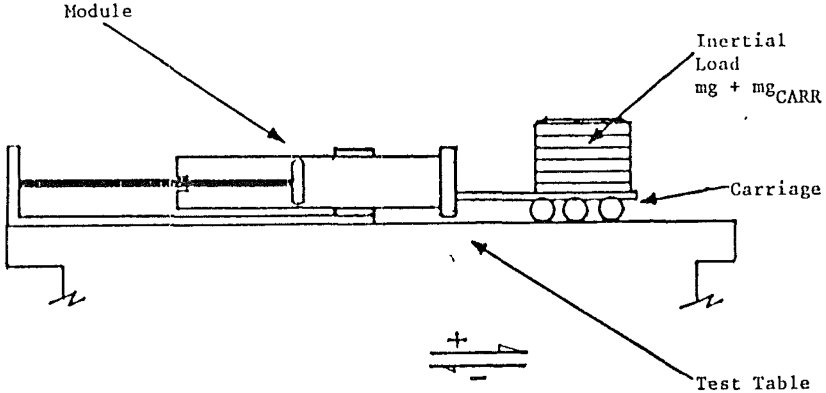


Figure (8.7.3.2) Load Test Configuration

Table (8.7.3.3) Experimental Conditions - Load Evaluation (4)

CONTROL CONDITION	DESCRIPTION	
Module	M/60110/600 Linear Unit	
Module Attitude	Horizontal	
Load Condition	Slideway	
Position Resolution	0.028 mm/pulse	
Proportional Valve	Solenoid Actuated	
Dither Frequency	50 hz	
Brake Mechanism	Not used	
Supply Pressure	5.5 bar	
Control Strategy	Milcom	
Version	Ver 2.50	
Control Parameters	Kvgain = 65	Klgpos = 9
	Kagain = 100	Klgneg = 9
	Deband = 6 (±0.17 mm)	Satpos = 650
	Spltum = 200 (2.13 ms)	Satneg = 375
	Maxneg = +60	
	Maxpos = -60	
	Inpnum = 20	
	Offpos = +17	
	Offneg = -17	
	Cntrng = 1000	
	Propst = 30	
	Kgnpos = +60	
	Kgnneg = -60	
	Lstcnt = 1000	

The standard static performance evaluation was conducted using a length of move of 84mm(3000pulses). The dynamic response evaluation included the monitoring of the transient response over various lengths of move in each direction.

8.7.4. EVALUATION RESULTS - LOAD TESTING

The results of the load evaluations of the Milcom version 2.50 control strategy are presented below.

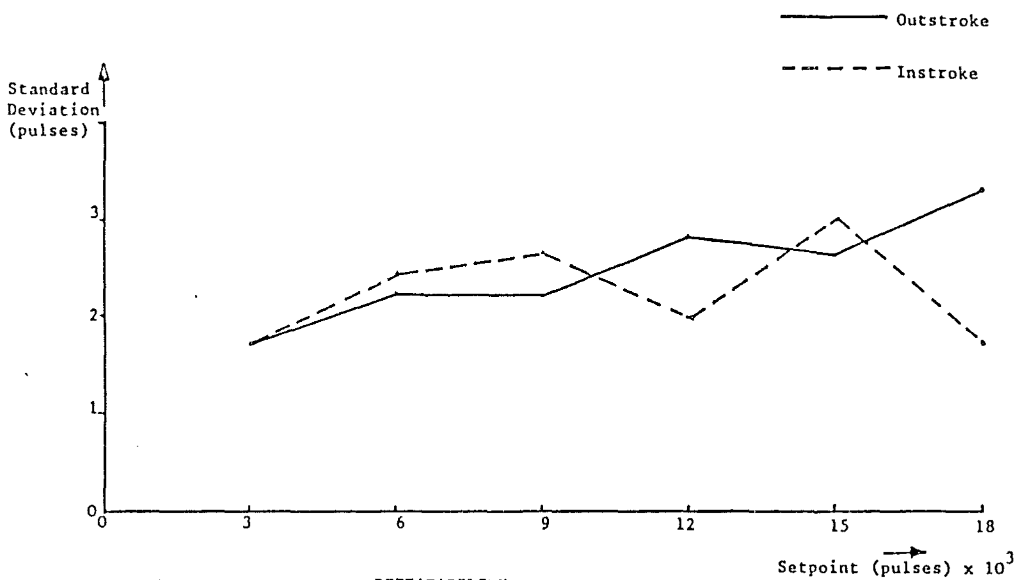
(a) LOAD EVALUATION (1) - 13kg payload

The results of a standard static performance evaluation are summarised in Figure [8.7.4.1] and Table [8.7.4.1]. A typical response is shown in Figure [8.7.4.2] with a summary of dynamic response given in Tables [8.7.4.2](a)and (b).

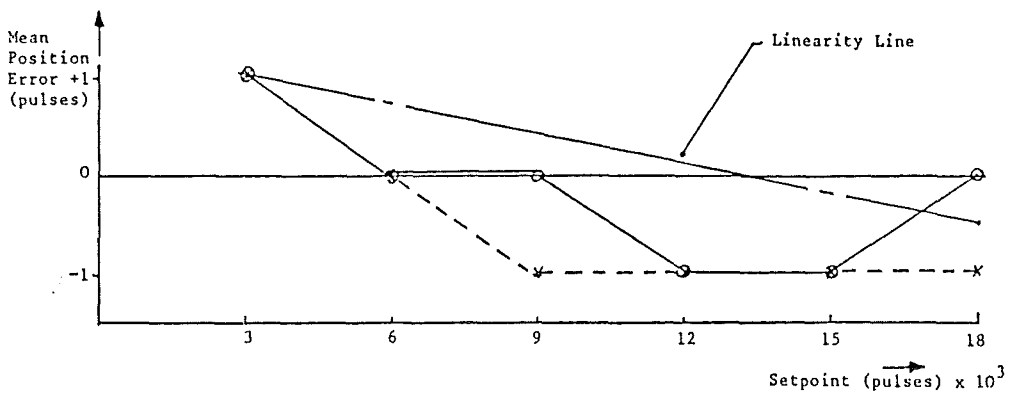
This evaluation of the M/60110/600 linear unit with a 13kg payload shows that static performance has not been adversely effected. The summary table shows that all quantitive static performance measures show good levels of performance. The results compare favourably with the Milcom evaluations on the M35 linear module.

The dynamic response of the system is quantified in terms of the summary tables and the transient response. These illustrate that a high static loading does considerably effect the dynamic response. In this evaluation the pay load configuration assists motion in the positive direction and acts against motion in the negative direction. The mean positioning time for an instroke move is increased by the order of twenty-five percent when compared to an equivalent outstroke move: due to the asymmetric nature of the payload. This asymmetry in the dynamic response can be balanced by appropriate selection of the bi-directional control system gains. Maximum

Figure (8.7.4.1) Load Evaluation (1)



REPEATABILITY



ACCURACY, LINEARITY AND HYSTERESIS

Table (8.7.4.1) Static Performance Summary

PARAMETER	OUTSTROKE (mm)	INSTROKE (mm)
Linearity	0.016	0.021
Repeatability	0.070	0.063
Hysteresis	0.009	
Accuracy	0.17	

Table (8.7.4.2) Dynamic Performance Summary

PARAMETER	MOVE OUTSTROKE (mm)	MOVE INSTROKE (mm)
Mean Positioning Time (ms)	1555	2006
Time Standard Deviation (ms)	N/A	N/A
Mean Positioning Velocity (mm/s)	53.9	41.3

Table (8.7.4.2)(b)

MOVE (mm)	DIR	V_p	POSITIONING TIME tp (s)	Vmax VELOCITY MAX (mms^{-1})	MEAN POSITIONING VELOCITY MPV (mms^{-1})	MOVE POSITION
83.8	O/S +	23	1.044	301.7	80.3	9000-12000
83.8	I/S -	14	1.542	183.6	54.3	12000-9000
279.4	O/S +	66	1.439	865.8	194.2	5000-15000
279.4	I/S -	26	1.958	341.0	142.7	15000-5000
530.9	O/S +	67	2.306	878.8	230.2	1000-20000
530.9	I/S -	28	2.942	367.3	180.4	20000-1000

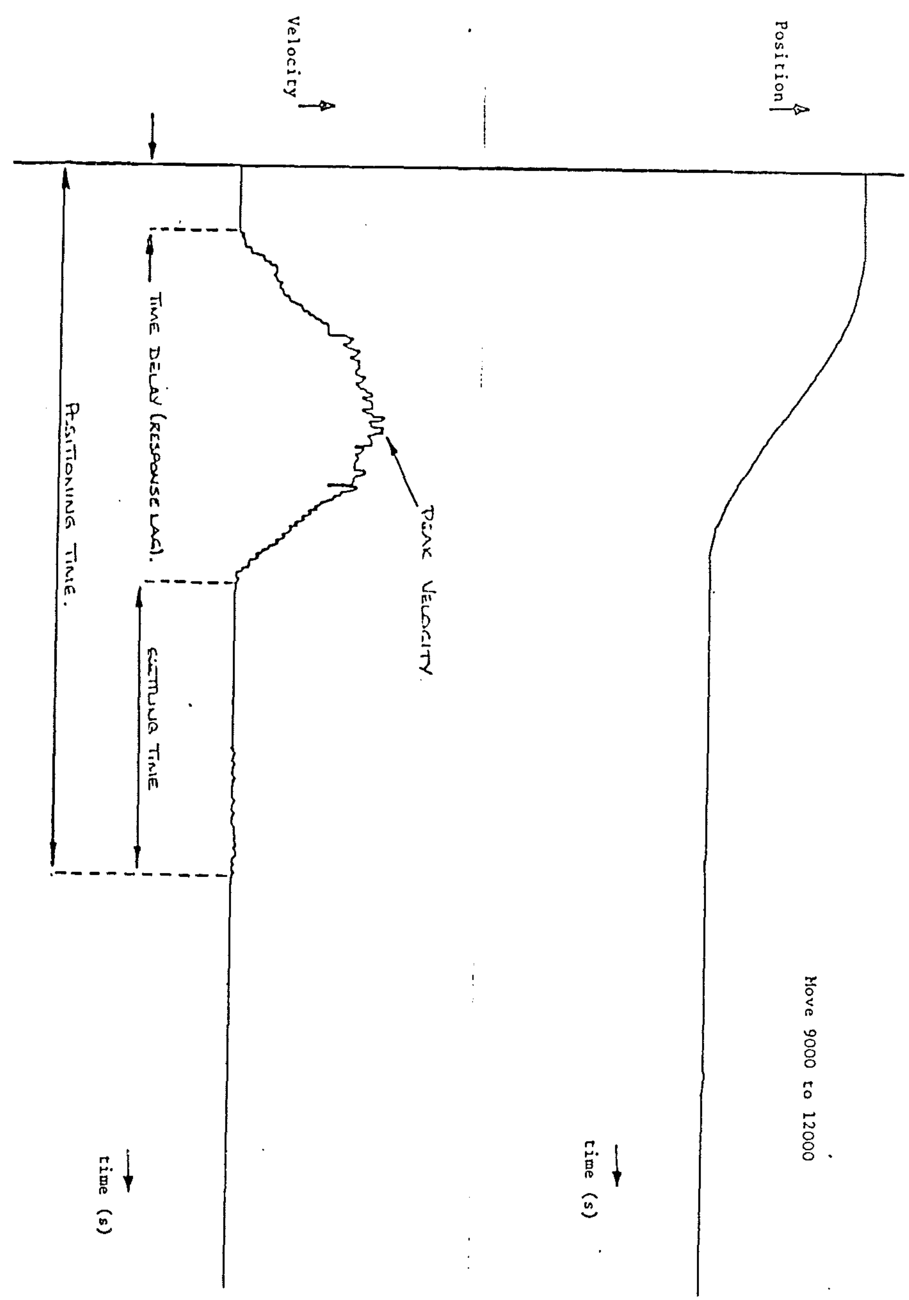


Figure (8.7.4.2) Dynamic Response

velocities in excess of 850mm/s were achieved in the outstroke direction for long moves, and this could be significantly increased as the controller saturation levels imposed for these evaluations were relatively low.

The transient response for any of the sample moves show the effect of the asymmetrical loading. For the instroke moves the response lag is significantly increased (in excess of 300 percent when compared to the outstroke move), and when motion takes place the mean velocity and maximum velocity are considerably lower when compared to those for the corresponding outstroke move. The settling phase for both instroke and outstroke moves appears to be very similar in duration and not significantly influenced by the load condition.

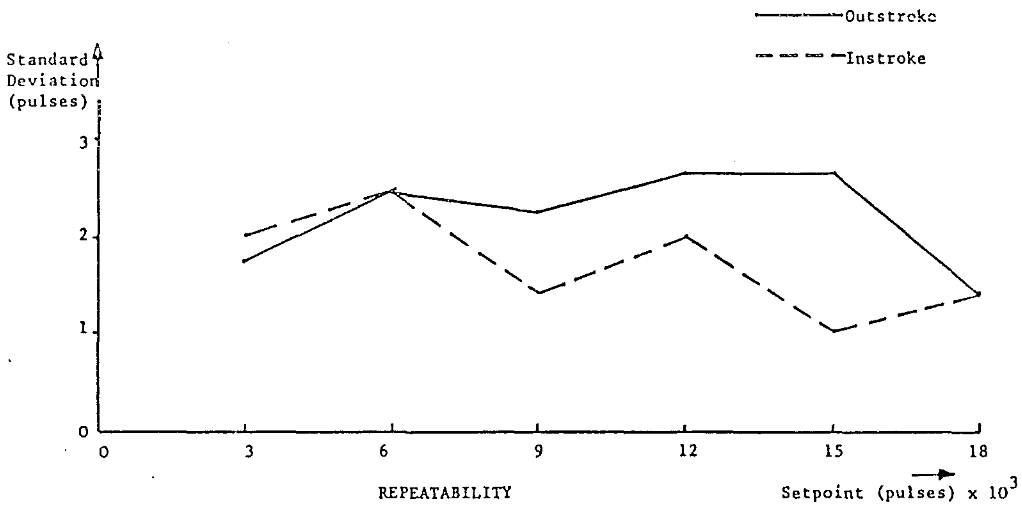
(b) LOAD EVALUATION (2) - 5kg payload

The results of a standard static performance evaluation are summarised in Figure [8.7.4.3] and in Table [8.7.4.3]. A typical transient response is shown in Figure [8.7.4.4] with a summary of dynamic performance given in Table [8.7.4.4](a) and (b).

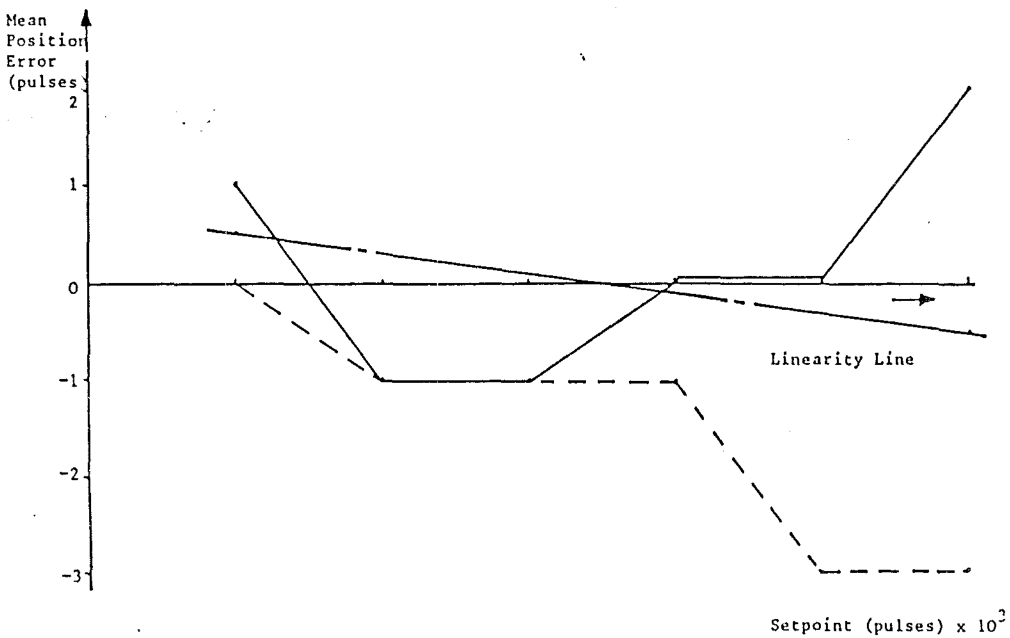
This evaluation of the M/60110/600 linear unit with a 5kg payload is designed to illustrate the effect of the reduced payload on the static and dynamic performance. The static performance criteria again show that results have not deteriorated with the application of the load and compare favourably with the best achieved in earlier evaluations.

The dynamic performance results (see summary tables and transient response), as would be expected, show an improvement over the previous evaluation. The asymmetric loading means that for identical moves the instroke motion is adversely influenced by the load: the positioning time is increased, the mean positioning velocity and the maximum velocity are lower than for the corresponding outstroke moves. However, the mean

Figure (8.7.4.3) Load Evaluation (2)



REPEATABILITY



ACCURACY, LINEARITY AND HYSTERESIS

Table (8.7.4.3) Static Performance Summary

PARAMETER	OUTSTROKE	INSTROKE
Linearity (mm)	0.027	0.042
Repeatability (mm)	0.061	0.048
Hysteresis (mm)	0.046	
Accuracy (mm)	0.17	

Table (8.7.4.4) Dynamic Performance Summary

PARAMETER	MOVE OUTSTROKE	MOVE INSTROKE
Mean Positioning Time (ms)	1.42	1526
Time Standard Deviation (ms)	N/A	N/A
Mean Positioning Velocity (mm/s)	59.0	54.9

Table (8.7.4.4) (b)

MOVE (mm)	DIR	V_p	POSITIONING TIME tp (s)	Vmax VELOCITY MAX (mms^{-1})	MEAN POSITIONING VELOCITY MPV (mms^{-1})	MOVE POSITION
83.8	O/S +	19	0.917	249.2	91.4	9000-12000
83.8	I/S -	17	1.221	223.0	68.6	12000-9000
279.4	O/S +	51	1.356	669.0	206.0	5000-15000
279.4	I/S -	40	1.737	524.7	160.8	15000-5000
530.9	O/S +	62	1.694	813.3	313.4	1000-20000
530.9	I/S -	51	2.015	669.0	263.5	20000-1000

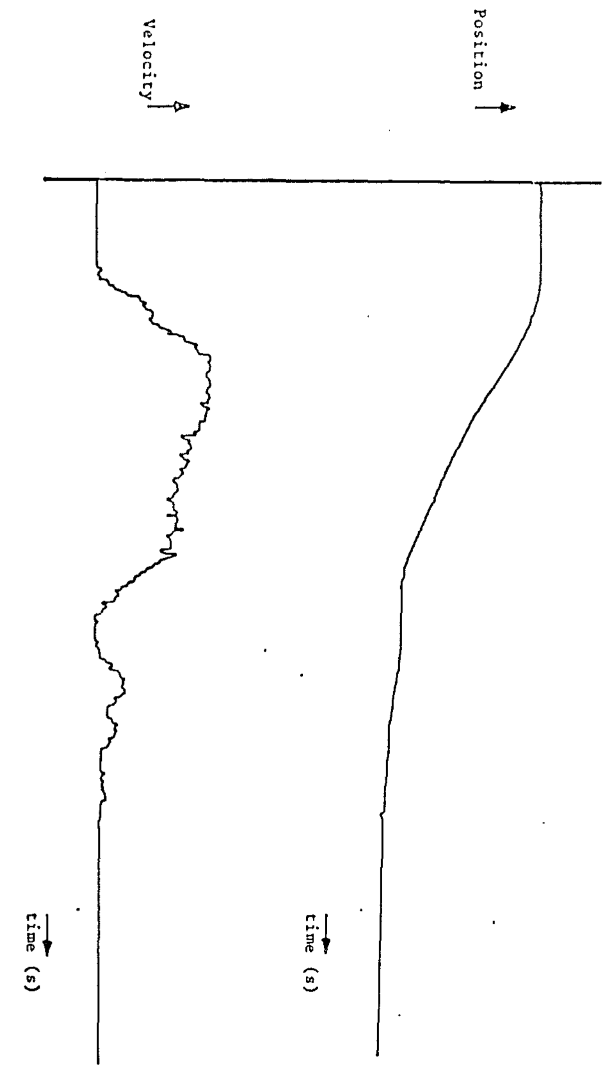


Figure (8.7.4.4) Dynamic Response

positioning velocities show an overall improvement compared with the 13kg payload evaluation. The response lag is a significant proportion of the positioning time for any instroke move.

(c) LOAD EVALUATION (3) - 0kg payload

The results of a standard static performance evaluation are summarised in Figure [8.7.4.5] and in Table [8.7.4.5]. A typical transient response is shown in Figure [8.7.4.6] with a summary of dynamic performance given in Tables [8.7.4.6](a) and (b).

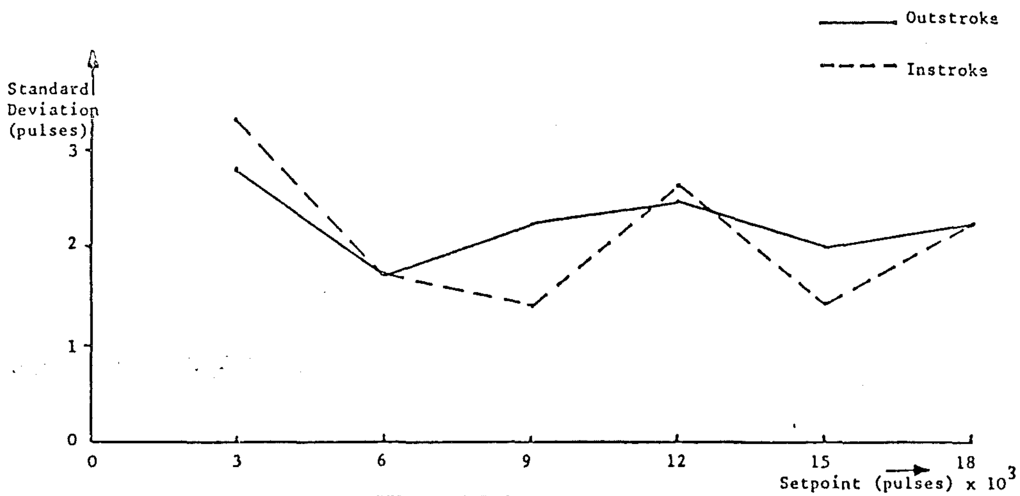
This evaluation of the M/60110/600 linear unit with zero payload (other than the intrinsic mass of the slideway) shows very good static performance has been achieved with respect to all the quantitive performance measures and compares favourably with the previous load evaluations.

With zero payload, the asymmetrical nature of the loading is removed and this is reflected by the dynamic response results. With a short move the 'response lag' is still a significant proportion of the positioning time (as was the case with the M35 module - Milcom ev(5)). This is caused by the command signal only reaching modest values when small errors in position are encountered as for short moves. An improvement in dynamic response results with the application of a 'front end' saturation control scheme applied until motion occurs.

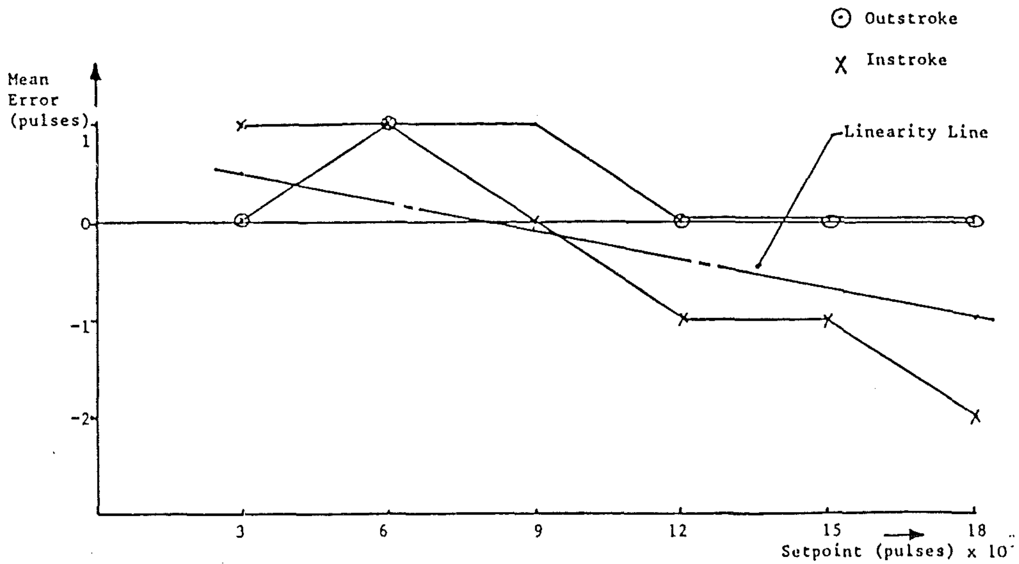
For corresponding instroke and outstroke moves the values for positioning time, mean positioning velocity and maximum velocity are similar: differences in the dynamic responses being accounted for by the asymmetry of the actuator and proportional valve. The levels of mean positioning velocity and maximum velocity achieved are correspondingly higher than the equivalent figures for the previous load evaluations.

(d) LOAD EVALUATION (4) - inertial payload

Figure (8.7.4.5) Load Evaluation (3)



REPEATABILITY



ACCURACY, LINEARITY AND HYSTERESIS

Table (8.7.4.5) Static Performance Summary

PARAMETER	OUTSTROKE	INSTROKE
Linearity (mm)	0.021	0.015
Repeatability (mm)	0.063	0.059
Hysteresis (mm)	0.028	
Accuracy (mm)	0.17	

Table (8.7.4.6) Dynamic Performance Summary

PARAMETER	MOVE OUTSTROKE	MOVE INSTROKE
Mean Positioning Time (ms)	1.25	1.28
Time Standard Deviation (ms)	N/A	N/A
Mean Positioning Velocity (mm/s)	67.0	65.5

Table (8.7.4.6) (b)

MOVE (mm)	DIR	V_p	POSITIONING TIME tp (s)	Vmax VELOCITY MAX (mms^{-1})	MEAN POSITIONING VELOCITY MPV (mms^{-1})	MOVE POSITION
83.8	O/S +	20	0.905	262.3	92.6	9000-12000
83.8	I/S −	18	0.993	236.1	84.4	12000-9000
279.4	O/S +	41	1.422	537.8	196.5	5000-15000
279.4	I/S −	47	1.055	616.5	264.8	15000-5000
530.9	O/S +	62	1.389	813.3	382.2	1000-20000
530.9	I/S −	58	1.849	760.8	287.1	20000-1000

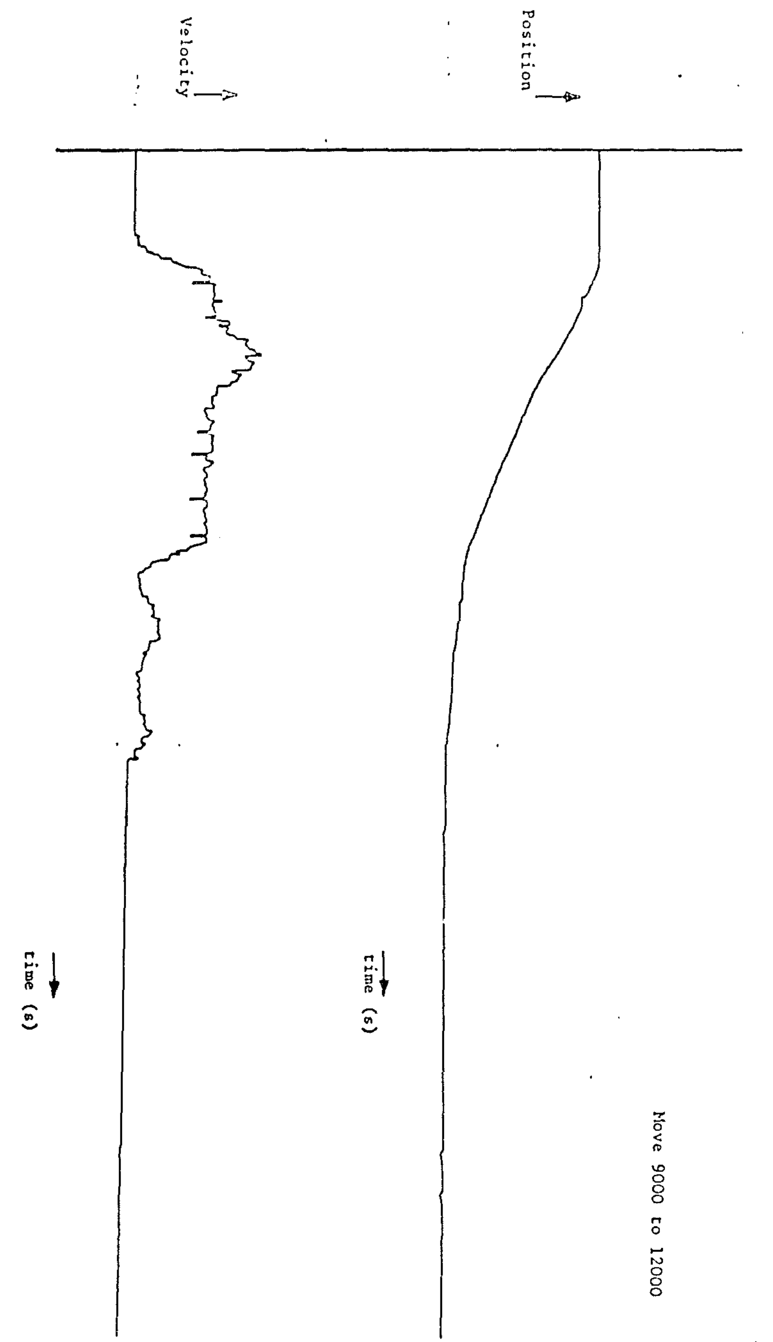


Figure (8.7.4.6) Dynamic Response

The test data from a standard static performance evaluation are summarised in Figure [8.7.4.7] and Table [8.7.4.7]. A typical transient response is shown in Figure [8.7.4.8] with a summary of dynamic performance given in Tables [8.7.4.8](a) and (b).

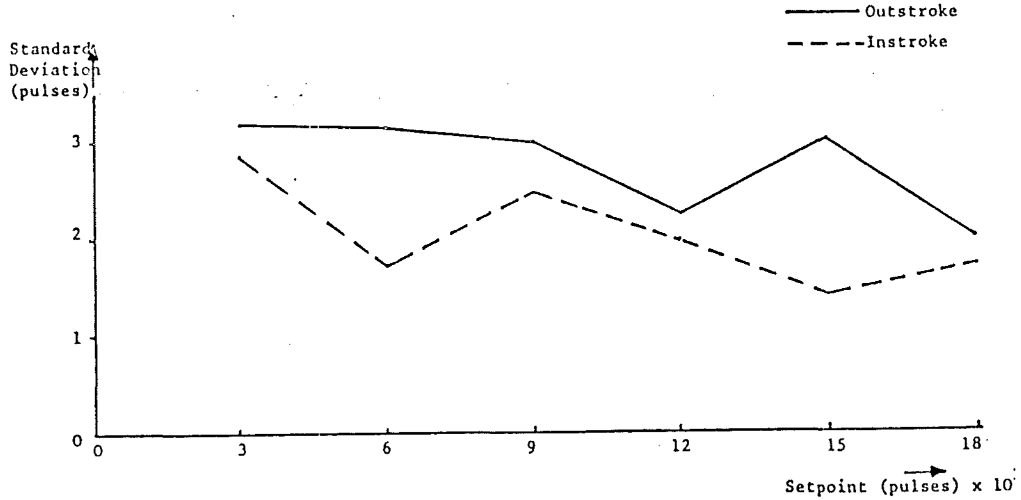
This evaluation of the M/60110/600 linear module with an inertial payload was conducted with the valve 'null' being set with the load in position. The static performance summary shows very good results are achieved with respect to all quantitive performance measures and compares favourably with the other load evaluations. It can conclude that inertial loading does not significantly influence static performance.

The dynamic response data illustrates that the response lag for an instroke move is substantially larger than the corresponding response lag for the outstroke move: this being particularly significant with short moves.

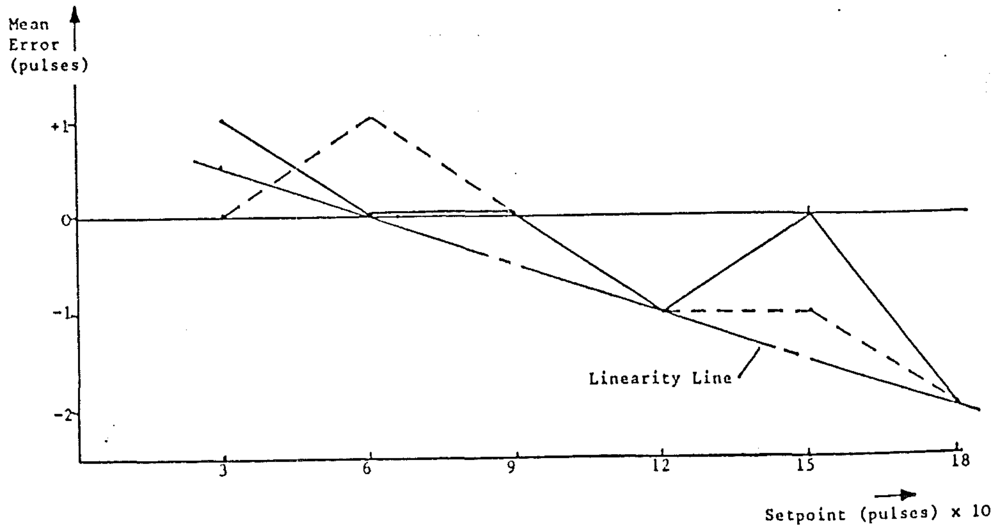
8.7.5. LOAD EVALUATION (MILCOM VERSION 2.50) CONCLUSIONS

It has been shown that variations in load system conditions do not adversely effect the static performance when using the minor loop compensation (Milcom Version 2.50) control strategy. However, the dynamic response is influenced by the load conditions but within limits this can be accomodated by the appropriate 'tuning' of the control system gains. Furthermore, a reduction in the 'response lag' by the introduction of 'front end' saturation control schemes (Milcom version 2.60) will improve the dynamic response under load particularly for short moves. From these load evaluations it would have to be concluded that such a pneumatic motion control system is appropriate for use in commercial positioning systems such as 'modular handling systems' for 'point-to-point' control of small

Figure (8.7.4.7) Load Evaluation (4)



REPEATABILITY



LINEARITY, ACCURACY AND HYSTERESIS

Table (8.7.4.7) Static Performance Summary

PARAMETER	OUTSTROKE (mm)	INSTROKE (mm)
Linearity	0.012	0.012
Repeatability	0.077	0.056
Hysteresis	0.014	
Accuracy	0.17	

Table (8.7.4.8) Dynamic Performance Summary

PARAMETER	MOVE OUTSTROKE (mm)	MOVE INSTROKE (mm)
Mean Positioning Time (ms)	1,275	1,503
Time Standard Deviation (ms)	N/A	N/A
Mean Positioning Velocity (mm/s)	65.65	55.69

Table (8.7.4.8) (b)

MOVE (mm)	DIR	V_p	POSITIONING TIME tp (s)	Vmax VELOCITY MAX (mms^{-1})	MEAN POSITIONING VELOCITY MPV (mms^{-1})	MOVE POSITION
83.8	O/S +	19	0.934	250	89.7	9000-12000
83.8	I/S –	17	0.910	223	92.1	12000-9000
279.4	O/S +	41	1.230	539	227.1	5000-15000
279.4	I/S –	43	1.113	565	251.0	15000-5000
530.9	O/S +	54	1.488	710	356.8	1000-20000
530.9	I/S –	56	1.746	736	304.1	20000-1000

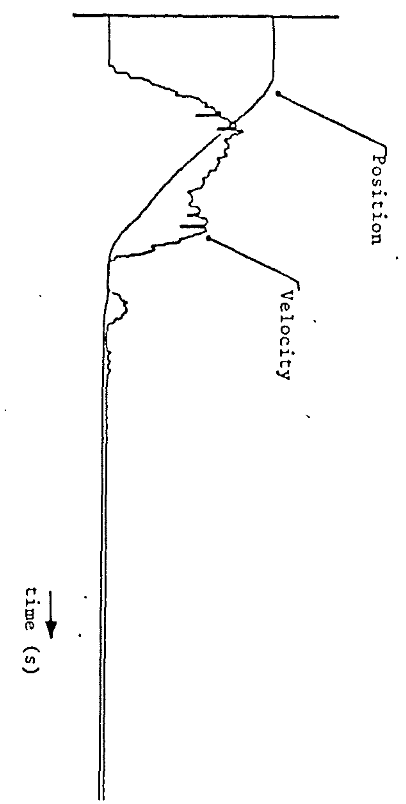


Figure (8.7.4.8) Dynamic Response

Move 9000 to 12000

and medium payloads.

It should also be noted that the magnitude of the allowable payload will be a function of the size and mechanical design of the actuator and slideway.

The control strategy described and evaluated in this research study though relatively complex in form (using minor loop compensation, non-linear gains and multiple control zones), is quite easy to 'tune' for a specified 'dynamic' and 'static' performance requirement. The control strategy is sufficiently 'robust' to be applicable to a wide range of modules, pneumatic actuators (Morgan [1985]) and load systems (as demonstrated in these evaluations).

The 'tuning' of control parameters is best achieved iteratively during installation/re-programming phases based on an empirically derived methodology. In the commercial implementation of this pneumatic motion control system only a limited sub-set of the control parameters need to be accessed by the user for 'fine-tuning'. These can be considered as parameters for determining the speed of response 'approaching' a setpoint (KGNPOS, KGNNEG setting the proportional error gain) and parameters for determining the 'damping ratio' during the 'settling phase' (where the velocity and acceleration loop gains are set using a single parameter). Other control parameters can be accessed (eg to improve the position 'deadband') but in general they can be considered as operating constants.

In Appendix [8] a control parameter tuning procedure is presented in flowchart form in Figure [8.7.5.1]. The final tuning tends to be an iterative procedure to match control parameters to the mechanical system, and performance required. The tuning procedure has been written with reference to Milcom version 2.50 and forms the basis of a 'single axis controller' set up procedure for the commercial pneumatic positioning system.

In the following section an assessment of the position feedback system of the M60110/600 linear module is presented to confirm the validity of performance evaluation data presented.

8.8. ASSESSMENT OF THE POSITION FEEDBACK SYSTEM OF THE M60110/600 LINEAR MODULE

The evaluation results presented in this thesis have been based on the feedback measurement of a position sensor which itself is an element within the control system. It was necessary to calibrate this sensor with an independant source of position measurement.

An assessment of the accuracy/linearity of the feedback system of the M60110/600 linear module was conducted. The M60110/600 module was used for a large proportion of the later static and dynamic performance evaluations and also it was felt necessary to calibrate the position feedback system currently employed. When applying digital control strategies that use state variable data (eg minor loop compensation) derived from a single feedback transducer the integrity of the velocity and acceleration is influenced by the accuracy/linearity of the feedback system.

8.8.1. TEST EQUIPMENT AND PROCEDURE

The feedback system of the M60110/600 module consists of a Hewlett Packard HED 5000 incremental encoder, with a rubber drive belt transmitting linear motion to a pinion wheel; thereby converting linear motion of the module to rotary motion of the encoder shaft.

The encoder quadrature pulse trains are processed directly by the encoder interface, providing an increase in the resolution by a factor of four. To calibrate this feedback system, it was necessary to use a secondary 'reference feedback' system as a standard, from which the accuracy and linearity of the M60110/600 feedback system can be determined. A Heidenhain LS513 incremental linear transducer was used as the 'reference feedback' system, which was interfaced to a duplicate encoder interface. Information from both encoder interface cards was processed by TMS9900 target systems. Details of the Heidenhain LS513 transducer are given in Chapter (5).

The M60110/600 has a position resolution of 0.028mm/pulse while the LS513 has a position resolution of 0.01mm/pulse; hence providing a reasonable increase in resolution to facilitate calibration. The LS513 linear transducer was mounted to comply with the manufacturers' specification. The sliding head of the LS513 transducer was coupled to the module using an aluminium carrier arm.

The evaluation software package described in Chapter (5) was used as a convenient method of conducting the calibration tests. A sequence program was produced which consisted of forty-two setpoints evenly distributed over the full-stroke of the module, each point separated by an interval corresponding to 500 pulses of the rotary encoder. Both outstroke and instroke tests were carried out. Using single step mode the module was moved to each setpoint: having initialised both the encoder feedback and LS513 feedback systems to zero at a datum 'home' position to ensure synchronisation. When 'in-position' the position recorded by the encoder was stored, as was the position recorded via the LS513: the module being moved sequentially to each setpoint.

It was then necessary to convert the rotary encoder pulse readings to equivalent LS513 readings, allowing the readings to be directly compared.

From the difference in these readings calibration could be achieved.

8.8.2. ACCURACY AND LINEARITY

The test results are shown graphically in Figure [8.8.1] which shows a plot of deviation in position readings against stroke position for both instroke and outstroke moves.

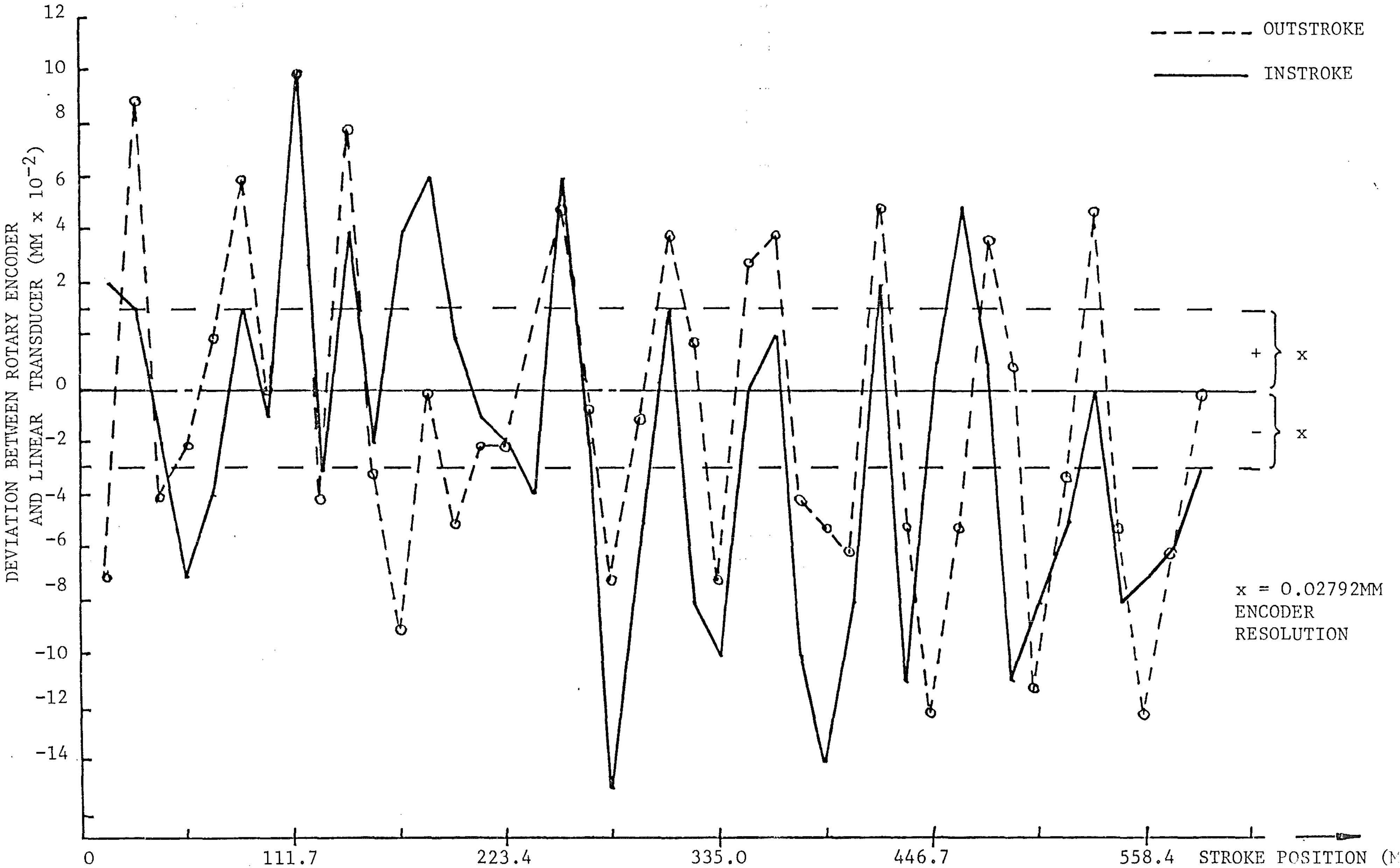
These results show that for outstroke motion the maximum deviation in position reading between the rotary encoder and the LS513 linear transducer corresponds to plus or minus 0.12mm and for the instroke motion the largest position deviation is +0.12mm and -0.15mm.

From Figure [8.8.1] it can be seen that the module feedback system (combination of encoder and transmission) exhibits a cyclic deviation with respect to the linear transducer, which is approximately sinusoidal in form and can be related to the toothed belt and pinion gear arrangement. The pattern of deviation is similar for both directions with a slight offset, which is attributable to transmission backlash.

The absolute accuracy attainable from a module is a function of the precision of the feedback system. The feedback system precision can also impose limitations on the position resolution and the accuracy with which velocity and acceleration data can be derived (errors in accuracy and linearity will influence the value of derived velocity and acceleration data).

A possible source of error in the feedback system is eccentricity between the pinion and the encoder shaft, or alternatively any ovality in the pinion/shaft assembly. This form of error can produce a cyclic sinusoidal error source, the diameter of the pinion determining the pitch of the error. Errors due to the alignment or tensioning of the belt transmission

Figure (8.8.1) Linearity and Accuracy Calibration



during assembly of the slideway will also influence accuracy and linearity. A source of backlash in the feedback system can be attributed to the large tooth pitch (on the pinion and tooth belt) used with this prototype module: tooth belts usually being designed to have several teeth engaged at any one time whereas with the Martonair modules the belt has not been used in conventional form.

Having discussed possible sources of error in the present feedback system (primarily connected with the transmission), it should be noted that for many industrial applications the drive belt and pinion provide an adequate feedback transmission and are very low cost. If greater absolute accuracy and linearity are required then a more elaborate transmission system must be employed, such as a steel or nylon rack and anti-backlash pinion (as employed in the prototype M35 linear module). Alternatively in high precision applications a linear displacement transducer could be used (such as the Heidenhain LS513) negating the requirement for a transmission, the penalty being an increase in cost.

8.9. ALTERNATIVE VERSIONS OF MINOR LOOP COMPENSATION CONTROL STRATEGY

During the course of this research study alternative versions of minor loop compensation control strategy were investigated. These control strategies are briefly described here; however formal evaluation results are not presented - evaluation results are given in Moore [1986]. Alternative control strategies were studied in an effort to improve the dynamic response (enhanced stability and improved speed of response), however Milcom version 2.50 was proved to be the most successful implementation of minor loop compensation. Greater improvements in dynamic performance were achieved using 'front end' control schemes.

8.9.1. MINOR LOOP COMPENSATION - MILCOM VERSION 3.0 - NON-LINEAR VELOCITY LOOP GAIN (POSITION ERROR DEPENDANT)

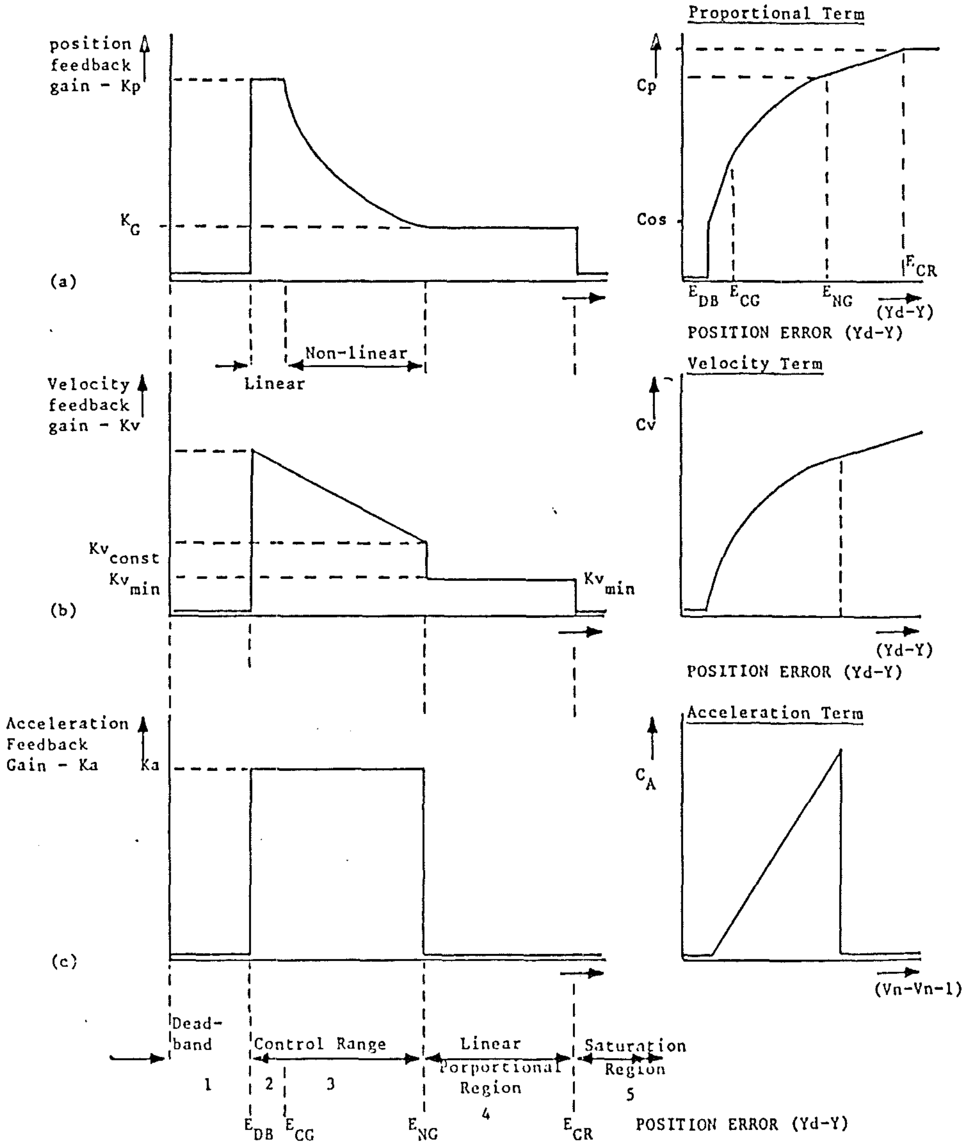
This control strategy is a variant of conventional minor loop compensation. Here the velocity feedback gain (Kv) is non-linear, and its value is a function of the position error (Yd-Y). In this way the damping present in the system can be increased as the setpoint is approached, but for larger positon errors (when the level of damping can be reduced) the velocity gain is reduced. The introduction of a non-linear feedback gain should provide enhanced stability when close to the setpoint, whilst allowing a faster dynamic response.

The relationship of the control system gains is shown in Figure [8.9.1.1]. As can be seen the velocity feedback gain is a function of the position error within the control range, and outside this region the velocity feedback gain becomes constant. Within the control range the velocity feedback gain is calculated by a straight line function (see Figure [8.9.1.1]). The ideal relationship between velocity loop gain and position error would be similar to the proportional gain/error characteristic. The non-linear characteristic of the velocity feedback gain would result in an increased velocity term as the proportional gain increased, as such a suitable ratio for stability can be maintained between the feedback loop gains.

The velocity feedback gain is calculated from a straight line function, as a compromise between taking a constant value and implementing an additional look-up table with its associated complexity. The velocity gain describing function, as implemented, produces a velocity feedback term (Cv), as shown in Figure [8.9.1.1], where even low velocities at small position errors produce a significant damping term.

Outside the control range,. the velocity loop gain takes a minimum value Kv(min). The acceleration gain Ka has a constant value within the control range. All other features of the control strategy are as described previously. The changes of significance in the control software are shown in flowchart form in Figure [8.9.1.2]. The evaluation of this control strategy is considered in Chapter (9).

Figure (8.9.1.1) Control Gains - Milcom Version 3.0



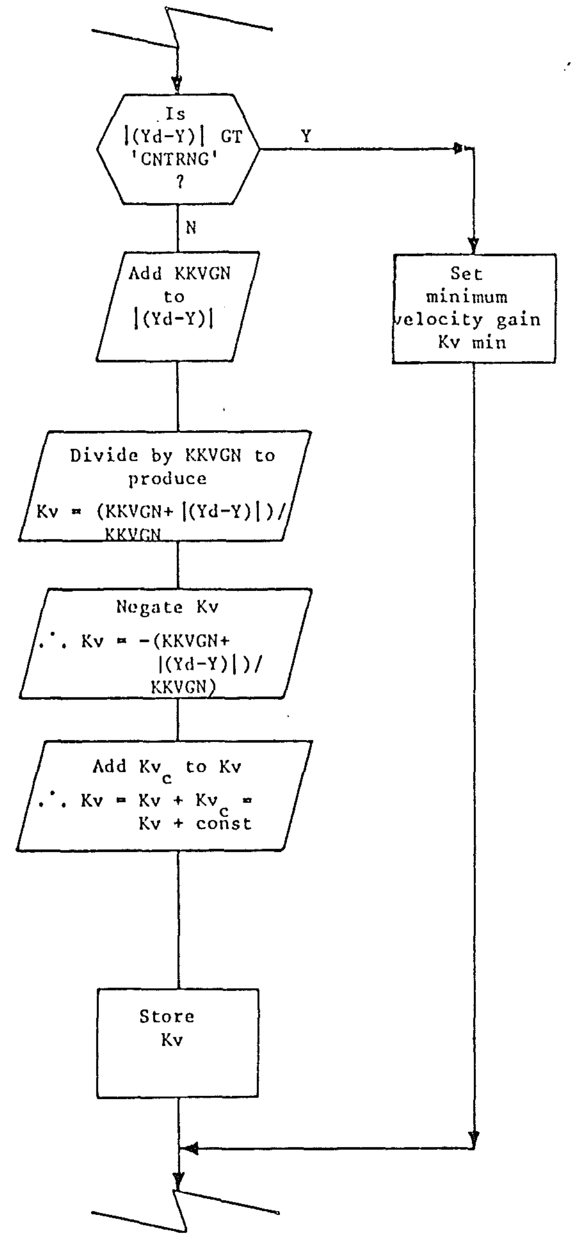


Figure (8.9.1.2) Minor Loop Compensation - Milcom Version 3.0 (position error related velocity gain)

8.9.2. MINOR LOOP COMPENSATION - MILCOM VERSION 4.0 - NON-LINEAR

VELOCITY LOOP GAIN (LOAD POSITION DEPENDANT)

The control strategies described so far in this Chapter, have made use of non-linear feedback gains to compensate for deficiencies in the dynamic and static characteristics of the drive system. Inspection of the open loop linearised model presented in Chapter (7) indicates that the drive stiffness will vary as a function of load position. As the relative chamber volumes of the actuator change, the stiffness and damping will change. It can be shown that the mid-stroke position (Ym) of a linear actuator is the least damped and at the extremes of stroke the damping increases to a maximum level, (Burrows [1969]).

The forward path gain Kf' can be shown to be

$$Kf' = Kp.K = \frac{Kp}{m}\left[\frac{4A^2\ Pq^2\ Gq^2}{Gp\{Vi^2 - A^2\ [Y-Ym]^2}\right]$$

For a given actuator, valve, and load mass 'm', with a constant proportional error gain Kp, then

$$Kf' = \frac{K'}{[Vi^2 - A^2\ [Y-Ym]^2\ ]}$$

where

$$K' = \frac{Kp}{m}\left[\frac{4A^2\ Pq^2\ Gq^2}{Gp}\right]$$

For a 25mm diameter cylinder with a 400mm stroke length (assuming actuator symmetry);

Piston Area $A = 4.6 \times 10^{-4}\ m^2$

therefore $A^2 = 21 \times 10^{-8}$

and $V_i = A.S/2 = 0.92 \times 10^{-4}\ m^3$

thus $V_i^2 = 0.85 \times 10^{-8}$

and $$Kf' = \frac{10^8 .K'}{[0.85-21(Y-Ym)^2]}$$

Constructing a table of position in stroke

Y-Ym	Kf'
0	$1.18 \times 10^8\ K'$
0.1m	$1.57 \times 10^8\ K'$
0.15m	$2.6 \times 10^8\ K'$
0.175m	$4.8 \times 10^8\ K'$
0.185m	$7.7 \times 10^8\ K'$
0.195m	$20 \times 10^8\ K'$
S/2=0.2m	

From which Figure [8.9.2.1] can be drawn showing the relationship between load position and the forward path gain.

High stiffness at the extremes of stroke should be beneficial , but additional damping and higher stiffness is obviously desirable about the mid-stoke position. In an attempt to increase the damping and improve stability in the mid-stroke a control strategy was implemented where the velocity feedback gain is made a function of load position.

An implementation of a non-linear velocity feedback gain (Kv) related to load position about midstroke is shown in Figure [8.9.2.2]. This allows an increase in the velocity gain (and hence increased damping) as midstroke is approached. The structure of the control strategy is shown in flowchart form in Figure [8.9.2.3].This particular control strategy was never fully evaluated due to time constraints.

The ideal velocity feedback gain describing function could only be implemented in the form of a look-up table or contour, but this adds to the complexity. It would be feasible to relate the proportional error gain and the acceleration gain to the load position in a similar way. In this way variation in forward path gain with load position could be compensated.

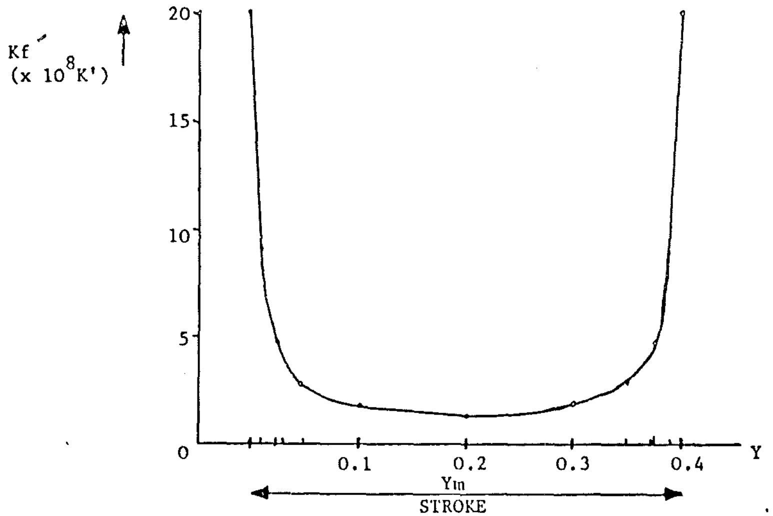


Figure (8.9.2.1) Variation of Forward Path Gain with load position in a pneumatic cylinder

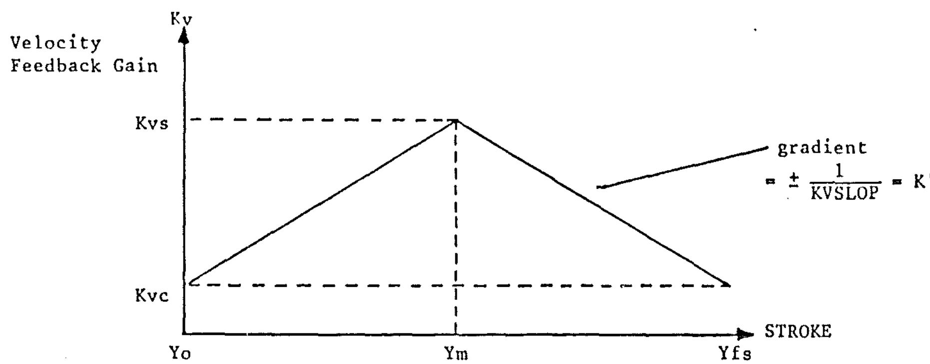


Figure (8.9.2.2) Load Position Related Velocity Feedback Gain

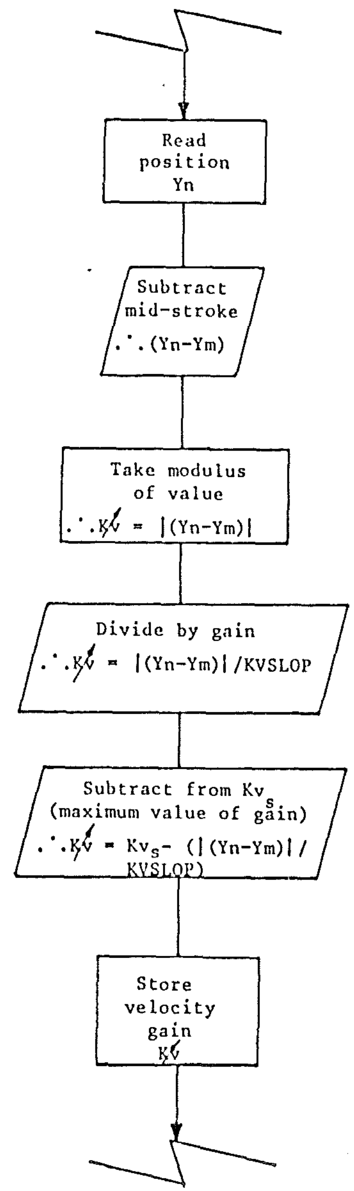


Figure (8.9.2.3) Minor Loop Compensation - Milcom Version 4.0 (load position related velocity gain)

8.10. FRONT END CONTROL SCHEMES APPLIED TO PNEUMATIC MOTION CONTROL

In this section several control strategies are described that are designed to improve the dynamic response characteristics of the pneumatic position control system. The minor loop compensation strategy is used with the addition of 'front end' control schemes in an effort to enhance the dynamic response characteristics. The approach is based on an outer decision loop which modifies the command issued to an existing closed loop system. This outer loop is designed to generate a set of initial (front end) control sequences at the start of any 'point-to-point' move.

Deficiencies in the dynamic response characteristics are mainly attributable to the 'time lag' after a command signal is issued, and 'rise time' of the transient response which can be sluggish for short moves or under the influence of load. These characteristics are attributable to dominant non-linearities caused by stiction, compressibility, hysteresis and saturation effects in the proportional valve. These non-linearities can make the dynamic response unpredictable which results in variable speeds of response and overshoot for different moves, though the strategy has a high degree of robustness and reliability.

8.10.1. FRONT END COMMAND SIGNAL 'SATURATION CONTROL' SCHEME

The multi-strategic minor loop compensation control scheme is illustrated in Figure [8.10.1.1]. The first application of a front end control scheme with this strategy incorporated the use of command signal 'saturation control' and is illustrated in Figure [8.10.1.2]. To reduce the time lag of the system the appropriate command signal saturation value (Csat) is applied until movement of the actuator is detected. This control approach

Figure (8.10.1.1) Minor Loop Compensation -

Algorithm Scheduling

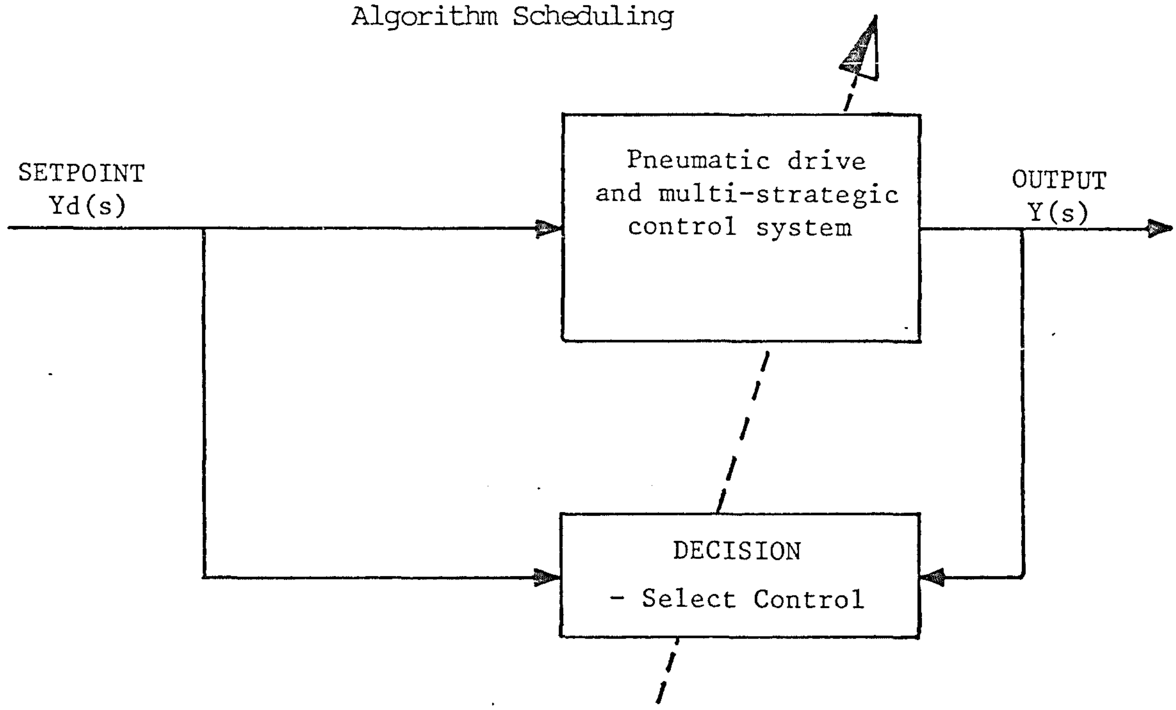
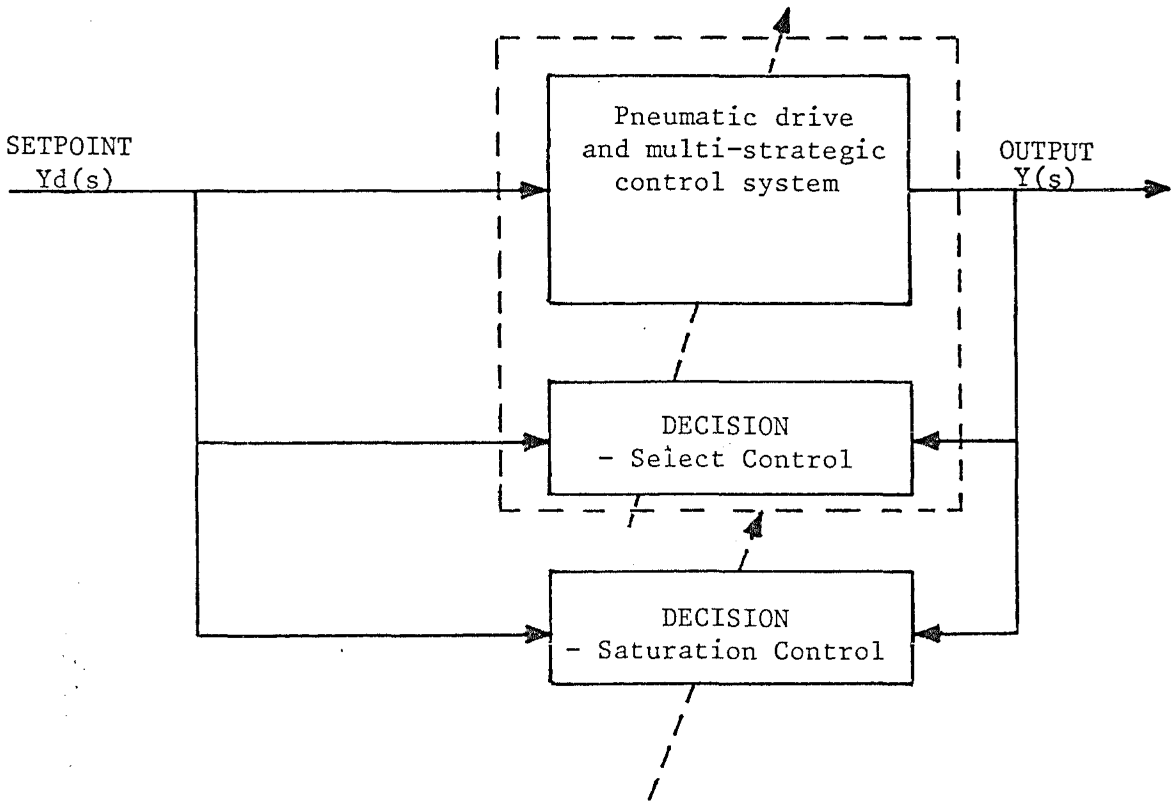
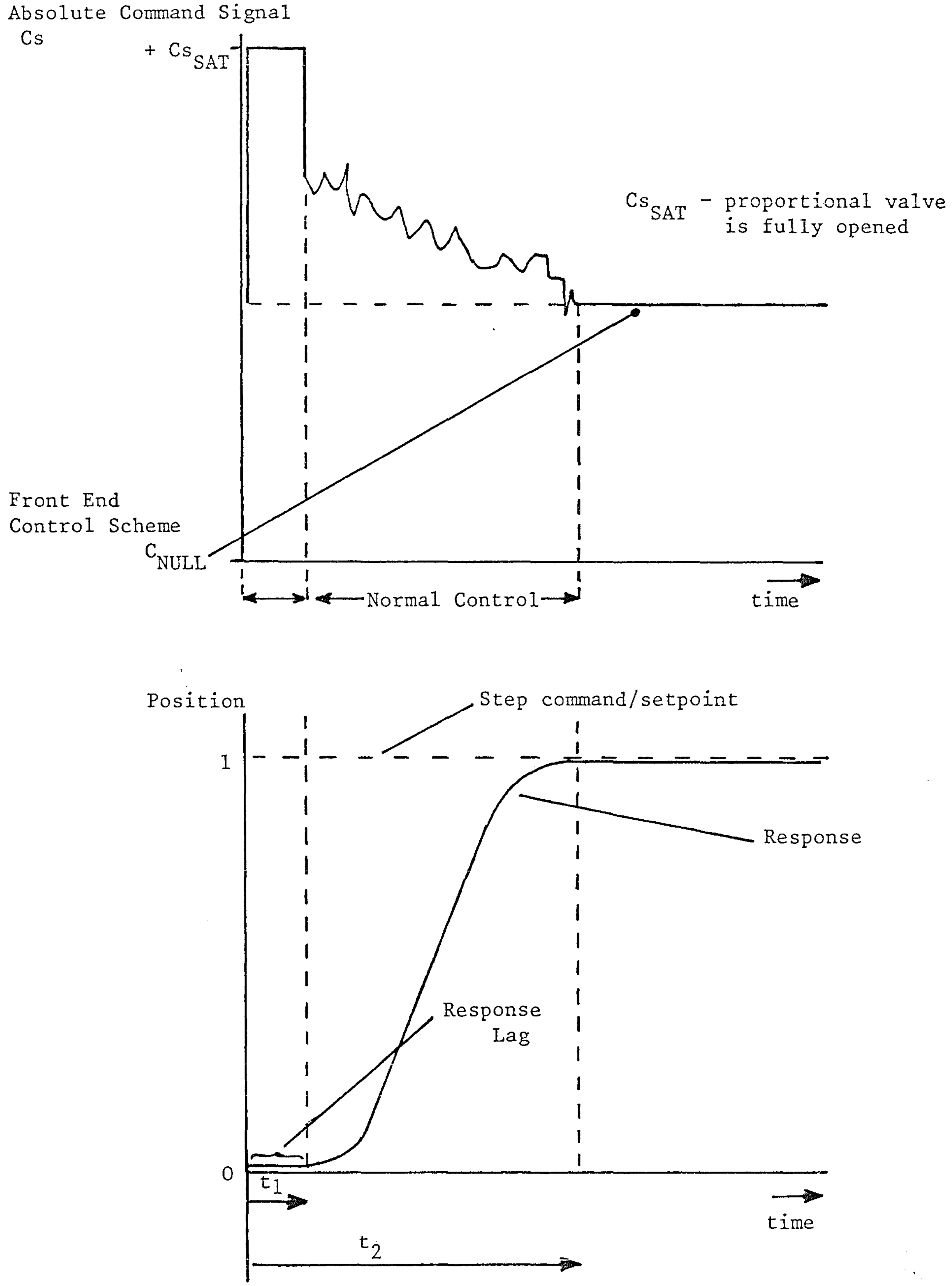


Figure (8.10.1.2) Minor Loop Compensation with Saturation Control



is illustrated in Figure [8.10.1.3], and was implemented in Milcom version 2.60, and is illustrated in flowchart form in Figure [8.10.1.4]. This front end control scheme must reduce the response lag to a minimum for any given load condition, actuator, valve and supply pressure, the maximum flowrate of the proportional valve being determined by the command signal saturation Csat when the valve is fully opened. Hence the fastest initial response possible to a step command will be achieved by the system when this command is applied. The time (t1) for which the command saturation is applied is determined by the time lag. The command cannot be maintained at saturation for a longer period, as at some later instant, overshoot and instability will occur, particularly for small moves.

Figure (8.10.1.3) Saturation Control



t_1 = response lag time for which command signal saturation

t_2 = tp = positioning time

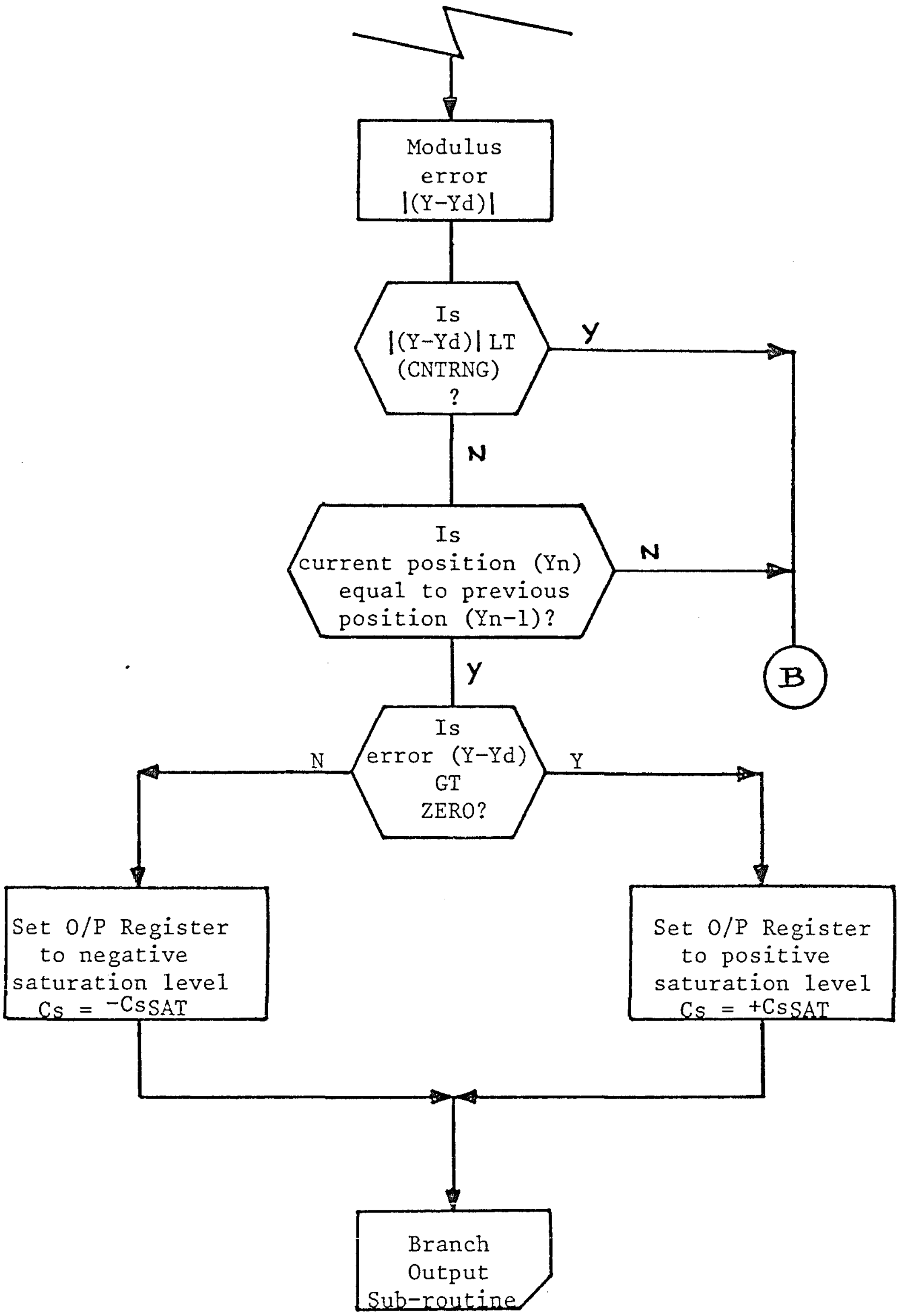


Figure (8.10.1.4) Milcom Version 2.60 - Saturation Control

8.10.2. FRONT END CONTROL SCHEMES INCORPORATING SETPOINT SWITCHING TECHNIQUES

Front end control schemes incorporating setpoint switching or modification techniques for pneumatic drives have been used in a parallel research study at the University (Nagarajan et al [1985]), where the control system described in this thesis was used and a logical decision is incorporated, external to the original realtime control software, to action an adjustment to the non-linear setpoint gain Ks. The approach is illustrated schematically in Figure [8.10.2.1].

A control scheme has been designed that combines 'setpoint modification' and 'command signal saturation control' to produce an improved dynamic response characteristic by minimising the 'time lag' and reducing 'rise time'. The control signal (Cs) for a typical move is shown in Figure [8.10.2.2], which shows that the control signal is modified as a logical function of the setpoint and selected features of the response.

In this control scheme (Milcom version 2.7) the control signal is initially saturated (Csat) until motion is registered: time t1. The setpoint to the controller is then adjusted for a time t2 producing a control signal greater than would be the case for the original RTC: thereby allowing the system to move faster towards its desired position. The magnitude of the setpoint modification and its duration are two control parameters to be selected to increase the speed of response. In this implementation of the control scheme, a fixed setpoint modification is made which corresponds to twice the original step command. The duration t2 is chosen to correspond to the time at which the position error (Yd-Y) reaches half its initial value. Many other criteria could be used to select these

Figure (8.10.2.1) Schematic Representation of a Front End Control Scheme - setpoint modification

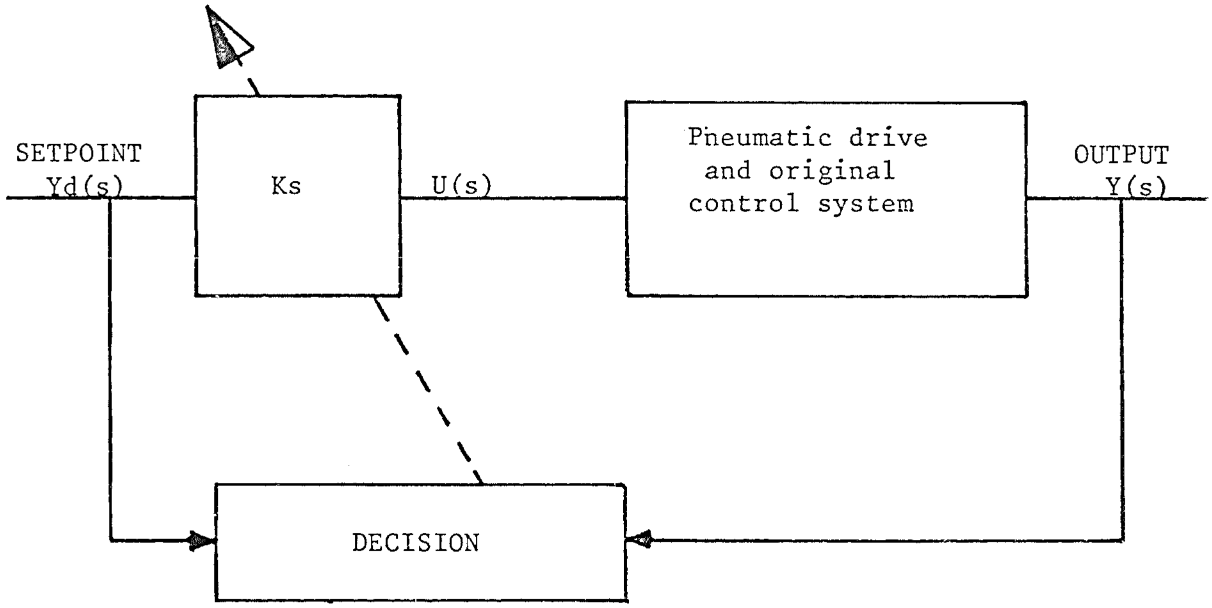
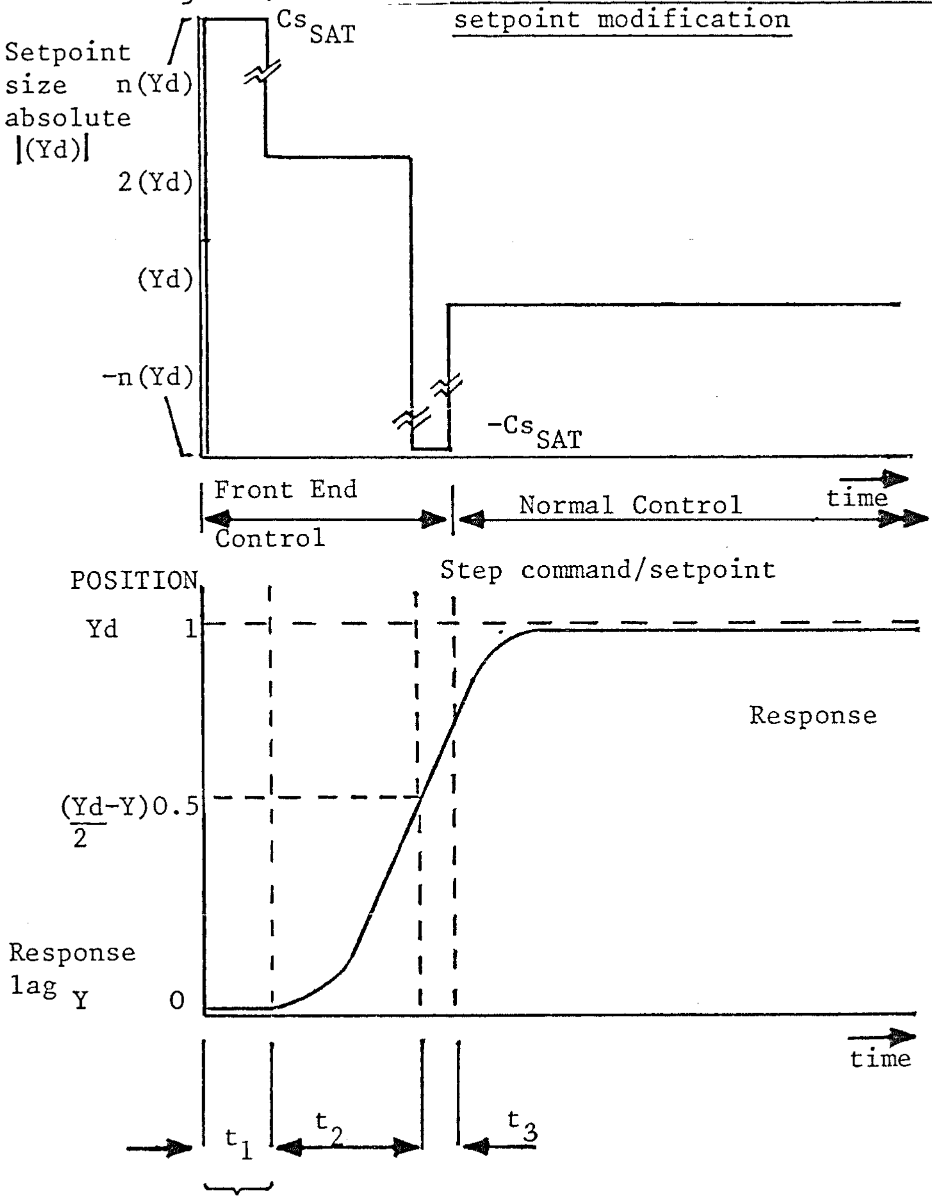


Figure (8.10.2.2) Front End Control Scheme incorporating setpoint modification



control parameters. This type of control scheme design was suggested by Astrom [1981] where it is employed as the first sample in a regulator designed for stable systems with monotone step response.

With these control parameters having a fixed relationship it is desirable to have a control parameter that can be used to adjust the amount of overshoot once an acceptable speed of response has been obtained. Here we use reverse command signal saturation (-Csat) for a duration t3, where the control parameter sets the percentage of the move over which this reverse or negative command signal saturation is applied.

The influence of 'Front-End' control schemes on dynamic response in pneumatic motion control is studied in Chapter (9), where results are presented.

8.11. LEARNING SCHEMES FOR PNEUMATIC MOTION CONTROL

The front end control schemes described in the previous section can be incorporated with a minimum of modification into a simple adaptive control scheme. This could have great potential for use in the control system when used in industrial applications, particularly where operating conditions and load conditions vary considerably. Such an adaptive scheme can be used to ensure that the quality of performance does not degrade with time or variation in operating conditions.

The aim of an adaptive control scheme is to adapt its control and 'learn' an optimised parameter level for each move, the criteria for adapting the control parameters is dependent upon the design of the adaptive system. The adaptive scheme can be used continuously 'on-line' or alternatively be used for fine tuning to learn the 'optimised' parameters to use subsequently.

Adaptive controllers are used when the plant to be controlled is unknown (namely the parameters of the plant are not known) and the plant parameters vary with respect to time in an unpredictive manner. Adaptive controllers work 'on-line' to generate 'adaptive controls', where they modify their own parameters in order to minimise the deterioration of plant performance due to variations in plant parameters. The adaptive control system, hence, has an 'adaptive mechanism' to modify the parameters of the controller by employing realtime control algorithms of appropriate design.

In the general scheme it is required to maintain plant performance as close as possible to a desired level. A performance index can be used to measure this performance. The performance of the plant is compared with a desired performance, dependant upon the deviation, the parameters are adjusted such that the deviation is always made a minimum. The adjustment of controller parameters is made based upon an adaptive algorithm with which the adaptive mechanism works. It is essential that the overall scheme (plant, controller, adaptive mechanism, and performance evaluation) is 'asymptotically' stable, that is, that modification of controller parameters is made in such a way that the performance deviation from the desired level tends towards zero. Various techniques can be employed to ensure asymptotic stability (Eveleigh [1976], Ogata [1978]).

An alternative adaptive control scheme is to have the desired performance specified in a 'reference model', known as a 'model reference adaptive control' (MRAC) scheme. The reference model (usually a software sub-system) is designed to achieve the specified performance desired from the input to the plant. The adaptive mechanism alters, 'on-line', the controller parameters so as to minimise a defined performance indice. When the adaption is complete the plant-controller combination behaves identically to the reference model (Shacloth and Buchart [1965]).

Another approach to adaptive control is to measure plant performance, on-line, and to modify the controller parameters on the basis of these measurements. Using an 'identifier', the input and output of the plant are measured and the set of plant parameters computed. The on-line parameter identifier is also a 'model reference' system. With this approach the plant parameters will be tracked as they vary with time (Kudvn and Narendra [1974]). A detailed survey of types of adaptive control schemes is given by Landau [1974].

8.11.1. A CONTROL STRATEGY WITH LEARNING

In the learning control scheme implemented in each run the control system evaluates its own performance and compares it with specified performances.

Depending on the deviation of the actual performance and the specified performance, the control parameters are modified. The modification of the parameter values can be based on an optimisation procedure such as steepest descent method. At the end of the trial runs, (or continuously), the software stores the optimised parameters, for use in the next run.

There are many parameters that affect performance, but to simplify the optimisation process only one parameter is chosen in this case. The parameter chosen is the duration t3, during which time the reverse command signal saturation is applied as part of the 'front end' control scheme. This type of adaptive scheme is illustrated in Figure [8.11.1.1].

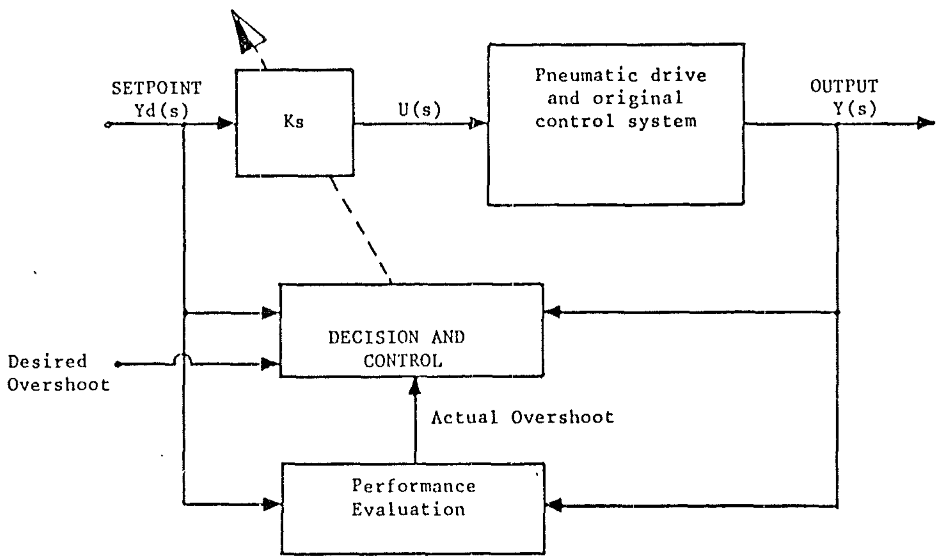
In this implementation with only t3 to be optimised a simple search procedure is sufficient, the search procedure flowchart is illustrated in Figure [8.11.1.2]. The routine modifies t3 in an effort to keep position overshoot within a specified tolerance. The duration of the learning phase depends on the increment by which t3 is modified and the specified tolerance.

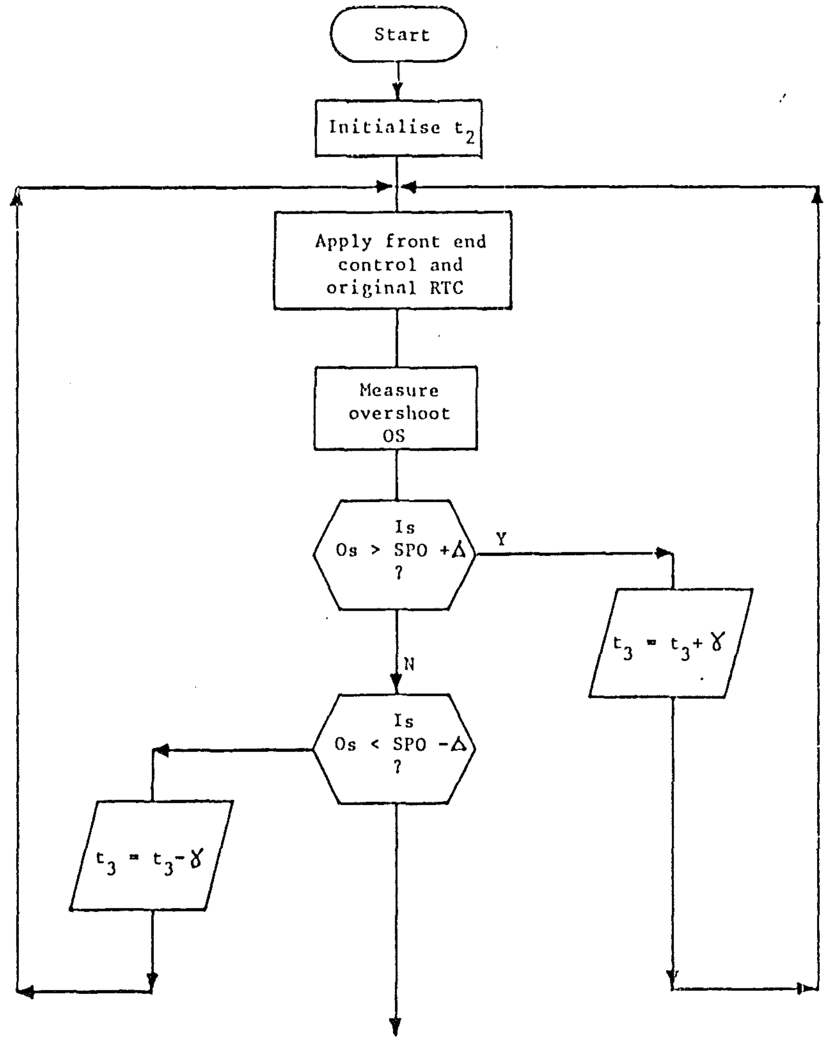
Position overshoot is the selected performance measure (or indice) chosen for this implementation but other indices could be used (see Chapter (5)). This type of learning scheme could be applied to any other selected control parameter, such as 'velocity loop gain' or 'proportional error loop gain'. A 'learning strategy' as described here is studied in Chapter (9), where results are presented.

8.12. CONCLUSIONS

In this Chapter the evolution of a motion control strategy for pneumatic servo-drives has been considered. A novel control strategy has evolved incorporating minor loop compensation and non-linear feedback loop gains.

Figure (8.11.1.1) Learning/Adaptive Control Scheme





Os = Overshoot
SPO = Specified overshoot
Δ = Tolerance factor to the specified overshoot
γ = A factor by which t_3 is modified

Figure (8.11.1.2) Search Procedure used in the

Adaptive Control Scheme

The additional use of 'front-end' control strategies and 'learning/adaptive' schemes can further improve the dynamic response and robustness of the control. Analysis of the dynamic response characteristics with various motion control strategies is presented in Chapter (9).

CHAPTER 9

PERFORMANCE COMPARISON OF THE MOTION CONTROL STRATEGIES EVOLVED

9. INTRODUCTION

In this Chapter a study of the dynamic response characteristics of the M60110/600 linear module is described for a range of motion control strategies. In the previous Chapter a series of performance evaluations were presented for a range of motion control strategies. The primary objective of those performance evaluations was to determine static performance data for the drive system; together with an analysis of the the dynamic response for the motion control strategies evolved.

The studies outlined in this Chapter attempt to analyse the performance of a pneumatic linear module in further detail and compare directly alternative control strategies. The dynamic response characteristics of a drive system will directly influence the acceptability of the drive for various manufacturing applications. In many industrial tasks short cycle times are an essential feature of any automation scheme (eg assembly, packaging and palletising where high speed point-to-point motion is a requirement).

The evolution of the control strategies described in Chapter (8) has resulted in the design of a pneumatic motion control system that is finding wide acceptance as an industrial drive with potential cost benefits over alternative drive technologies. Milcom versions 2.50 and 2.60 have been proven to be robust control strategies resulting in good static and dynamic performance. However, the dynamic response of a pneumatic drive can be further enhanced with the addition of 'front-end' control schemes that result in a faster and more predictable dynamic response characteristic. This study of the dynamic response presented here will quantify the beneficial influence of such 'front-end' control schemes when applied to a pneumatic drive.

A short study of control strategies which incorporate a learning scheme, as described in Chapter (8), are also presented. The learning schemes are simple in concept to keep the additional processing time necessary to a minimum. Such learning schemes can be used in one of two ways;

(i) A learning scheme is used as part of an initialisation phase when the pneumatic drive system is being set up. The control parameters are set at default values and then the necessary program sequence is taught to the actuator. The actuator then runs through this sequence in an 'initialisation mode' in which the control strategy optimises the response of each move by successively modifying one control parameter. Once a specified level of performance has been achieved for each setpoint then the learning stops. In this way the learning scheme has resulted in a specified dynamic response and the necessary 'tuning' of control parameters has been greatly simplified.

(ii) Alternatively, a learning scheme can be used adaptively 'on line', that is the learning is always active within the control strategy. Any detrimental change in performance will be accomodated by the strategy self adapting the chosen control parameter/s until performance returns to a specified level. In some applications such an approach would be most desirable, perhaps where large variations in ambient temperature are encountered, or external forces will change.

The limited study presented here will serve to illustrate the potential of 'learning schemes' when applied to a pneumatic servo drive.

9.1. MOTION CONTROL STRATEGIES

For the purpose of this study, the 'programmable actuator' comprised a Martonair linear module M60110/600 (with a 600mm stroke operating at a 6 bar supply pressure) and the load table as described in Chapter (5). In addition, a digital storage oscilloscope and pen plotter were used to provide storage, measurement and hard copy facilities to analyse the experimental data generated.

As indicated previously in Chapter (8), a number of control strategies for pneumatic motion control have been investigated. Some of these control strategies are described and experimental data produced during dynamic performance evaluations is referenced. The evaluation data is presented in the form of transient responses and tabulated performance indice relating to pre-defined moves of 56mm, 112mm and 168mm (which represent a cross section of point-to-point move categories). By selecting identical moves (and position in stroke) a direct comparison can be made with respect to selected performance criteria for each strategy: a sampling interval of 2.133 milliseconds being used for all evaluations. Of major importance in comparing different control strategies is the selection of appropriate performance criteria. In practise, the performance of a drive system will ultimatley be application dependant but generally with positioning mechanisms, such as robot modules, good static performance can be identified with satisfactory repeatability/accuracy while good dynamic performance requires fast positioning (and hence short cycle times) coupled with adequate damping (critical damping usually being optimal). Performance indices used in this study to quantify the dynamic response include settling time, positioning time, maximum velocity, mean positioning velocity and absolute error integral. These indices are defined in Table [9.1].

9.2. PROPORTIONAL ERROR CONTROL

To achieve conventional proportional error control, position information (measured by the encoder) is obtained once each sampling interval and the controller generates a command signal in proportion to the position error,

where $Cs = Kp(Yd - Yn)$ ---(9.1)

Cs = command signal

Yd = setpoint or desired position

Yn = quantised position after 'n' samples

and Kp = position error gain

Using only position information, loop closure in software may be achieved in several ways (Weston et al [1984], Moore et al [1985], Moore et al [1984]), but here this was achieved in two ways; viz

(i) selecting a constant position loop gain (Kp), and

(ii) selecting a non-linear position loop gain (Kpn) as a function of position error.

To study the effect of varying the position loop gain an evaluation was conducted by selecting a gain (Kp) to achieve a critically damped transient response. Subsequently, the gain was increased in stages and the changes in response monitored and analysed.

A non-linear position loop gain was also investigated, the non-linear gain being achieved by storing a proportional term look-up table in memory (Moore et al [1984]) and selecting and scaling this index as a function of position error. In this way, high stiffness can be achieved close to

Performance Indices	
Settling Time (ts)	it is the time required for the response to reach and remain within a specified tolerance band of its final value (usually defined as a percentage of move length)
Positioning Time (tp)	the time required for the response to reach and remain within the specified position 'deadband' (which defines an acceptable steady state error used in the software to flag "in-position")
Maximum Velocity (Vmax)	this is the peak value of velocity measured in any point to point move
Mean Positioning Velocity (MPV)	this is the mean velocity level during any point to point move
Absolute Error Integral (AEI)	this is the sum of errors measured for any point to point move

Table (9.1)

setpoint without inducing instability, ie the gain reduces as the error increases (Weston et al [1984]).

9.2.1. ASSESSMENT OF THE EFFECT OF PROPORTIONAL ERROR CONTROL

The effect of increasing the magnitude of a linear position loop gain is illustrated in Figure [9.2.1]. This figure shows the response of the module for two step inputs (setpoints), in each case a critically damped response is achieved with gain (Kp=K1). However, when the gain is increased (Kp=10K1) the response becomes faster and overshoot occurs, and as the gain is further increased (Kp=45K1) the response becomes unstable. The performance indices are listed in Table [9.2.1] from which it can be seen that the maximum velocity achieved increases as the gain is increased and the absolute error integral is reduced until the module reaches an unstable condition. The proportional error gain K1=6.35 volts/m.

These results illustrate clearly the severe control problems associated with a pneumatic drive where there is little inherent natural damping, low values of stiffness and natural frequency occur and the pneumatic time delay is significant at low operating pressures due to the compressible nature of the drive medium. Furthermore, the 'proportional' valve controlled actuator system is highly non-linear.

Applying a low position loop gain results in a sluggish response, positioning can only be achieved when using a large deadband which only allows an absolute position error of 3.4mm or better to be achieved. Increasing the loop gain reduces the necessary deadband but results in overshoot and ultimately instability.

By careful design a non-linear position loop gain results in a faster response, a smaller position deadband and reduced overshoot (see Chapter

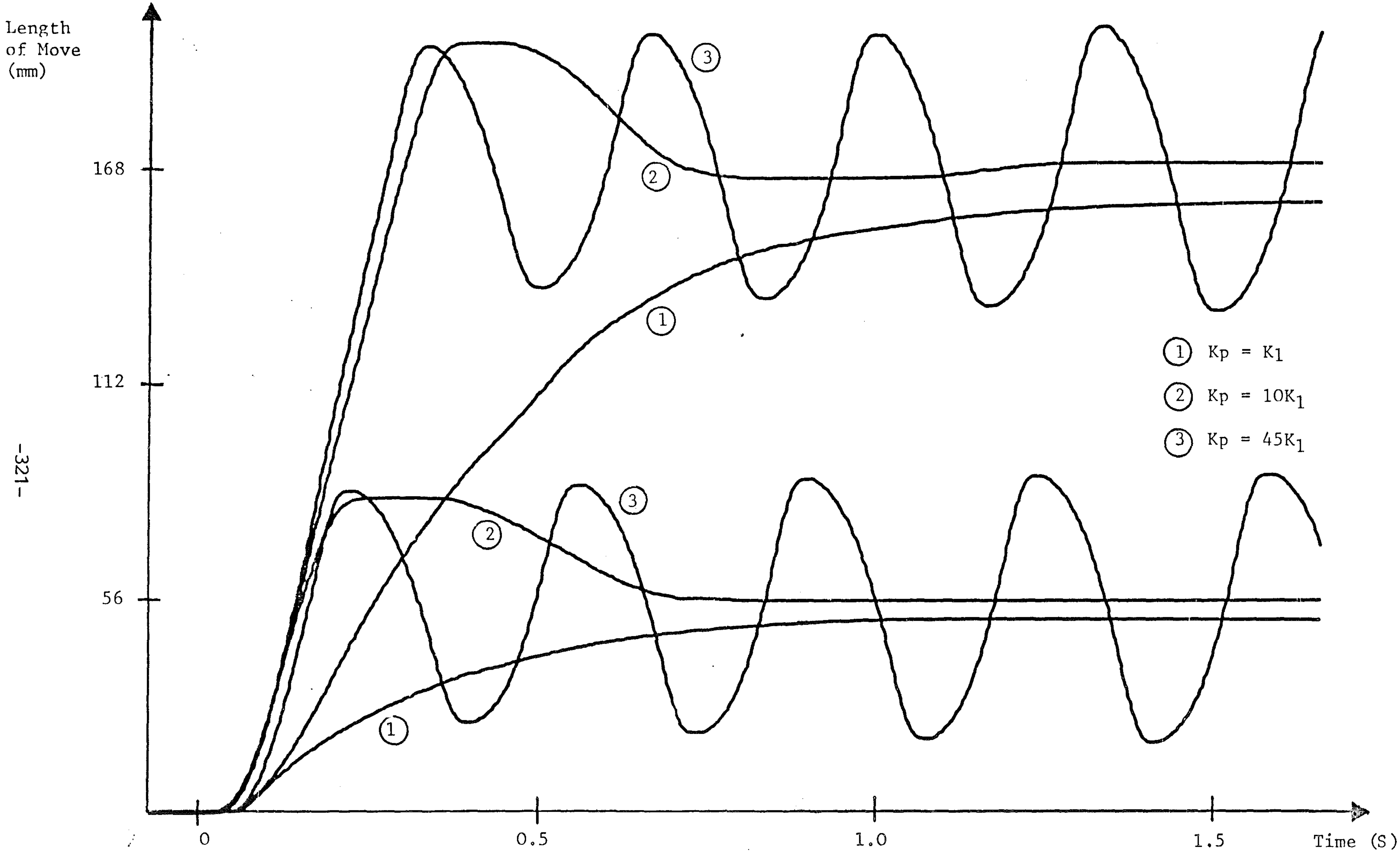


Figure (9.2.1) Transient Response
Proportional Error Control
Proportional Loop Gain Kp
$K_1 = 6.35\ V.m^{-1}$

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mm x 10^3$)	MPV (mms^{-1})	Length of Move (mm)
$Kp = K_1$	-	1340	157.6	12.86	41.79	56
	1200	1650	341.4	37.51	262.08	168
$Kp = 10K_1$	1410	1870	748.4	6.24	29.95	56
	973	1530	853.5	17.23	109.80	168
$Kp = 45K_1$	The response is oscillatory					

Table (9.2.1)
Module Performance With Proportional Control

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mm x 10^3$)	MPV (mms^{-1})	Length of Move (mm)
Using Kp	913	542	183.82	7.45	61.34	56
Using Kp_n	494	166	682.76	2.85	113.36	
Using Kp	1032	610	315.12	15.55	108.53	112
Using Kp_n	299	258	748.41	8.51	374.56	
Using Kp	1024	657	485.81	24.72	164.06	168
Using Kp_n	597	382	735.28	19.44	281.71	

Table (9.2.2)
Module Performance With Kp and Kp_n Proportional Control

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mm x 10^3$)	MPV (mms^{-1})	Length of Move (mm)
$Kv = K_2$	910	1480	787.80	4.16	37.84	56
	745	1360	800.90	17.00	123.53	168
$Kv = 3K_2$	277	393	722.15	2.92	142.49	56
	382	574	827.19	15.53	292.68	168
$Kv = 8K_2$	523	973	590.85	4.12	57.55	56
	589	1120	761.54	19.38	150.00	168

Table (9.3.1)
Module Performance Using Minor Loop Compensation with a Constant Velocity Gain (Kv)

(8)). A comparison of linear and non-linear loop gains is illustrated in Figure [9.2.2] and Table [9.2.2] where the magnitude of each gain was selected to produce a near critically damped response. A further improvement in drive stiffness could be achieved with an increase in system damping (Weston et al [1984]).

9.3. MINOR LOOP COMPENSATION

Minor loop compensation (Weston et al [1984], Moore et al [1985]), in the form of velocity and acceleration state feedback, can be used to introduce a controlled damping term into the control system. The state variable data is derived from the position feedback using software algorithms.

By using this velocity and acceleration information, compensation can be achieved in several ways (Moore et al [1984]), the additional damping allowing the drive stiffness to be increased whilst still maintaining stability. Here the command signal (Cs) is formed using the relationship;

$$Cs = Kp(Yd-Y) - Kv.V - Ka.A$$

where Kv = velocity loop gain

Ka = acceleration loop gain

With reference to the linearised model of the drive described in Chapter (7) (Weston et al [1984]), the velocity and acceleration loops can be considered to adjust the coefficients of the characteristic equation: the coefficients being dependant on the magnitudes of the loop gains Kp, Kv and Ka.

In this analysis a large position loop gain (Kp=75K1) was used with a constant acceleration loop gain (Ka). The velocity loop gain (Kv) was implemented in two ways, viz

(i) as a constant loop gain (Kv), and

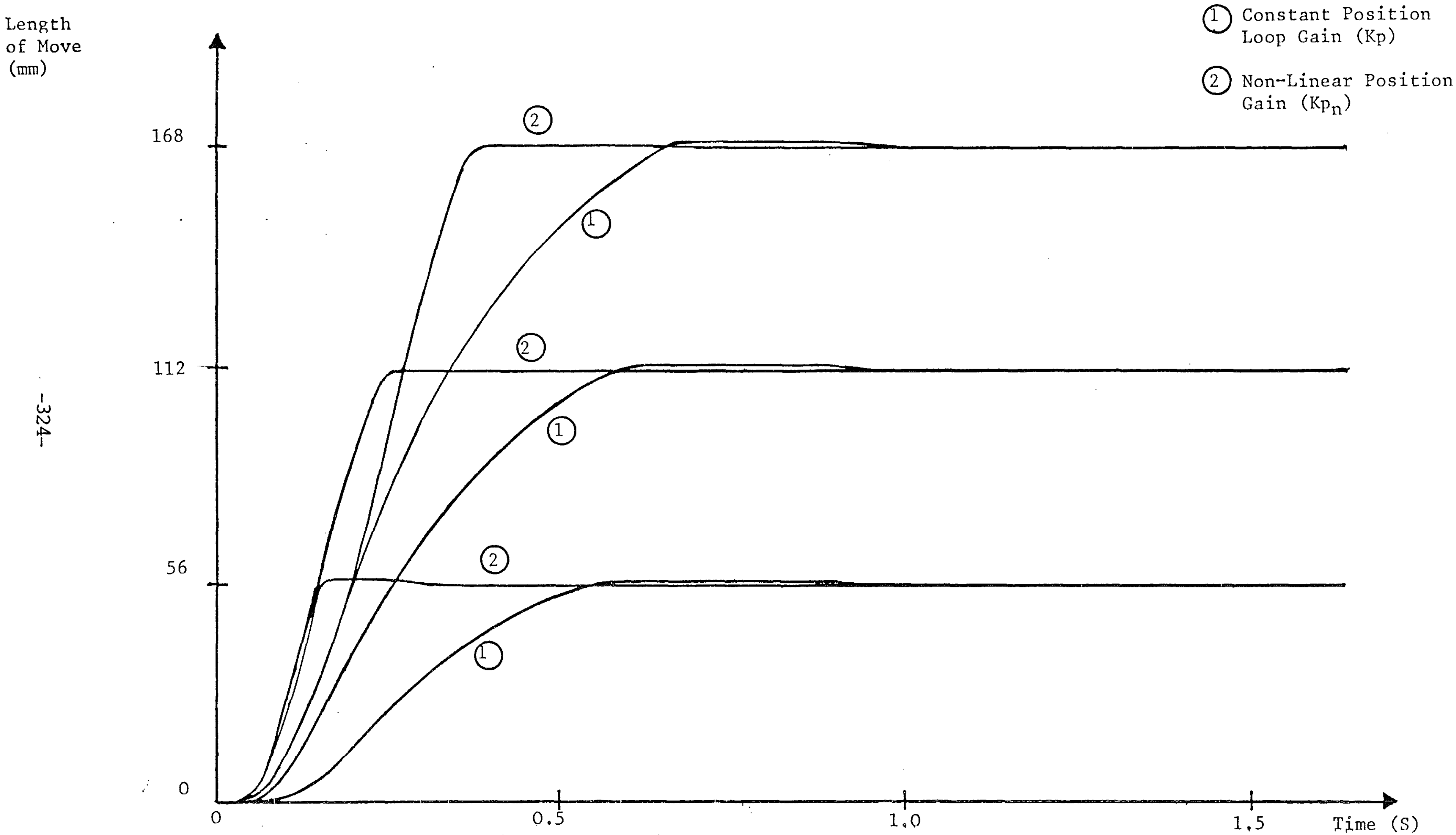


Figure (9.2.2) Transient Response
Proportional Error Control
Comparison of Constant and Non-Linear Position Loop Gains.

(ii) as a non-linear loop gain (Kvn), where the velocity gain is selected as a function of position error (see Chapter (8) for details).

9.3.1. ASSESSMENT OF THE EFFECT OF MINOR LOOP COMPENSATION

The effect of increasing the magnitude of a constant velocity loop gain is illustrated by the transient responses of Figure [9.3.1]. The response is oscillatory with a low velocity loop gain (Kv=K2) while as the velocity gain is increased (Kv=3K2) we approach a critically damped condition. A further increase (Kv=8K2) results in an overdamped, sluggish response. Table [9.3.1] lists the performance indices for each response. The critically damped response results in an absolute error integral of lowest value and the highest mean positioning velocity, thus illustrating a significant improvement in dynamic response which is directly attributable to the use of minor loop compensation.

The velocity loop gain, $K2 = 1.8 \times 10^{6}$ volts/ms^{-1}

Resulting from the stabilising effect of minor loop compensation, the allowable increase in drive stiffness enables an absolute position error of 0.2mm or better to be achieved (Weston et al [1984], Moore et al [1984]). Static performance positioning repeatability figures of this order have also been demonstrated for the complete family of linear modules now marketed by Martonair (see Chapter (8) for details of static performance evaluations).

When a non-linear velocity loop gain is used, the magnitude of the gain is a function of the position error (Yd-Y). With position errors less than Ecrg the velocity loop gain is determined by,

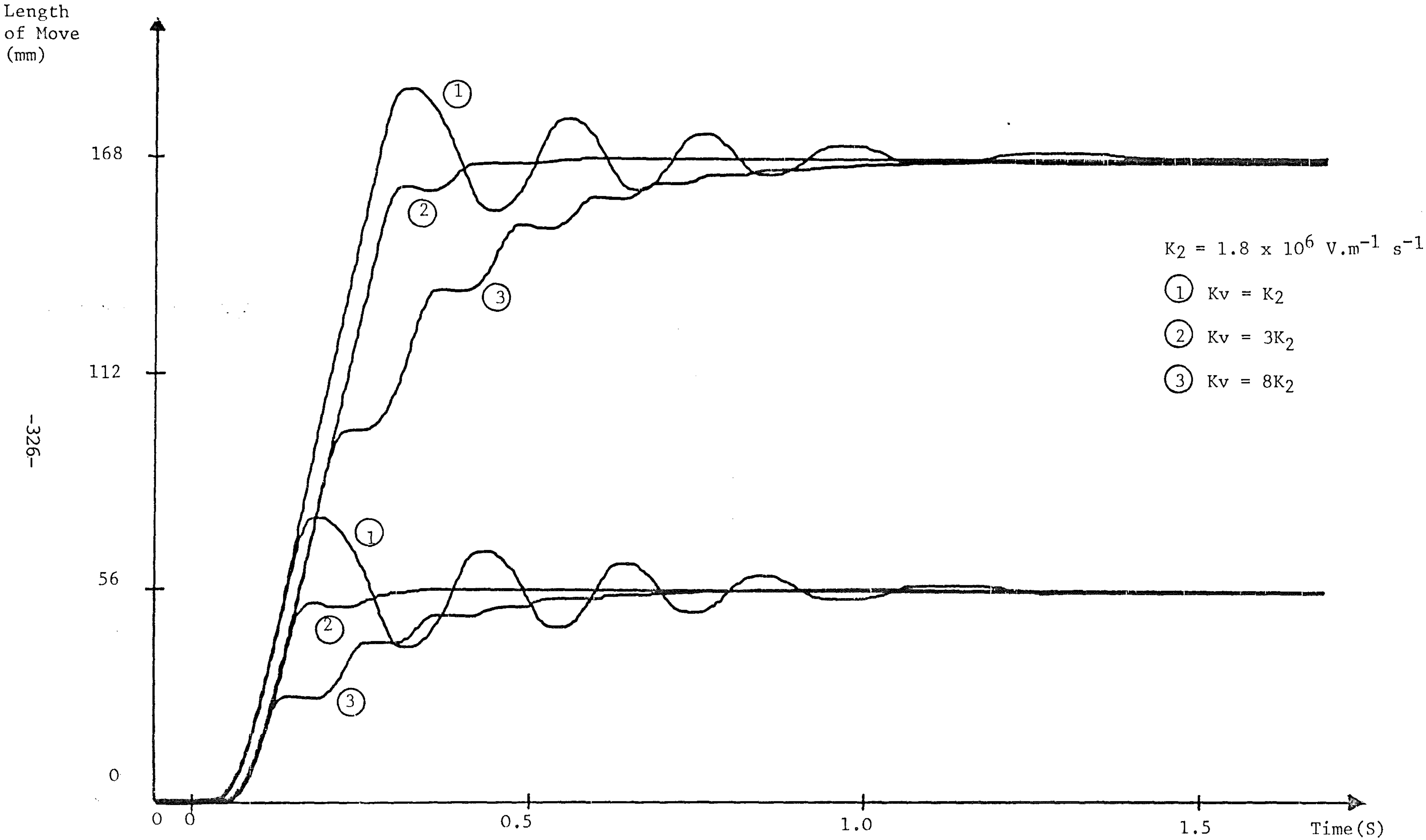


Figure (9.3.1) Transient Response
Minor Loop Compensation
Illustration of the Influence of the Velocity Loop Gain.

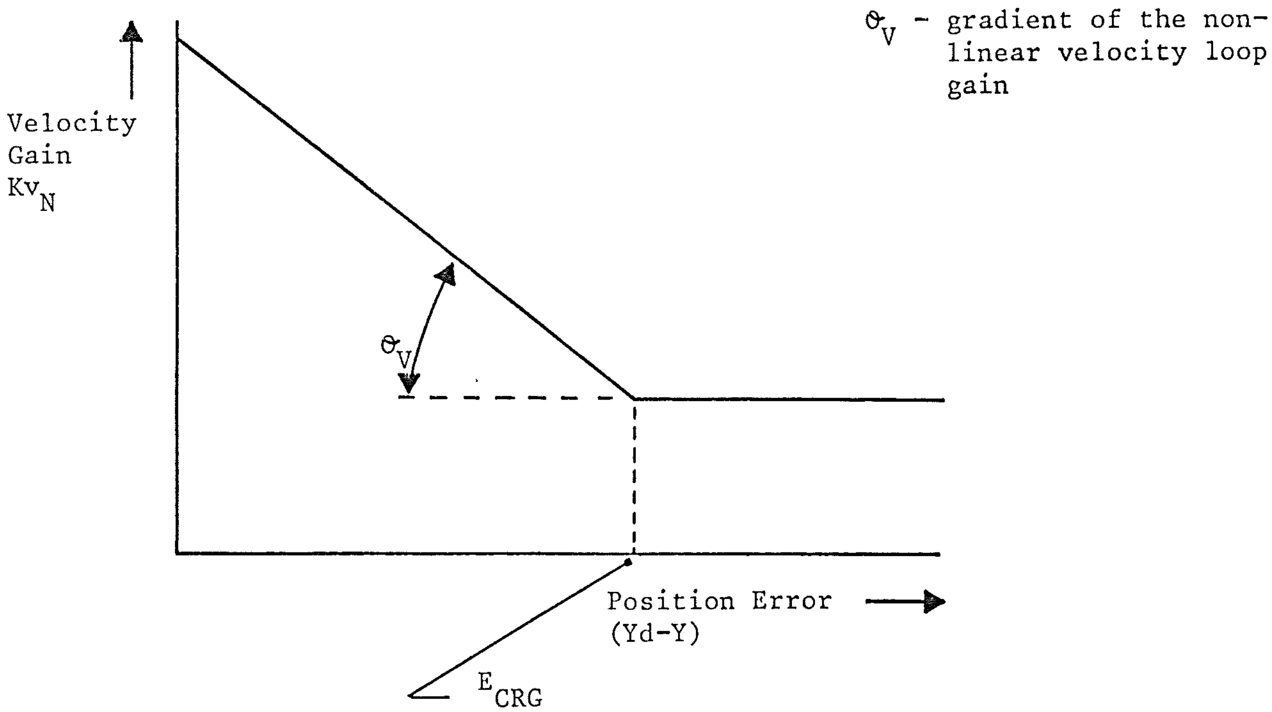


Figure (9.3.2) Non-linear Velocity Loop Gain

$Kvn = \Theta v.(Yd-Y)$

where Θv is the gradient (see Figure [9.3.2]).

For position errors greater than or equal to Ecrg the velocity loop gain takes a constant value. Use of such a non-linear relationship allows viscous damping in the system to increase as the setpoint is approached ensuring that stability is maintained but allowing a fast dynamic response.

The response of the module when using a non-linear velocity loop gain is illustrated in Figure [9.3.3]. The form of the response is similar to those when using a constant loop gain. An increase in the magnitude of the non-linear velocity loop gain results in a stabilised response, a further increase in the non-linear velocity loop gain results in an over damped response. The associated performance indices for this evaluation are listed in Table [9.3.2]. By using a non-linear velocity loop gain one can attempt to accomodate for variation in the system transfer function with load position as identified in the linearised model (Weston et al [1984]) enabling a faster dynamic response and enhanced stability to be achieved.

Use of minor loop compensation and non-linear control system gains has resulted in a significant improvement in the dynamic response of the pneumatic drive and a dramatic increase in drive stiffness (resulting in good accuracy and repeatability (Weston et al [1984, Moore et al [1985])) which has enabled such a drive system to be exploited commercially (Morgan [1985]). However, deficiencies in the dynamic response are still identifiable in terms of variable time lags (due to the influence of friction), particularly for small moves, and unacceptable overshoots with change in load. In order to improve the performance further, additional 'front-end' controls were implemented as an outer loop acting in conjunction with the realtime control strategy (Milcom version 2.5).

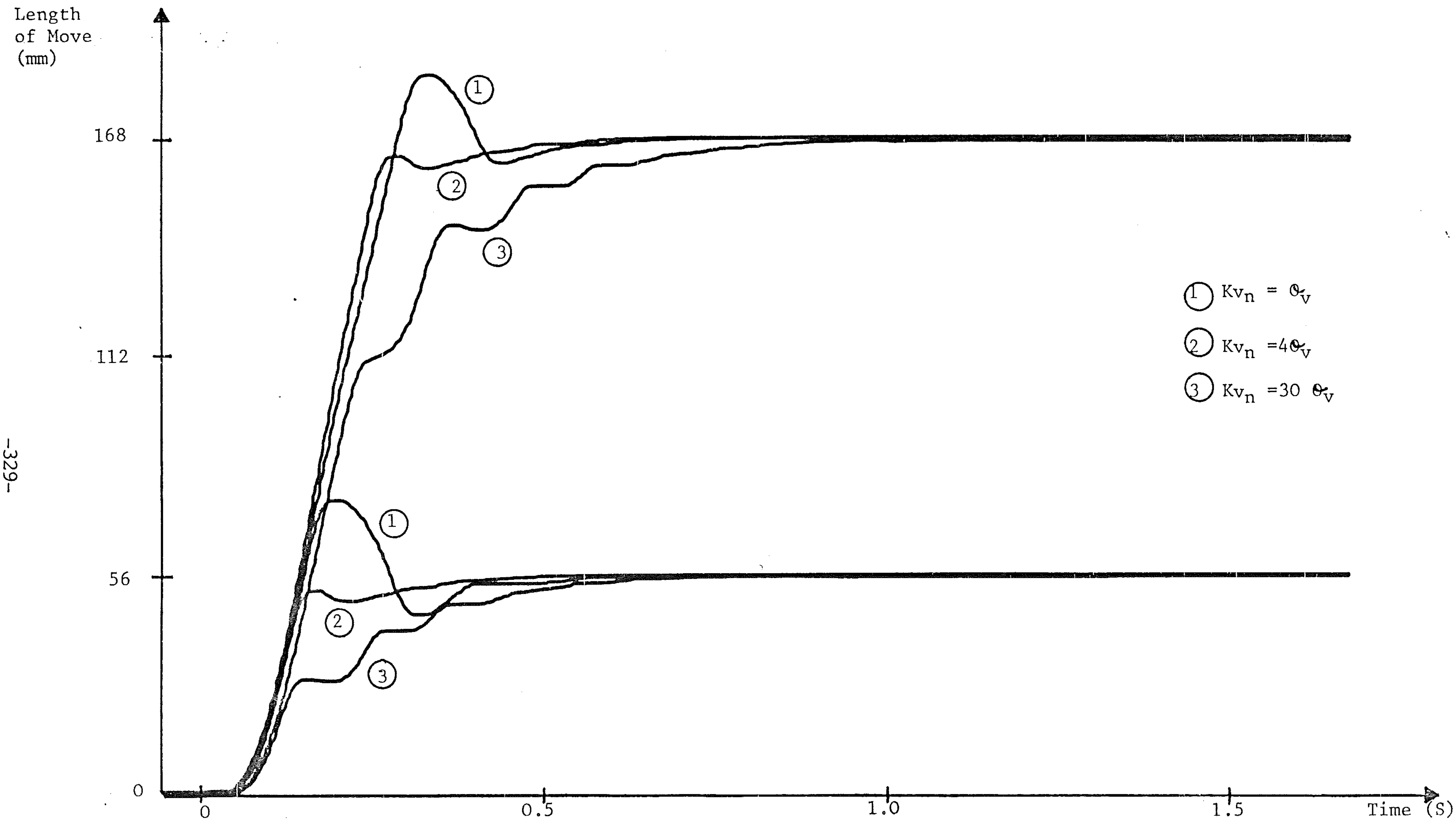


Figure (9.3.3) Transient Response

Minor Loop Compensation

Illustration of the Influence of a Non-Linear Velocity Loop Gain.

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
$Kv_n = \theta v$	499	739	800.93	3.64	70.98	56
	405	840	879.71	15.30	200.00	168
$Kv_n = 4\theta v$	365	595	774.67	2.98	94.12	56
	384	802	905.67	14.68	206.48	168
$Kv_n = 30\theta v$	561	853	551.46	4.75	65.65	56
	597	1080	827.19	19.27	155.56	168

Table (9.3.2)
Module Performance Using Minor Loop Compensation with a Non-Linear Velocity Gain (Kv_n)

Parameters	Time Lag (ms)	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
No saturation	173	1220	1314	160	19.03	43.4	56
Saturation	38	70	747	200	6.85	76.3	
No saturation	109	1276	1545	200	29.75	73.8	112
Saturation	36	976	1122	240	20.28	102.6	
No saturation	62	1289	1762	267	44.32	97.05	168
Saturation	41	1084	1273	307	34.10	134.33	

Table (9.4.1)
Performance on Applying the Saturation Control Scheme

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
Original RTC	531	872	210.00	7.56	64.20	56
Setpoint scheme	247	700	393.90	3.40	80.00	
Original RTC	625	1047	315.12	16.23	106.97	112
Setpoint scheme	341	743	630.24	9.15	150.74	
Original RTC	651	1092	485.81	24.39	153.85	168
Setpoint scheme	448	751	840.32	16.26	223.70	

Table (9.4.2)
Performance on Applying the Setpoint Modification Control Scheme

9.4. FRONT-END CONTROL SCHEMES

Several 'front-end' control schemes have been implemented during this research study aimed at improving the dynamic characteristics of pneumatic servo drives. The approach is based on an outer decision loop which modifies the command issued to an existing closed loop drive. The outer loop is designed to generate a set of initial (front-end) control conditions at the start of any 'point-to-point' move (Nagarajan and Weston [1985]). A logical decision is incorporated, external to the existing control scheme, to action an adjustment to the command signal. The command signal is modified as a logical function of the setpoint and selected features of the response.

The front-end control schemes implemented include;

(a) a saturation control scheme, and

(b) a setpoint modification scheme.

9.4.1. SATURATION CONTROL SCHEME

The saturation control scheme is applied when a new setpoint is generated and involves the issuing of a maximum (saturation condition) command signal for a short duration (t1) to overcome stiction in the actuator and load system. As soon as movement is detected the normal control scheme becomes operative. With a proportional control valve (Weston and Morgan [1984]), a saturation command will normally be identified with a maximum flow condition (ie the spool is displaced by a maximum amount relative to the valve sleeve), see Chapter (6) for a full description of the proportional valve.

The effect of applying a saturation control scheme is illustrated in

Figure [9.4.1] which shows the resulting transient responses both with and without 'saturation control'. The effect is particularly advantageous as the pneumatic time lag is significantly reduced and results in a more predictable and faster response for all categories of move. The performance indices listed in Table [9.4.1] illustrate the improved dynamic response achieved in terms of a reduced absolute error integral, increased mean positioning velocity and a reduced time lag when compared to the original control: the improvement being particularly significant for short moves.

9.4.2. SETPOINT MODIFICATION SCHEME

The setpoint modification scheme includes the saturation control scheme previously described with additional switching (or modification) stages. The use of setpoint modification for stable systems with monotone step responses has been analysed by Astrom [1980].

To summarise, after initial experimentation a scheme was implemented which involves four switching stages as follows:

Stage 1: Saturation control is applied for a period (t1) to overcome stiction.

Stage 2: A fixed control level is applied which corresponds to twice the original command signal for the desired setpoint. The duration (t2) of this stage of control is selected to correspond to the time taken for the module to reach half the distance to be traversed for any selected move.

Stage 3: A negative saturation command is applied for a

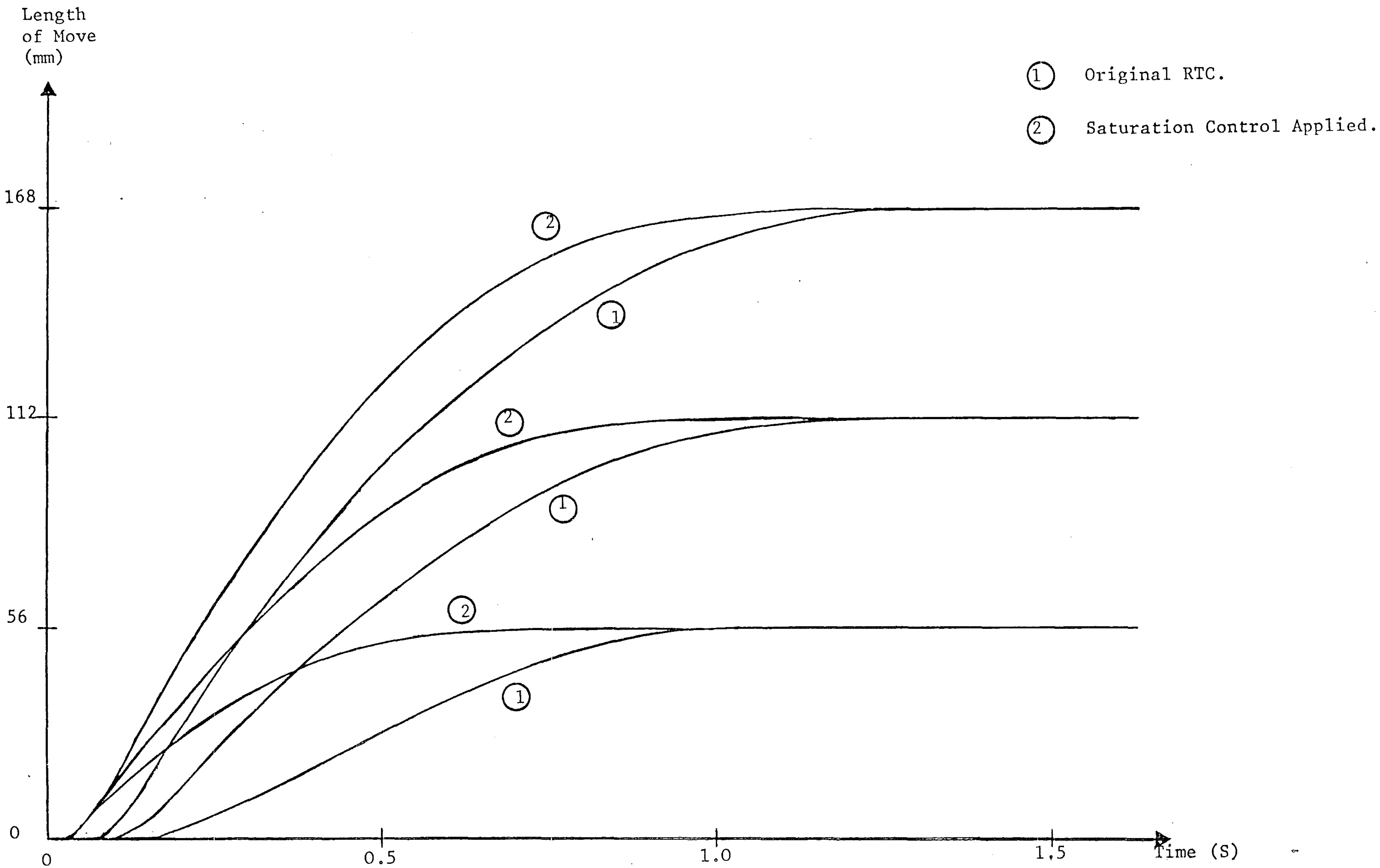


Figure (9.4.1) Transient Response
Front End Control Schemes
Comparison of Control Schemes with & without 'Saturation Control'

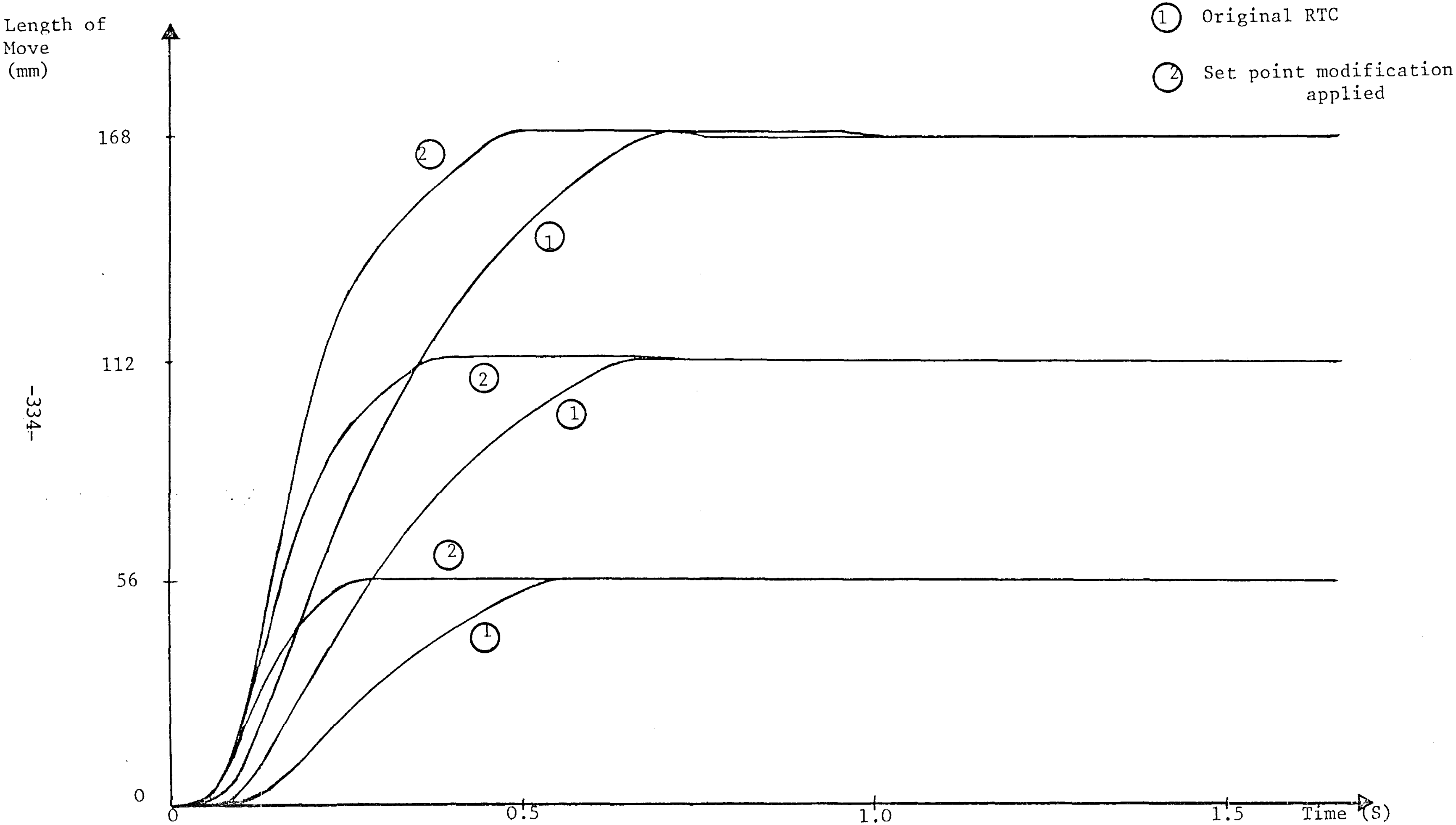


Figure (9.4.2) Transient Response
Front End Control Scheme
Comparison of Control Schemes with and without 'Setpoint Modification'

short time (t3) which decelerates the drive as it approaches its setpoint thereby reducing overshoot. Typically the duration (t3) would be selected to correspond to the time taken for the module to traverse one eighth of the desired stroke.

Stage 4: Normal realtime control is applied for the rest of the move, ie the current setpoint is used in calculating the command signal for the drive.

The benefits of using a setpoint modification scheme are illustrated in Figure [9.4.2] which shows a series of transient responses both with and without setpoint modification. These responses show that the time lag before movement commences is reduced to a relatively small reasonably constant value (as indicated previously with saturation control) and that discrete logical switching of the setpoint/command signal has resulted in a much faster response (reduced rise time) and an improvement in the overall quality of the dynamic resonse. The results are illustrated in Table [9.4.2] and show a significant reduction in the Absolute Error Integral index and corresponding increase in the Mean Positioning Velocity value when introducing setpoint modification.

The combination of saturation control and setpoint modification as a 'front-end' control scheme for pneumatic servo drives has resulted in a faster and more predictable dynamic response whilst maintaining stability and good static performance. Good static performance is assured by incorporating stage 4 switching where a well proven realtime control strategy is used in final positioning. A typical command signal (Cs), when 'front-end' control schemes are applied, for a point-to-point move is

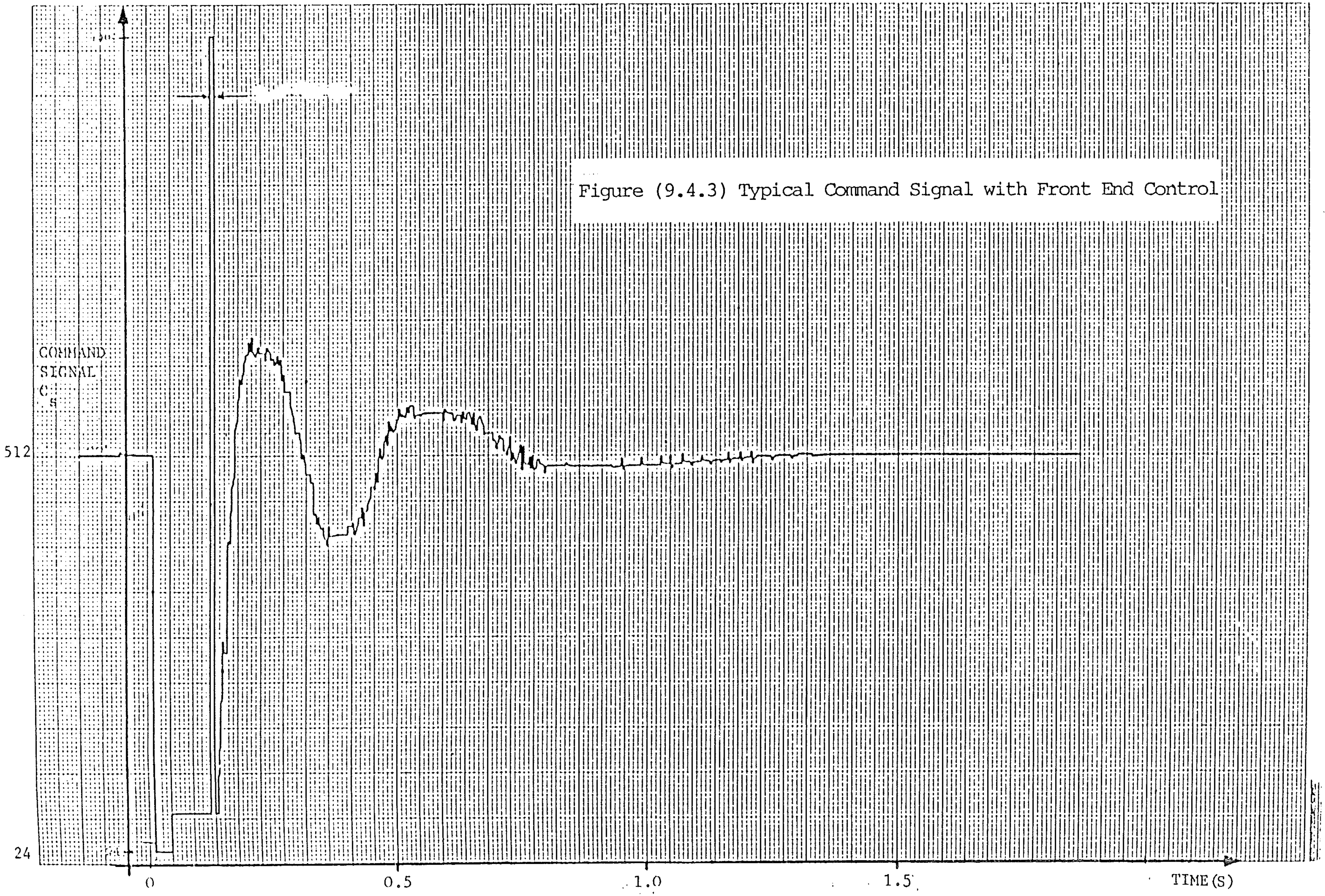


Figure (9.4.3) Typical Command Signal with Front End Control

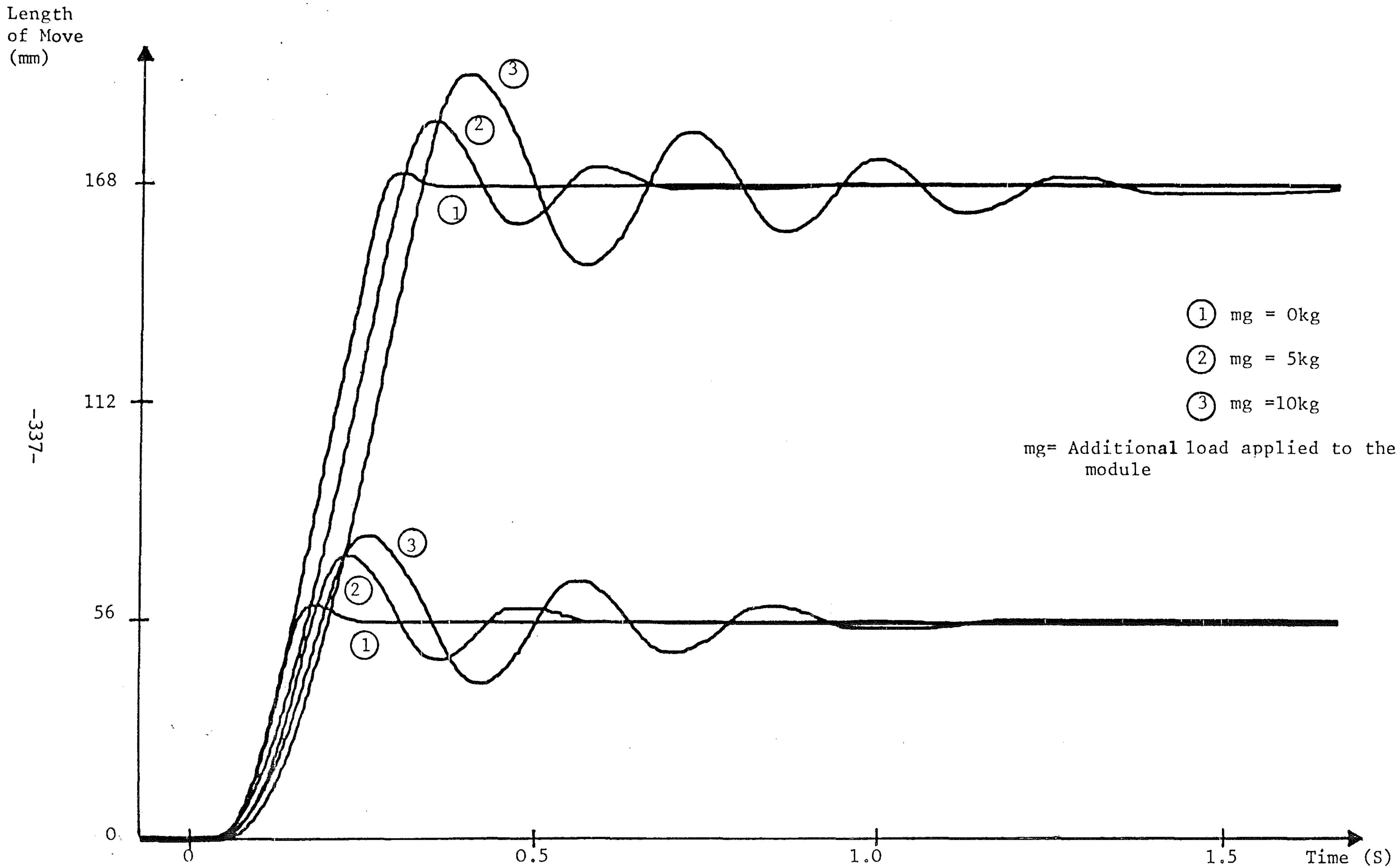


Figure (9.5.1) Transient Response
The Influence of increasing the inertial load.

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
Mg = 0 Kg	213	582	774.70	2.90	96.22	56
	314	653	800.90	15.48	257.27	168
Mg = 5 Kg	435	1080	682.76	4.00	51.85	56
	525	896	840.32	17.05	187.50	168
Mg = 10 Kg	1130	1450	656.50	5.53	38.62	56
	1080	1230	853.50	22.04	136.59	168

Table (9.5.1)
Performance on Increasing the Loading

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
$Kp = K_3$	834	1330	512.07	5.28	42.11	56
	680	1420	787.80	20.92	118.31	168
$Kp = \frac{1}{4}K_3$	196	580	643.37	3.42	96.55	56
	324	742	905.97	17.42	226.42	168
$Kp = \frac{1}{10}K_3$	1210	973	722.12	5.22	57.55	56
	582	953	879.71	19.17	176.29	168

Table (9.5.2)
Performance on Increasing the Position Loop Gain when Loaded with 5 Kg

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	AEI ($mmx10^3$)	MPV (mms^{-1})	Length of Move (mm)
$Kv = K_4$	1190	1300	709.02	6.97	43.08	56
	943	1480	853.45	21.90	113.51	168
$Kv = 3K_4$	215	543	682.76	3.42	103.13	56
	313	651	919.10	17.32	182.79	168
$Kv = 6K_4$	904	1030	420.16	6.39	54.37	56
	834	1070	814.06	23.27	157.01	168

Table (9.5.3)
Performance on Increasing the Velocity Loop Gain when Loaded with 5 Kg

illustrated in Figure [9.4.3].

9.5. THE EFFECT OF LOADING

It is important to consider the influence of load on the dynamic response of a pnematic drive and the effectiveness of any control strategy will necessarily depend upon an ability to deal with the load disturbances which can occur in a given application.

To evaluate the effect of inertial loading on dynamic response, several experiments were conducted as outlined below.

(i) The effect on the dynamic response of increasing the inertial load: the control parameters originally being selected to give a high quality dynamic response with no additional inertial load (the intrinsic unloaded moving mass of the module tested being 3.5kg). The transient response was then studied using standard test moves with various inertial loads added. The results are illustrated in Figure [9.5.1] which shows a series of transient responses obtained, while Table [9.5.1] lists the measured performance indices.

(ii) The effect on the dynamic response of changing the position loop gain: the module was loaded with 5kg and the transient response analysed for various values of position loop gain (Kp). The transient responses obtained are illustrated in Figure [9.5.2], while Table [9.5.2] lists the measured performance indices (K3 = 1428 volts/m).

(iii) The effect on the dynamic response of changing the velocity loop gain: the module was loaded with 5kg and the transient responses analysed for various values of the velocity loop gain (Kv). The transient responses obtained are illustrated in Figure [9.5.3], while Table [9.5.3] lists the measured performance indices.

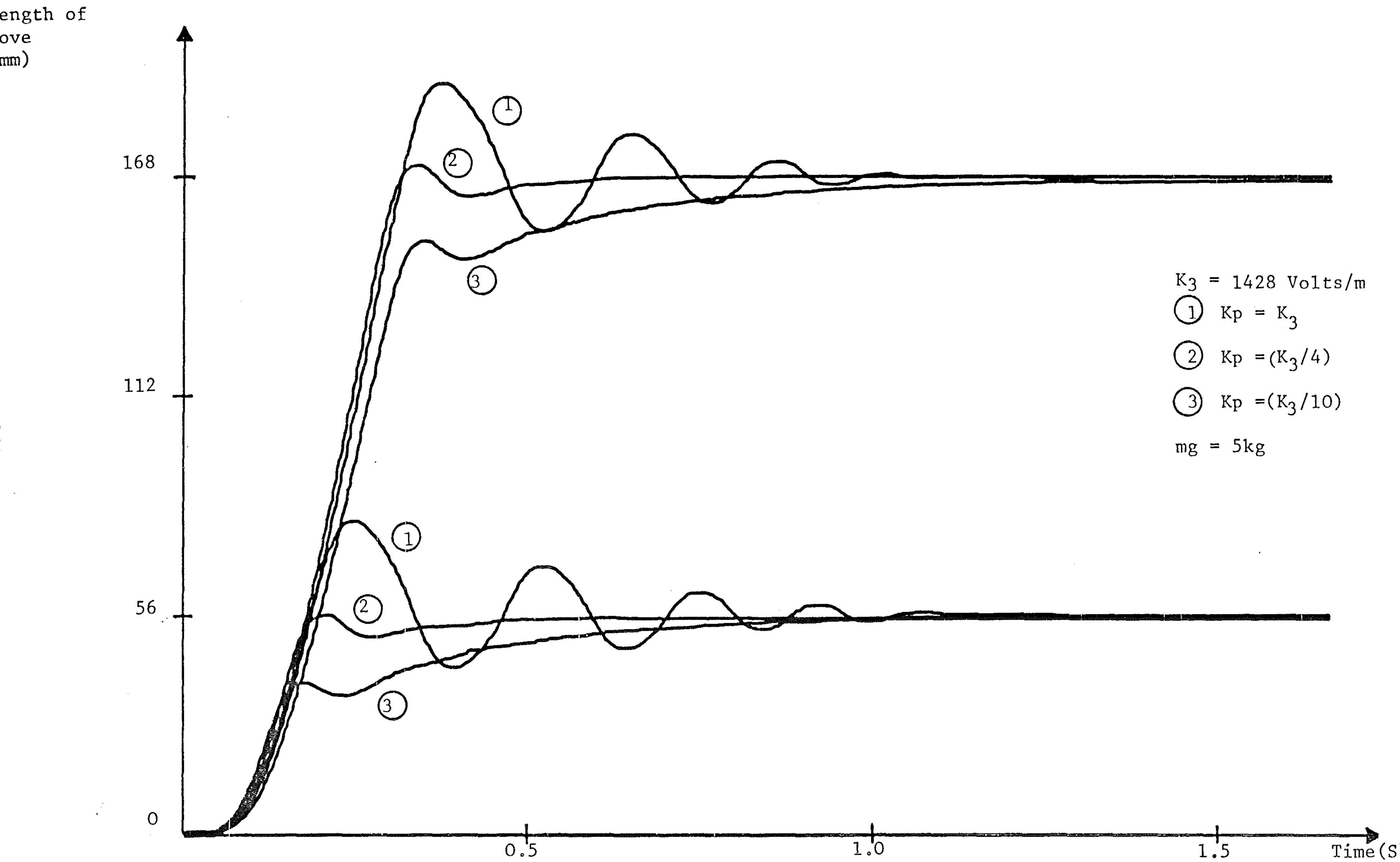


Figure (9.5.2) Transient Response
Influence of changing the Position Loop Gain (Kp) when the Module is loaded.

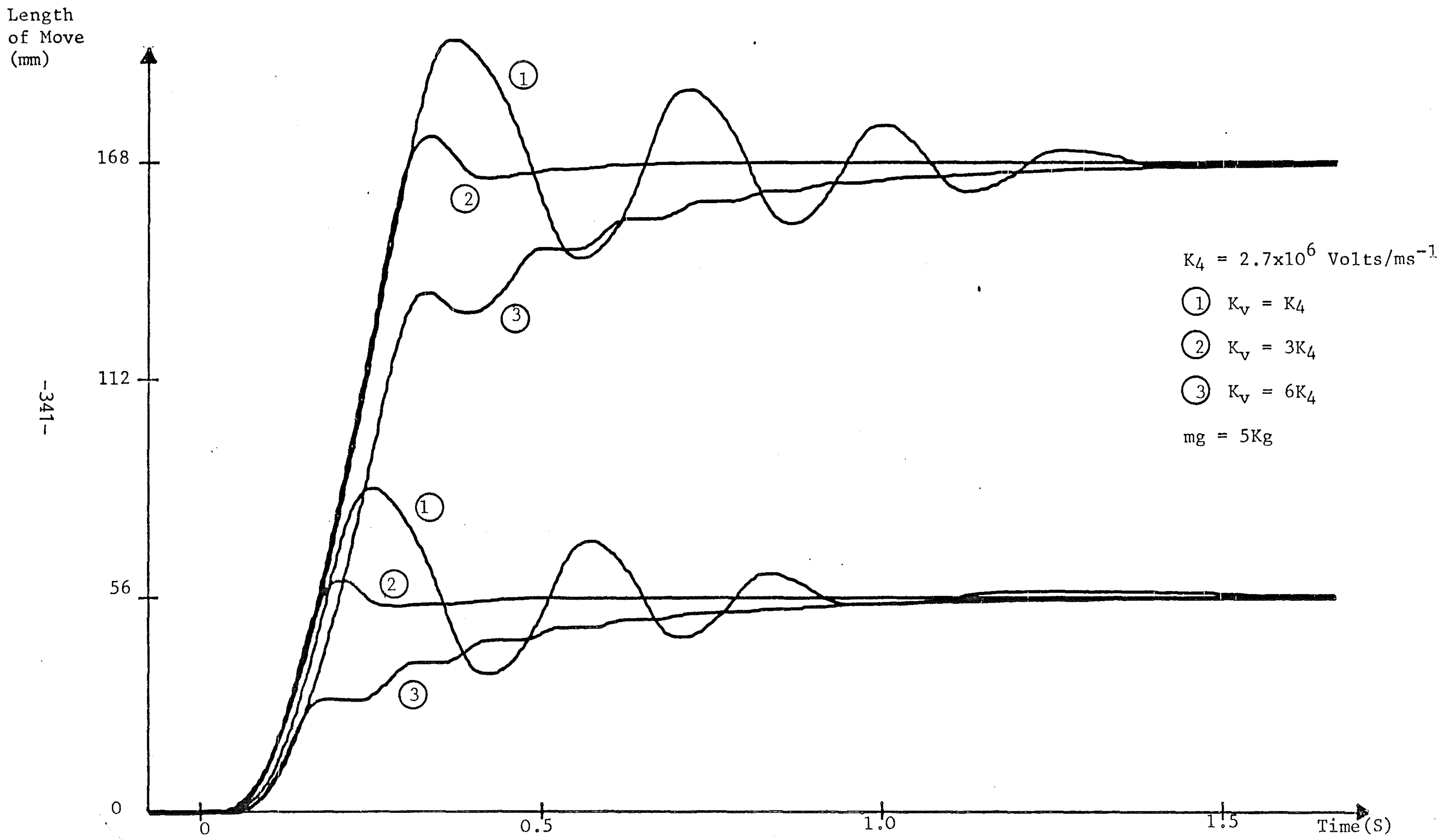


Figure (9.5.3) Transient Response
Influence of changing the Velocity Loop Gain (K_v) when the Module is loaded.

The velocity loop gain $K4 = 2.7 \times 10^{6}$ volts/ms^{-1}

9.5.1. ASSESSMENT OF THE EFFECT ON THE DYNAMIC RESPONSE OF LOADING

The effect of increasing the inertial load on a pneumatic drive is to increase the rise time and overshoot of the response (see Figure [9.5.1]). By correctly selecting control system gains the overshoot can be reduced to a specified level but this will result in a slower response (see Figures [9.5.2] and [9.5.3]). In many industrial applications the variations in payload will be relatively small and the control system gains can be selected to give an acceptable dynamic response for a 'known load' condition. In application areas where a wide variation in payload is to be encountered a 'range' of control system gains may be necessary to produce acceptable dynamic responses.

An alternative approach to the problem is to use a learning scheme (Nagarajan and Weston [1985]) whereby the pneumatic drive adapts its control parameters with respect to an evaluation of its own performance. The learning procedure can continue until it has produced an optimised parameter set for each move and associated payload thereby achieving a dynamic response of specified quality.

9.6. LEARNING SCHEMES

Simple learning schemes have been implemented in software which allow a specifed control parameter to be adapted dependant upon a measured performance. Such an adaptive mechanism is active until a specified quality of response is achieved. A learning scheme of this type can be used as part of an 'initialisation' phase to produce an optimised parameter set to simplify any pre-installation tuning of a module, alternatively the learning scheme can be maintained continuously 'on-line' to rectify any drift/degradation in performance. Such a control scheme is illustrated in Chapter (8).

In this evaluation, to simplify learning and reduce associated realtime processing, only a single parameter is adapted: the chosen parameter being the reverse saturation time (t3) in the setpoint modification scheme. By adapting the time (t3) the damping can be adjusted until a satisfactory response is achieved.

The result of applying a learning scheme of this type is illustrated in Figure [9.6.1] which shows the transient responses from a number of trials after 1, 3 and 6 learning sequences respectively. Here an optimised set of control parameters were chosen with the module unloaded. Subsequently, an additional inertial load of 5kg was applied which resulted in under damped responses before the learning scheme was included. Thus as the selected parameter is adapted, overshoot is reduced to within a predetermined level: the associated performance indices are given in Table [9.6.1] which shows how the dynamic responses have been improved.

Various forms of learning schemes are currently being investigated and methods of evaluating and describing the necessary dynamic response are being studied. Performance indices such as 'absolute error integral' or 'absolute error time integral' can be used to obtain a better quantitive measure of the dynamic response thereby allowing improved learning procedures to be devised.

9.7. CONCLUSIONS

It has been demonstrated in this research study that the dynamic response characteristics of a pneumatic motion control system can be dramatically

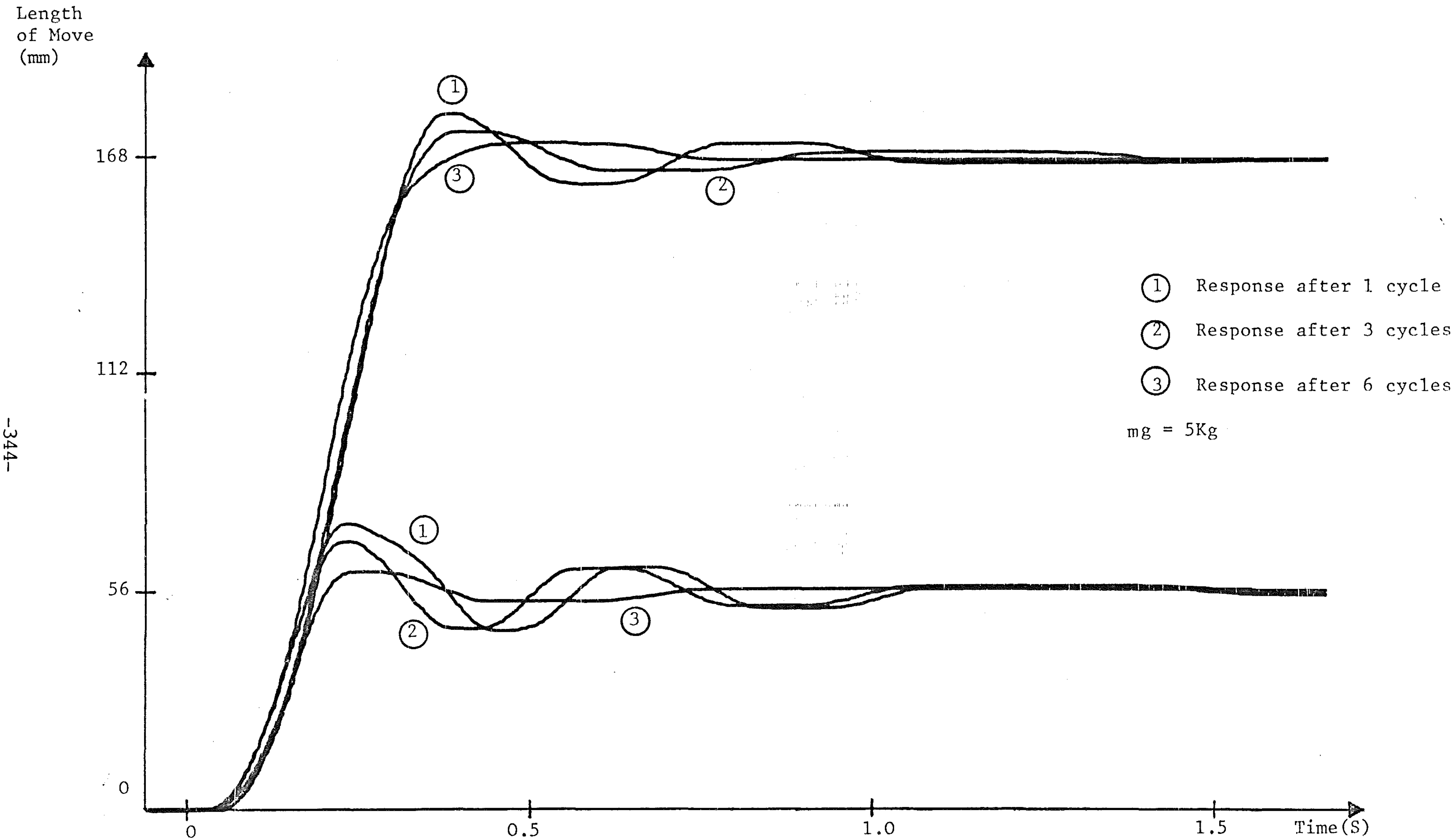


Figure (9.6.1) Transient Response
Learning Schemes
Illustration of the influence of learning on the response

Parameters	t_s (ms)	t_p (ms)	V_{max} (mms^{-1})	MPV (mms^{-1})	Percentage Overshoot
168 mm Sequence 1	1410	642	18.6	119.1	8.1
Sequence 3	1170	354	17.76	143.6	6.2
Sequence 6	972	343	16.23	172.8	1.5
56 mm Sequence 1	1570	1030	4.53	35.7	31.6
Sequence 3	1370	968	4.04	40.9	28.2
Sequence 6	740	608	3.39	75.7	6

Module Performance on Applying the Learning Scheme

Table (9.6.1)

improved by applying appropriate software based motion control strategies.

Application of 'minor loop compensation' and 'non-linear' loop gains allows the stiffness of a pneumatic drive to be significantly increased whilst maintaining stability. This enables high accuracy and repeatability to be achieved coupled with satisfactory dynamic characteristics.

Results of tabulated indices for various control strategies have been presented together with transient responses. To illustrate further the performance improvements achieved by incorporating digital compensation it is instructive to consider at one extreme 'proportional error' control and at the other 'minor loop compensation with non-linear loop gains and set point modification' control. For 'short' moves, when using a proportional error control scheme, the static performance can be characterised by positioning errors of 3.4mm or better while typical dynamic performance characteristics are critical damping with a positioning time of 1.4 seconds: such an arrangement will have very limited application in manufacturing industry. However, by including minor loop compensation with non-linear loop gains and set point modification the corresponding static positioning errors are 0.2mm or better with critically damped responses demonstrating positioning times of less than 0.6 seconds. Furthermore, load disturbance effects can be improved by incorporating learning schemes. The dramatic performance improvements involved can make the use of computer controlled pneumatic servos in manufacturing very attractive with cost performance ratios which match or better alternative drive technologies.

CHAPTER 10

COMMERCIAL EXPLOITATION OF PNEUMATIC MOTION CONTROL

10. INTRODUCTION

This Chapter describes the commercial exploitation that has resulted from the successful evolution of pneumatic motion control systems in this research study. Pneumatic motion control is not as yet generally accepted as an alternative drive system within industry. However the commercial availability of proprietary pneumatic motion control systems will further enhance the acceptance of such technology in appropriate applications (Morgan [1985]).

The commercial exploitation of pneumatic motion control can be considered within categories which include;

(a) robot module drives,

(b) "intelligent" actuators,

(c) special machinery drives,

(d) single axis controllers, and

(e) distributed control systems.

To achieve commercial exploitation it has been necessary to package a "pneumatic motion control" system in a way that ensures;

(i) cost advantages are achieved,

(ii) easy integration with other manufacturing equipment and systems,

(iii) "user friendly" set up and operational facilities are provided,

(iv) good performance characteristics can be achieved with a range of "intelligent" actuators and module types with appropriate "tuning", and

(v) pneumatic motion control systems can be used in "standalone" applications, or integrated within larger automation schemes with distributed control where supervisory control facilities are provided by either proprietary PLC's or computer systems.

The control system architecture adopted for the commercial exploitation of pneumatic motion control was the result of a parallel research study at Loughborough (Thatcher et al [1983], [1986]). Specific consideration was given to maintaining hardware and software modularity within the overall control architecture, and allowing easy integration of the system into the manufacturing environment (Weston et al [1983], [1984]).

The potential of any motion control system can only be realised when it is complimented by mechanical hardware of appropriate design, and a control system architecture which exploits its features to the best advantage. The design of the software for supervisory control functions (Thatcher et al [1983], Thatcher et al [1984]) and the "local motion control" (Thatcher et al [1985]) has been the subject of a concentrated research study at the University.

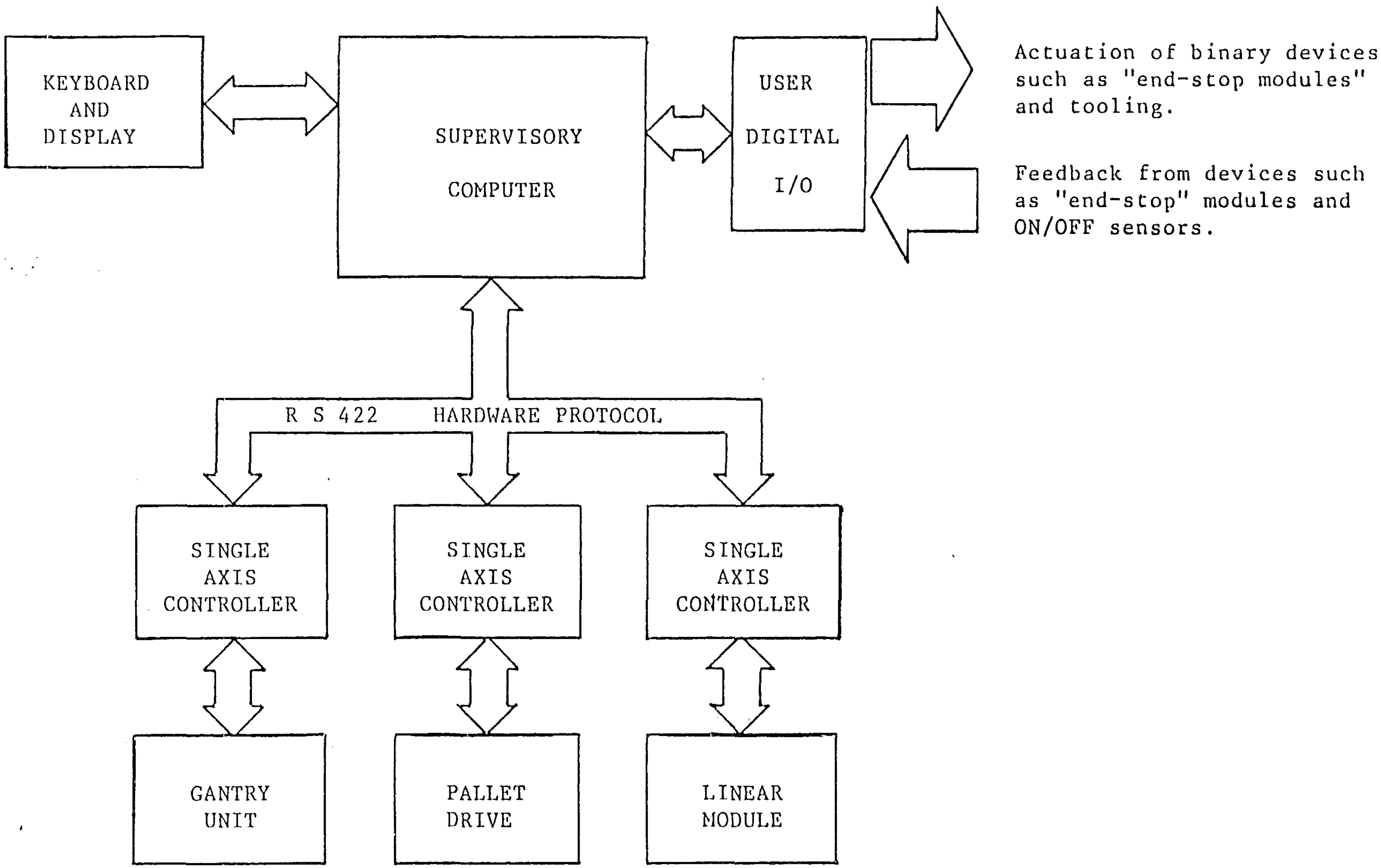
The commercial availability of new families of robot modules, "intelligent" actuators and associated control system modules will accelerate the introduction of distributed automation schemes within manufacturing industries.

10.1. CONTROL SYSTEM ARCHITECTURE

A distributed multi-processor control system architecture has been adopted to compliment the "modular" design philosophy in the mechanical elements (see Figure [10.1.1]).

The addition of servo-controlled modules and "intelligent" actuators will

Figure (10.1.1) Distributed Control Architecture



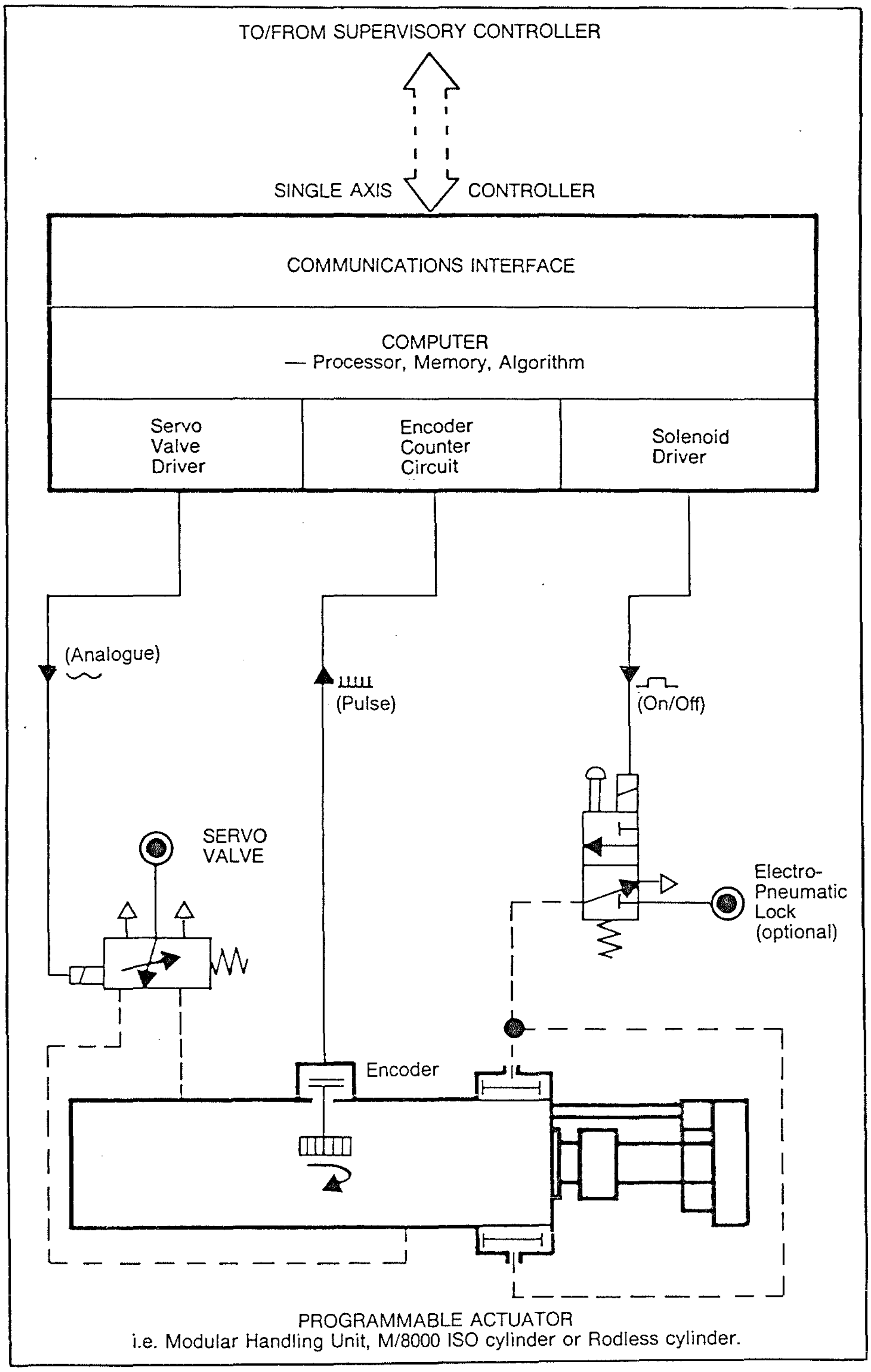
allow industrial users to configure manipulators and machinery which incorporate appropriate combinations of servo-controlled and end-stop modules as dictated by the requirements of each application area. The number of modules/actuators and their individual features will be application dependant. A manipulator comprised of only end-stop modules can be conveniently controlled by a conventional programmable logic controller (PLC). However, in many applications, especially those involving frequent re-programming, manipulators/machinery which comprise one or more servo-controlled axes cannot easily be controlled in this way and require a control system offering "user friendly" programming and sequence control facilities which are commonly provided in the control systems of conventional industrial robots.

The particular need for a supervisory control system which may be required to control at one extreme a single servo-controlled module or "intelligent" actuator and at the other extreme a 'distributed manipulator' comprising more than one multi-axis group of servo-controlled modules (in which the groups must operate in a co-ordinated manner to achieve a given manufacturing function) has led to the adoption of the distributed control architecture as shown in Figure [10.1.1].

Each servo-controlled module is assigned a local processor or "single axis controller" (Thatcher et al [1985]) which is responsible for the realtime control of the module in response to position commands as supplied by the "supervisor". Figure [10.1.2] illustrates the functions of a single axis controller when interfaced to a programmable positioning unit.

A distributed control architecture results in certain significant advantages that can be gained as follows:

(i) even when large numbers of servo-controlled modules are required, Figure (10.1.2) Functional Elements of the 'Single Axis Controller' within a System



sufficient processing power is available to execute the realtime control functions necessary to achieve satisfactory drive performance. An approach of using a single processor to implement both supervisory and servo-control functions would severely limit the maximum permissible number of servo-controlled axes;

(ii) modularity is conserved and the need for redundancy is reduced as it is not necessary to use a high cost and powerful computer controller capable of controlling a large number of servo-controlled modules when in many application areas only one or two such modules may be required;

(iii) an elegant division of software functions is achieved between

(a) fast realtime control strategies which can conveniently be assembler derived to maximise execution speed and minimise hardware costs,

and (b) supervisory software can be conveniently implemented in a high level language with advantages of transportability, reduced system engineering costs and improved documentation; and

(iv) division of responsibility when designing and configuring the automation scheme.

10.2. SINGLE AXIS CONTROLLERS

A microprocessor based controller known as a "single axis controller" (SAC) is now available in commercial form which utilises the motion control system described within this thesis (Milcom version 2.60).

A servo-controlled axis is known commercially as a "programmable positioning unit". The basic operation of a unit interfaced to a SAC is shown in Figure [10.1.2]. Clearly, for a programmable positioning unit to have industrial applicability it is essential that the "single axis controller" can be easily interfaced to existing and new manufacturing plant in such a way that system controllers or supervisors (PLC's, computers, etc) can readily exchange data with the SAC (Thatcher et al [1985]). For this reason the SAC design incorporates several types of interface.

The SAC microprocessor hardware, shown in Figure [10.2.1] and Plate [10.2.1], has been developed into a full commercial product by Nutek(London)Ltd from a design based on a Loughborough University prototype. The SAC incorporates a TI 9995 microprocessor, which is a device which is software compatible with the TI 9900 processor (which was used for the prototype evaluation system) and this facilitated the transfer of motion control software to the production SAC's. The TI 9995 has an external 8 bit data bus, its internal 16 bit architecture and other pipe lining features ensured sufficient speed of operation to execute the realtime control strategy. The 40 pin package of the TI 9995 and considerable on-board features facilitated an inexpensive production implementation.

The production version of the SAC hardware consists of two half euro-sized printed circuit boards. The first board contains the CPU, memory, supervisory computer/teach pendant interface and incremental encoder interface. On the second resides the proportional amplifier circuit, PLC interface and the 'home' sensor interfaces. A backplane connects these boards. The power supply resides in half single euro-size and provides sufficient power for up to three SAC's, so that it is possible to mount up to three SAC's and PSU in a single 19 inch rack.

In operation each SAC is required to move its associated

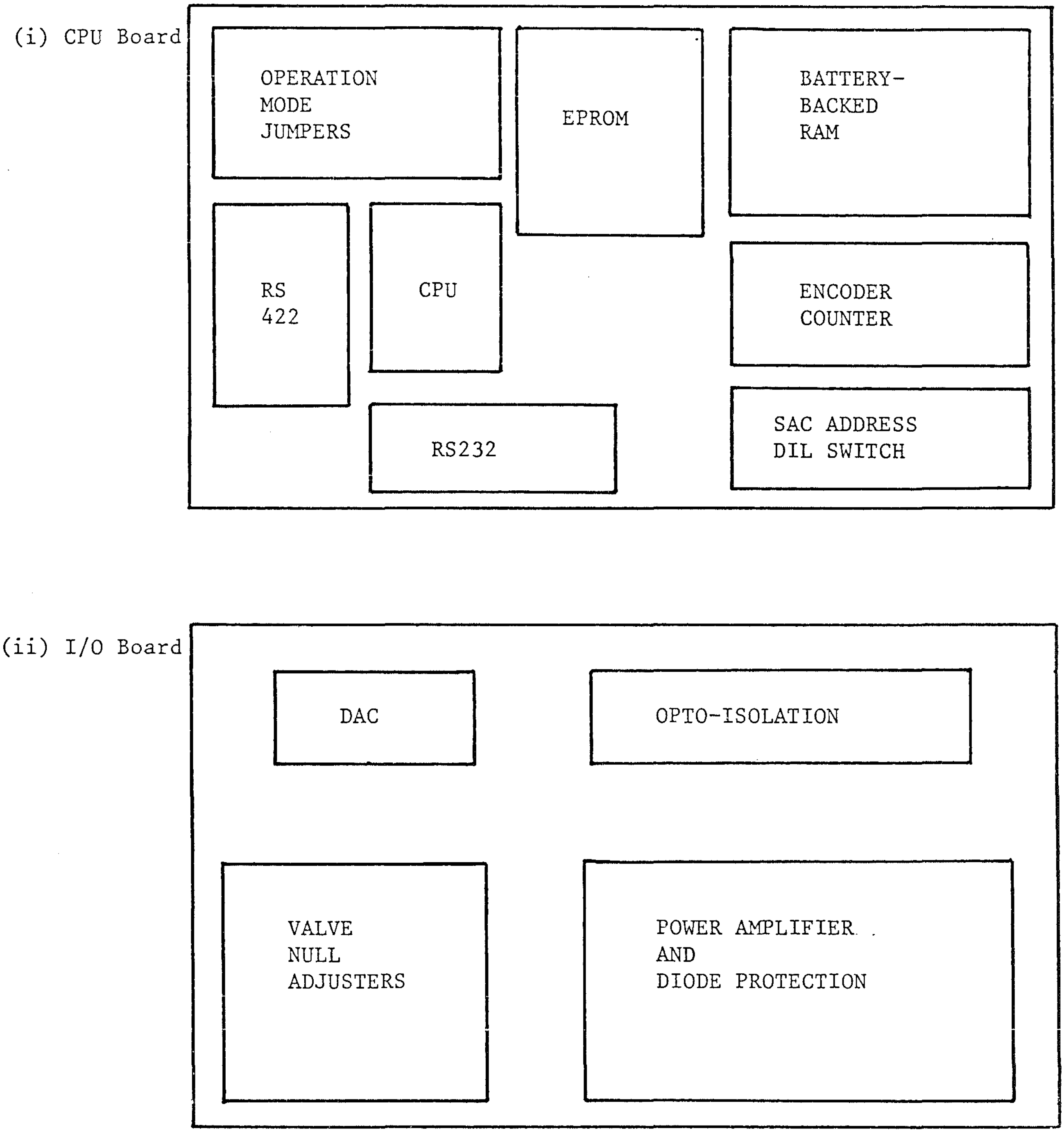


Figure (10.2.1) Single Axis Controller
Board Configuration

module/intelligent actuator to a series of setpoints. The setpoints may be stored locally in the SAC's battery backed RAM or remotely in the supervisory controller. The setpoints can be acquired using either one of two methods. If a hand held terminal is connected to the RS 232 port and teach/set mode selected the hand held terminal can be used as a teach pendant.

The module can be driven in either direction at variable speed until the desired position is achieved, at which point the SAC is instructed to store the desired position as a setpoint. Alternatively numerical values corresponding to desired setpoint positions can be keyed into the SAC's memory. When running, the setpoints can be selected in any order by the PLC or intelligent controller. An intelligent controller can also obtain the setpoints from the SAC for editing, storage, monitoring and subsequent down loading.

The second method of providing setpoints involves the use of an intelligent supervisory controller attached to the RS 232/422 port, from which commands to the SAC requiring it to move its module in either direction at variable speed can be transmitted. On reaching the desired position, the supervisor may store the setpoint or the SAC can be commanded to store the setpoint locally.

Each SAC has two interfaces, see Figure [10.2.2], from which supervisory controllers can control the actions of the SAC and its associated module. The PLC interface comprises a parallel "bus" and associated software drivers which facilitate direct connection to PLC's and similar devices by implementing a set of communication functions.

The serial interface provides both RS232 or RS422 communication data links (RS422 being appropriate for systems which require more than one

Figure (10.2.2) Single Axis Controller Interfaces

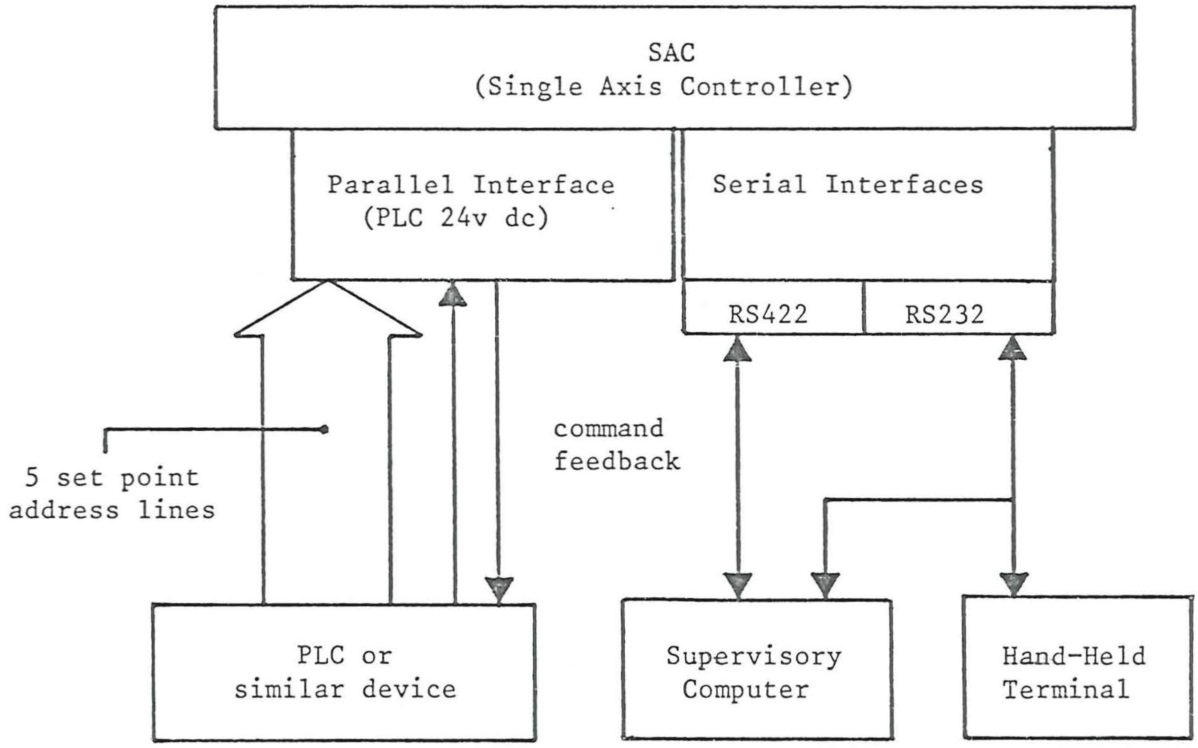
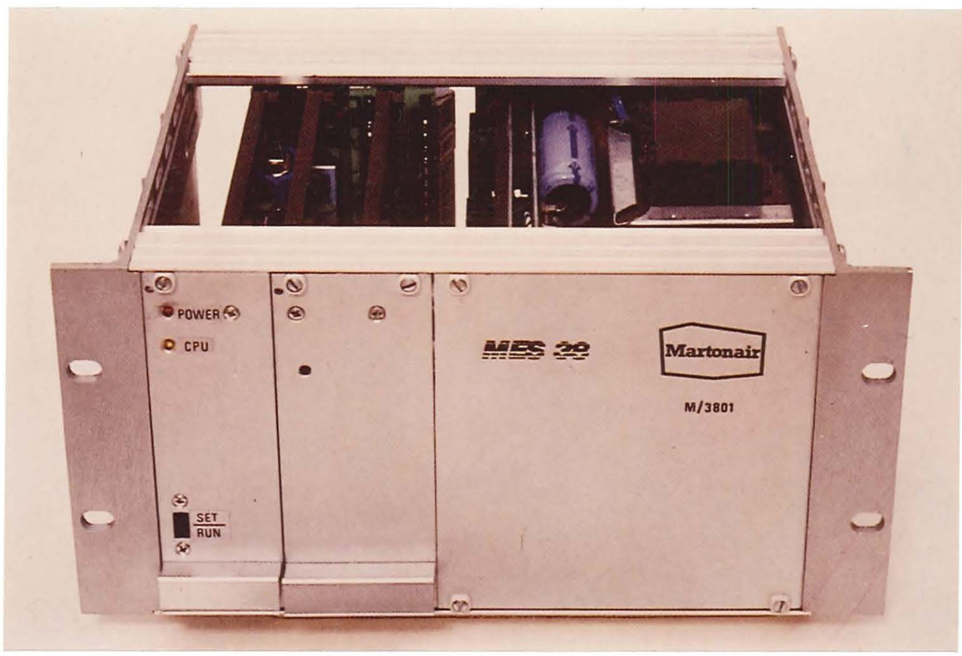


Plate (10.2.1) Single Axis Controller



pneumatic servo-drive). The same software protocol is used for both versions. Devices using the serial interface follow a master/slave relationship, where supervisors send messages to the SAC using a message format of fixed length.

A unique address is set on each SAC with an address DIL switch so that each SAC can determine if the message is for itself or another SAC. Many SAC instructions exist, examples are those which command a SAC to load or return a setpoint value, move to a setpoint, change or read a control parameter, etc..(Thatcher et al [1985]).

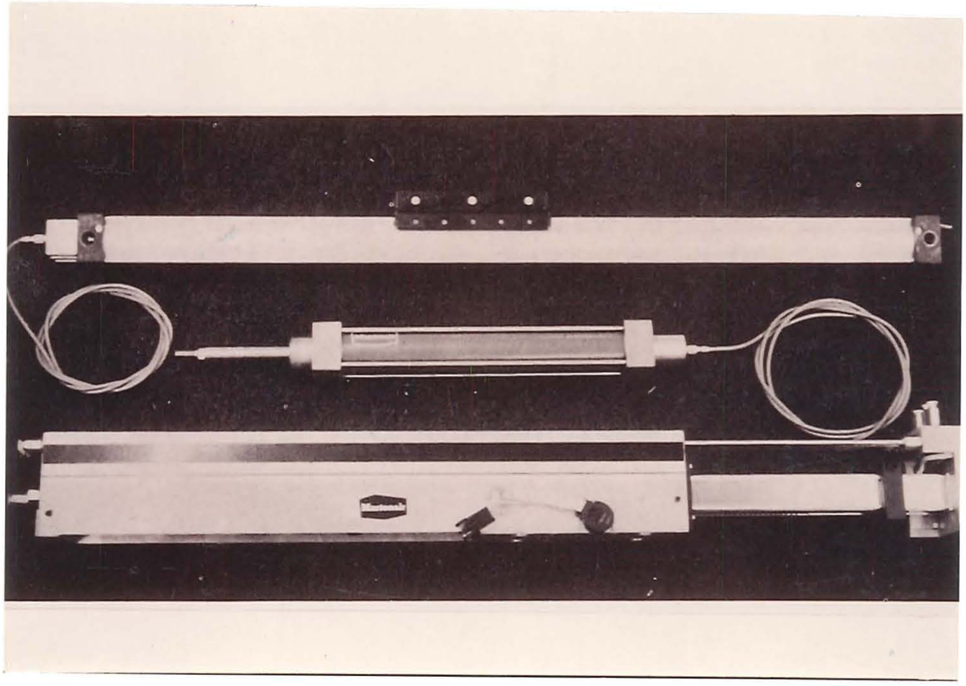
When the supervisor requires data from a SAC (eg SAC status, error conditions, parameter values), it issues a command to request that information and then awaits the reply.

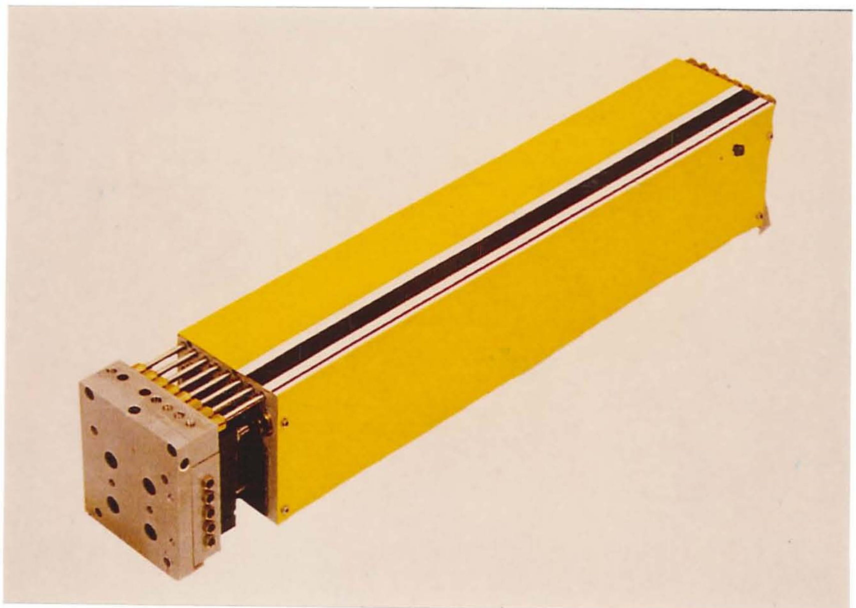
The software design incorporated in the SAC includes two interrupt service routines and a background scanning task. The highest priority interrupt eminates from the TI 9995's on-board timer. The service routine to this interrupt constitutes the motion control software. The background scanning task monitors the operation mode status which informs the SAC as to which of the interfaces is currently active and whether the Teach/Set or Run mode has been selected.

10.3. PROGRAMMABLE ACTUATORS

The pneumatic motion control system described in this thesis is being marketed by Martonair Ltd in a number of forms. Much of the early Martonair product development work and the corresonding LUT research and evaluation effort has concentrated on programmable variants of the M/60000 series of linear handling modules (Martonair [1985], of which the M/60110/600 was used for the latter series of performance evaluations), since often in

Plate (10.3.1) Programmable Actuators





applications requiring programmable positioning it is appropriate to utilise off the shelf mechanical modules which can easily be included within the machine structure involved. However, other generic forms of actuator have been considered and LUT and Martonair have evaluated versions of both the M/8000 ISO standard cylinder and the M/45000 rodless cylinders in which integral position sensors have been fitted. A standard actuator with the addition of a position sensor and axis motion controller is termed an "intelligent actuator"; the approach has been applied to both linear and rotary devices. The position sensor used in these actuators is an incremental rotary encoder, the encoder is physically small and is neatly embodied in the actuators. In the M60000 modules the encoder is driven by a pinion engaging with a rubber drive belt which is attached to the module slideway. The two cylinders referred to use an integral nut and lead screw arrangement to drive the encoder which is set in a small housing mounted on the end cover. SAC's incorporating the Milcom motion control software can be used to control "intelligent actuators" which should find application as cost effective drive elements in "special purpose machinery". The range of actuators and modules to which servo-control can be utilised illustrates the robustness of the control strategy. Linear and rotary "intelligent actuators" have been evaluated at LUT.

10.4. APPLICATION OF PNEUMATIC MOTION CONTROL

A study of the potential applications for pneumatic motion control in the form of modular robots/work handling systems and "intelligent actuators" has been conducted both at the University (Harrison et al [1985]) and at Martonair (Morgan [1984]), aimed at achieving the following objectives:

(i) to categorise potential application areas for the family of Martonair/LUT pneumatic motion controllers used in combination with both standard modules and "intelligent actuators",

(ii) to evaluate the systems engineering requirements when configuring systems,

(iii) to assess the behaviour and capabilities of servo-controlled pneumatic modules under representative conditions,

(iv) to investigate, for a range of applications, the requirements for sequencing and programming facilities, and

(v) to improve the effectiveness of future research into modular motion control systems by more accurately establishing and categorising the requirements of industrial applications.

Some fifty companies were contacted who returned questionnaires concerning the application of modular robots and "intelligent actuators". Twelve of these companies were willing to participate in detailed application studies where potential application areas were identified and classified.

The LUT application studies have been complemented by a parallel Martonair activity which has seen the introduction of module/SAC combinations and "intelligent actuator"/SAC combinations in a range of industrial applications (Weston et al [1984]).

Some example application classifications of pneumatic motion control include;

(a) work handling where modules are used to configure stand-alone manipulators, for picking and placing components,

(b) "customised automation" where the modules and "intelligent actuators" can be considered as "higher order building blocks" which OEM's and system builders can use to adapt standard machinery. An example of these is a programmable bar feed unit (utilising a standard module) which has been fitted to a saw machine used in the preparation of billets from bar stock,

(c) "special purpose machines", constructed from pneumatic robot modules, "intelligent" actuators and associated control modules, such as palletising units, which allow a component to be picked up or set down at any location within a two-dimensional array, and

(d) assembly systems, where distributed manipulators (constructed from modules and/or "intelligent actuators") can exhibit a number of advantages (including higher operational speeds, concurrent motions and reduced costs) when compared with conventional pedestal mounted robots.

10.5. CONCLUSIONS

This chapter has described the implementation and commercial exploitation of pneumatic motion control resulting from the research study described in this thesis. The commercial availability of pneumatic motion control systems will advance the application of such technology within industry. Since the marketing introduction of pneumatic motion control systems by Martonair in April 1985 a small number of competitive systems have started to appear as the potential for commercial exploitation is recognised; these include systems from

(i) Robert Bosch have introduced a programmable pneumatic drive using binary (on/off) control valving and a brake mechanism (pneumo-mechanical control),

(ii) NEL have developed a programmable piston air-motor to be marketed by Motron, which utilises a proportional control valve, designed for high torque applications, and

(iii) Schrader Bellows have demonstrated a pneumo-mechanical programmable drive which utilises binary valving and an electro-magnetic brake system; are now known to the author and serve to illustrate the developing awareness of such technology within industry.

The research study described in this thesis has provided a foundation from which the initial introduction of pneumatic motion control systems in robot modules and "intelligent actuators" within manufacturing industry has resulted. However, the author acknowledges that there remains considerable scope for future research effort in this field.

CHAPTER 11

CONCLUSIONS

This thesis has described the design and evolution of motion control systems for pneumatic drives. The use of microprocessor based controls has allowed a family of low cost pneumatic servo-drives to be produced which exhibit excellent cost performance ratios, thereby allowing them to find widespread application in manufacturing industry. Much of this work has centred on devising and testing appropriate motion control strategies which have incorporated the use of various control techniques which include;

(i) "gain scheduling",

(ii) "non-linear" feedback coefficients (gain contours),

(iii) "control algorithm scheduling",

(iv) "front-end" control, and

(v) learning.

The design approach has been largely experimentally based with an evaluation facility devised to enable the success of any control implementation to be established. Embodied in this work has been the need to define performance measures which can provide the necessary quantitative assessments of how each control strategy overcomes the problems associated with a compressible drive medium in terms of low stiffness, little inherent damping and the highly non-linear characteristics associated with the control components.

The motion control strategies have evolved with respect to specified design criteria (which relate to static and dynamic performance characteristics, to be achieved using "low cost" control components) and demonstrate significant performance improvements over conventional linear proportional control approaches with respect to both static and dynamic performance of the pneumatic servo-drive. The success of the work has furthered the industrial application of pneumatic motion control with the drive technology evolved leading to exploitation in commercial form of "single axis controllers" for pneumatic servo-drives which can be used in conjunction with robot modules, "intelligent actuators" and "special purpose" machine drives (for translation and rotary motion).

The resultant pneumatic drive systems exhibit for many categories of industrial task, a cost performance ratio comparable to or better than alternative drive systems based on hydraulic or electric actuation. However, much work remains in maximising the usefulness of pneumatic servos: for example the implementation of learning/adaptive control schemes can further enhance the performance and robustness of pneumatic servo-drives and facilitates a greater level of user friendliness with respect to control parameter tuning. However, the work reported in this thesis has provided a framework for future work, not least through the availability of commercial pneumatic servo systems and components; thereby also allowing them to gain industrial acceptability.

The describing equations of compressible fluid power systems are highly non-linear and hence analysis is complex. However, linearised analysis has been used by a number of researchers in attempting to develop a pneumatic system model which is appropriate over a range of operating conditions. In general a linearised model offers advantages of simplicity and can be analysed using a range of well proven techniques. Linearised analysis was first used by Shearer [1954] to develop a model for a valve controlled pneumatic actuator and load, operating about its' midstroke position. The analysis has been extended by Burrows [1969] to account for motion of the load about any initial position, and shows that the use of velocity and pressure feedback can be used to stabilise low pressure (less than 10 bar) pneumatic servo-drives.

A linearised model is presented, derived from Burrows [1969] analysis, and has been extended in this research study with the objective of formulating a system model that can be used;

(i) to gain an understanding of the physical system and as a means of establishing a theoretical foundation for the open-loop system, to be used in the ensuing research program,

(ii) in analysis of the closed loop system behaviour and illustrate the generalised effect of changes in controller feedback design (from which new controller designs can be formulated), and

(iii) specifically in controller design (eg selecting controller gains to simplify tuning of robot modules and "intelligent" actuators).

However, as a result of studies here it was found to be impractical, using linearisation techniques, to make specific use of the model for controller design due to;

(i) the inadequate nature of the model in representing the highly non-linear system, and

(ii) the difficulties embodied in establishing system parameters. For these reasons the ensuing research program was largely formulated on practical investigations, where the evaluation system is used to test an hypotheses.

The linearised model can be used as an aid in formulating the design of appropriate control approaches, but its application here is limited to illustrating the generalised effect of changes in feedback design. The model is not good enough to represent the true system behaviour because of simplifications made within the analysis and the limitations of linearised analysis. The model and its interpretation need to be viewed with care when direct comparison with the physical system is made. The linearised model has been used to qualify practical findings as to the limitations of conventional proportional control (using the Root Locus technique for generalised stability assessment). The analysis is then extended to incorporate velocity and acceleration minor loop feedback and is qualified by experimental findings which show that by appropriate selection of the minor feedback loops stability can be achieved. However, it must be remembered that the model only approximates to the physical system (which is highly non-linear), and that many assumptions are embodied within its formulation. A reader wishing to extend the use of this model should proceed with caution because;

(a) the saturation-type non-linearities (which occur when input demand exceeds a given level) are not considered within the model,

(b) linearisation techniques are effective only if the system is operating away from the threshold conditions and the input is not sufficiently large to cause saturation, (and when high gains are encountered in control loops a saturation condition is registered for very small changes in operating conditions),

(c) the proportional valve demonstrates asymmetry in its characteristics which are not embodied within the analysis,

(d) friction forces (and the non-linearities therein introduced) are ignored, in particular those associated with actuator seals and bearings have a significant effect,

(e) the difficulties associated with establishing representative system parameters for use within the model, and

(f) the linear model is only valid for small perturbations about the initial position.

The research study described in this thesis has resulted in a pneumatic motion control system with significant merits which include;

(i) accuracy and repeatability for point-to-point positioning of payloads can be achieved to 0.2mm or better for a range of robot modules, "intelligent" actuators and payloads (which is commensurate with many industrial application requirements) through the application of novel non-linear feedback loop coefficients (gain contours) which result in significant increases in drive stiffness as the setpoint is approached. Non-linear loop gains, achieved using "gain scheduling" techniques and look-up tables, enable very significant increases in drive stiffness (high gain) to be attained which reduces not only steady state errors, it also reduces the effects of non-linearities due to Coulomb friction, hysteresis, backlash, etc., thus improving system accuracy;

(ii) stability is achieved by utilising 'minor loop compensation' in the form of velocity and acceleration feedback. Velocity feedback has a similar effect on stability as increasing viscous damping acting on the load, but unlike viscous damping it does not dissipate energy. Acceleration feedback further improves system stability and is most useful in conjunction with velocity feedback. Regenerative, or positive acceleration feedback has been found most effective in achieving minor loop compensation in this study with a pneumatic drive (where previous studies known to the author advocate negative feedback);

(iii) satisfactory dynamic response characteristics are achieved through the implementation of novel realtime "control algorithm scheduling" techniques allowing control strategies to be selected which can cater for the major changes in operating conditions which occur during load and valve spool movements;

(iv) non-linearities (due to backlash, hysteresis, seal friction and stick-slip friction effects), and asymmetry in the control components (ie proportional valve and cylinder) can be reduced through the novel use of software controlled offsets and high loop gains, which can also reduce the thresholds relating to the static and dynamic deadbands;

(v) speed of response can be improved and the pneumatic time delays minimised through the application of 'front-end' control schemes to the original realtime control strategy (Moore et al [1986]).

Front-end control is an approach which can be used to improve the dynamic characteristics of pneumatic servo-drives. The strategy utilises an outer decision loop which modifies the command issued to an existing close loop drive. The outer loop generates a set of initial (front-end) control sequences at the start of any "point-to-point" move. The front-end control schemes decribed are based on simple logical strategies, and as such are relatively easy to implement. Such a strategy results in an improved quality of response, however highly oscillatory systems would cause problems (Astrom [1980]), so the response must be over damped under its rated load condition before applying a front-end strategy. Such a control approach includes;

(a) "saturation" control which minimises the pneumatic time delay in addition to making the response to step demands more predictable. "Saturation" control is where a maximum command signal (corresponding to a valve "saturation") is applied until the system responds to the input. Such an approach results in a very significant improvement in response characteristics (pneumatic drives are characterised by a "response lag" caused by the delay in pressure rise within the actuator). Under "saturation" control the response lag is minimised for a given valve, actuator, load and supply pressure. The "response lag" is particularly significant for small moves when the command signal is consequently small, (the command signal being a function of error).

(b) "setpoint switching" techniques which result in a faster speed of response (ie reduced rise time) whilst stability is maintained, and good static performance achieved by finally reverting to the original control strategy. Setpoint switching involves the use of a setpoint co-efficient, by increasing this co-efficient from unity for a proportion of the response the setpoint is adjusted accordingly. As such the system moves faster towards its desired position, before reverting to a unity co-efficient. The magnitude of change in the setpoint co-efficient and the proportion of the response time over which the modification takes place are control parameters that have to be selected; and

(c) "negative saturation" control can be used to ensure overshoot is maintained within specified levels. "Negative saturation" involves the use of a discrete command (as the setpoint is approached) which corresponds to reverse "valve saturation", which increases damping. The duration of the "negative saturation" can be incorporated within a "learning" strategy (where an adaptive algorithm is used) to optimise the final response characteristics without having to modify system control parameters directly;

(vi) learning/self tuning control strategies have been shown to further enhance the characteristics of a pneumatic servo. Learning schemes have been incorporated within the control system design which allow certain characteristics of the dynamic response of a drive to be optimised. Such learning schemes can further extend the application areas and enhance the benefits associated with pneumatic motion control. Such a control approach can be used in one of two modes;

(a) to simpliy control parameter tuning during an 'initialisation phase' to achieve a specified dynamic performance level (eg overshoot within five percent), and alternatively

(b) to be used 'continuously' on-line (adaptive control) to rectify any changes in performance as time progresses due to external load variations or drift in plant conditions. The use of "learning" in a "self tuning" control scheme for pneumatic drives could be a major area of subsequent study, where performance "optimisation" and simplified system "initialisation and set-up" are two tangible benefits that should result;

(vii) robustness of the control strategy has been demonstrated in a number of ways;

(a) through a series of evaluations for which performance measures were derived to facilitate the partially automated testing of control strategies. These evaluations yielded statistically valid performance data,

(b) the control strategy has been successfully applied to a number of robot modules and "intelligent" actuators which demonstrate a wide variation in mechanical characteristics with respect to actuator size, intrinsic mass, slideway friction and ratio of valve port area to actuator volume. Modules include M60000 series (25 and 50mm diameter units); intelligent actuators include ISO standard pneumatic cylinders with integral leadscrews and encoder, Lintra rodless cylinders with rack and pinion encoder feedback, Origa rodless cylinders with integral leadscrew and encoder, and SMC rotary vane actuators with tooth belt encoder feedback,

(c) performance testing of modules under "load" has been undertaken (inertial loading and mass loading have been considered). The static performance of modules does not deteriorate under rated load conditions. However, dynamic performance characteristics are influenced by variation in payload. The influence on the dynamic response can be minimised with the use of "front-end" schemes, and similarly adaptive control can accommodate variations within the rated load spectrum. For most industrial applications of the M60000 series modules the Milcom control strategy is sufficently robust to accommodate the load variations encountered without any noticable performance deterioration by appropriate tuning. Additionally, a feature unique to the pneumatic servo-drive configuration described in this research study, allows "known payloads" to be "counterbalanced" by automatically re-setting the nominal "valve-null" when the system encounters a static load (eg positioning a "mass" in a vertical orientation) which can be a very significant advantage (the use of "active" counterbalance systems based on pneumatic drives could be a significant area for future study), and

(d) through a series of example industrial application studies the reliability and robustness of the control strategy has been confirmed;

(viii) the motion control system is applicable in a range of applications requiring point-to-point control (other than robotic systems): eg machine elements, process control valve operation applications, etc..; and

ix) the control parameter "tuning procedure" is reasonable, but could be an area of subsequent study (eg utilising computerised simulation and "self tuning" strategies). To "tune" for most types and sizes of the pneumatic actuators discussed in (vii) is not difficult, however to optimise characteristics is more difficult. Within the "Single Axis Controller" implementation it is only necessary for the user to access three "user setup parameters" for setting (a) approach speed, (b) damping and (c) final speed, for each direction. With access via a "password" the user can modify "system parameters" if necessary. The tunability is commensurate with proprietary dc electric drive axis motion control systems available (ie Galil, Electrocraft, Quin, etc.), where fine tuning tends to be an iterative process. However, there remains considerable scope for further work generally in this area.

For industrial applications the pneumatic servo drive will only find wide acceptance if it can maintain a cost:performance advantage over alternative drive technologies. From the component cost perspective pneumatic systems should be cheaper than their electric or hydraulic counterparts. It can be argued that the operational cost of pneumatic actuators can be neglected as the pneumatic network within a manufacturing plant would be required to service other functions. The efficiency of pneumatic systems is low (due to cooling of the fluid, use of positive displacement actuators, and difficulty in achieving satisfactory sealing), however, operational efficiency is not necessarily of prime importance in selection of a drive system, but where efficiency is an important criteria a pneumatic drive would not be selected.

Future developments that could further enhance the performance and acceptance of pneumatic motion control systems could include;

(a) enhanced "self-tuning" and adaptive control schemes,

(b) optimised mechanical system components, namely a family of fast response two-stage proportional valves with symmetric and linear operational characteristics; reduced friction in actuators by utilising PTFE seals/wipers and coated actuator barrels; improved transducer resolution and linearity (eg low cost linear scales); and functional development of "intelligent" actuator technology (eg microprocessor axis controls integrally embodied within the actuator),

(c) use of computer aided design/simulation packages for control system design (eg component orientated modelling, non-linear system models, etc..), and

(d) the use of pneumatic motion control systems in applications requiring a speed control and/or contouring capability. These are categories of motion control that cannot be addressed by the present generation of pneumatic servo-drives.

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REFERENCES

Abdel Gadir A.A., (1984), "Design and Evaluation of a Pneumatic Servo-Controlled Gantry Modular Unit", MSc Thesis, LUT.

ACE Controls Inc., (1982), "Industrial Shock Absorbers", Data Sheet AS-P6.

Analog Devices Instruments and Systems, (1980), "RTI 1240/1241 Users Reference Manual".

Anon, (1978), "Pneumatic Systems for the Fast, Gentle Push", Nov., pp 96-100, Modern Materials Handling.

Anon, (1983), "Robotic and Fluid Power", Dec., pp 6-12, Hydraulic and Air Engineering.

Anon, (1984), "Turning the Key to Automation - How a Turnkey Systems Service Unlocks the Future", vol.11, pt.81, pp 17-21, Applied Pneumatics.

Anshin S.S., (1978), "Acceptance Tests for Industrial Robots", Machines & Tooling, Vol.49, No.7, pp 11-13.

Araki K., (1971), "Compensation of a Force-Feedback Pneumatic Servo-Valve", Vol.7, Pt.4, pp 366-375, Trans.Soc.Instrum.Control Eng. (Japan).

Araki K., (1981), "Effects of Valve Configuration on a Pneumatic Servo", Paper F2, pp 271-291, April, Int.Fluid Power Symp, Cambridge, U.K.

Archer J.R. & Blenkinsop P.T., (1983), "Actuation for Industrial Robots", pp 85-89, publication no.C306/83, I Mech E. Conf. Electric versus Hydraulic Drives.

Astrom K.J., (1980), "A Robust Sampled Regulator for Stable Systems with Monotone Step Responses", Automatica, pp 313-315.

Astrop R., (1982), "Robot Revolution Gains Momentum", Feb., pp 14-19, Machinery and Production Engineering.

Astrop R., (1982), "Robot Applications: A Trio of Success", Feb., pp 24-29, Machinery and Production Engineering.

Backe W., (1983), "Servopneumatischer Linearantneb", RWTH Aachen (Dissertation (1979)).

Barber N.T., (1983), "The Future of DC Servo-Drives", pp 59-66, publication no. C303/83, I Mech E. Conf. Electric versus Hydraulic Drives.

Belforte G., D'Alfio N., Quagliotti F., (1981), "Variable Positioning Systems of Pneumatic Cylinders", Paper F1, pp 259-270, 6th Int.Fluid Power Symp., Cambridge University, U.K., April 8th-10th.

Bender E. & Kotschenreuther P., (1980), "Position Control Loops with Pneumatic Actuators", pt.2, pp 85-90, Process Automation.

Botting L.R., Eynon G.T., Foster K., (1969-70), "The Response of a High Pressure Pneumatic Servo-Mechanism to Step and Sinewave Inputs", Vol.184, Pt.1, No.54, Proc.Instn.Mech.Engrs.

Brown D.E. & Ballard R.L., (1972), "Digital Computation for the Analysis of Pneumatic Actuator Systems", Proc.Instn.Mech.Engrs., Vol.186, pp 881-889.

Brown R.A., (1983), "Designing Custom Robots for In-Plant Use",13th Int.Symp.on Ind.Robots, Chicago, pp 20-45 to 20-56.

Burrows C.R., (1968), "Non-linear Pneumatic Servo-Mechanisms", PhD Thesis, University of London.

Burrows C.R., (1969), "Effect of Position on the Stability of Pneumatic Servo-Mechanisms", Vol.11, No.6, Journal of Mechanical Eng.Science, pp 615-616.

Burrows C.R., (1969) "Further Study of a Low Pressure On-Off Pneumatic Servo-Mechanism", Vol.184, Pt.1, pp 849-857, No.45, Proc.Instn.Mech.Engrs.

Burrows C.R., (1972), "Fluid Power Servomechanisms", Van Nostrand Reinhold

Chitty A. & Lambert T.H., (1976), "Modelling a Loaded Two-way Pneumatic Actuator", Vol.9, January, pp19-25.
Controls Technology Corp. USA, (1984), Data Sheets 101, 202, 302, 501/2/84.

Cutland M.J., (1967), "Response of a Pneumatic Servo-Mechanism", MSc(Eng) Thesis, University of London.

Diaz R.J., (1980), "Robotic Actuators: A Technology Assessment", Int.Comp.Tech.Conf.on Advances in Comp. Tech., pp 225-272.

Ditchfield K., (1984), "An Investigation into the

Open-Loop Response of Pneumatic Spool and Sleeve Valves", Undergraduate Project Report, LUT.

Drazan P.J. & Jeffrey M.J., (1976), "Microprocessor Control and Pneumatic Drive of a Manipulating Arm", Paper D2, ppD2-9, March 24th-26th, 6th Int.Symp.Ind. Robots, Nottingham, UK.

Drazan P.J. & Jeffrey M.J., (1978), "Some Aspects of an Electro-Pneumatic Industrial Manipulator", pp 396-404, Conf.Ind.Robot Tech., Stuttgart, W.G.

Drazan P.J., Jeffrey,M.J. & Zarek J.M., (1977), "Recent Developments in the Design of a Novel Electro-Pneumatic Robot", pp 143-150, Oct 14th-17th, Conf.Theory & Practise of Rob. & Man. & 2nd Proc.Int. CISM-IFToMM, Warsaw.

Drazan P.J., & Kennett R., (1975), "Opto-Pneumatic Manipulating Arm", Vol.2, pt.1, pp7-10, March, The Industrial Robot, also 4th Int.Symp.Ind.Robots, Tokyo 1974.

Drazan P.J. & Thomas P.B., (1978), "Simulation of an Electro-Pneumatic Manipulator System", paper C5, pp C5-59, Sept.13th-15th, 5th Int.Fluid Power Symp., Durham, UK.

Drazan P.J. & Toulis V., (1980), "Control of Robot Dynamics", March, Conf.of Swiss Ass.for Automation, Zurich, Switzerland.

Drazan P.J. & Zarek J.M., (1978), "Controllability of an Electro-Pneumatic Manipulator", pp 283-300, September, Conf.3rd CISM-IFToMM, Udine.

Dugan R., (1975), "A State-of-the-Art Cobalt-Rare Earth Permanent Magnet Motor for the Machine Tool Industry", IEEE-IAS Machine Tool Conference, Milwaukee, Wisconsin.

Dynamic Valves Inc., USA, (1984), Data Sheet SGG 7-2-84, Models 130, 140 and 160.

Ekere N.N., (1984), "The Use of a New Servo-Valve for Pneumatic Positioning", MSc Thesis, LUT.

El-Ibiary Y.M., Dodds D.E. & Nikiforuk P.N., (1979), "Load Position Control with On-Off Hydraulic Valves", Hydraulics and Pneumatics, October, pp 149-151.

Emessem Solenoid Company Ltd., (1983), Data Sheet, Product G RF 035.

Engelberger J., (1980), "Robots in Practise", Kogan Page.

Eun T., Cho Y.J. & Cho H.S., (1982), "Stability and Positioning Accuracy of a Pneumatic On-Off Servo-Mechanism", Vol.3, Paper WP8.3.30, pp1189-1194, June 14th-16th, American Control Conf., Arlington.

Eveleigh, (1976), "Adaptive Control and Optimal Techniques", McGraw-Hill.

Fitch E.C.,(1966), "Fluid Power and Control Systems", McGraw-Hill Inc., USA.

Gradetskii V.G., Paroi A.A., (1981), "Industrial Robots with Gradual Braking of the Pneumatic Actuating Mechanism Motion", Vol.1, No.3, pp 3-5, Soviet Engineering Research.

Hammett G.G., (1984), "Low Cost Servo-Actuator Design", pp 4-45, Int.Symp.on Ind.Robots, Detroit.

Harrison R., Weston R.H., Moore P.R. & Thatcher T.W., (1985), "The Industrial Application of Pneumatic Servo-Controlled Modular Robots", Nat.Conf. on Prod.Res., Nottingham, Sept.9th-11th

Hattori M., (1980), "Effect of Dither on Non-Linear Characteristics of a Position Control System - Effect on Jerking Motion due to Varying Torque", pp 143-149, Electr.Eng.Japan(USA), Vol.100, No.4, (July-Aug).

Heer E., (1981), "Robots and Manipulators", Nov., pp 42-49, Mechanical Engineering.

Heidenhain GMBH, WG, (1984), Data Sheets 217 391 91. 30.4/83.E.

Herion UK Ltd., (1984), Data Sheets 7501499.06.03.83 VDH, /06.83, 7501273.06.03.83.

Hewlett Packard, (1982), Data Sheet 5953-7774(9/82).

Huhne G., Neugebauer A. (1974), "The Performance of Novel Pneumatic Industrial Robot Drives for Point-to-Point and Continuous-Path Controls", Paper B6, B6-73, 2nd Conf.on Ind.Robot Technology, University of Birmingham, U.K., March 27th-29th.

Ingersoll Engineers, (1980), "Industrial Robots: The Report by Ingersoll Engineers, Vol.1, An Industrial Appreciation", DoI/N.E.L.

Ingersoll Engineers, (1980), "Industrial Robots: A Worldwide Literature Survey, Vol.2, Applications R&D Bibliography", DoI/N.E.L.

Ivlev V.I., (1983), "Economical Pneumatic Servo-Drive for a Robot Arm", pp 68-70, Vol.3, No.9, Sov.Eng. Research.

Kato S., Marui E., Matsubayashi T. & Suzuki M., (1972), "Some Considerations on Characteristics of Pneumatic Servo-Mechanism", Vol.15, pt.80, pp225-235, Feb., Bull J.S.M.E.

Koliskor A.Sh. & Kochenov M.I., (1978), "Methods for Checking the Operating Accuracy of Industrial Robots", Machines & Tooling, Vol.49, No.8, pp 8-10.

Kopp V.Ya., Pogorelev B.V. (1982) "Dynamics of a Digital Pneumatic Rotation Drive of a Manipulator", Vol.2, No.2, pp 7-9, Soviet Engineering Research.

Korolev V.A., Sergeev S.M., (1979), "Point-to-Point Robots for Assembly", MS 79-255, pp 1-20, 9th ISIR, Washington, USA.

Korolev V.A., Sergeev S.M., Jurevich E.I., (1982), "Application of Pneumatic Industrial Robots for Assembly", pp 101-114.

Kozyrev Y.G., (1978), "Constructing a Standard Series of Industrial Robots", vol.49, no.7, pp 3-10, Machines and Tooling.

Kozyrev Y.G. & Anshin S.S., (1977), "Industrial Robots", vol.48, no.11, pp9-12, Machines and Tooling.

Kraynin G.V., Ivlev V.I. & Dyuffel O.V., (1982), "Utilisation of an Observer Model for Improving the Dynamic Characteristic of a Pneumatic Tracking Actuator", pt.5, pp 50-55, Mech.Science.

Kreinin G.V.,(1978), "Pneumatic Drives for Industrial Robots", Vol.49, No.7, pp 31-35, Machines and Tooling.

Kristztiniez P., Elek I.,(1976) "Positioning Problems of Pneumatic Industrial Robots", Paper E3, E3-29, 3rd Conf.on Ind.Robot Technology, University of Nottingham, U.K., March 24th-26th.

Kudvn, Narendra, (1974), "An Identification Procedure for Descrete Multivariable Systems", IEEE Transactions on Automatic Control, AC19, No.5, pp 549-552.

Kuo B.C., (1974), "Theory & Applications of Stepper Motors", West Publishing Co.

Landau, (1974), "A Survey of Model Reference Adaptive Techniques - Theory and Applications", Automatica, Vol.10, pp 353-379.

Larsen W.W., (1982), "The Evolution of Pneumatics into Robotics", pp 31-35, Vol.12, No.1, Automation and Control, New Zealand.

Mannetje J.J., (1981), "Pneumatic Servo Design Method Improves System Bandwidth Twenty-Fold", Vol.28, pt.6, pp 79-83, June, Control Engineering.

Mariuzzo C.L., Martins Meto A.F., Soares A.M.C. & Mansour W.M., (1980), "Analysis and Design of a Class of Pneumatic Positioning Servos", pp 455-462, Conf. Brasileiro de Automatica, Rio de Janeiro.

Mariuzzo C.L., Martins Meto A.F., Soares A.M.C. & Mansour W.M., (1980), "Mathematical Models for Pneumatic Actuators", pp 445-453, Conf.Brasileiro de Automatica, Rio de Janeiro.

Martonair Ltd, (1982), Catalogue and Data Sheets, pp 2.3.129 - 2.3.134.

Martonair, (1984), "Modular Handling Units", Publication No.s/1/138, Issue 3, 10/84.

Martonair, (1984), "Modular Handling Units", publication No.s/1/143, Issue 1, 3/84.

Mayer E.A., (1981), "Dual Solenoid Valve Interfaces Electronics Pneumatics", pp 47-49, Vol.49, Pt.6, Automotive Engineering.

McCloy D. & Martin H.R., (1973), "Control of Fluid Power : Analysis and Design", 2nd revised edition, Longman, pp 185-273.

McLennan Servo Supplies Ltd., (1983), "Stepper Motor Systems", product catalogue 982-12.

Moore P.R., (1986), "Pneumatic Motion Control Systems for Modular Robots", Internal Department Report.

Moore P.R., Ssenkungo F.W., Weston R.H. & Thatcher T.W., (1986), "Control Strategies for Pneumatic Servo-Drives", submitted to 'International Journal of Production Research' for publication.

Moore P.R., Weston R.H.& Thatcher T.W., (1984), "Control of Pneumatic Servo-Drives Using Digital Compensation", IASTED Int.Sym.on Telecommunication and Control, Halkidiki, Greece, Aug., proceedings pp 264-268.

Moore P.R., Weston R.H. & Thatcher T.W., (1985), "Compensation in Pneumatically Actuated Servo-Mechanisms", Trans.Inst.Measurement and Control, July-Sept., Vol.7, pt.4.

Moore P.R., Weston R.H., Thatcher T.W. & Gascoigne J.D., (1983), "Modular Robot Systems", 2nd IASTED Int.Sym.on Robotics and Automation, Lugano, Switzerland, June, Proceedings pp 165-169.

Morgan G., (1985), "Programmable Positioning of Pneumatic Actuators", Applied Pneumatics, May, pp 16-20.

Morozov A.I., (1973), "Selection of Braking Systems for Auto. Pneumatic Drive", Part 9, pp 24-26, Melch.and Avton.Proiz (USSR).

Nagarajan R. & Weston R.H., (1985), "Front End Control Schemes for Pneumatic Servo-Driven Modules", IME pt(B).

Nagrath I.J. & Gopal M., (1982), "Control Systems Engineering", Wiley.

Nazarczuk K., (1979), "Modelling of Pneumatic Drives" Vol.2, pp 1258-1261, July 8th-13th, Conf.Thy.of Mach & Mechanisms, New York, ASME.

Neuhaus R., (1981), "Electro-Pneumatic Servo-Drive with Closed Loop Position Control", Vol.25, pp 120-125, Oelhydraul & Pneum (Ger).

Nguyen H.T., Pachnicke E., Anders P., (1984), "Servo Pneumatic Cylinder Drive for Positioning Purposes", pp 137-157, Kolloquium.

Noodleman S., "Factors in Applying Rare Earth-Cobalt Magnets to D.C. Motors", Centre d'Information du Cobalt, Brussels, (Reprinted from COBALT 1974.4, pp 69-73).

Ogata, (1978), "Modern Control Engineering", Prentice Hall.

Ottinger L.V., (1982), "Robot System's Success Based on Maintenance", vol.14, no.6, June, pp38-43, IE.

Ottinger L.V., (1982), "Robots and Other Technologies in the Automated Factory", vol.14, pt.9, Sep., pp 26-32, IE.

Paroi A.A., (1981), "Calculating the Co-ordinates at Which to Start Braking of the Piston on the Pneumatic Drive of an Industrial Robot", Vol.1, No.7, pp 7-10, Soviet Engineering Research.

Pietzsch L., Huhne G., Neugebauer A. (1974), "Advantages of Novel Electro-Pneumatic Servo-drives for Industrial Robot and Influence of the Kinematics on the Positioning Errors". IBP Pietzsch GmbH, 7505 Ettlingen/Karlsruhe, GFR.

Pluhar K., (1980), "Pneumatic Motors and Positioners Offer Simplicity and Reliability", vol.27, pt.2, Feb., Control Engineering.

Portin D.L. & Chen P.I., (1981), "A Linear Fluidic-Pneumatic Position Servo", Vol.13, pt.1, pp 19-36, March, Journal of Fluid Control.

Potter R.D., "Practical Applications of a Limited Sequence Robot", pp 55-63.

Prokofev V.N. et al (1976) "Hydraulics and Pneumatics in Industrial Robots", Vol.56, No.5, pp 16-20, Russian Engineering Journal.

PTM Industries Corp., USA, (1984), Data Sheets Model 101, 202, 302, and 501.

RHP Precision Linear Products, (1984), RHP Linear Motion Products, publication no.C730 and no.S774.

Richardson B.A., "Application of the Microprocessor to Assembly Machines", pp 187-197, Wickman Automated Assembly Ltd., Coventry.

Richardson B.A., "The Use of Standard Units in the Design and Manufacture of Assembly Machines", pp353-355, Wickman Automated Assembly Ltd., Coventry.

Riley F., (1982), "The Use of Modular Flexible Assembly Systems as a Halfway Path between Special Design and Robots", 3rd Int.Conf. on Assembly and Automation, May.

Robert Bosch GMBH, (1984), Data Sheet WE/11/83.

Robotics Industry Directory, (1982/83/84), Technical Data-Base Corp., USA.

Rooks B.W., Tobias S.A., "A Simple Accurate

Positioning System for an Industrial Robot", pp 371-384.

Salihi A., (1983), "An Investigation into the Use of Distributed Processing in Glass Container Manufacture", PhD Thesis, Loughborough University.

Shacloth, Buchart, (1965), "Synthesis of MRAC by Lyapunor Second Method", Proc. of IFAC on Theory of Self Adaptive Control System, pp 145-152.

Shearer J.L., (1954), "Continuous Control of Motion with Compressed Air", ScD Thesis, Massachusetts Inst. of Tech.

Shearer J.L., (1956) "Study of Pneumatic Processes in the Continuous Control of Motion with Compressed Air", Vol.78, pp 233-234, I and II Trans.Am.Soc.of Mech.Engrs.

Shearer J.L., (1957), "Non-linear Analog Study of a High Pressure Pneumatic Servo-Mechanism", Vol.79, No.3, p 465,Trans.Am.Soc.of Mech.Engrs.

Schwenzer R., (1984), "Static and Dynamic Behaviour of State Controlled Pneumatic Drives", Vol.28, pt.3, pp 154-161, Oelhyd.& Pneum.(Ger).

Surnin B.N. et al, (1978), "Design Features of Modular Type Robots", vol.49, no.7, pp 17-20, Machines and Tooling.

Taft C.K. & Herrick B.M., (1981), "A Low Power Electro-Pneumatic Servo-Valve", pp 197-201, October 21st-23rd, 37 Nat.Conf.Fluid Power, Chicago, USA.

Taft C.K. & Herrick B.M., (1981), "A Proportional Piezoelectric-Electro-Pneumatic Servo-Valve Design", Vol.1, June 17th-19th, WA-8A, Control Conf., Charlottesville.

Tal J., (1984), "Motion Control by Microprocessors", Galil Motion Control Inc., USA.

Taylor M., (1984), "Lower Order Robots - Will They Play the Dominant Role in Non-Dedicated Automation?", Applied Pneumatics.

Texas Instruments, (1978), "Assembly Language Programmers Guide".

Texas Instruments, (1980), "Microprocessor Pascal System User Guide".

Texas Instruments, (1985), "The TTL Data Book".

Texas Instruments, (1980), "TM 990/100M/101M Microcomputer User Guide".

Texas Instruments, (1977), "TMS 9900 Data Manual".

Texas Instruments, (1976), "TMS 9901 Data Manual".

Texas Instruments, (1980), "TX 990 Operating System".

Thatcher T.W., (1983), "Optimisation of Dither Frequency and Amplitude", unpublished internal report, LUT.

Thatcher T.W., (1983), "Proportional Amplifier Design and Evaluation", unpublished internal report, LUT.

Thatcher T.W., Weston R.H. & Moore P.R., (1983), "Modularity in Workhandling Systems", UK Robotics Research Conf.IEE London, Dec., proceedings pp 28-34.

Thatcher T.W., Weston R.H. & Moore P.R., (1983), "Software System for Modular Robots", Conference on Advanced Software in Robotics, Liege, Belgium, May, proceedings pp 48-54.

Thatcher T.W., Weston R.H. & Moore P.R., (1984), "Software for Modular Robots", in Advanced Software in Robotics Ed A. Danthine & M. Geradin, North Holland Press.

Thatcher T.W., Weston R.H. & Moore P.R., (1984), "Software Structures for Modular Robots", Int.J.Prod. Res., Jan., vol.22, no.1, pp 71-84.

Thatcher T.W., Weston R.H., Moore P.R. & Harrison R., (1985), "Single Axis Controllers for Modular Workhandling Systems", IASTED Int.Symp.on Robotics and Automation, Lugano, Switzerland, June, proceedings pp 240-243.

Thatcher T.W., Weston R.H., Moore P.R. & Harrison R., (1985), "Supervisory Control of Single Axis Controllers for Modular Robotic Systems Using a Serial Interface", Nat.Conf.on Prod.Res., Nottingham, Sept.9th-11th.

Turnbull D.E., (1976), "Fluid Power Engineering", Butterworth & Co.(Publishers)Ltd., London.

Vaughan D.R., (1965), "Hot Gas Actuators. Some Limits on the Response Speed", pp 113-119, March, Trans.of

ASME, Journal of Basic Eng.

Viersma T.J. & Blok P., (1969), "Stabilisation of Hydraulic and Pneumatic Servo-Mechanisms", Vol.13, No.1, Jan., Olhydraulik and Pneumatik.

Vodopyan P.O., (1981) "Improvement of Hydro-Pneumatic Drives for Industrial Robots", Vol.1, No.3, pp 25-27, Soviet Engineering Research.

Vodopyan P.O., Oksenenko A.Ya., (1978), "Hydro-Pneumatic Drives in Industrial Robots", Vol.49, No.12, pp9-12, Machines and Tooling.

Walker D.G.W., (1983), "Electric and Hydraulic Drives for Heavy Industrial Robots", pp 15-19, publication no.C298/83, I Mech E. Conf. Electric versus Hydraulic Drives.

Warnecke H.J. & Schraft R.D., (1982), "Industrial Robots : Application Experience", copyright I.F.S. Publications Ltd.

Watton J. (1984), "The Generalised Response of a Servo-Valve-Controlled, Single Rod, Linear Actuators and the Influence of Transmission Line Dynamics", Vol. 106, June, pp157-162, Journal of Dynamic System, Measurement and Control, ASME.

Wauer G., (1980), "Electro-Pneumatic Servo-Drives as Position Drives for Manipulator Systems", Vol.24, No.7, pp552-555, July, Oelhydraul and Pneum (German).

Weston R.H. & Moore P.R., (1986), "Modular Robotic Systems", in Encyclopaedia of Systems and Control, Editor-in-Chief Madan Singh, Pergamon Press, to be published June-Aug.

Weston R.H., Moore P.R. & Thatcher T.W., (1983), "Design of a Family of Modular Robots", ICPR Conf., Windsor, Canada, August, Proceedings TD5d.

Weston R.H., Moore P.R. & Thatcher T.W., (1984), "Design of Modular Robotic Systems", Microprocessors and Microsystems, Feb., Vol.8, no.1, pp 16-20.

Weston R.H., Moore P.R., Thatcher T.W. & Morgan G., (1984), "Computer Controlled Pneumatic Servo-Drives", IME pt.B, Dec., Vol.198B, no.14, pp 175-281.

Weston R.H. & Morgan G., (1984), "A New Family of Robot Modules and their Industrial Application", UK Robotics Research Conf., IEE London, Dec.

Whited J., (1981), "Control System for A.C. Induction Motors", patent application.

Whited J., (1981), "Control System for Synchronous Brushless Motors", patent application.

Yurevich E.I., (1981), "Design, Testing and Reliability of Machines Standardization Problems in Industrial Robotics", vol.1, no.8, Soviet Engineering Research.

APPENDICES

APPENDIX (3.0) ROBOT MODULE DETAILS

Table (3.3.1)

MODULE DATA TABLE	LUT LINEAR MODULE No 1
Actuator	Martonair Cylinder ∅38 mm
Stroke	300 mm
Slideway	Twin hardened and ground bars in oilite bushes
Transducer	Ferranti Optical Encoder - 500 p/rev Type 24ST
Transducer Transmission	Rack and pinion
Brake Mechanism	Pneumatic clamp arrangement - positive 'on' action
Carriage Frame	Mild Steel
Additional Features	Sliding carriage with ACE hydraulic dampers
Transmission	Direct acting

Plate (3.3.1) LUT Linear Module No.1

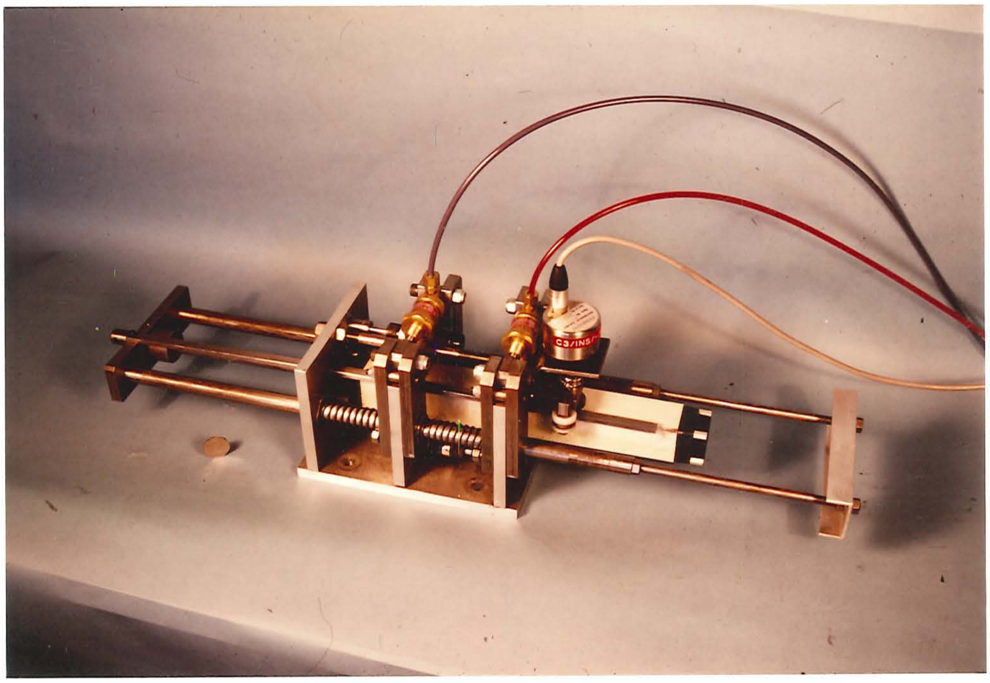
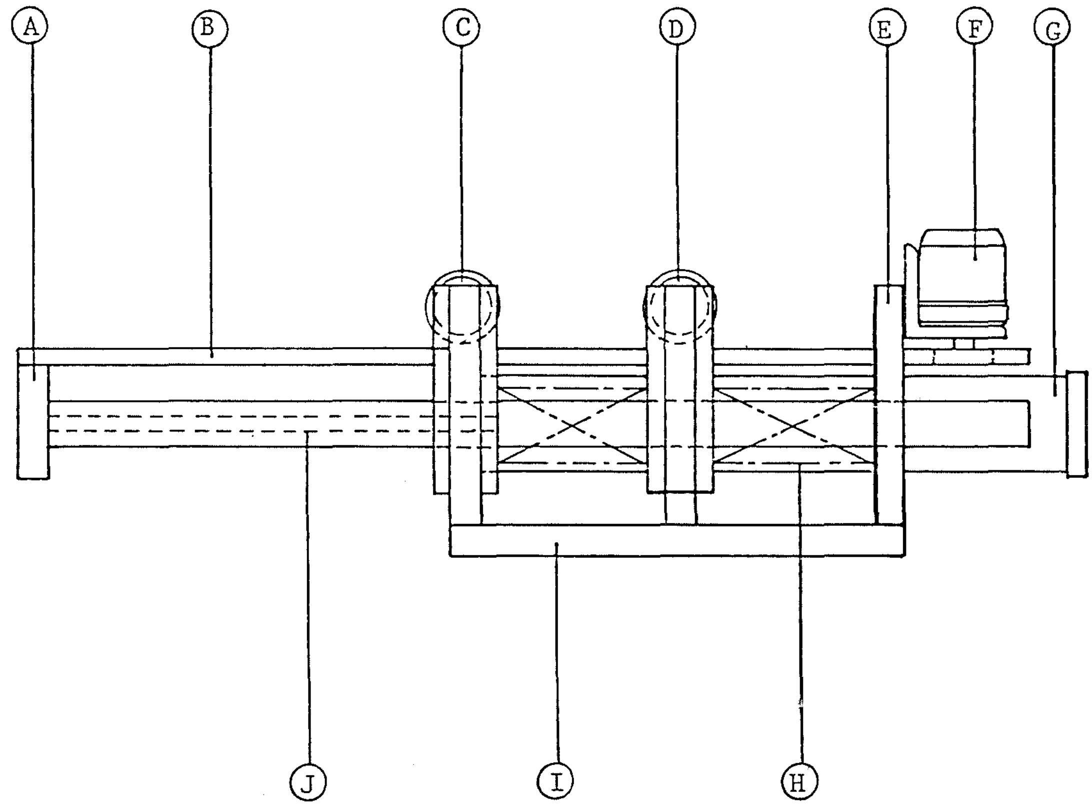


Figure (3.3.1)

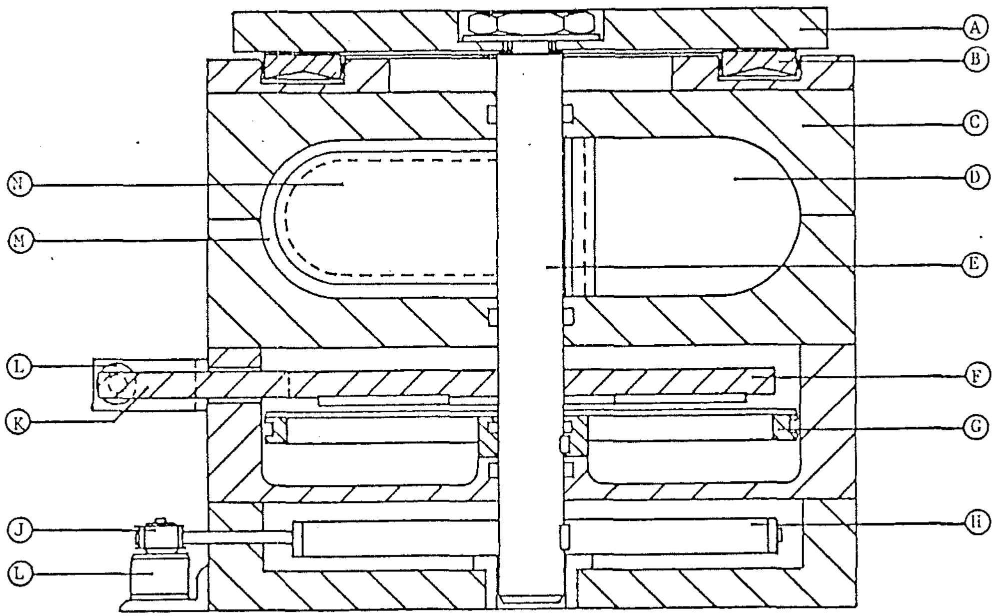
LUT Linear Module No 1



Part No	Description
A	Mounting plate
B	Rack-encoder transmission
C	Brake actuator
D	Sliding carriage actuator
E	Carriage U-Frame
F	Encoder and mounting
G	Pneumatic cylinder
H	Centralising springs
I	Base plate
J	Slideway bars

Figure (3.3.2)

LUT Rotary Module No 1



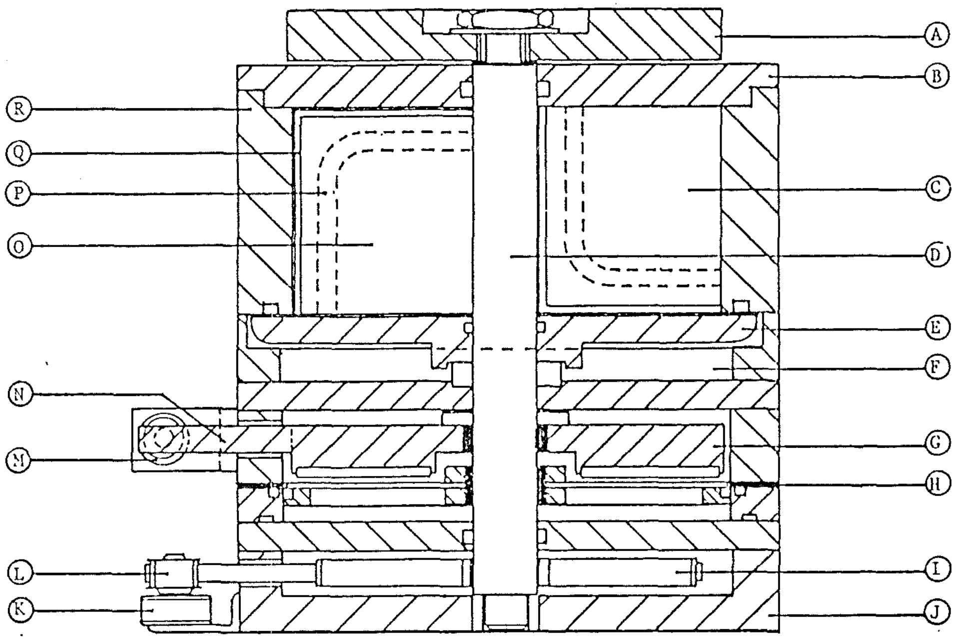
Part No	Description
A	Mounting plate
B	Brake mechanism
C	Actuator housing
D	Fixed actuator vane
E	Shaft
F	Sliding carriage and ferodo insert
G	Diaphragm
H	Encoder transmission - pulley wheel (1)
I	Encoder and mounting
J	Pulley wheel (2) and tooth belt
K	Carriage arm
L	Hydraulic damper
M	'O' ring vane seal
N	Rotating vane

Table (3.3.2)

MODULE DATA TABLE	LUT ROTARY MODULE No 1
Actuator	Oval shaped brass vane actuator in an aluminium housing ('O' ring seal)
Stroke	270 deg
Slideway	Steel shaft in phosphor bronze bushes
Transducer	Philips optical encoder - 250 p/rev
Transducer Transmission	Tooth belt and pulley wheel
Brake mechanism	Pneumatically actuated CI annular ring - positive 'on' action
Carriage frame	Aluminium
Additional features	Rotary sliding carriage with ACE hydraulic dampers actuated by a diaphragm and friction pad
Transmission	Direct acting

Figure (3.3.3)

LUT Rotary Module No 2



Part No	Description
A	Mounting plate
B	Frame top plate
C	Fixed vane of actuator
D	Shaft
E	Actuator plate
F	Brake chamber
G	Damper carriage with Ferodo insert
H	Diaphragm
I	Encoder transmission - pulley wheel (1)
J	Base plate
K	Encoder and mounting
L	Pulley wheel (2) and tooth belt
M	Hydraulic damper
N	Carriage arm
O	Rotating vane
P	'O' ring seal
Q	Phosphor bronze vane insert
R	Actuator frame

Table (3.3.3)

MODULE DATA TABLE	LUT ROTARY MODULE No 2
Actuator	Square brass vane in cylindrical aluminium housing (p/bronze insert seal)
Stroke	270 deg
Slideway	Steel shaft in p/bronze bushes
Transducer	Philip optical encoder - 250 p/rev
Transducer Transmission	Tooth belt and pulley wheels
Brake mechanism	Pneumaticly actuated annular ring
Carriage frame	Aluminium
Additional features	Sliding carriage and dampers
Transmission	Direct drive

Table (3.3.4)

MODULE DATA TABLE	M35 ARM
Actuator	Stainless steel Ø25 mm Cylinder RM8025 Ø12 piston rod
Stroke	400 mm
Slideway	Aluminium extrusion square section with stainless steel face inserts, in pre-loaded roller bearings
Transducer	Hewlett Packard HEDS 5000 - 500 p/rev 0.025 mm/pulse
Transducer Transmission	Rack and Pinion (1 rev = 50 mm) R47YG-2 AP2MS40-25
Brake Mechanism	Wrap spring on cylinder rod - positive 'off'
Carriage frame	Aluminium block
Additional features	Hydraulic dampers at the ends of stroke
Transmission	Direct Drive

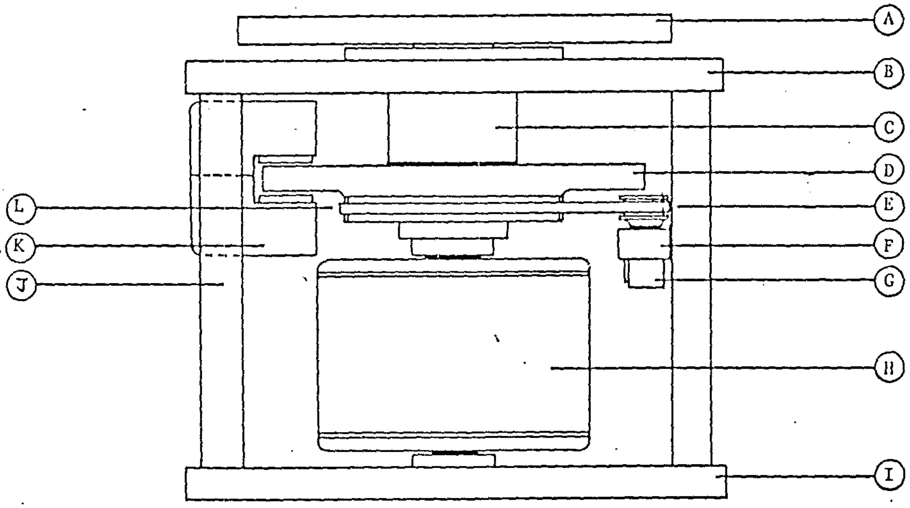
Table (3.3.5)

MODULE DATA TABLE	BOSCH 480 MODULE
Actuator	Aluminium extrusion through rod - annular Ø50 mm
Stroke	480 mm
Slideway	Twin bar hardened and ground in plain bearings
Transducer	Ferranti 24 ST - 500 p/rev 0.025 mm/pulse
Transducer Transmission	Rack and Pinion
Brake Mechanism	Pancake cylinder clamp
Carriage frame	Aluminium block
Additional features	Telescopic air feeds in s/s pipe and proximity sensor and connections
Transmission	Direct Drive

Plate (3.3.4) M35 Arm Module

Figure (3.3.6)

LUT Rotary Module No 3



Part No	Description
A	Mounting plate
B	Top plate
C	Drive coupling
D	Brake disc
E	Encoder Transmission - pulley wheel (1)
F	Encoder Mounting block
G	Incremental Encoder
H	SMC vane actuator
I	Base plate
J	Support bars
K	Caliper brake mechanism
L	Encoder Transmission - pulley wheel (2) and tooth belt

Table (3.3.6)

MODULE DATA TABLE	LUT ROTARY MODULE No 3
Actuator	SMC rotary vane actuator CRB80
Stroke	270^{o}
Slideway	Ball races SKF 16003 SKF 7210B/7208B
Transducer	Hewlett Packard HEDS-5000 500 p/rev 0.045 deg/pulse
Transducer Transmission	Tooth belt and DAVAL Pulley wheels 18XL037 72XL037
Brake Mechanism	Disc Brake GA5001
Carriage frame	Aluminium and BDS
Additional features	Aluminium face plate
Transmission	Direct drive

Plate (3.3.6) Rotary Module No.3

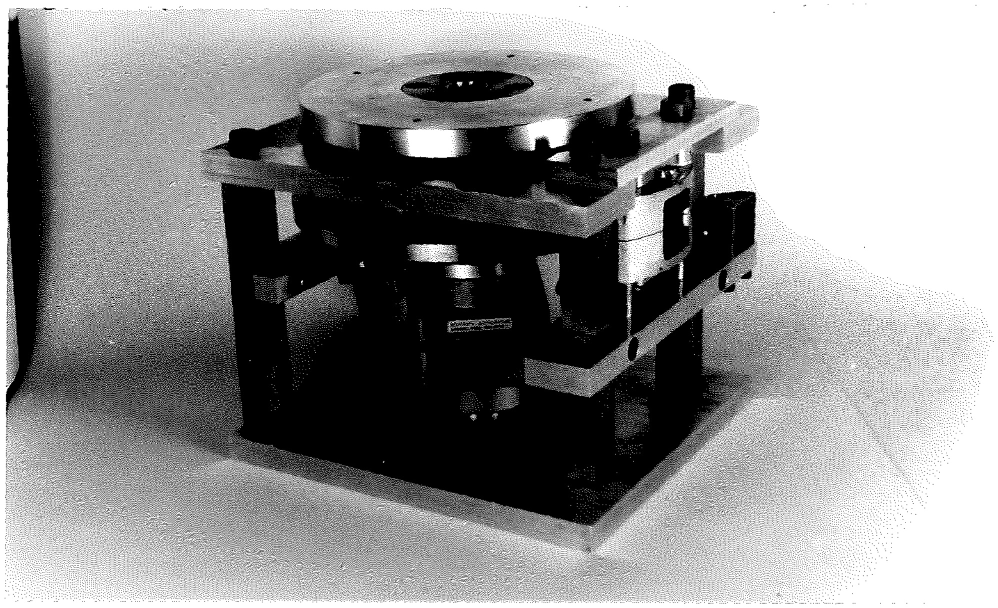
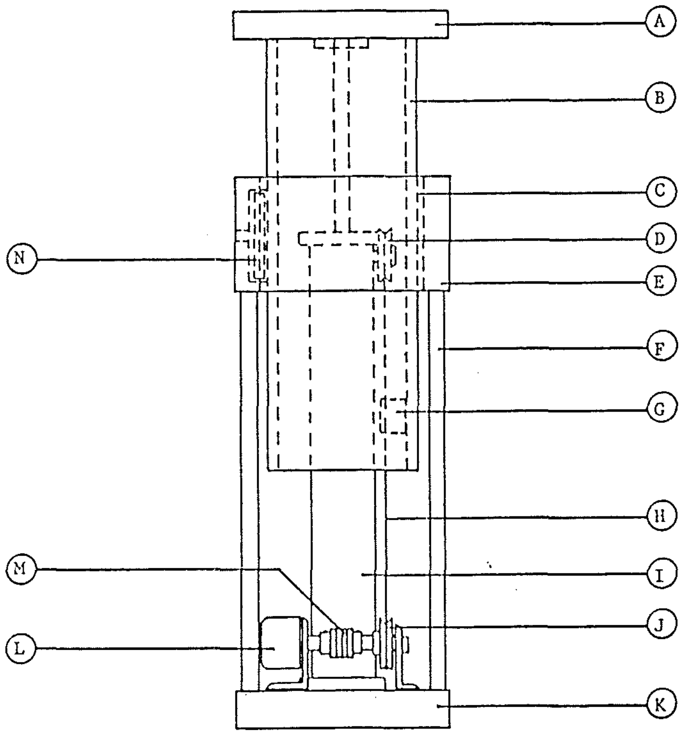


Figure (3.3.7)

LUT Linear Module No 2



Part No	Description
A	Mounting plate
B	Tubular slideway
C	Glacier bearing
D	Encoder pulley
E	Bearing block
F	Slideway tie bars
G	Encoder cable connector
H	Encoder cable
I	Actuator
J	Encoder pulley
K	Base plate
L	Ferranti encoder
M	Flexible coupling
N	Brake mechanism

Table (3.3.7)

MODULE DATA TABLE	LUT LINEAR MODULE No 2
Actuator	Martonair Cylinder Ø50 mm
Stroke	500 mm
Slideway	Mild steel tube 125 mm O/D in Glacier plain graphite bearing
Transducer	Ferranti 24 ST - 500 p/rev 0.032 mm/pulse
Transducer Transmission	Capstan pulley wheels
Brake mechanism	Wedge action, pneumatically actuated with Ferodo liner
Carriage frame	Aluminium block and tie rods
Additional features	Actuator mounted within the tubular slideway
Transmission	Direct drive

Table (3.3.8)

MODULE DATA TABLE	GANTRY MODULE M/60400
Actuator	Aluminium extrusions, Ø50 mm
Stroke	1000 mm
Slideway	Aluminium extrusion with s/s face plates and plastic bearing inserts
Transducer	Hewlett Packard HEDS 5000 500 p/rev 0.028 mm/pulse
Transducer Transmission	Fixed tooth belt and pulley wheel pinion
Brake mechanism	Direct acting pancake cylinde
Carriage frame	Aluminium
Additional features	Inductive proximity sensors, hydraulic dampers at stroke ends
Transmission	Direct drive via cable from piston to carriage

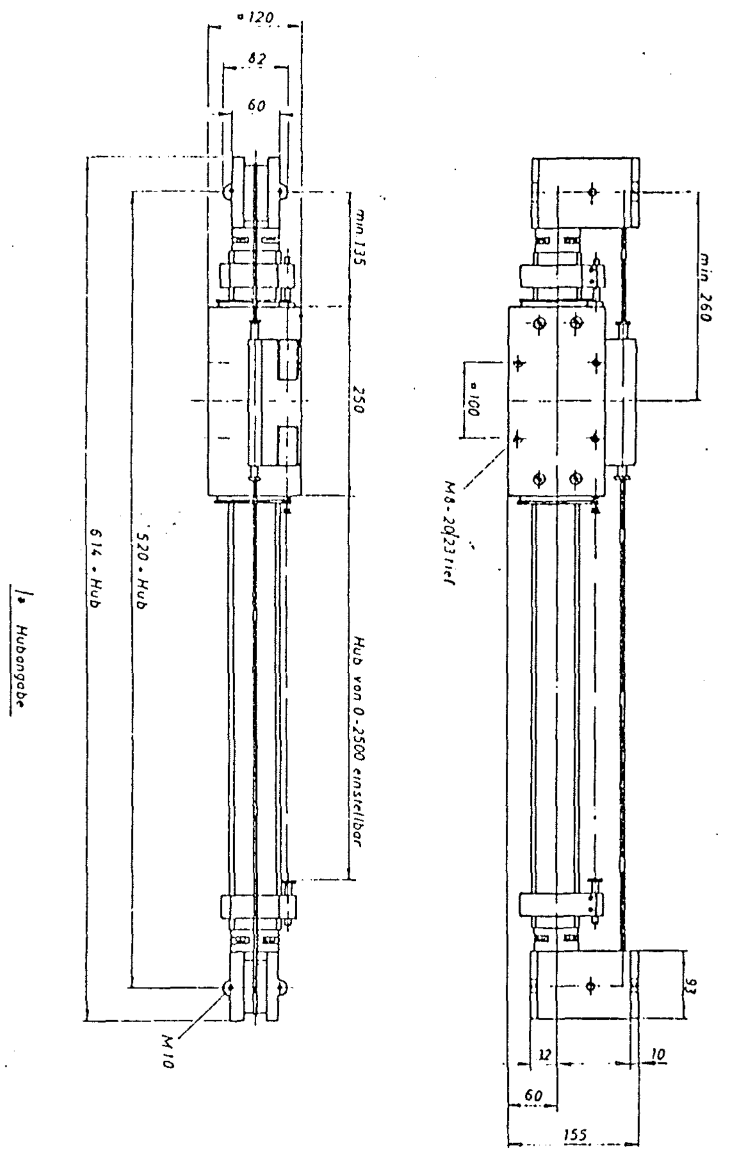


Figure (3.3.8) Gantry Module

APPENDIX (8.0) CONTROL STRATEGY DESCRIPTION
- MILCOM VERSION 2.50

8.7.1. MILCOM VERSION 2.50 - CONTROL STRATEGY DESCRIPTION

The control strategy is detailed in Figures [8.7.2] and [8.7.5]. Figure [8.7.2] shows the functional elements of the realtime control, and the detailed description of the software is given in flowchart form in Figure [8.7.5]. The relationship between the control system gains as a function of error is shown in Figure [8.7.3], from these it can be seen that the strategy involves the use of distinct control regimes. The control regime used is a function of the error in position. The limits of individual control regimes are determined by control parameters and saturation levels.

The control strategy within each control zone is described by equations (8.6.4.1) through (8.6.4.5). The control zones are defined as;

Zone (1) : DEAD-BAND

In control zone (1) where $/(Yd-Y)/ < /Edb/$

then $Cs = Cnull$

When the absolute position error, $/(Yd-Y)/$, is less than $/Edb/$ the system is defined as 'in-position', namely it has achieved its setpoint (Yd) within the specified tolerance. The 'deadband' is symmetrical about the setpoint.

Whilst the module remains within the 'dead-band' of the setpoint the valve is kept at null, but if any drift in position occurs realtime control will compensate for this drift and move the module 'in-position'.

Zone (2) : CONSTANT GAINS FOR ALL FEEDBACK LOOPS

In control zone (2) where:

$$Edb <= /(Yd-Y)/ < Ecg$$

then the command signal can be described by,

$$Cs = Kp_{1} (Yd-Y) - KvV - KaA + Cos + Cnull$$

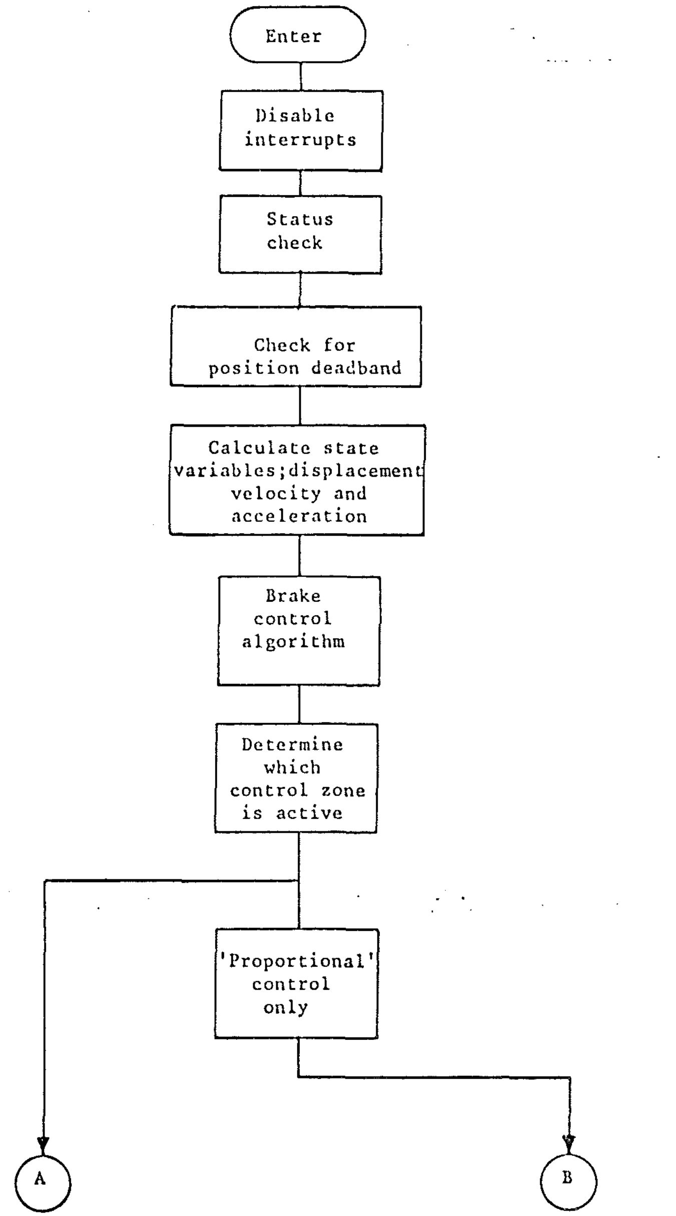


Figure (8.7.2) Sequential Flow Diagram for the Real Time Control

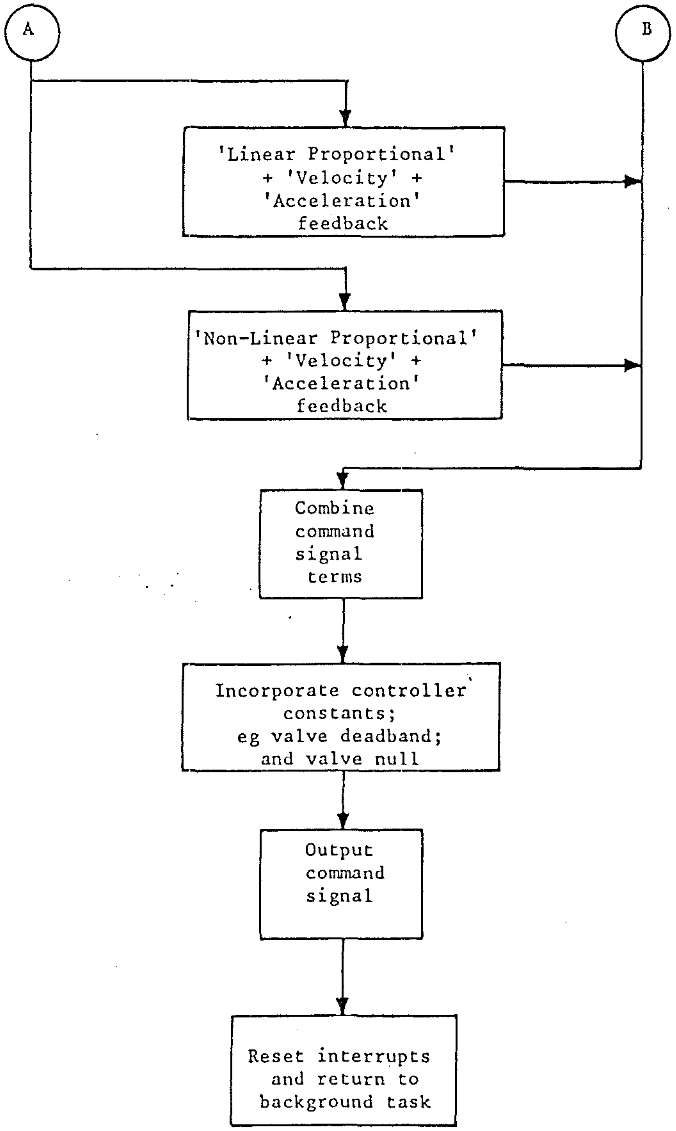
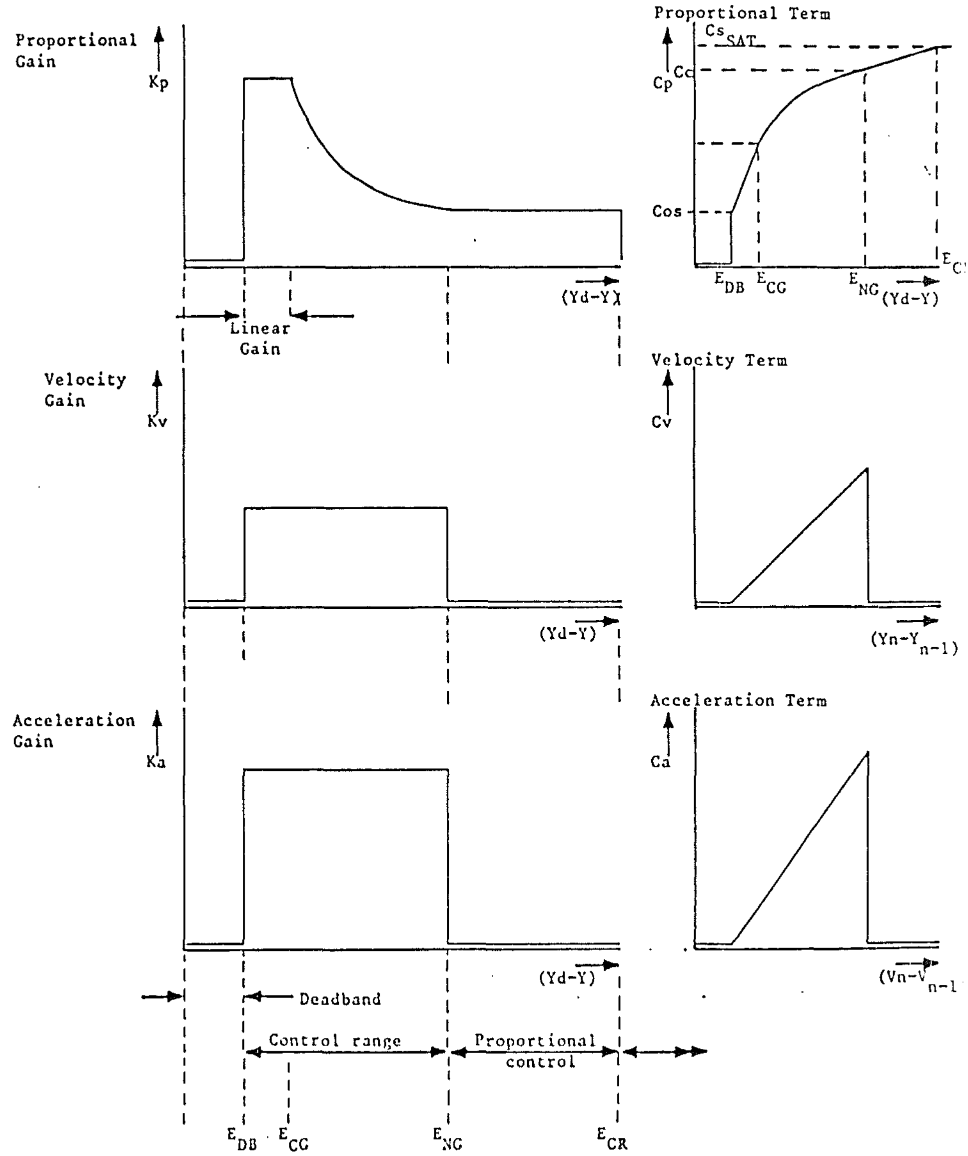


Figure (8.7.2) continued

Figure (8.7.3) Control Gains - Milcom Version 2.50



The position error has a high value of gain Kp(1), the gain having a constant value throughout this control zone. The proportional gain can take a high value in the region, due to the stabilising effect introduced by the velocity and acceleration feedback terms, and will demonstrate a linear relationship with displacement from the setpoint.

The limits of the control zone are defined by the control parameters DEBAND and LSTCNT in counter pulses, where DEBAND defines the inner limit relative to the setpoint, and LSTCNT defines the outer limit.

The proportional gain can take a different value for each direction of move, the gain being selected using the control parameters KLGPOS and KLGNEG for positive and negative moves respectively. The velocity gain (Kv) is selected using the control parameter KVGAIN, the gain being equal for both positive and negative moves. The acceleration gain (Ka) is selected using the control parameter KAGAIN, the gain being equal for both positive and negative moves.

ZONE (3) : NON-LINEAR PROPORTIONAL TERM WITH LINEAR MINOR LOOP GAINS

In control zone (3) where

$$Ecg \leq /(Yd-Y)/ < Eng$$

then the command signal can be described by,

$$Cs = Kp_{nl}\ (Yd-Y) - KvV - KaA + Cos + Cnull$$

Control zone (3) is similar to control zone (2) but the proportional gain is non-linear; this produces a non-linear relationship between position error and the proportional term. The use of a non-linear gain Kp(nl) allows a high gain close to the setpoint, producing a high static stiffness, but it allows a reduced gain for large errors.

The non-linear gain Kp(nl) is used to generate a proportional term contour or look-up table. The ratio of the position error (Yd-Y) to the active range of control zone (3) (Eng-Ecg), determines the proportional term (Cp).

Dependant upon the selected magnitude of the proportional term, values are accessed from the contour and scaled. The scaling is set with the control parameters MNGCTP and MNGCTN for the positive and negative moves respectively. The active range of this control zone is determined by the control parameters CNTRNG and LSTCNT, set in counter pulses.

The contour currently implemented approximates to a segment of a quantised circular function. As the circular function approaches the 'null' a tangent to its circumference will approximate to a quantised linear function with a gain dependant upon the radius of the circle defined (see Figure [8.7.4]).

The magnitude of the contour can also be altered. Thus, once the desired Cp contour has been defined, quantised and stored in memory, the gain of the proportional loop can be adjusted, independent of the shape of the contour. Thus it is possible to match the requirements of a given set of valve, actuator, slideway and load conditions.

ZONE (4) : PROPORTIONAL POSITION CONTROL

In control zone (4), where

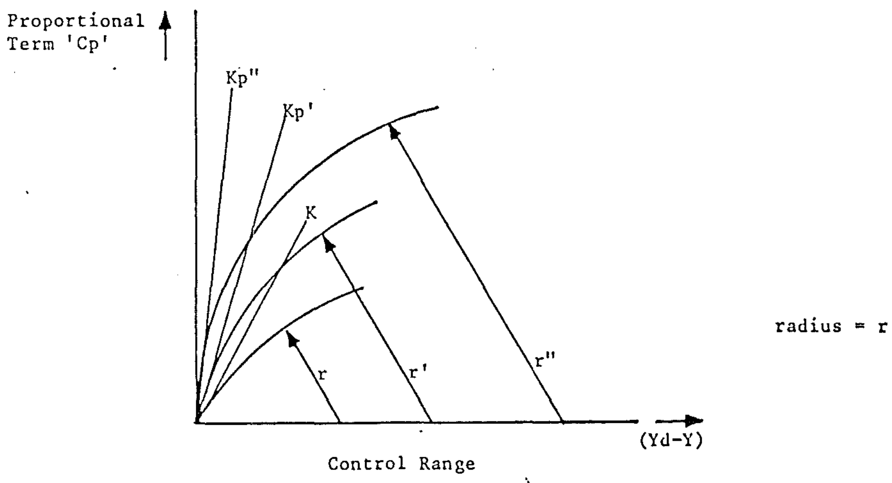
$$Eng \leq /(Yd-Y)/ < Ecr,$$

the command signal can be described by

$$Cs = Kp(Yd-Y) + Cnull + Cc$$

In control zone (4) position loop feedback only is present. The proportional error gain Kp is selected using the control parameters KGNPOS and KGNNEG, which set the proportional gain for positive and negative moves

Figure (8.7.4) Influence of Proportional Term Contour Scaling on the Absolute Gain at the Setpoint



If r" > r' > r

.'. Kp" > Kp' > Kp

where r is determined by the contour scaling factor (CSF) which is set using the control parameters 'MNGCTP' and 'MNGCTN' for the positive and negative halves.

respectively where the chosen values of gain remain constant.

It can be seen that the command signal includes a controller constant (Cc). This constant is selected using the control parameter PROPST, and is set at an absolute value within the DAC range. This term is necessary to prevent a controller output discontinuity in the transition between control zones (3) and (4).

ZONE (5) : SATURATION

In control zone (5), where

/(Yd-Y)/ >= Ecr ,

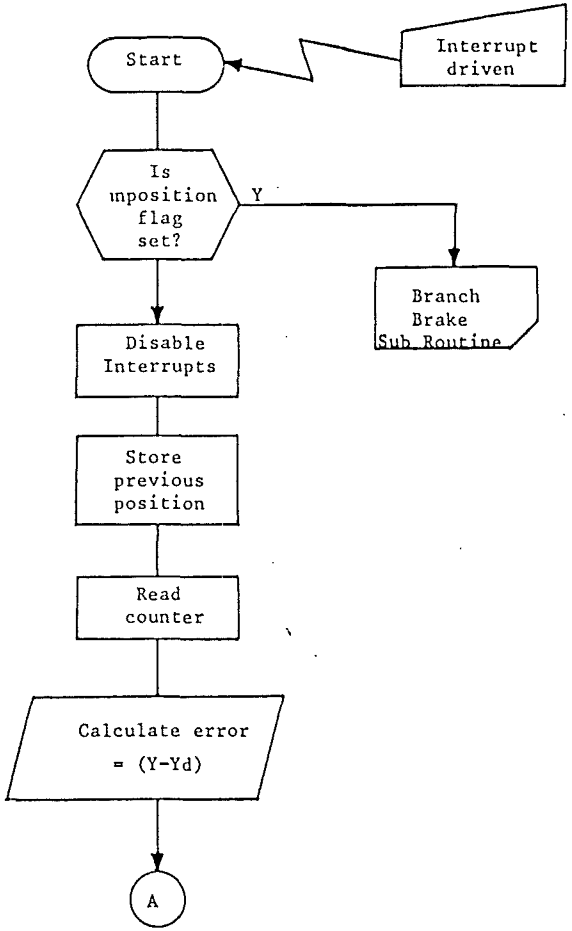
the command signal can be described by

Cs = Csat

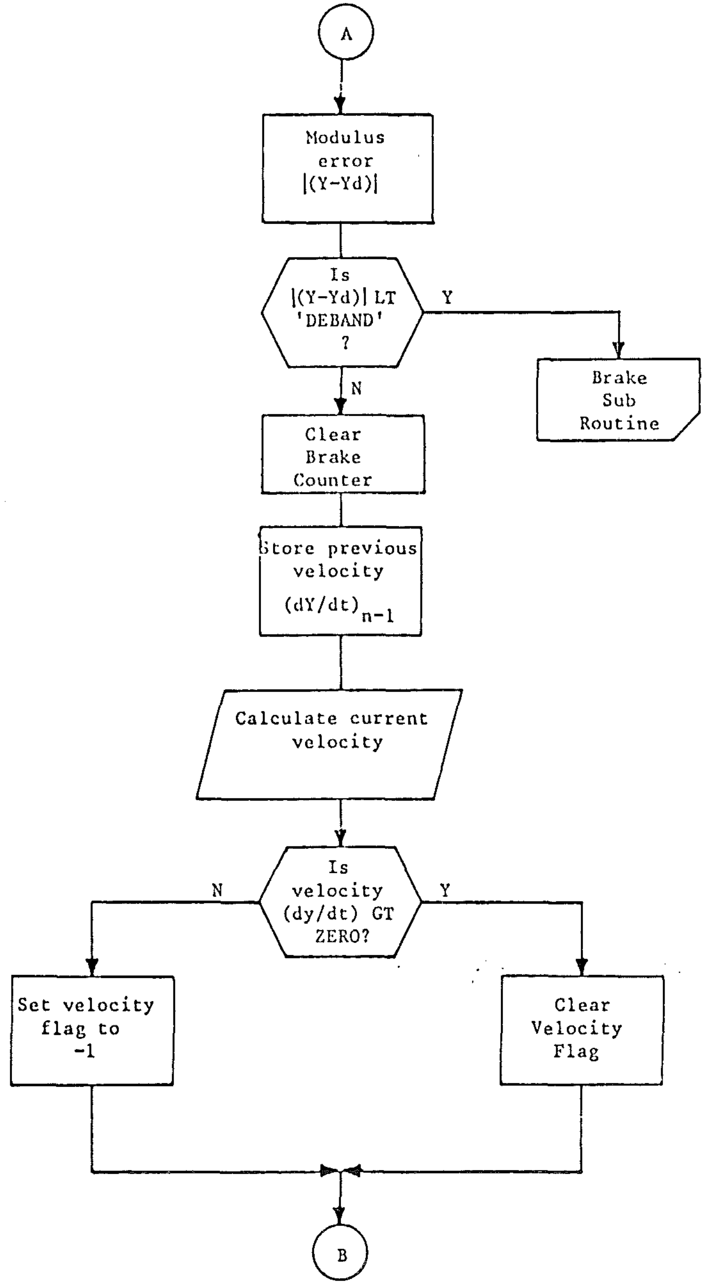
The command signal is a constant, Csat, which determines the upper limit of the command signal in either direction of motion. This constant is selected using the control parameters SATPOS and SATNEG which set the saturation levels for positive and negative moves respectively.

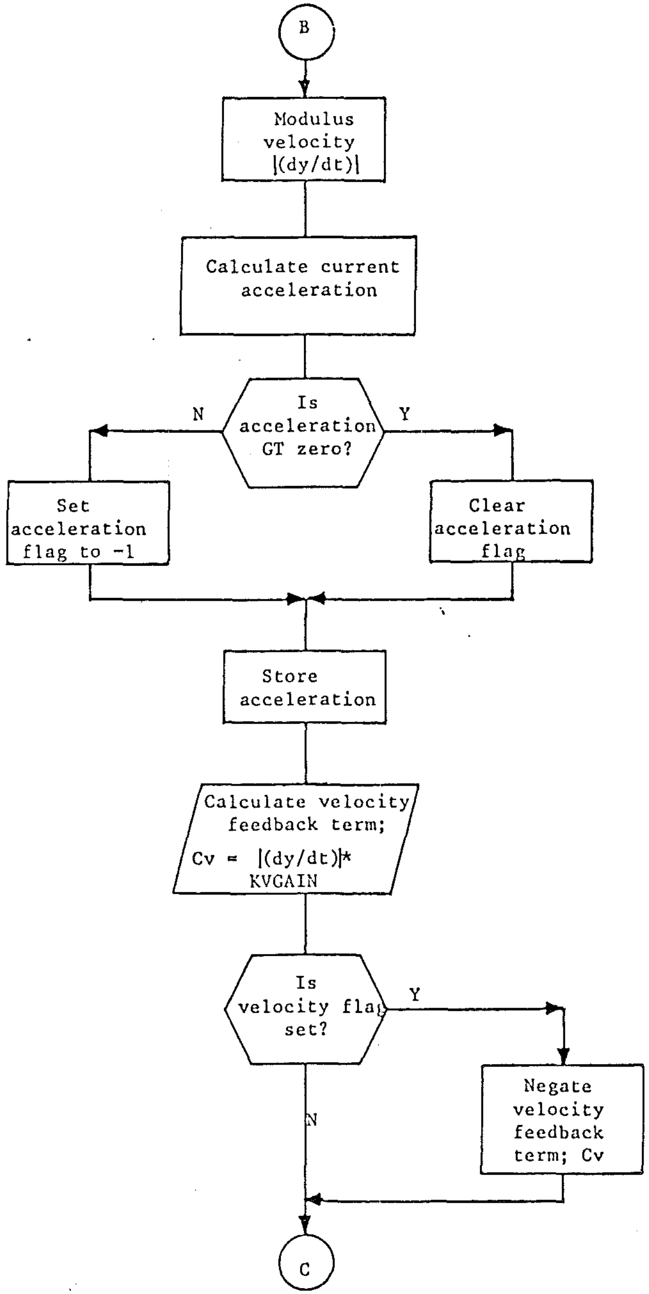
This control strategy was fully evaluated on a prototype linear module under load.

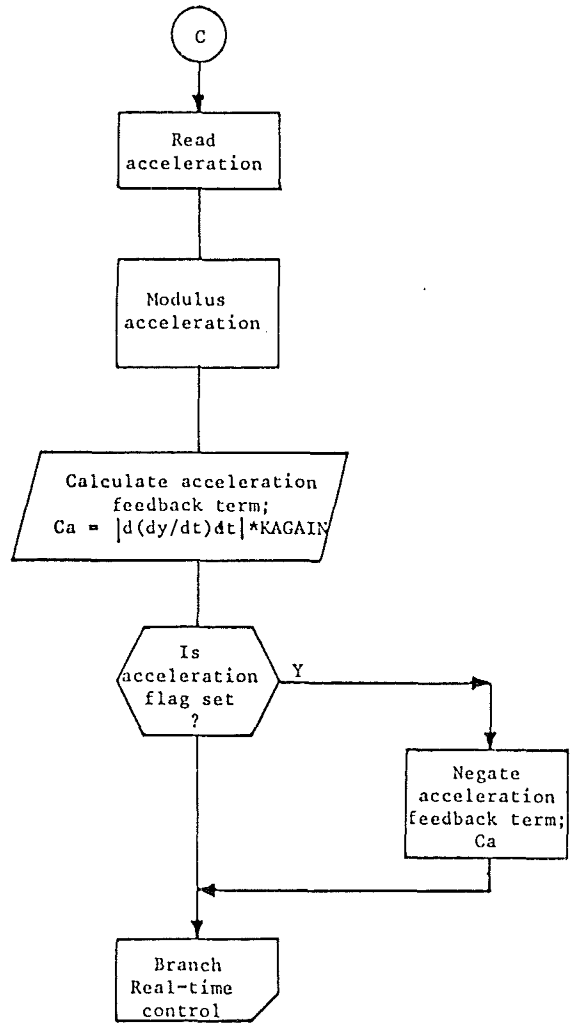
APPENDIX (8.1) CONTROL STRATEGY FLOWCHART
- MILCOM VERSION 2.50
Figure (8.7.5)

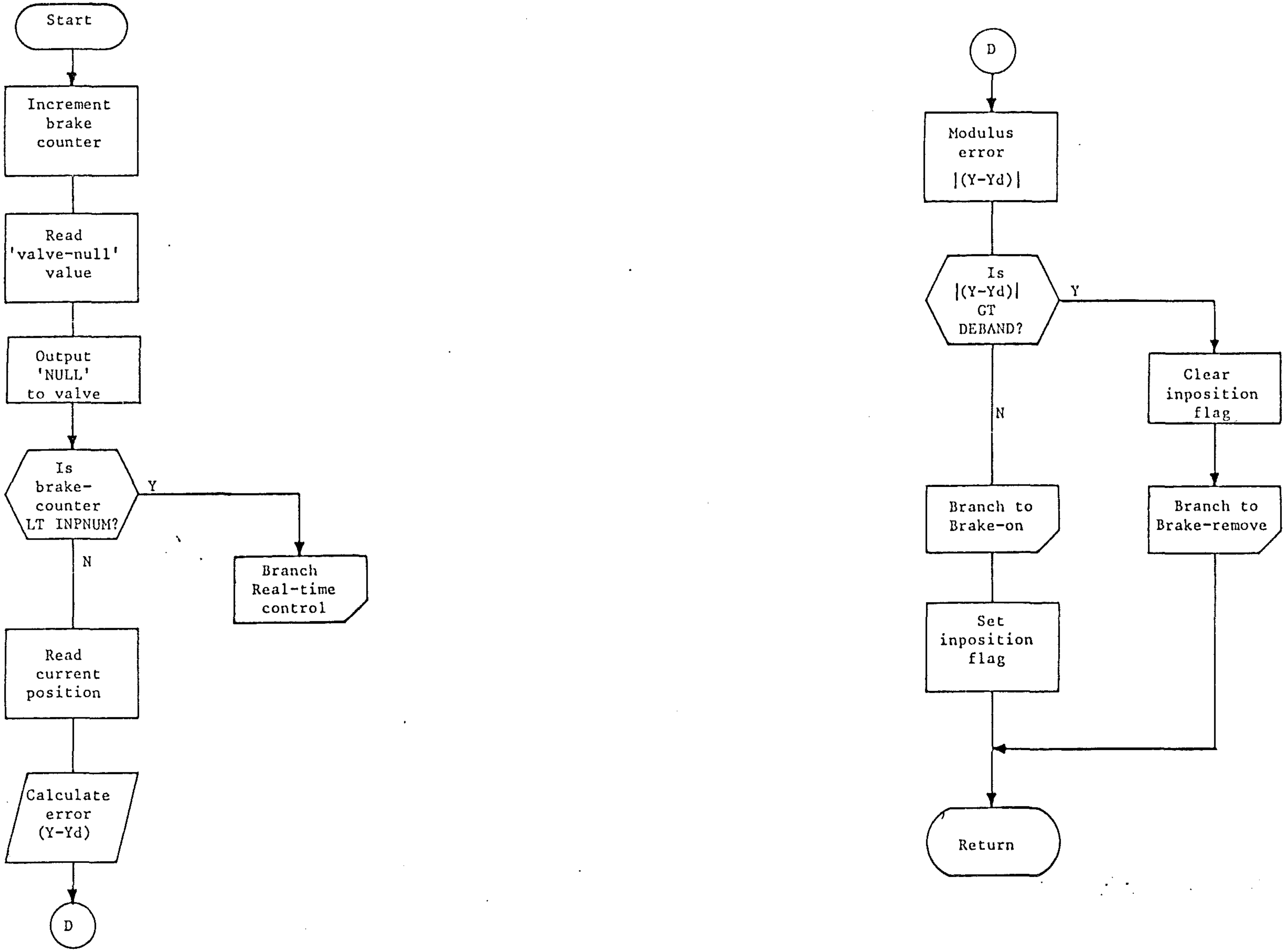


Real Time Control Flowchart - Milcom Version 2.50

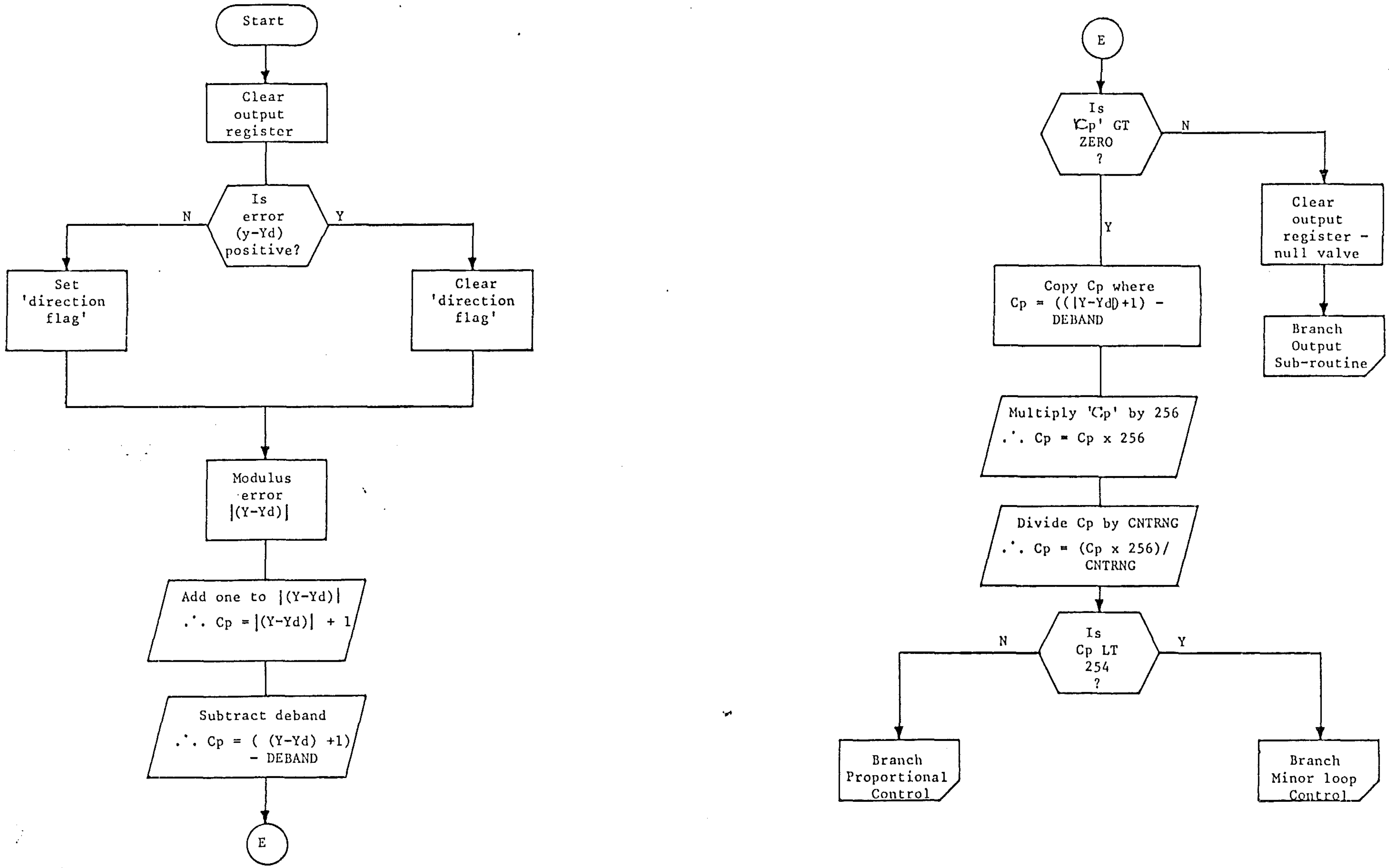



B
Modulus velocity |(dy/dt)|
Calculate current acceleration
Is acceleration GT zero?
N
Y
Set acceleration flag to -1
Clear acceleration flag
Store acceleration
Calculate velocity feedback term; Cv = |(dy/dt)|* KVGAIN
Is velocity flag set?
Y
N
Negate velocity feedback term; Cv
C

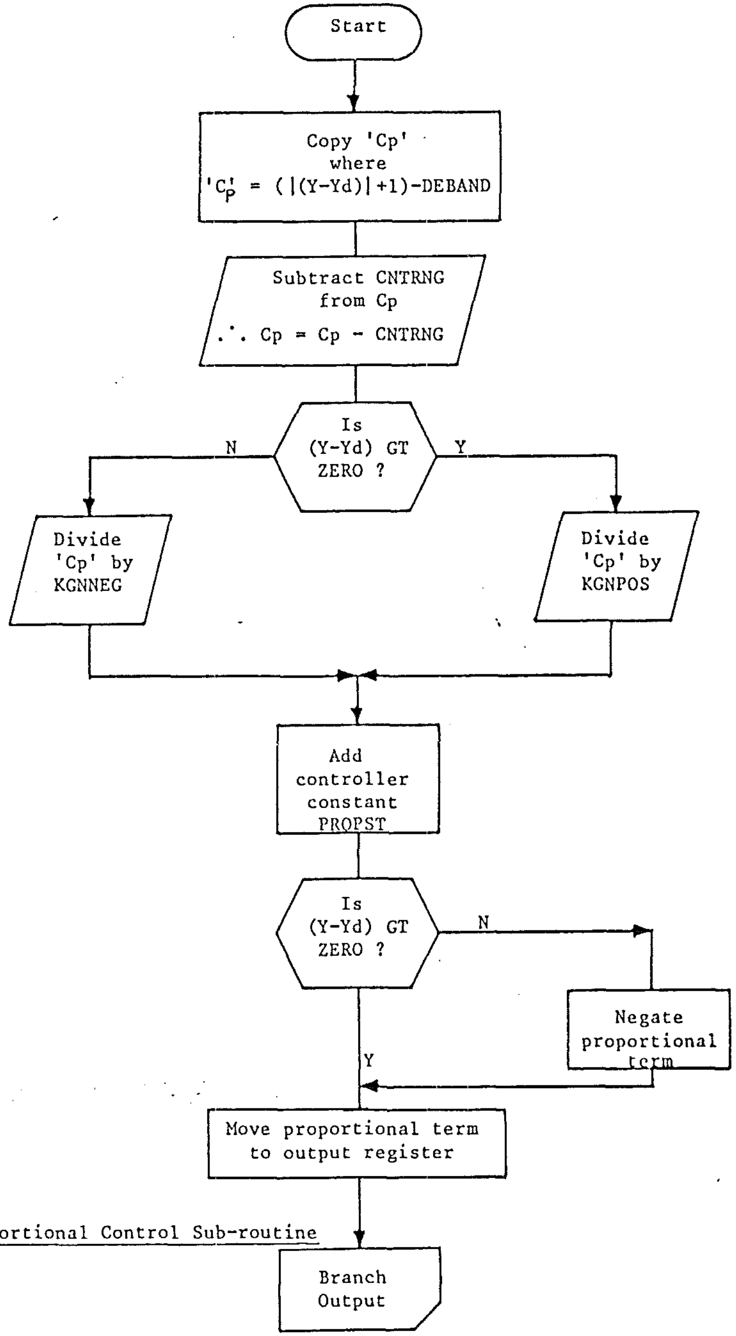

C
Read acceleration
Modulus acceleration
Calculate acceleration feedback term; Ca = |d(dy/dt)dt|*KAGAIN
Is acceleration flag set ?
Y
Negate acceleration feedback term; Ca
Branch Real-time control



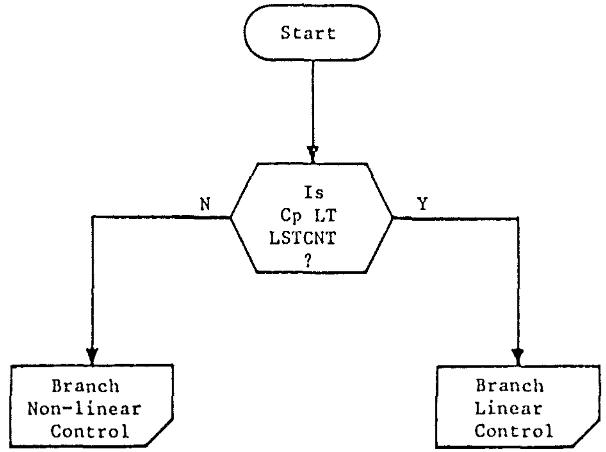
Brake Sub-routine



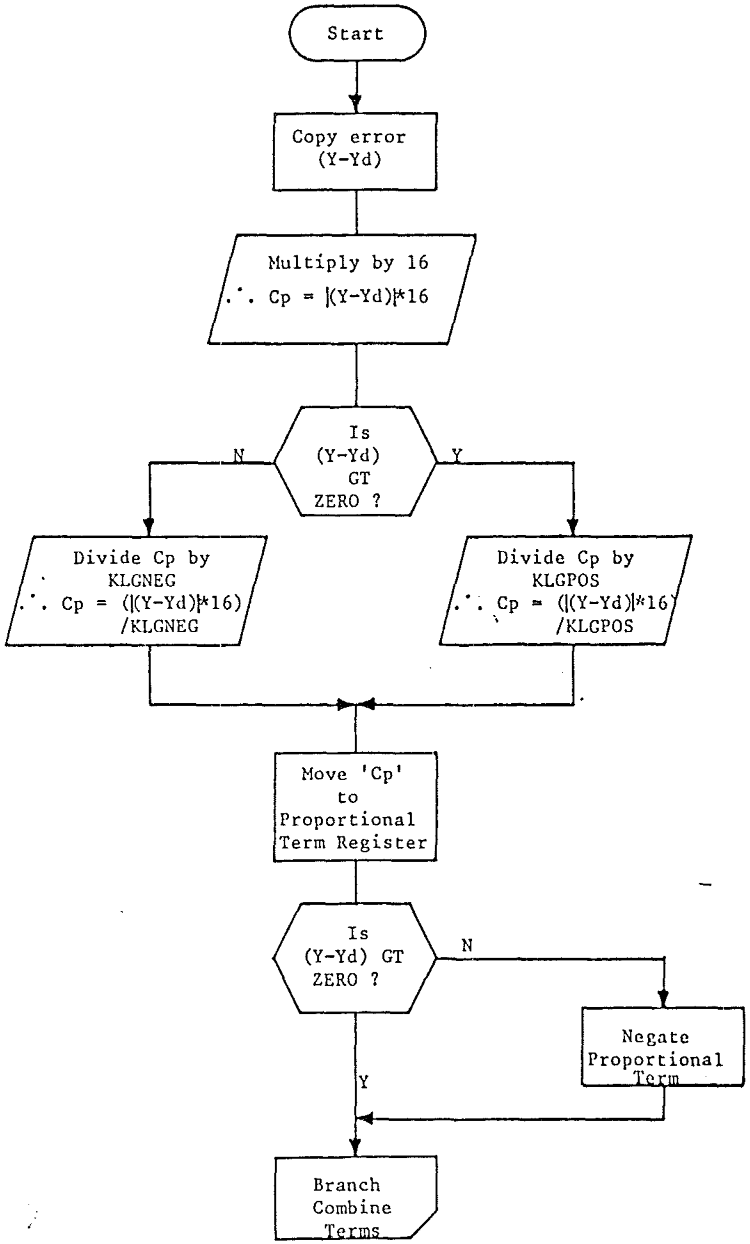
Realtime Control



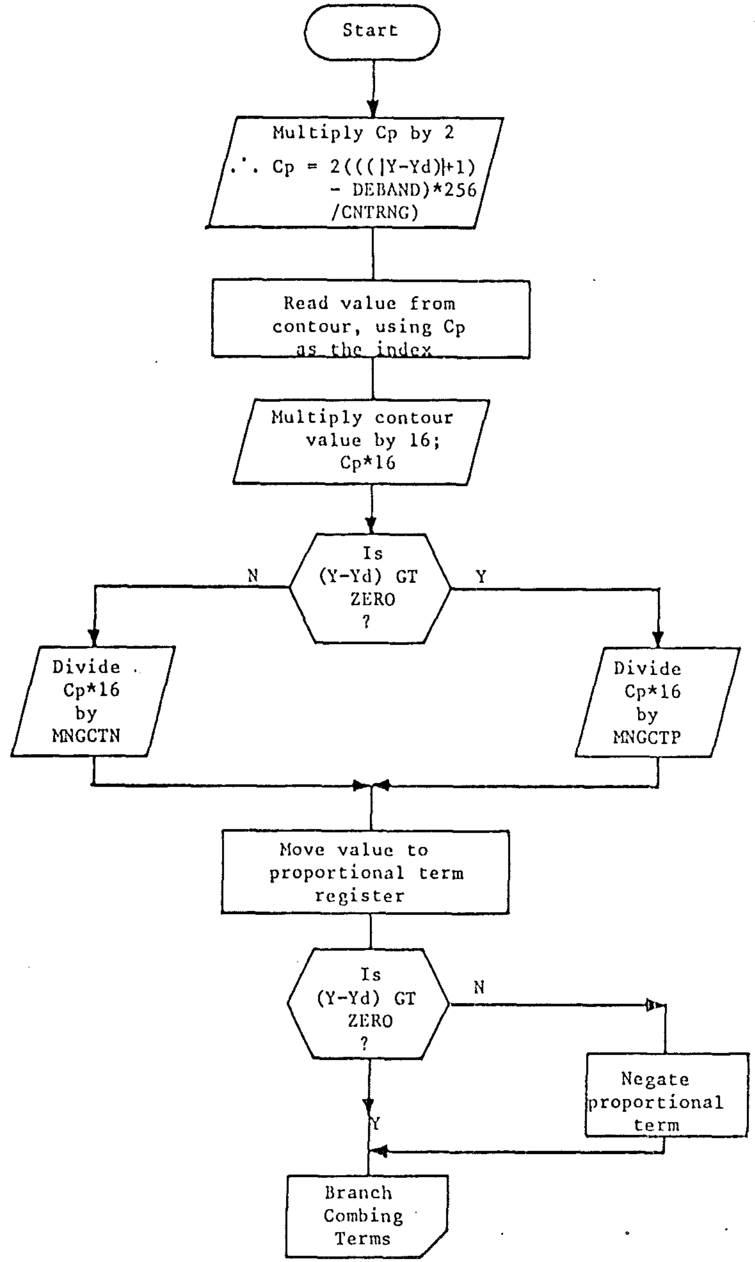
Proportional Control Sub-routine



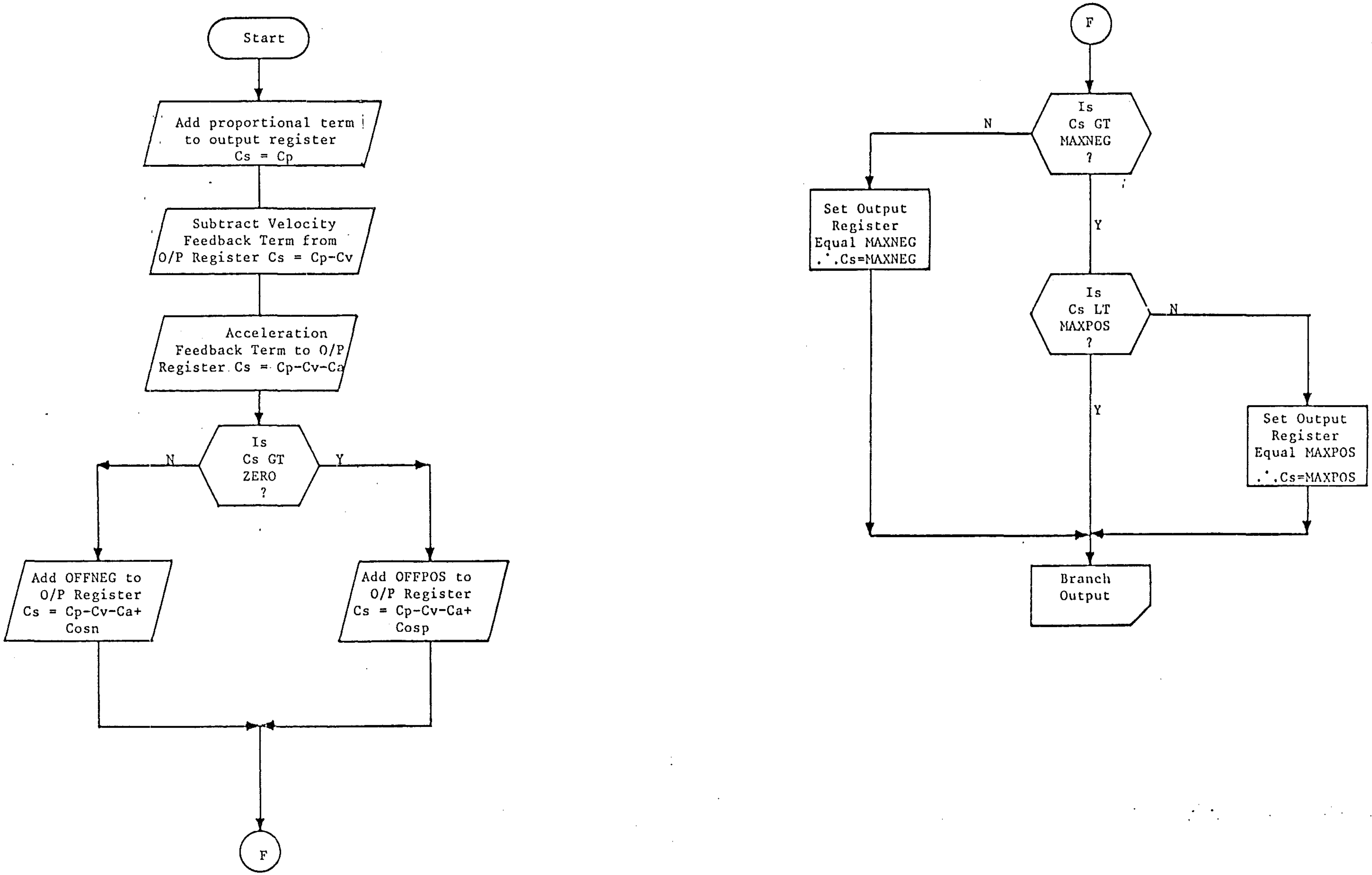
Minor Loop Control



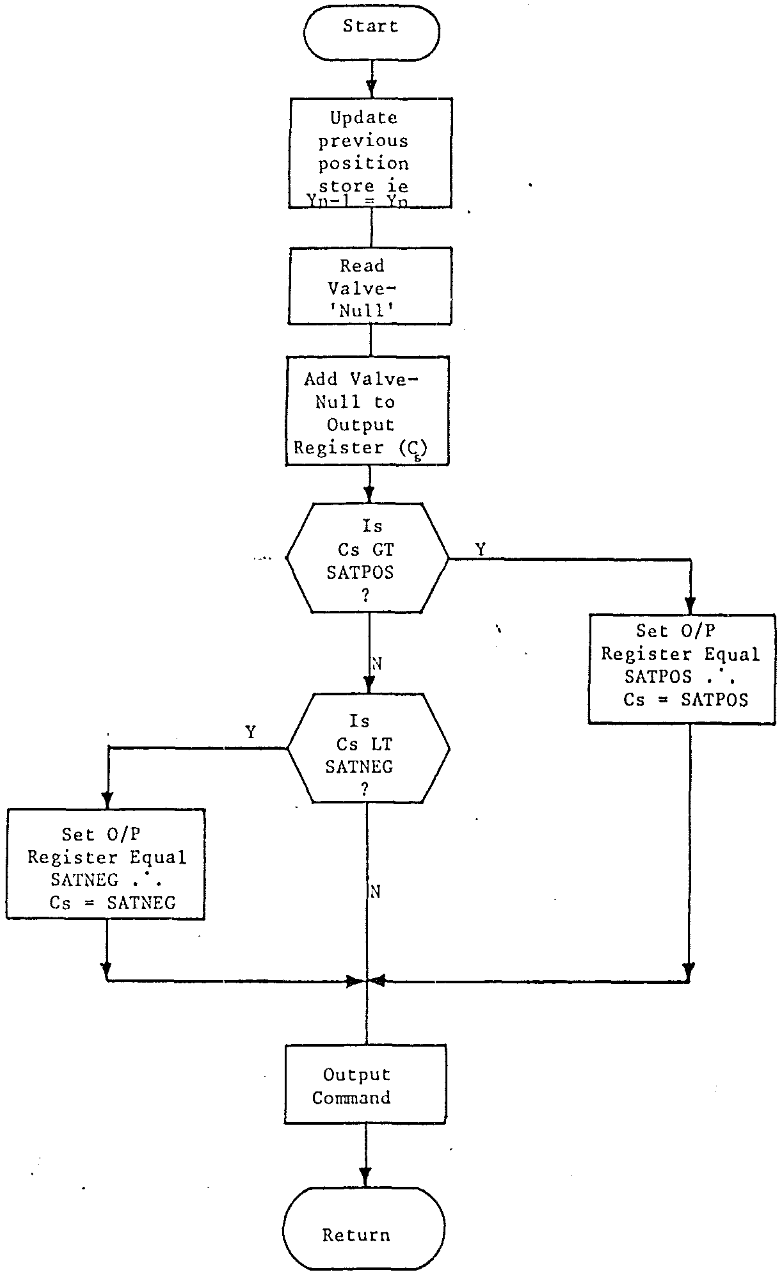
Linear Control Sub-routine



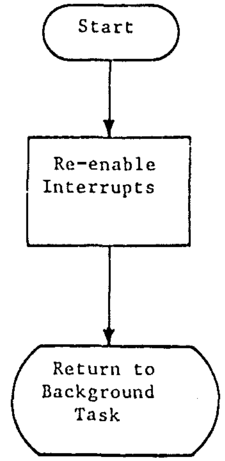
Non-linear Control Sub-routine



Combine Terms



<u>Output Sub-routine</u>



<u>Return Sub-routine</u>

APPENDIX (8.2) CONTROL PARAMETER TUNING PROCEDURE

Figure (8.7.5.1) Control Parameter Tuning Procedure

- reference Milcom Version 2.50

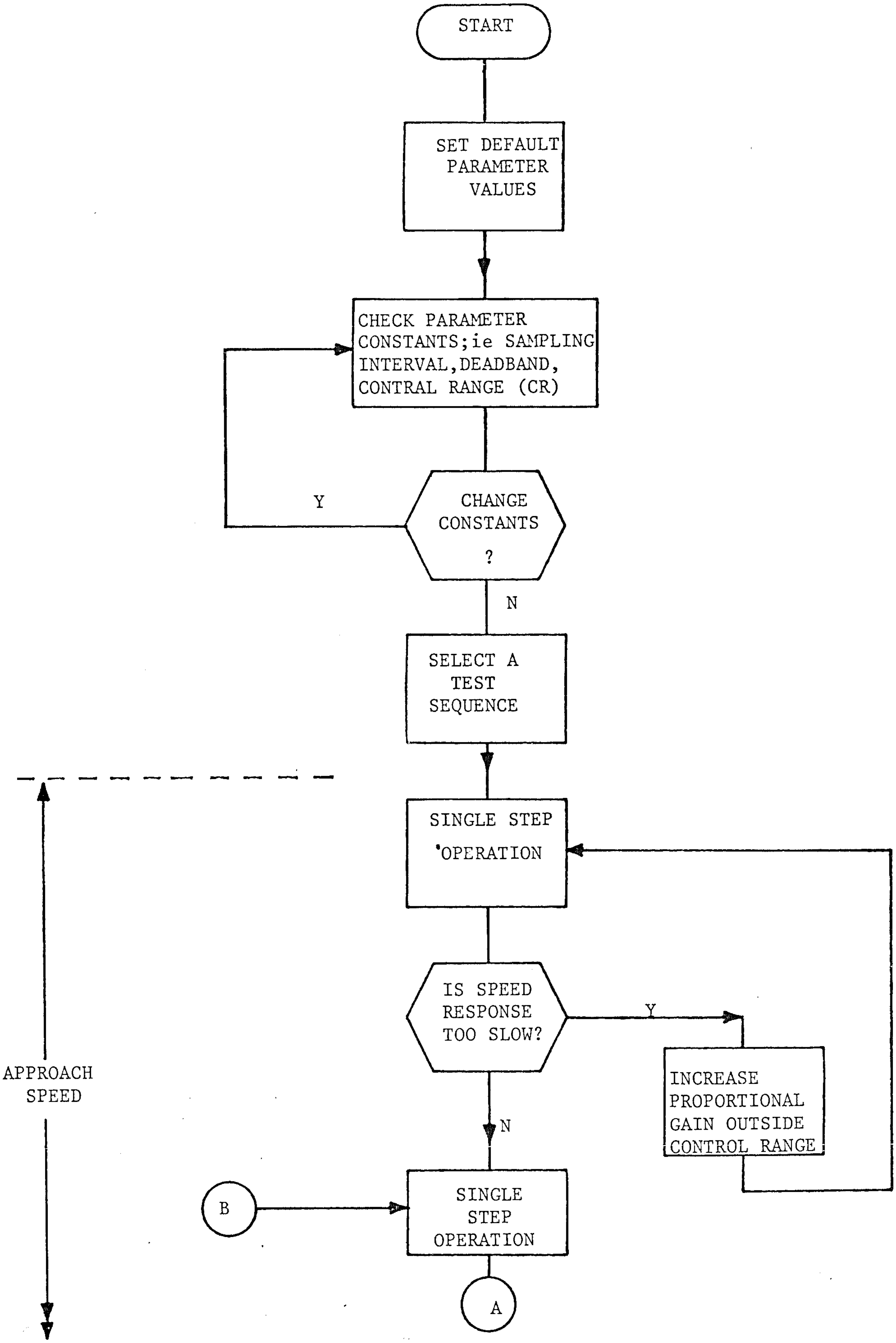
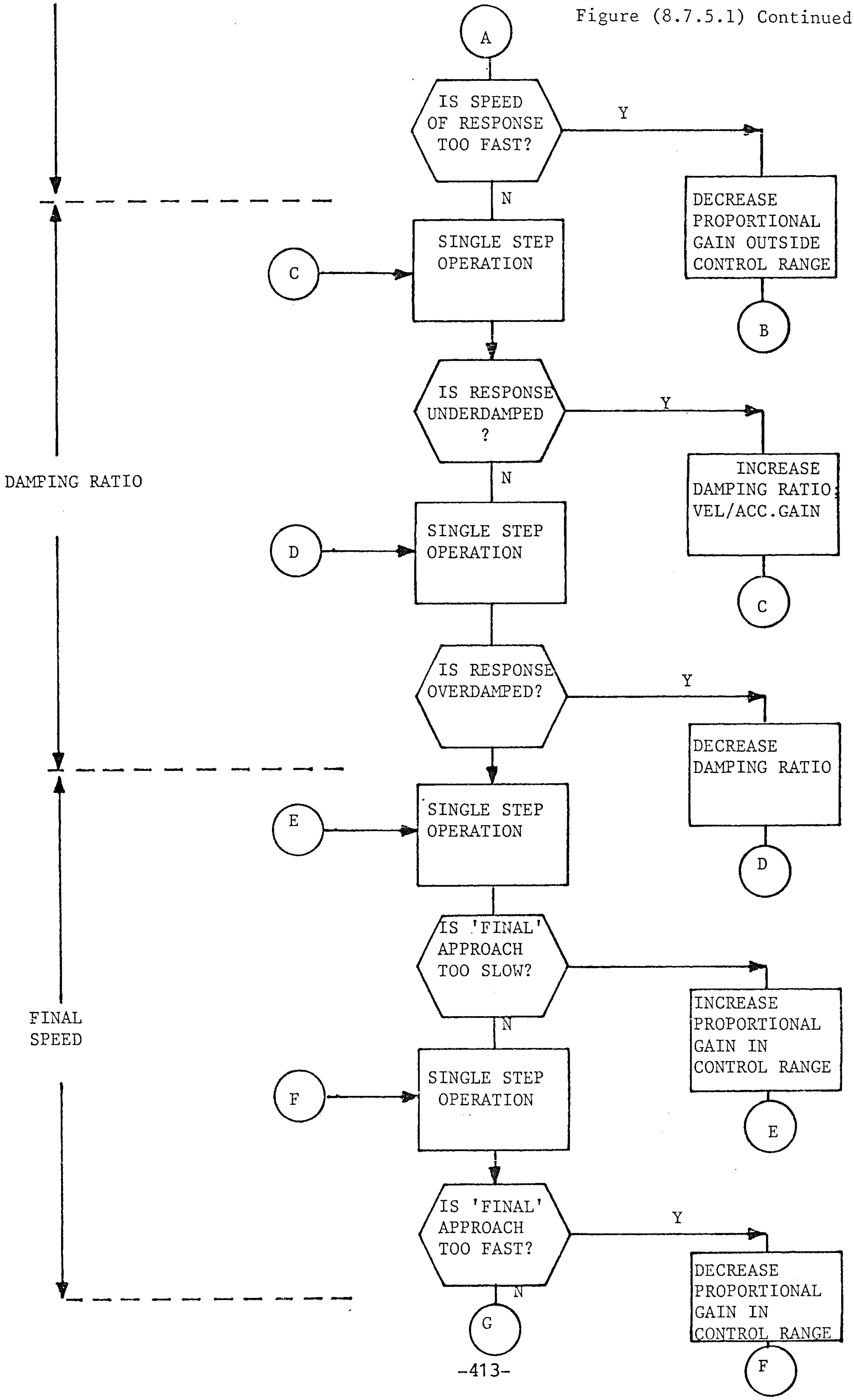
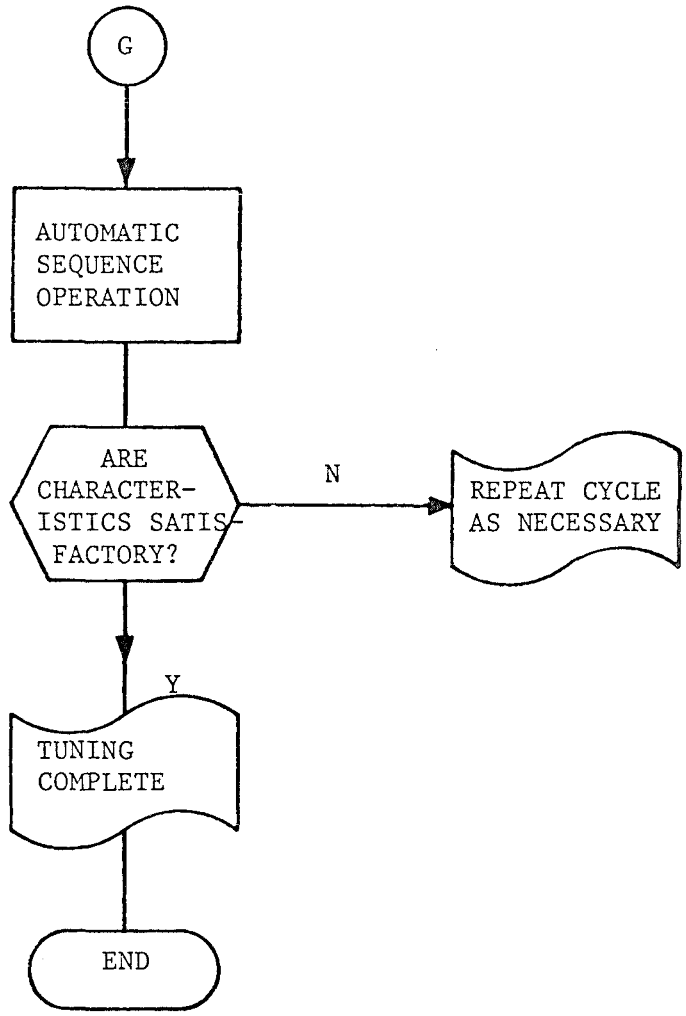


Figure (8.7.5.1) Continued





N.B.: To successfully 'tune' control parameters for variation in 'module' characteristics it is simply necessary to select a suitable 'Approach Speed', a 'Damping Ratio' and a 'Final Speed'.