Development of design procedures for reciprocating mechanisms using hydraulic actuators controlled by rotary valves and a performance evaluation of actuator designs used in tufting machines

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DEVELOPMENT OF DESIGN PROCEDURES FOR RECIPROCATING MECHANISMS USING HYDRAULIC ACTUATORS CONTROLLED BY ROTARY VALVES AND A PERFORMANCE EVALUATION OF ACTUATOR DESIGNS USED IN TUFTING MACHINES

by

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A Doctoral Thesis

Submitted in partial fulfilment of the requirements for the award of the Degree of Doctor of Philosophy of Loughborough University of Technology, U.K.

July 1982

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SYNOPSIS

The presented work represents a continuation of the investigations into the possible application of miniature hydraulic actuation and control techniques to high speed reciprocating mechanisms in manipulative machinery. Earlier research in this area led to the design of knitting, sewing and tufting machines which employed actuators and rotary valves developed with the aim of exploiting some of the advantages that hydraulic mechanisms have over their mechanical equivalents. Tests performed on these machines indicated that more benefits could be derived if:

a) Procedures could be established for designing reciprocating mechanisms using hydraulic actuators controlled by rotary valves. These procedures could then be used to design a valve-actuator system to meet any specified dynamic performance requirement dictated by the operational requirements of a machine.

b) Methods of programming hydraulically actuated mechanisms were investigated so as to provide complete control over individual, or groups of actuators used in manipulative machines, for example to enable production of wide ranging patterns in textile fabric producing machines.

These are the main areas of investigation covered by this work. It involved:

i) Advancing by computer aided techniques, the design procedures for rotary valve-hydraulic actuator combinations to meet specific dynamic requirements, and
ii) Investigations into methods of sequential programming of hydraulic actuator positions and the development of techniques for interfacing the programming system with the hydraulically actuated mechanism.

In order to illustrate the significance of the variables involved in the design of high speed miniature hydraulic actuators for a specific machinery application and to obtain a practical performance evaluation of such mechanisms, the developed techniques were used as a basis for the design of a needle position control system which would enable a tufting machine to produce a wide range of patterned fabric. The work is presented in five main sections:

SECTION ONE:

A review of earlier work on rotary valves and miniature high speed reciprocating actuators. Research work in this field has been going on since 1968 in the Department of Mechanical Engineering of Loughborough University of Technology. This research has been biased towards the design of actuating systems applied to industrial hydraulic textile machinery. Advantages offered by these systems over solid member drive mechanisms can be evaluated in terms of increased speed of operation, ease of sequential programming and design flexibility. These advantages have been demonstrated in the work done and reported by Priestley, Garside and Cameron on:

1. Investigations into miniature hydraulic actuation techniques for needle control on circular weft knitting and sewing machines.
2. Investigations into miniature hydraulic actuation and control techniques for use on high speed reciprocating mechanisms, and
3. An hydraulic actuation technique for needle, looper and knife control in a candlewick tufting machine. This work is reviewed, first to indicate the state of the technology when the work reported in this thesis began, and secondly to show that the further work presented forms part of the continuing research studies into the development and application of hydraulic actuation techniques. Also, by showing the significant variables that can limit system performance, this review indicates possible areas that require further investigation.

SECTION TWO:

Computer aided design of rotary valves. The application of rotary valves to control actuators describing time/displacement profiles necessary for weft knitting had earlier been investigated. Garside(2,5) developed a mathematical technique for balancing the hydraulic pressure forces and Cameron(5,3) improved on this when he put forward new techniques of constructing an inherently balanced valve. Work presented in this section deals with the development of a computer program which processes input data specifying the diameter of the bobbin of the rotary valve, axial positions of supply and exhaust lines to the bobbin, the dynamic cycle characteristics in terms of acceleration, constant velocity, cushioned deceleration and dwell times of individual actuators and the positional phase relationships between them. The output copy from the program is a geometric development of the balanced bobbin, showing the required pressure and exhaust regions to be machined in order to meet the specified dynamic requirements.
The usefulness of this valve design technique is that the map produced can then be drawn full size. The full size map can be wrapped round a cylindrical bar and the sculpturing of the bobbin to produce the required pressure and exhaust regions can thus be greatly simplified with the contours acting as machining guidelines.

Factors to be considered in the design of the valve-actuator system are illustrated, together with an indication of how the program was developed and how it can be applied. Flow charts are given and an illustration of the basic elements of curves and coordinates of points separating the pulse generating sections cut on the bobbin is included in Appendix One.

SECTION THREE:

Design procedures for Actuator/Rotary valve combinations. Certain parameters necessary for the implementation of the design procedure of a rotary valve for application to a prescribed task will have to be specified. Other parameters will have to be calculated from given data. This section first briefly deals with factors that must be considered in the specification of the problem. Equations of motion of actuators under different operating conditions are then derived and an indication is given of how they can be used to calculate the time increments of the cycle for actuator acceleration, constant velocity and deceleration. Finally a relationship between the diameter of the valve, the speed at which it is driven, and the resulting duration of the pulses is presented to illustrate the connection between the relevant variables. The derived expression
indicates factors that must be taken into account in the specification of the problem, the design and operation of the system and shows that it is possible to select the diameter of the bobbin and the speed at which it is to be driven so that the actuator it drives moves with the specified motion to meet specified dynamic requirements.

SECTION FOUR:

The use of limited rotary stroke valves to control tufting machine needle positions for patterned fabric production. To produce patterned fabric using an hydraulic tufting machine, it is necessary to control needle positions. Investigations were therefore carried out into brake and clutch techniques that are currently being used with rotary and linear reciprocating mechanisms. The object was to find out whether any of these techniques could be adopted for patterned tufted fabric production. Some of these techniques are presented in Section Four. Considered in conjunction with the review of earlier work presented in Section One, these investigations showed that one of the areas requiring further detailed investigation was in the development of suitable hydraulic/electronic interface connection methods to make full use of the advantages offered by electronic and hydraulic components. In Section Four a method of using a small electrical signal to control the position of hydraulically driven high speed reciprocating actuators is presented. The technique involves the use of limited rotary stroke control valves and illustrates the simplicity with which fluid flow can be stopped or reversed
in certain selected portions of an hydraulic circuit in response to an electrical signal. On an hydraulically actuated tufting machine, control of needle positions can be achieved by controlling fluid flow using this technique. This makes it technically feasible to produce a wide range of patterns by programmed needle loop-miss combinations and transfer of tufting action to a different set of needles to effect yarn colour changes. Conditions for easy patterning, advantages offered by the technique and an expression to be used in selecting a suitable electrical solenoid and for the sizing of the rotary stroke valve are presented.

SECTION FIVE:

Summary of main findings and conclusions. To obtain a performance evaluation of actuator and valve designs, rotary stroke valves were incorporated into the design of an experimental prototype tufting machine:

i) To control needle reciprocation thus making it possible to program the loop-miss sequences.

ii) To control the needle transfer actuator that effects yarn colour changes.

By programming needle reciprocation and transfer, a wide range of patterned fabric can be produced. Section Five is a summary of the main findings and conclusions drawn from experimental results and observed performance of the prototype machines designed and constructed in the Department.
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<th>Definition</th>
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<td>a</td>
<td>-</td>
<td>Subscript referring to the actuator</td>
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<tr>
<td>A</td>
<td>m²</td>
<td>Effective working area of the actuator</td>
</tr>
<tr>
<td>A₀ₐ, A₁, A₂...</td>
<td>m²</td>
<td>Fluid flow cross-section area in different sections of an hydraulic circuit</td>
</tr>
<tr>
<td>Aₖ</td>
<td>m²</td>
<td>Wetted surface area of the actuator</td>
</tr>
<tr>
<td>Cₚ</td>
<td>m</td>
<td>Radial clearance</td>
</tr>
<tr>
<td>d</td>
<td>m</td>
<td>Actuator port diameter</td>
</tr>
<tr>
<td>D</td>
<td>m</td>
<td>Diameter of the bobbin of the valve or diameter of the valve housing</td>
</tr>
<tr>
<td>D₀</td>
<td>m</td>
<td>Outer diameter of a thick-walled cylinder</td>
</tr>
<tr>
<td>E</td>
<td>N/m²</td>
<td>Modulus of elasticity of material</td>
</tr>
<tr>
<td>f</td>
<td>-</td>
<td>Pipe friction factor</td>
</tr>
<tr>
<td>F</td>
<td>N</td>
<td>Force</td>
</tr>
<tr>
<td>g</td>
<td>m/sec²</td>
<td>Acceleration due to gravity</td>
</tr>
<tr>
<td>h₁,₂</td>
<td>-</td>
<td>Roots of nonhomogeneous second order differential equation</td>
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<tr>
<td>Hₗ</td>
<td>m</td>
<td>Head loss due to bends, fittings, changes in flow section etc</td>
</tr>
<tr>
<td>k</td>
<td>-</td>
<td>Resistance coefficient due to pipe entrance, pipe exit, bends, fittings etc.</td>
</tr>
<tr>
<td>Kᵥ</td>
<td>Nsec/m³</td>
<td>Combined linearized viscous damping coefficient</td>
</tr>
<tr>
<td>K</td>
<td>Nsec²/m⁴</td>
<td>Constant of proportionality relating the total pressure loss to the square of fluid velocity</td>
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<tr>
<td>L</td>
<td>m</td>
<td>Valve land width</td>
</tr>
<tr>
<td>m</td>
<td>Kg</td>
<td>Mass of actuator and load</td>
</tr>
<tr>
<td>Symbol</td>
<td>S.I. Unit</td>
<td>Definition</td>
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<td>--------</td>
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<td>------------</td>
</tr>
<tr>
<td>m₁, m₂, m₃...</td>
<td>Kg</td>
<td>Mass of fluid in circuit sections 1, 2, 3, ... respectively</td>
</tr>
<tr>
<td>M</td>
<td>Kg</td>
<td>Effective circuit mechanism mass</td>
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<tr>
<td>n</td>
<td>-</td>
<td>Number of pulses required during each rotation of the valve</td>
</tr>
<tr>
<td>N</td>
<td>rad/sec</td>
<td>Valve rotational speed</td>
</tr>
<tr>
<td>P, P*</td>
<td>N/m²</td>
<td>Piezometric pressure, pressure difference, effective system pressure or nominal system pressure</td>
</tr>
<tr>
<td>Q</td>
<td>m³/sec</td>
<td>Volume flow rate</td>
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<tr>
<td>R, r</td>
<td>m</td>
<td>Radius</td>
</tr>
<tr>
<td>Rₐ</td>
<td>N</td>
<td>Actuator external work or resistance or spring constant</td>
</tr>
<tr>
<td>Rₐ</td>
<td>m</td>
<td>Radial displacement at inner surface of thick-walled cylinder due to internal pressure increase</td>
</tr>
<tr>
<td>t</td>
<td>sec</td>
<td>Time</td>
</tr>
<tr>
<td>T</td>
<td>sec</td>
<td>Total duration of valve pulses during each rotation of the valve</td>
</tr>
<tr>
<td>Tₒ</td>
<td>Nm</td>
<td>Torque</td>
</tr>
<tr>
<td>Tₕ</td>
<td>m</td>
<td>Wall thickness of a thick-walled cylinder</td>
</tr>
<tr>
<td>Tₚ</td>
<td>sec</td>
<td>Time taken to clear the actuator ports, turn through the valve land widths and through the pulse generating sections of a valve. It is equal to the period of the cycle</td>
</tr>
<tr>
<td>u, v</td>
<td>m/sec</td>
<td>Linear fluid velocity, actuator velocity or velocity of a solid boundary</td>
</tr>
<tr>
<td>x, y</td>
<td>m</td>
<td>Linear displacement</td>
</tr>
<tr>
<td>Xₐ</td>
<td>m</td>
<td>Actuator displacement or amplitude</td>
</tr>
<tr>
<td>δ, Δ</td>
<td>-</td>
<td>Differential operators</td>
</tr>
<tr>
<td>θ</td>
<td>rad</td>
<td>Angular displacement</td>
</tr>
<tr>
<td>ρ</td>
<td>Kg/m³</td>
<td>Fluid mass density</td>
</tr>
<tr>
<td>τ</td>
<td>N/m²</td>
<td>Shear stress</td>
</tr>
<tr>
<td>Symbol</td>
<td>S.I. Unit</td>
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<tr>
<td>--------</td>
<td>-----------</td>
<td>------------</td>
</tr>
<tr>
<td>ν</td>
<td>-</td>
<td>Poisson's ratio (=½ for metals)</td>
</tr>
<tr>
<td>μ</td>
<td>Nsec/m²</td>
<td>Fluid coefficient of viscosity</td>
</tr>
<tr>
<td>ω, Ω</td>
<td>rad/sec</td>
<td>Angular velocity</td>
</tr>
<tr>
<td>1,2,3...</td>
<td>-</td>
<td>Subscripts referring to different parts of the actuator or valve</td>
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<tr>
<td>(AR)</td>
<td>-</td>
<td>Ratio of mechanism accelerations at two successive time instants ( t_n, t_{n+1} )</td>
</tr>
<tr>
<td>(VR)</td>
<td>-</td>
<td>Ratio of mechanism velocities at two successive time instants ( t_n, t_{n+1} )</td>
</tr>
<tr>
<td>(AH)</td>
<td>Kg</td>
<td>Total mechanism mass (( = M )) — notation introduced because of Fortran Computer Language</td>
</tr>
<tr>
<td>( N_k )</td>
<td>N</td>
<td>Normal friction force</td>
</tr>
<tr>
<td>η</td>
<td>-</td>
<td>Coefficient of surface friction</td>
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INTRODUCTION

Hydraulics - the science of liquid flow - is a very old discipline which has commanded increased interest in recent years, especially in the field of hydraulic actuation and servo control. Hydraulic actuation devices and control components are found in many mobile, airborne and stationary applications where they offer unique features compared to other types of control. These features (Merritt (8) 1.1) are fundamental and account for the wide use of hydraulic control systems. The flexibility, compactness and increased speed of operation of hydraulic components, has prompted organisations such as the National Research Development Corporation, Newport Precision Engineering Limited, The Bonas Machine Company Limited, Matramatic Company Limited, Courtaulds' Educational Trust Fund to support research at Loughborough University of Technology, especially for the development of machines using hydraulic means for knitting, sewing and tufting.

This research has demonstrated that the exclusion of hydraulic actuation from the field of high speed reciprocating mechanisms has ignored potentially successful alternative techniques. This is supported by the successful development of a circular weft knitting machine and the subsequent incorporation of hydraulically actuated mechanisms into the design of a lockstitch sewing machine which gave consistent and reliable stitch formation at the required stitching speeds for a specific industrial application. In these machines, individual needles were powered by miniature hydraulic actuators, resulting in superior cycling rates due to higher accelerations and
velocities. To obtain the correct displacement/time profile suitable for the knitting action for example, the control and sequencing of individual needles was obtained by rotary valves. This basic actuator/valve concept, although conceived initially to meet the specific knitting application, has considerable and perhaps better potential in the general field of high speed manipulative machinery and it is in this type of application, that hydraulic actuation can compete with and offer advantages over and above those offered by solid member devices. Further evidence of this claim is indicated by some of the findings presented in this thesis.

Advantages offered by miniature hydraulic actuation and control techniques are described in detail in the reports of the work done so far at Loughborough (Garside (2), 3,5,4;1; Cameron (5)). These features are considered as having great potential in the textile industry as shown by the results of tests carried out on prototype knitting and sewing machines designed and constructed in the Department. This work, parts of which have been reported by the researchers at International conferences, (Priestley (3,4)), is summarised here to illustrate the state of the technology at the commencement of the work presented in this thesis.

To develop these actuation and control techniques, two areas required further detailed investigation:
a) Development of design procedures for reciprocating mechanisms using hydraulic actuators controlled by rotary valves.
It is thought one of the reasons why hydraulic controls are not as widely used as their electrical equivalents is the lack and difficulty of obtaining basic design procedures. Therefore one of the objects of this research was to indicate factors that must be taken into consideration in the specification of the problem, and design and operation of an hydraulic control system. When using rotary valves, the problem will normally be to design a valve-actuator system such that the actuator moves with prescribed motion so as to perform a specific task. In this respect, theoretical analysis was necessary to show what performance could be available when using valve-actuator combinations. By showing the relationship between the significant system performance limiting variables, the analysis would give information that would be useful when drawing up a specification, designing or operating the system. For example the analysis gives information regarding the durations of the pulses that can be generated by a valve. From this, the optimum diameter of the valve and speed at which it must be driven can be selected. To simplify valve design and manufacture, the relative positions and durations of the pulses can also be used to obtain a geometric development of the balanced bobbin showing the pressure/exhaust regions to be cut on the bobbin of the valve. In this work a procedure for designing reciprocating mechanisms using hydraulic actuators controlled by rotary valves was established.

b) Methods of sequential programming of hydraulic actuator positions and the development of techniques for interfacing the programming system with the hydraulically actuated mechanism.
Use of electronic devices for controlling industrial processes is becoming widely spread. To make full use of the advantages that can be offered by electronic/hydraulic system combinations, studies into possible interfacing methods to meet the rapidly expanding demands of industrial automation were required.

Investigations were therefore carried out into methods of programming hydraulically actuated mechanisms so as to provide complete control over individual, or groups of actuators. This led to the development of an electro-mechanical-hydraulic technique using limited rotary stroke control valves for patterned fabric production on an hydraulic tufting machine. The control valves are used to:

i) Control needle reciprocation. This makes it possible to program the loop-miss combination.

ii) Control the needle transfer actuators. This provides a facility for programming yarn colour changes.

A wide range of patterns is therefore possible since the pattern produced depends on the input to the electrical solenoids driving the control valves.

The work presented in this thesis is in five sections:

1. In Section One, previous work on rotary valves and miniature high speed reciprocating actuators is reviewed and areas that require further investigation identified.

2. Section Two discusses the development of an algorithm for producing maps (geometric development of the balanced bobbin)
of pressure/exhaust regions to be cut on the bobbin of the rotary valve powering actuators that move with prescribed motions to perform specified phase related tasks.

3. The connection between the diameter of the rotary valve, the durations of the required pulses and the speed at which the valve is driven is discussed in Section Three. Information obtained from this discussion must be taken into account when drawing up a specification, designing and operating the system.

4. The work presented in Section One and Section Four showed that there was need to develop suitable hydraulic/electronic interface connection methods. In Section Four, an hydraulic tufting machine was taken as a specific example of where a method of controlling needle reciprocation and yarn colour changes was required. The technique put forward involves the use of limited rotary stroke control valves and is part of the system that was developed to produce patterned fabric with an hydraulically actuated tufting machine. The advantages of applying this alternative as opposed to other techniques that can be used for the purpose are indicated, together with a note on the relevant variables to be considered if this alternative is to be properly applied.

5. Section Five is a summary of the main findings and conclusions drawn from experimental results and observed performance of prototype mechanisms designed and constructed in the Department.
Appendix One gives an illustration of the basic elements of curves and coordinates of points separating the pulse generating sections cut on the bobbin. This is intended to clarify certain points made in the main body of the presented work. Expressions that can be used to estimate some of the hydraulic system parameters in circuits involving reciprocating mechanisms controlled by rotary valves are derived and presented in Appendix Two. Photographs included in Appendix Three show components of the needle actuating mechanisms and the looper-knife mechanisms for the experimental prototype tufting machines.
SECTION ONE

A REVIEW OF EARLIER WORK ON ROTARY VALVES AND MINIATURE HIGH SPEED RECIPROCATING ACTUATORS

1.1 Introductory Summary

Research and development studies in the field of hydraulic actuation and control techniques have been going on since 1968 in the Department of Mechanical Engineering at Loughborough University. These studies have led to the development of an hydraulic circular weft knitting machine and an hydraulic sewing machine. In both these cases, miniature high speed reciprocating actuators were powered by rotary valves. These valves have separate regions machined round cylindrical bobbins, the regions being connected either to pressure or exhaust. Rotation of the bobbins in their housings causes the actuators connected to ports in the stationary body, to move in predetermined phase related sequences to perform the desired tasks.

In this section, the development of these reciprocation control techniques is reviewed in chronological order, first to indicate the state of the technology when the presented work started, secondly to show that what is presented represents a combination of a 'complement to' and a 'follow on' to earlier work, and lastly to identify those parameters limiting system performance and therefore to indicate areas that require further investigation.

1.2 The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine

This work, submitted by J D Garside in March 1970 for the Degree of Master of Technology of Loughborough University of Technology, involved
an examination of the possibility of replacing the existing mecha-

nical cam drive of needles on circular weft knitting machines with
alternative methods of actuation, and led to the design and develop-
ment of a miniature hydraulic actuator capable of cycling at 50 Hertz.
This actuator could be made to describe the required time-displacement
profile (Figure 1.01) needed for the knitting action. The profile can
be considered as a plane linear reciprocating movement of amplitude
approximately equal to 25.4 mm. This sort of movement can be obtained
using either an electrical solenoid, a pneumatic jack, or an hydraulic
jack, but because a large force can be obtained from a small area using
normal pressures, the hydraulic jack option was chosen as being the
most likely to give the required cycling rate of 50 Hertz.

As a result of an examination of the control aspect of the hydrau-
lic actuator, a rotary valve in which an inlet groove was connected
to an outlet port for a fixed duration, was designed, built and tested
with a particular port arrangement to suit the required application.
The rotary valve gave a train of pulses in required sequence, the pul-
eses being obtained by revolving a bobbin within a cylinder and take-off
ports made at desired intervals round the cylinder. In this manner,
stroke variation, either a not-knit (miss) stroke of amplitude 0.0 mm,
a tuck stroke of 12.7 mm or a knit stroke of 25.4 mm was achieved by
using compound rams inside the actuator (Figure 1.01) so that the par-
ticular stroke could be selected by directing oil pressure from the
supply valve to the appropriate ports for the required time intervals.
Figure 1.02 shows a single actuator connected to the rotary valve and
gives the sequence of pulses required at the ports to produce the
knitting action.
A DIAGRAM TO SHOW THE IDEALISED TIME–DISPLACEMENT PROFILE REQUIRED FROM A MINIATURE HYDRAULIC ACTUATOR USED TO KNIT

A COMPOUND RAM MINIATURE HYDRAULIC ACTUATOR

FIG 1.01
Sequence of hydraulic pulses required at the ports of the compound ram miniature hydraulic actuator to produce a knitting action

<table>
<thead>
<tr>
<th>Action</th>
<th>Pressure</th>
<th>Exhaust</th>
</tr>
</thead>
<tbody>
<tr>
<td>Knit (t₁)</td>
<td>1, 2</td>
<td>3, 4</td>
</tr>
<tr>
<td>Tuck (t₂)</td>
<td>1, 4</td>
<td>2, 3</td>
</tr>
<tr>
<td>Miss (t₃ + t₄)</td>
<td>3, 4</td>
<td>1, 2</td>
</tr>
</tbody>
</table>

FIG 1.02
1.3 An Investigation into Miniature Hydraulic Actuation Techniques for Needle Control on Industrial Knitting and Sewing Machines

This work, submitted by Garside in May 1972 for the Degree of Doctor of Philosophy of Loughborough University of Technology, demonstrated how high speed linear motions normally obtained from mechanical drives (involving cams, pulleys, linkages, gears etc) can be produced by miniature hydraulic actuation techniques. This was verified by the building and testing of probably the first ever hydraulic knitting and sewing machines. These investigations were directed towards the creation of novel hydraulic control systems for application to textile machinery. The hydraulic actuators used were purpose designed so as to increase the speed of production of conventional knitted fabrics while providing only limited patterning capabilities. The work was presented in four main parts:

i) Using technical knowledge gained in his previous research work, the first part involved the application of basic high speed actuation techniques outlined in his thesis, referred to above (Garside (1)). Since the actuator and rotary valve control mechanism previously designed and tested as a single unit had demonstrated that a time/displacement profile suitable for the formation of a plain stitch could be obtained by using hydraulic devices, design and development studies of a hydraulic circular weft knitting machine were carried out. This dealt with the knitting machine aspect and verified that a multi-actuator rotary valve system could operate with the desired time/displacement profile and in the correct sequence. This was then used as the basis for the design and
development of a prototype ninety-six needle, single feeder circular weft knitting machine. This machine was tested to obtain an assessment of the advantages offered by hydraulic knitting techniques. It demonstrated how a series of miniature hydraulic actuators could be controlled to operate in sequence to form stitches at speeds in excess of existing mechanical cam systems. In addition, this new approach to knitting resulted in the discovery of certain features which have great potential in certain textile machines (Garside (2), Sections 3.5 and 4.1).

ii) In the second part of his work which served as an introduction to hydraulic sewing techniques, Garside converted a mechanical lockstitch machine by replacing the needle and thread take-up mechanisms by two miniature hydraulic actuators controlled by a rotary valve which was coupled to the end of the hook driving shaft, thus establishing a phase relationship between the rotary hook motion and hydraulic linear motions. This work was to demonstrate the flexibility offered by applying miniature hydraulic actuators to a sewing machine and proved that stitches could be formed successfully.

iii) The third part was a detailed design study and analysis of pulse-generating rotary valves which were used as control mechanisms in the previous applications. Although rotary valves had previously been used in other applications, no evidence could be found of their being used for control of high
speed reciprocating mechanisms. Thus having established the definite potential of the basic ideas, this warranted the development of a design procedure to optimize their overall performance under specified conditions. A mathematical model method for analysing the pressure distribution around the bobbin was therefore evolved. The model was based on related hydrodynamic bearing theory.

The results of the analysis were used to design compensating pads which had the effect of balancing the resultant internal hydrostatic pressure forces, since the most desirable running condition is when the valve behaves like an unloaded shaft rotating in a concentric cylinder. With the hydrostatic forces balanced, it was also possible to predict the torque required to rotate the bobbin in its cylinder.

iv) The last part was concerned with the design of a twelve-feeder hydraulic circular weft knitting machine controlled by an integral actuator-rotary collar valve to generate pulses. It demonstrated how a series of twelve knitting time-displacement profiles could be created by ninety-six actuators positioned in a circular configuration.

1.4 An Investigation into Miniature Hydraulic Actuation and Control Techniques for use on High Speed Reciprocating Mechanisms

This research, presented by D Cameron in October 1979 for the Degree of Doctor of Philosophy of Loughborough University of Technology, relates to an actuation and control concept whereby miniature hydraulic
mechanisms are sequenced by rotary valves. The actuators and valves which had been developed for the knitting machine application, although adequate to demonstrate the feasibility of the basic concept, were known to have certain operating deficiencies. The objectives of this research were to advance the technology, to further develop the dynamic control techniques and to improve the engineering of linear actuators sequenced by rotary valves and their associated hardware. Achievement of these objectives would then enable the techniques to be applied to high speed reciprocating motions used in a wide range of manipulative machinery. This represented a continuation of the investigations by Garside in earlier research.

Based on theoretical analyses and the results of investigations into the performance limitations of previous valve and actuator designs, improved actuator and valve designs were conceived, analysed, manufactured and tested. Theoretical equations of motion of mechanisms were derived for a generalized actuator/valve circuit and correlated with the measured performance of various prototype mechanisms. The research was presented in five parts:

i) The first part involved the design concept and theoretical analysis of miniature cushioned actuators. Two new concepts of miniature actuators were presented; one a single-acting module and the other, a double acting version (Figure 1.03). Both designs incorporated a novel method of hydraulic cushioning, had no seals and were capable of compound movements. The various operating modes obtainable from the actuators were outlined, together with their general operating characteristics. An analysis to obtain the theoretical motion of different mechanism configurations when
A DOUBLE ACTING ACTUATOR WITH HYDRAULIC CUSHIONING

SINGLE ACTING ACTUATOR WITH HYDRAULIC CUSHIONING

\[
\frac{A_1(x-x_d)d^2}{A_1 + A_2} > x_d d^2 > x d^4
\]

FIG 1.03 (Cameron (5) 1.3)
reciprocated by the actuators was presented. The derived equations of motion were analysed, their practical application discussed and the relative importance of the system parameters identified.

ii) The second part related to some of the practical tests carried out on the actuators to establish their functional operation and to compare their actual dynamic performance with that predicted theoretically. Prototype mechanisms whose designs were based on the concepts outlined in the first section, were manufactured and tested to determine the correlation between actual cyclic performance and theoretical predictions as well as the consistency of operating characteristics. The essential points that must be observed to obtain accurate correlation between theoretical predictions and measured performance were identified. To enable the actuating concepts to be applied successfully, a guide to the selection of the system parameters, based on theoretical considerations and the practical evidence was presented.

iii) The third section involved a new design concept and analysis of a 'Universal' self-balanced rotary valve plus a technique for applying it to multi-actuator circuits (Figures 1.04, 1.05). The requirement to reduce the driving torque of previous valve designs and the need to extend the flexibility of operation of these valves led to a new valve design. An inherently self-balanced valve having an increased pulse sequence selection facility and improved performance was therefore presented. An analysis of theoretical valve losses to minimise leakage and drag losses
SELF BALANCED PULSE GENERATING ROTARY VALVE

(Cameron 5) 3.4

FIG 1.05
against specific operating parameters was also presented.

iv) In the fourth section Cameron gave results of tests carried out on rotary valves to establish the correlation between measured and theoretical performance. Prototype valves designed and balanced as outlined in the third section were tested and compared with previous valve designs and theoretical predictions. The validity of the design was confirmed by the sequencing of both single actuator and multi-actuator circuits. The compactness of the new design was demonstrated by the building of a small valve module having a restricted pulse selection facility.

v) In the last part, as a specific example, the design and development of a mattress sewing machine which embodied the hardware and techniques developed, was presented as an example of the flexibility and predictability of the system elements. Also designs of hydraulically actuated mechanisms for the control of the needle, looper and knife of a candlewick tufting machine were illustrated.

1.5 An Hydraulically Actuated Tufting Machine with Limited Patterning Capability

By the time the investigations presented in this thesis started, further research work directed to developing high speed patterning systems for tufted fabric production on hydraulically actuated machines had been undertaken in the Department of Mechanical Engineering at Loughborough University by Mr Priestley. Success with the knitting and sewing machines had led to the conclusion that with hydraulically
actuated mechanisms the overall design concept could include features that are not readily obtainable on conventional mechanical tufting machines; for example that of programmable patterning and the 'running on' of the looper mechanism with the needle retracted at the end of a stitching run. These could be achieved simply by preventing needle descent and/or by changing needles at the appropriate times. By incorporating simple on-off switching in the basic hydraulic control circuits, both these features could be obtained without decoupling the mechanical drives. Because the final solution would involve the simultaneous and/or sequential operation of a large number of hydraulic devices, one requirement was to devise a system for the manipulation of miniature hydraulic actuators reciprocating at high speed. Consistent and accurate position control was also necessary. These are some of the features that were built into the first hydraulically actuated tufting machine.

The patterning possibilities offered by an hydraulically actuated tufting machine are not feasible with mechanical machines for two major reasons:

i) Hydraulic actuation allows the use of mechanical and/or hydraulic stops to prevent needle descent while the normal cycle of power transmission continues. Patterning can therefore be achieved at much higher speeds. The possible use of solenoid operated latches for example allows the sequence of needle patterning to be controlled electronically. This vastly increases the range of possible sequences and allows a sequence to be easily changed when required.
ii) The use of hydraulic actuators with their flexible power transmission allows individual looper-knife and needle actuating units to be positioned without mechanical constraints. The looper-knife and needle actuating units can even be moved from side to side, or backwards and forwards during continuous stitching, if a desired pattern can be produced by doing so.

In addition to needle patterning, there will be commercial application for looper patterning. A second looper actuator could be positioned on the opposite side of the needle with the looper inserted through the yarn loop in the direction of fabric feed. This looper could then be designed to produce loop pile rather than cut pile. It would operate with a parallel tipped looper instead of a hooked one, and without a knife. Both cut pile and loop pile actuators could be equipped with pattern control, using interlinked sequencing so that when the cut pile looper stops, the loop pile looper starts, and vice-versa. Intermixed patches of cut and loop pile could then be produced to a pre-set pattern.

These considerations pointed to the need to investigate practical aspects of programming, so as to provide complete control over individual, or groups of actuators. Suitable hydraulic/electronic interface connection methods had to be developed because these principles of actuation and control can be applied to a wide range of machines and for many industrial applications. The stitches of a chain stitch machine for instance are made in a similar way to those of a candlewick tufting machine. Another example is the need to extend the range of knitted fabric from the basic knitted structure. This can be done if a programming technique for producing a tuck and miss stitch is
developed.

On the prototype tufting machine developed in the Department, the needle, drawing yarn with it, moves down through the fabric and immediately, the looper slides in between the needle and the yarn. When the needle withdraws upwards, the looper holds the yarn thus forming and holding a loop. The backing fabric feed mechanism moves up and lifts the backing fabric off the working surface, advances it forward through a stitch length and then drops down leaving the backing fabric on the working surface. When the looper withdraws, the knife cuts a loop, thus forming a tuft and withdrawal of the backing fabric feed mechanism completes the cycle. This phase relationship is shown in Figure 1.06 and Figure 1.07 shows the diagrammatic arrangement of the looper unit, tufting head and backing fabric feed mechanism. The needles are driven by an hydraulically cushioned double acting piston while the loopers are driven by a pair of opposed single acting pistons. Both the needle and looper mechanisms are supplied by balanced pulse generating rotary valves of the type designed by Cameron.

In order to make one line of tufts on the fabric, it is necessary to have yarn passing through the eye of the needle, a looper to form the loops and a knife to cut the loops. Only one loop can be made in each line of tufting per needle penetration.

If patterned tufting is to be produced on a high speed tufting machine by the manipulation of the various components in a predetermined sequence defined by the pattern and unique to each design, certain requirements have to be met. For each row this requires information for the individual control of the needle carrying the yarn
A diagram showing the phase relationship of the time-displacement profiles of the needle, looper-knife combination, and the backing fabric feed mechanism on the 'Hydratuft', rotary valve-controlled hydraulic tufting machine.

FIG 1.06
The needle driving mechanism uses an hydraulically cushioned double acting piston.

The looper mechanism employs two opposed single acting pistons with hydraulic cushioning.

The backing fabric mechanism allows variation of pile height.
(each needle has to adopt a position determined by the pattern) and information regarding the position of the fabric relative to the actuator powering the needle. Therefore, a system of information acquisition, storage and processing, together with the necessary interface connections, are required to control these devices. It was therefore decided to carry out a survey of some of the methods that are currently used in the textile industry for similar purposes. The findings are given in Section Four. For the tufting machine under consideration, the following methods of programming the loop-miss sequence to give limited patterning capability were investigated.

1. The needle actuator piston movement controlled directly, using a solenoid operated valve to:
   i) Open a pressure port direct from an accumulator to the lower part of the cylinder to prevent the needle actuator piston descending.
   ii) Close the exhaust passage just after the needle begins to descend to prevent fluid below the piston from being displaced.

   Both these methods require almost instantaneous operation of the valve to keep the needle travel during stitching acceptably small. Also, unless a complex set of pressure compensating ports is provided around the valve, if conventional hydraulic servovalves are to be used, the solenoid would have to move the valve against full system pressure at some time in the cycle of operation.

2. Use of a latch mechanism as a mechanical stop. This method has the advantage that the latch can be moved, in advance, ready to drop under a stop as soon as the actuator reaches the upper limit
of its movement. The latch could either be normally in, requiring a positive signal to allow stitching, or normally out, requiring a positive signal to prevent needle descent. This latter option is the one that was adopted.

The latch-in motion can be controlled by:

i) The latch pushed in by a hydraulic piston controlled by a rotary valve. This is fast acting but it is complex and the additional masses can limit the speed of operation and use of a rotary valve without any other control will limit the number of possible patterns.

ii) An hydraulic piston used to latch-in, this piston being actuated by a pressure pulse controlled by a solenoid valve. This has the advantage of electrical control but the extra complication of hydraulic piston movement greatly limits the speed of system operation.

iii) The latch pushed in directly by a solenoid operated pushrod. Again this is a very simple method and has the advantage that it will be initiated by an electrical signal. Therefore a wide range of sequences can be set on the programming control circuit. However, it requires a high thrust solenoid to produce the necessary movement of the latch and the configuration means that patterning will not be available beyond a certain cycling rate due to the limitations on the speed of response of the latching mechanism.
A solenoid of appropriate size capable of fast response was available. Therefore a mechanism, latched in directly by a solenoid-operated pushrod was used. It was then possible to make preliminary investigations into some of the factors like spring-latch configuration and yarn tension that affect successful tuft formation. The latch mechanism is shown in Figure 1.08. If a selected needle down-stroke is to be prevented, then the latch must be pushed against the moving bar stop sometime during the preceding cycle of the needle, ready to latch-in as soon as the needle bar stop clears the top of the latch. An electronic circuit designed to control the stitch/miss operation of the machine enabled patterned tufting to be achieved at speeds in excess of 1000 stitches per minute.

1.6 Conclusion

The foregoing sets out the state of hydraulic actuation and control technology when studies presented in this thesis began. Work by Priestley, Garside and Cameron not only highlighted the advantages of flexibility, adaptability and controllability of hydraulics, it also provided some analysis and design techniques for the proper application of miniature hydraulic actuators. As a result of these research and experimental development studies, new designs of linear actuators and rotary valves were conceived, analysed, manufactured and tested. They indicated that it is possible to produce a rotary valve multi-actuator system in which individual actuators can be controlled through specified time-displacement profiles in predetermined sequences, to an acceptable degree of accuracy and consistency of performance, at speeds far in excess of those obtainable by using
DIAGRAM TO SHOW OPERATION OF THE NEEDLE ACTUATOR PROGRAMMING LATCH

- Rotary valve
- Manifold
- Latch
- Needle actuator
- Needle bar stop
- Solenoid operated push rod
- Spring plunger
- Needle bar
- Latch pivot
- Needle

FIG 1.08
cam or linkage drives for similar manipulative work. Films showing the prototype experimental machines under operating conditions are available in the Department of Mechanical Engineering.

In addition to the textile machines investigated, there are other fields where machinery design and performance is limited by the solid member mechanisms employed. For example machines for packaging, automatic component handling and assembly, machine tool controls and certain continuous motions operating in phase-related sequences. It was therefore necessary to develop and establish analysis and design techniques that would provide a sound design basis for applying hydraulic actuators and rotary valve combinations to improve design and performance of machines in these categories. For this, detailed theoretical and practical investigations were required to yield the necessary design, manufacturing and operating methods for the correct application of actuator/valve ideas to reciprocating mechanisms. It was intended that this work would result in design procedures enabling determination of correct geometries for actuators and valves to meet particular operating specifications. This was realized because it is believed that one of the reasons why hydraulic controls are not as widely used as their electrical equivalents is the difficulty experienced by non-specialist hydraulic power engineers in setting up readily usable design procedures. This is perhaps partly due to the complexity of hydraulic control analysis where no single law exists to accurately describe the hydraulic resistance of passages to flow. For this seemingly simple problem, a number of empirical formulae and coefficients may need to be used.

These studies went a long way in suggesting possible solutions
to some everyday industrial problems and in providing a basis for further developments. They indicated that more benefits could be derived if design procedures were established and practical aspects of programming investigated.

The rotary valve is one of the most important components of the systems so far investigated. Results of tests performed on the knitting, sewing and tufting machines indicate that it is very important to optimize its overall performance because this affects the performance of the actuators and of the system as a whole. Garside (2,5) used a mathematical model method of analysing the pressure distribution around the bobbin to design compensating pads, so balancing the internal hydrostatic forces and making it possible to predict the torque required to rotate the bobbin in its cylinder under various operating conditions. Cameron (5,3) improved on this and presented a valve having an increased pulse selection facility and improved performance. The need to increase the pulse selection facility for application to multi-actuator circuits and flexibility of operation warranted further investigations into rotary valve design with a view to developing an accurate design procedure applicable to various actuator/valve combinations for the actuation of machine components operating in cyclic phase related sequences.

The decision to concentrate on the fundamental actuation problem to the exclusion of programming in previous research, was influenced by the fact that in the textile industry highly successful methods of programming needles by electromagnetic means had already been developed for use on conventional cam operated machines. Now that
hydraulic systems could be designed to operate at speeds far in excess of existing mechanical drives, it became necessary to investigate possible programming methods suitable for use with these devices. If full advantage was to be taken of miniature hydraulic actuation techniques, detailed analysis and design studies had to be made into data handling techniques and electronic/hydraulic interface connection methods to meet the ever-present and rapidly expanding demands of industrial automation. The prototype tufting machine, with its limited patterning capability, suitable for over-tufting and straight runs, provided a good starting point for these studies. On this machine, patterning can be achieved at 1200 loops per minute, while straight runs are possible at speeds up to 1600 loops per minute. Investigations were therefore required into patterning systems with a view to increasing the patterning capability, while maintaining the high speed of operation of the machine.

1.6.1 Experimental development testing of the hydraulically actuated tufting machine

In parallel with the author's investigations, exhaustive tests were carried out on the experimental prototype tufting machine, first by Isham, Ref. 6 and then by Durrant, Ref. 7, final year students whose reports were submitted in partial fulfilment of the requirements for the award of the Degree of Bachelor of Technology. Results presented in these reports enabled the author to make a performance evaluation of actuator designs used in tufting machines. In Section 5.2, a comparison is given of the operating features of hydraulically actuated candlewick tufting machines and conventional solid member mechanism types.
SECTION TWO

COMPUTER AIDED DESIGN OF ROTARY VALVES

2.1 Introductory Summary

As illustrated in Figures 1.02, 1.04 and 1.05 a rotary valve consists of a bobbin revolving within a cylinder around which supply, take-off and drain ports are appropriately connected, resulting in a device capable of generating a series of cyclic pulses required to control an external system. The application of rotary valves to obtain the correct time/displacement profiles suitable for knitting, sewing and tufting had earlier indicated that the sequencing of individual actuators or groups of actuators can be achieved if this technique is correctly applied. Although different actuator time/displacement profiles require different valve configurations, in general, a rotary valve can be designed for each individual profile or group of profiles.

The need to increase the pulse selection facility for application to multi-actuator circuits and to have greater flexibility of operation, warranted further investigations into rotary valve design, in order to develop a simple and accurate design procedure applicable to various actuator/valve combinations, for driving system components operating in phase-related sequences. Further study was required to enable the general design criteria to be resolved and a procedure specified so that functionally efficient valves can easily be designed and operated for industrial applications.
For example, out-of-balance forces can be generated by the pressure and exhaust sections cut on the bobbin of the valve for pulse generating purposes. The most desirable running condition is when these forces are minimised. Therefore a method of construction had to be devised so that these forces are balanced, making the valve equivalent to an unloaded shaft running in a concentric cylinder. Also a method of predicting the flow and power requirements for the valve had to be evolved so that if necessary, the drive mechanism for the bobbin can be specified at the design stage.

A properly designed rotary valve must be capable of generating the series of cyclic pulses required to control the external system. This implies that the dynamic characteristics of the system, as well as the relative positioning of the controlled system components must be well defined before a suitable valve can be designed. This is so because it is desirable to position the valve as close to the controlled system as possible in order to minimize the pressure drop in the connecting lines and minimize the total effective system mass. Also, the positioning of the system components has a considerable effect on the geometry and relative positioning of the pressure and exhaust sections cut on the bobbin. It is therefore possible to break down the algorithm for producing an efficient rotary valve system into a number of steps. Firstly, the controlled system parameters and characteristics must be defined. Secondly these must then be operated upon to produce the desired result. Lastly flow and power requirements for the valve can then be predicted and the drive mechanism for the bobbin specified.
A note on the determination and specification of system parameters and characteristics is presented in Section Three. Also given in that section is an indication of how flow and power requirements for the valve can be predicted. In the present section it is assumed that information in terms of the diameter of the bobbin; position of supply and exhaust lines to the bobbin; number, position and individual time increments for acceleration, constant velocity, deceleration, and dwell of the actuators is known. Working on this information, a map showing the required pressure and exhaust regions to be cut on the bobbin so as to meet the specified dynamic requirements, can then be obtained by using the computer program developed.

Important design, construction and operating points are considered first and then by taking specific examples an algorithm for producing the map is developed. Flow charts and a typical example of a computer printout showing the contours are given, together with a note on how to use the program. An indication of the basic elements of curves and coordinates of points separating the pulse generating sections cut on the bobbin is included in Appendix One where a specific example of how to use the developed program is given.

A pulse $GP(I,J)$ is the time during which an actuator port remains connected to a pressure or exhaust section on the valve. During this time, the actuator starts from rest, moves until it attains a maximum velocity with which it moves until it is retarded and brought to rest by cushioning. It then stays in this position for some time before its motion is reversed. Therefore $GP(I,J)$ is equal to the sum of acceleration time, constant velocity time, deceleration time, and dwell
time. Acceleration time, constant velocity time, and deceleration time can be calculated using the equation of motion, Section 3.3, and the dwell time has to be specified, Section 3.2. Here it is assumed that GP(I,J) is known and can therefore be input.

2.2 Preliminary Rotary Valve Design Considerations

To design a functionally efficient rotary valve, the layout of the sections cut on the bobbin must be considered in addition to the parameters and characteristics of the system to be controlled. The geometry and relative positioning of these sections is largely governed by the pulse requirements of the system. For example if the system is to be supplied with a pressure pulse at a given instant, the system pressure port must be opposite a pressure section machined on the bobbin. Therefore the first consideration must be the specified time/displacement profiles to be described by the actuators, because these determine the geometry and relative positioning of the pressure and exhaust sections.

Profiles that can be described by actuators of the double acting reciprocating type, with possible application to multi-actuator circuits are shown in Figure 2.01. These profiles indicate the pressure requirements at one of the two ports of an actuator. Pressure requirements at the other port are the exact opposite, that is to say, if one port is pressurized, the other must be exhausted. In order to identify the steps that could be followed so as to produce an acceptable design, it was decided to use these profiles to study the design of a rotary valve with the objective of producing a geometric development of the
FIG 2.01
Displacement-time profiles of double-acting, reciprocating type actuators with possible multi-actuator circuit application.

- **A**: One actuator, pressure and exhaust pulses of equal duration.
- **B**: One actuator, pressure and exhaust pulses of unequal duration.
- **C**: Two actuators acting as one, producing stepped motion e.g. that for the knit-tuck-miss cycle.
- **D1**: Pressure pulses of unequal duration.
- **D2**: Exhaust pulses of unequal duration.
- **E**: Two actuators, pressure and exhaust pulses of unequal duration.
- **F**: Three actuators, pressure and exhaust pulses of equal duration with phase difference.
- **G**: Three actuators, pressure and exhaust pulses of unequal duration.
- **H**: Four actuators, pressure and exhaust pulses of equal duration with phase difference.
- **I**: Four actuators, pressure and exhaust pulses of unequal duration.

Taking the duration of the smallest pulse at one of the two ports of an actuator (pressure or exhaust) as one unit, then the pulse sequence for the different actuators will be as follows (e.g. 6P 2E means that a pressure pulse lasting for 6 units of time is followed by an exhaust pulse lasting for 2 units of time for each bobbin rotation. A unit in this case is 1/4 of the period.)
balanced bobbin showing the supply/exhaust sections to be cut on the bobbin. This was to be achieved by obtaining the plane coordinates (say X and Y) of the boundaries separating the pressure and exhaust sections. Also actuator port connection points were to be indicated so that the prescribed phase relationships could be maintained. While doing this three general points were noted:

2.2.1 Relationship between the required pulse sequence, the map of sections generating the pulses and position of actuator supply ports along the axis of the valve

If the pulse sequence at one of the two ports of an actuator is 1P 4E 2P 1E for example (P = pressure, E = exhaust) the sequence at the other port must be 1E 4P 2E 1P, because while one port is pressurized, the other must be exhausted. This means that a map of the sections for one port must be the same as that of the other port with only a phase difference. For the profiles considered in Figure 2.01 the sequences at the ports of the actuator must be as given in the table below.

<table>
<thead>
<tr>
<th>Profile</th>
<th>Port A</th>
<th>Port B</th>
</tr>
</thead>
<tbody>
<tr>
<td>A, D, F, H</td>
<td>1P 1E</td>
<td>1E 1P</td>
</tr>
<tr>
<td>B</td>
<td>2E 1P 2E 1P</td>
<td>2P 1E</td>
</tr>
<tr>
<td>C 1</td>
<td>2P 1E</td>
<td>2P 1E</td>
</tr>
<tr>
<td>2</td>
<td>2E 1P</td>
<td>2P 1E</td>
</tr>
<tr>
<td>E 1</td>
<td>2E 1P 2E 1P</td>
<td>2P 1E 2P 1E</td>
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<tr>
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<td>4E 2P</td>
<td>4P 2E</td>
</tr>
<tr>
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<td>2P 1E 2P 1E</td>
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<td>3P 1E 2P 2E</td>
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<td>6P 2E</td>
</tr>
<tr>
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<td>2E 6P</td>
<td>2P 6E</td>
</tr>
<tr>
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<td>1P 4E 2P 1E</td>
<td>1E 4P 2E 1P</td>
</tr>
<tr>
<td>4</td>
<td>3P 1E 3P 1E</td>
<td>3E 1P 3E 1P</td>
</tr>
</tbody>
</table>
For the profiles considered in Figure 2.01, whose pulse sequences are given in the table above, two cases arise. One is when the pulse sequence at one port is the same as that at the other port, and the other case is when the sequences are different.

When the sequence is the same (Profiles A, D, F and H with a sequence 1E, 1P, 1E, 1P) the maps will be identical with only a phase difference. This means that the pressure and exhaust ports of an actuator can be connected in line on the same circumference round the valve if they are appropriately phased. This is indicated in Figure 2.02c for example. Figure 2.02 shows maps of the pressure/exhaust sections required to give some of the time-displacement profiles specified in Figure 2.01. A very important and useful point to note about this is that if the actuator ports are displaced through the same angle as the pulse generating sections, the pulses remain in the same sequence. As a result of this, ease of bobbin machining can be achieved. Figure 2.02a, d, e and f illustrate this point by indicating two possible section arrangements in each case.

When the sequence at one port is not the same as that at the other port, the two actuators ports cannot be connected in the same line on the same circumference round the valve. The connection must be done as indicated in Figure 2.02a for example, where each one of the two ports of an actuator is on a different circumference.

If reduction of inertia and efficient material utilization are taken into account these points have considerable implications if the size and power requirements of the valve are to be kept to a minimum.
Possible arrangements of the pressure/exhaust sections to produce pulses required for the time/displacement profiles specified for each cycle.
A related case is that indicated in Figure 2.01c. In this case the pulse sequence for one actuator is the reverse of that for the other meaning that although the two ports of an actuator cannot be connected in the same line, it is possible to connect one port of an actuator and the other port of the second actuator in the same line on the same circumference round the valve as indicated in Figure 2.02b. Instead of four actuator channels (2 for each actuator), two would be enough.

2.2.2 Methods of constructing an hydrostatically balanced valve

The most desirable running condition is when the out of balance forces generated by the pressure/exhaust sections are minimised. Taking a one actuator system with pulse sequence 2P 1E for example, a possible section arrangement might be as indicated in Figure 2.03a. This results in an unbalanced system which increases the valve driving power requirements because the pressure forces at diametrically opposite points (e.g. 0 and π) are not equal and the resulting force system will tend to turn the bobbin about one end of its axis. There are a number of ways by which force balance can be achieved and therefore valve driving power requirements cut down:

i) Garside's mathematical model method

Garside used a mathematical model method of analysing the pressure distribution round the bobbin to design compensating pads, so balancing the internal hydrostatic forces and was able to obtain a 50% reduction in the power requirements for the valve drive (Garside (2), Section 5.7).
FIG 2.03

A possible pressure-exhaust section arrangement to produce a balanced rotary valve.
ii) **By using 'balancing' channels**

Another way is to double the number of actuator channels required. If the extra channels are appropriately phased, a pressure system (similar to Cameron's system shown in Figures 2.04 and 2.05) can be created that results in no net moment tending to turn the bobbin about one of its ends.

iii) **Cameron's method of sculpturing the bobbin**

By sculpturing the bobbin and creating pressure/exhaust sections on the bobbin surface, thus reducing the bobbin-cylinder contact area, Cameron obtained a further reduction in the power requirements of the two valves that he designed and used in his experimental work. In these valves, the forces at diametrically opposite points are not equal but the force system is such that no net moment is produced (Figures 2.04 and 2.05).

iv) **Sculpturing the bobbin and doubling the number of pulses per rotation**

Using the same sculpturing technique, another way of achieving force balance is by doubling the number of pulses per bobbin rotation as indicated in Figure 2.03b where the forces at diametrically opposite points are equal and opposite. Although this may necessitate increasing the diameter of the bobbin, or reducing its speed of rotation in order to avoid attenuation of the actuator amplitude of travel while operating at high speeds, this technique may be useful if machining considerations, ease of design procedure and bobbin drive mechanism are taken
ROTARY VALVE SUPPLYING FOUR ACTUATORS OPERATING IN PHASE—PRESSURE AND EXHAUST PULSES OF EQUAL DURATION

THE 'UNIVERSAL' BALANCED ROTARY VALVE - ONE PRESSURE AND ONE EXHAUST PULSE PER REVOLUTION OF BOBBIN ROTATION, RELATIVE DURATION OF PULSES BEING DEPENDENT UPON THE AXIAL POSITIONS OF THE PORTS SUPPLYING THE ACTUATOR

FIG 2.04
Cameron's 'universal' balanced rotary valve

Pressure system at diametrically opposite points

Cameron's system

PROPOSED SYSTEM

FIG 2.05
into account. A comparison of the possible pressure/exhaust section arrangements for the time/displacement profiles of Figure 2.01e is indicated in Figure 2.03c. Figures 2.04 and 2.05 compare this technique with that adopted by Cameron. Valve designs based on doubling the pulses per revolution are presented. It is however recognized that from a manufacturing point of view it may be easier to construct a valve with lands parallel and perpendicular to the bobbin axis as in Figure 2.03e rather than the V shape of Figure 2.04.

2.2.3 Position of pressure supply and return lines

All valves require a pressure supply and return line. It may be better to provide a central supply section with return lines on the outside as indicated in Figure 2.03d for the time/displacement profiles specified in Figure 2.01e. Figure 2.03e shows a development of the bobbin.

2.3 Development of the Algorithm for Producing Geometric Developments of Balanced Bobbins to Meet Specified Dynamic Requirements

The profiles to be described by the actuators must be suitable to carry out repetitive cycles of operation in a machine, e.g. for the processes of knitting, sewing or tufting. Therefore having noted the general points indicated in the section above, it was decided to study in more detail, the design of a rotary valve to power up to eight actuators operating in sequence and describing different time/displacement profiles. These studies were aimed at providing more information about the increased pulse selection facility and multi-actuator circuit/
rotary valve design. Figure 2.06 indicates the profiles, pulse sequences and phase relationships prescribed for the actuators.
In order to produce a map showing the pressure and exhaust regions to be cut on the bobbin, a number of steps were taken:

i) 'New' pulse sequences were derived so that the total for each actuator was the same. This was necessary since all eight were to be controlled by the same valve. The sum of each sequence \( \text{SUM}(I) \) was found, the greatest sum of all sequences, \( \text{SUMC} \), was obtained and a multiplication factor \( F(I) = \frac{\text{SUMC}}{\text{SUM}(I)} \) for each actuator derived to yield the new pulses. Individual actuator sections were then laid out (Figure 2.07) taking phase relationships into account.

ii) Actuator connection points were moved by shifting all sections so that as far as possible, pressure sections started on the Y axis, Figure 2.08a. In doing this it was noted that displacement of the actuator ports and pulse sections through the same angle leaves pulses in the same sequence and phase relationship. The purpose of shifting the connection points was to minimize the number of corners, thereby shortening the regions separating the exhaust and pressure sections on the bobbin. This was to simplify bobbin machining as can be seen by considering Figure 2.09g, h and i for example, which gives possible section arrangements for an actuator with a pulse sequence 5P 3E.
FIG 2.06
PHASE RELATIONSHIP, PULSE SEQUENCES AND TIME/DISPLACEMENT PROFILES SPECIFIED
INDIVIDUAL ACTUATOR PRESSURE AND EXHAUST SECTIONS

ORIGINAL INDIVIDUAL ACTUATOR PULSE SEQUENCES

<table>
<thead>
<tr>
<th></th>
<th>1P</th>
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</tbody>
</table>

DERIVED INDIVIDUAL ACTUATOR PULSE SEQUENCES

<table>
<thead>
<tr>
<th></th>
<th>4P</th>
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<td>8</td>
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</tr>
</tbody>
</table>

FIG 2.07
FIG A

2.08 EFFECT OF NUMBER AND POSITION OF SUPPLY AND EXHAUST LINES ON BOBBIN MANUFACTURE
Effect of number and position of supply and exhaust lines on bobbin manufacture

FIG 2.09
iii) A decision on the position of the pressure supply and return lines had to be taken. The positions of these lines is of major importance as far as actuator performance and ease of valve machining are concerned. For multi-actuator circuit applications there are two most likely patterns. These are given in Table T2.2. The first is to alternate the supply and return lines such that one line can supply or exhaust up to two actuators, Figure 2.08b. The second pattern is when one line is used to supply or exhaust up to four actuators, Figure 2.08c.

In both cases it was judged better to have exhaust sections at both ends of the bobbin. While the pattern in Figure 2.08c might result in a smaller number of lines separating the pressure/exhaust sections and therefore in less contact area between the bobbin and cylinder body, ease of bobbin manufacture, actuator performance and other boundary conditions, had to be considered when selecting the positions of the supply and return lines. The provision of supply and return lines will often be a simple drilling operation. Therefore, overall it was felt that alternating the pressure and return lines with one line supplying or exhausting up to two actuators would require the simplest manufacturing method.

2.3.1 Constraints set in the development of the algorithm

As a result of the above preliminary considerations, it was decided to develop a digital computer program that can be used to produce a print-out of a map showing the sections to be cut on the bobbin for pulse generation purposes. The following constraints were set:
SUGGESTED PATTERNS OF EXHAUST, ACTUATOR AND SUPPLY SECTION ARRANGEMENTS FOR MULTI-ACTUATOR CIRCUIT APPLICATIONS

**Actuators**

<table>
<thead>
<tr>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
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<th>17</th>
<th>18</th>
<th>19</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>Eab SE</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td>Eab Sab E</td>
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<td>Eab Sab E</td>
<td>Eab Sab E</td>
<td></td>
</tr>
</tbody>
</table>

**Number of actuators in circuit**

**Table T 2.2**
i) The valve was to be used to control actuators of the double acting reciprocating type describing the time/displacement profiles given in Table T2.3.

ii) The valve power requirements were to be minimized by employing the sculpturing technique and the valve was to be balanced by doubling the number of pulses per bobbin rotation.

iii) The pressure and exhaust sections were to be arranged such that one line supplied or exhausted up to two actuators (Table T2.2, e.g. Figure 2.08b).

iv) The bobbin was to have the minimum number of corners to make machining easier. Therefore the positions of the port connection points were to be calculated in order to maintain the prescribed phase relationships and sequences.

2.3.2 Flow charts and output of the developed algorithm

Flow charts for the developed algorithm are given in Figures 2.10 to 2.13 and Figure 2.14 shows a computer print-out of the pressure and exhaust regions. A note on how to use the program (ROVA) is given in Section 2.4 and an indication of the basic elements of curves and coordinates of points separating the pulse generating sections cut on the bobbin is given in Appendix One.
TABLE T2.3
PRESCRIBED TIME-DISPLACEMENT PROFILES OF ACTUATORS USED IN THE COMPUTER PROGRAMME DEVELOPED

<table>
<thead>
<tr>
<th>ACTUATOR (I)</th>
<th>NGP(I)</th>
<th>STAI(I)</th>
<th>GP(I,J)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>0</td>
<td>1.1</td>
</tr>
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</tr>
<tr>
<td>10</td>
<td>8</td>
<td>1</td>
<td>2.1.2.1.1.4.2.1</td>
</tr>
</tbody>
</table>
GENERAL FLOW CHART FOR THE DEVELOPED PROGRAMME USED TO OBTAIN A MAP OF THE PULSE GENERATING SECTIONS CUT ON THE BOBBIN

DECLARE COMMON P, K, XX, YY, DY, N1, N2, ISA, ISB, SUMX, ISTA(I), ISTB(I), SUMB(I), SUMA(I), SUMA(I), SUMB(I), X(I, J), Y(I, J), N3GP(I), N4GP(I), A(I, J), B(I, J), NSEP(I), NPLT(I)

DIMENSION F(I), SUM(I), EVEN(I), PLUS(I), SHIFT(I), IDD(I), PSD(I), XPU, YPU, GP(I, J), NGP(I)

CALL C1051N

INPUT
NGP(I), IDD(I), ISTA(I), GP(I, J), DY

FLOW CHART 2.1

OUTPUT
N3GP(I), N4GP(I), SUMF, SUMX, EVEN(I), A(I, J)

FLOW CHART 2.2

OUTPUT
NPLT(I), B(I, J), PSD(I), ISTB(I)

FLOW CHART 2.3

OUTPUT
X(I, J), Y(I, J)

CALL WINDOW (2)

CALL AXIPLO(- - - - - -)

DO AA
I=1, MAX

DO BB
J=1, NPLT(I)

XPU= X(I, J)- EVEN(I)
YPJ = Y(I, J)

XPU < 0: STOP

XPU > 0: STOP

XPU = 0: STOP

CALL GRASYM(- - - - - -)

CALL GRAPOL(- - - - - -)

CALL DEVENT

STOP

FIG 2.10
FLOW CHART (2.1) FOR THE DERIVATION OF ACTUATOR PULSES A(I,J) FROM GIVEN PULSES G(I,J)

If all actuators are to be controlled by the same rotary valve, the sums of individual actuator pulses (J) must be the same. Therefore sum G(I,J) for each actuator [SUM(I)], find the greatest sum [SUMC], derive multiplication factor F(I) for each actuator to maintain all pulses in given proportions, hence derive actuator pulses A(I,J).

FLOW CHART (2.4) FOR THE SUMMATION OF ELEMENTS OF TWO-DIMENSIONAL REAL ARRAYS

FUNCTION TOTAL(XO, NG, A, I)

A — ELEMENTS WHOSE SUM IS REQUIRED
I — ROW NUMBER
NG — NUMBER OF ELEMENTS IN THE ROW
XO — INITIAL VALUE OF SUM
TOTAL — FINAL VALUE OF SUM

FIG 2.11
FLOWCHART(2.2) FOR THE DERIVATION OF ACTUATOR PULSES (I,J) TO GIVE THE MOST SUITABLE PRESSURE-EXHAUST SECTION ARRANGEMENT ON THE BOBBIN OF THE ROTARY VALVE FOR CONTROL OF ACTUATORS DESCRIBING PRESCRIBED TIME-DISPLACEMENT PROFILES

To ease manufacture of the bobbin, actuator channels should have a minimum number of corners and cross-drilling should be avoided. Take an actuator A with derived pulse sequence (A1, J), select a pulse sequence (B1, J) for that sequence calculate the number of actuator channel corners NSEP(N) and note the minimum number of corners NPLOT(I) for each actuator. Regenerate the pulses (B1, J) that give the minimum number of actuator channel corners for each actuator. Finally calculate the distance P(SO(I)) through which the other actuator port connection point must be shifted to maintain the required phase sequence.
NOTE: ABSS—Absolute value of \((S-MM)\)  
ABSE—Absolute value of \((E-NN)\)  
ABSV—Absolute value of \((V-JJ)\)

FLOW CHART (2·3) FOR THE CALCULATION OF CO-ORDINATES \(X(I,J), Y(I,J)\) OF ACTUATOR CHANNEL CORNERS FROM THE DERIVED PULSE SEQUENCES \(A(I,J), B(I,J)\)

START  
INPUT \(A(I,J), B(I,J), \text{SUMX, ISTA(I), ISTB(I), DY} \)

DO AA  
\(I = 1: \text{MAX} \)

\(R=I \)

\(E = R/2.0 \)

\(NN = 1/2 \)

\(\text{X}(I,J) = 0.0 \)

\(\text{Y}(I,J) = (2I-0.5) \times \text{DY} \)

\(0.1 < \text{ABSS} < 0.1 \)

ISTB(I) = 1  
ISTA(I) = 1  

\(\text{SUMA(I)} = \text{SUMA(I)} + A(I,N1+1) \)

\(\text{SUMB(I)} = \text{SUMB(I)} + B(I,N2+1) \)

\(M = \text{NPLT(I)} \)

yes  
\(XX = \text{SUMX} \)

\(\text{SUMA(I)} = \text{SUMA(I)} + A(I,N1+1) \)

\(\text{SUMB(I)} = \text{SUMB(I)} + B(I,N2+1) \)

\(N1 = N1 + 1 \)

\(N2 = N2 + 1 \)

DO BB  
\(J = 1: N3 \times M3+1 \)

STOP

FIG 2.13
FIG 2.14 A computer printout showing the pressure and exhaust regions for the prescribed profiles
2.4 Using the Developed Program to Get a Computer Print-out of the Geometry of the Pressure and Exhaust Regions

2.4.1 Introduction

From given input information, the program produces a developed view of the profile of the bobbin of a rotary valve used to control actuators placed side by side, describing different displacement/time relationships. The value of this lies in the fact that with a full size map, it can be wrapped round a cylindrical bar and the sculpturing of the bobbin can thus be greatly eased with the contours acting as rough machining guidelines.

In addition to controlling actuators placed in an axial configuration and describing different motions, such a valve can be used to control a very large number of actuators connected in line round the circumference and describing the same motion with phase differences. The limit to that number will depend on how close the connections can be made. This technique could be useful in Garside's multifeeded hydraulic circular weft knitting machine which is controlled by a rotary collar valve that generates the correct sequence of pulses. Another application could be in the hydraulic tufting machine, where rotary valves are used for the looper and needle mechanisms. If the pressure losses are taken into account, it is possible to drive both mechanisms off the same valve.

Here, the problem specified in Table T2.3 is taken as an example to explain various terms and indicate how the algorithm can be used.
2.4.2 Definitions of variables used in the program

i) NGP = Number of actuator pulses per cycle. This must be specified for each actuator. For example from Table T2.3 NGP(1) = NGP(2) = NGP(3) = 2, NGP(4) = NGP(5) = NGP(6) = NGP(7) = 4, NGP(8) = NGP(9) = 6 and NGP(10) = 8.

It is assumed that pulse sequences are such that the two ports of each actuator cannot be connected in line on the same circumference round the valve. Therefore two actuator channels A and B, one for each port are required. The B pulses are derived from the given A pulses. The number of possible pulse sequences for the B pulses will be equal to NGP(I). If the A pulses are 1, 2, 3, 4, 5, for example, the B pulses could be 2, 3, 4, 5, 1; 3, 4, 5, 1, 2; 4, 5, 1, 2, 3; 5, 1, 2, 3, 4; or 1, 2, 3, 4, 5. The one that results in the minimum number of actuator channel corners (NPLOT(I)) is selected.

Note:

It is necessary to use N4GP(I) and N3GP(I) because the number of pulses per cycle is to be doubled for balancing the bobbin and the B pulses are to be shifted in relation to the A pulses in order to achieve the most suitable section arrangement that gives the minimum number of actuator channel corners and therefore eases bobbin manufacture.

ii) IDD. This indicates whether the actual number of actuator pulses per cycle is odd or even. IDD(I) = 1 if the number is odd and zero if that number is even. For all actuators NGP(I) is input as an even number and IDD(I) is used to differentiate between the two cases.
iii) Since it is assumed that the required pulse sequence is such that the two ports of any actuator cannot be connected in line on the same circumference round the bobbin and therefore two pulse channels A and B, one for each port are required for each actuator, ISTA(I) or ISTB(I) is used to indicate the nature of the first A or B pulse. It is 1 if the first pulse is pressure and zero if the first pulse is exhaust. The convention is adopted since in general not all ports pressurize or exhaust at the same time. In the example studied for instance, the first A pulse was exhaust for actuators 1, 4, 5, 7, 8 and was pressure for actuators 2, 3, 6, 9 and 10 as was prescribed by the phase relationship.

Note: On input

ISTA(I) = 1 if the first A pulse is pressure and IDD(I) = 0
ISTA(I) = 0 if the first A pulse is exhaust and IDD(I) = 0
ISTA(I) = 0 if the first A pulse is pressure and IDD(I) = 1
ISTA(I) = 1 if the first A pulse is exhaust and IDD(I) = 1

iv) DY. This is the actuator channel width. It is not necessary that DY be made the same for each actuator, although it was kept constant in this case. It was also assumed that one line supplies or exhausts up to two actuators with supply and return lines alternating (Table T2.2). In general the axial spacing of the actuators can be varied by simply varying the Y coordinate of the origin of each contour(Y(I,1)).

v) GP. These are the specified individual actuator pulses per cycle and they represent the durations of each pulse. (Sum of acceleration, constant velocity, deceleration and dwell times).
Note:
While inputting GP(I,J), (J = 1, N4GP(I)) the first and last pulses are added if IDD(I) = 1 and the second pulse becomes the first. This is why NGP(I) is always even and ISTA(I) takes on the other value. For example:
Actuator 3, sequence 2,1,2 is changed to 1,4 and ISTA(3) = 0
Actuator 7, sequence 1,1,2,3,3 is changed to 1,2,3,4 and ISTA(7) = 1
Actuator 9, sequence 1,1,2,1,2,1,2 is changed to 1,2,1,2,1,3 and
ISTA(9) = 0

vi) The form of each curve will depend on whether the first A or B pulse is pressure or exhaust and also on the position of the actuator along the axis of the valve, I even or odd, i.e. whether it is the 2nd, 4th, 6th, ... or the 1st, 3rd, 5th ... actuator along the axis counting from one end of the valve bobbin. It also depends on the relative sums of the pulses at a point, \( \Sigma A(I,J) \) and \( \Sigma B(I,J) \) as well as on whether the curve was previously going up or down. The convention used is that \( K = 1 \) if the curve was going down and \( K = 2 \) if it was previously moving up.

\[
\text{SUM}(I) = \text{sum of the GP}(I,J) \ N4GP(I) \text{ pulses}
\]
\[
\text{SUMA} = \text{sum of the first A}(I,J) \ N1 \text{ pulses}
\]
\[
\text{SUMB} = \text{sum of the first B}(I,J) \ N2 \text{ pulses}
\]
\[
\text{SUMA} = \text{sum of the first A}(I,J) \ (N1+1) \text{ pulses}
\]
\[
\text{SUMB} = \text{sum of the first B}(I,J) \ (N2+1) \text{ pulses}
\]
\[
\text{ISA} (\text{or ISB}) = \text{number of A (or B) pulses to be added to SUMA, (or SUMB) for the next calculation}
\]
\[
\text{SUMF} = 0.5 \times \text{SUMC} = \text{circumference of the bobbin. Generally P, NSEP(N) and NPLT(I) refer to the number of points on a contour.}
vii) XX, YY values of the X and Y coordinates to be used in the next
calculation as the origin of an element of the curve. X(I,J),
Y(I,J) are the calculated values of the coordinates of the
actuator channel corners.
XP(J), YP(J) are calculated values of the coordinates of the
actuator channel corners derived from X(I,J), Y(I,J) by changing
from a two-dimensional array to a one-dimensional array required
by the graph plotter.

viii) SHIFT, PLUS, EVEN. In cases where IDD(I) = 1, it is necessary
to shift the origin of a contour to maintain the required phase
sequence. This is done by employing a factor EVEN(I) which is
equal to the amount of shift required. EVEN(I) = 0 when IDD(I) = 0.

2.4.3 Procedure for using the developed program

The procedure for using the program is given in the general flow
chart, Figure 2.10 and consists of the following steps:

1. DECLARE COMMON

Generally, a variable named and used in one segment of a complete
program will be kept quite distinct from a variable of the same name in
any other segment. However, if the name occurs as an actual parameter
of a FUNCTION or SUBROUTINE, then the value will be transmitted between
the segments. To keep variables in different segments the same as each
other, (they may or may not be given the same name) a COMMON declaration
in each of the relevant segments is used. So altering the value of any
one variable, alters the values of the others. Essentially, COMMON
enables the same storage location to be used in different segments, with or without the same name being used in each segment.

2. **DIMENSION**

   It is also necessary to employ subscripted variables (e.g. representing arrays of numbers, matrices, vectors) in the program. This requires a facility by means of which a single name is given to a set of storage spaces and then reference made to these using both the name and the particular number of the storage position within that name.

3. **CALL C1051N**

4. **INPUT** NGP(I), IDD(I), ISTA(I) and DY (I = 1, MAX)

5. **FLOW CHART 2.1 BETWEEN DO AA AND CC CONTINUE**

6. **FLOW CHART 2.2 BETWEEN DO AA AND AA CONTINUE**

7. **FLOW CHART 2.3 BETWEEN DO AA AND AA CONTINUE**

8. **CALL WINDOW (2)**

9. **CALL AXIPL(0) (-----)**

10. **GENERAL FLOW CHART, Figure 2.10, BETWEEN DO AA AND STOP**

11. **END.**

**Note:**

i) C1051N, WINDOW(2), AXIPL(-----), GRASYM (-----) and GRAPOL (-----) are standard subroutines available at the University Computer Centre and can be called when producing graphs. Other subroutines for the same purpose are also available and can be used depending on the format required. If only the X and Y coordinates are
required without having to produce a map, then these do not need to be called.

ii) By employing write statements it is possible to output various quantities like $\text{SUMX}$, $\text{EVEN(I)}$, $\text{A(I,J)}$, $\text{X(I,J)}$, $\text{Y(I,J)}$ etc at any point in the program.

iii) BLOCKS ZZZ01-ZZZ14 give the basic elements of curves and coordinates of points separating the pulse generating sections cut on the bobbin. These blocks can be written into the program as subroutines together with the function $\text{TOTAL}$ and are then called by appropriate statements when required. In Flow Chart 2.2 where it is required to generate the most suitable pulse sequence for the B pulses, BLOCKS ZZZ01-ZZZ14 are written to output $\text{ISA}$, $\text{ISB}$, $\text{NSEP(N)}$ and $\text{K}$ while in Flow Chart 2.3, where they have to be given other names, they have to output $\text{X(I,J)}$ and $\text{Y(I,J)}$ as well.
SECTION THREE
DESIGN PROCEDURES FOR ACTUATOR-ROTARY VALVE COMBINATIONS

3.1 Introductory Summary

As with conventional mechanical machines, the basic components of an hydraulic tufting machine are a looper unit, a tufting head and a backing fabric feed mechanism. A diagrammatic arrangement of these units is given in Figure 1.07. The tufting action is as follows:

Each needle draws yarn as it moves downwards through the fabric. When the needle reaches full travel, the looper moves in between the needle and yarn. When the needle withdraws, the looper holds the yarn thus forming a loop. The fabric is then advanced through a stitch length and when the looper withdraws, a knife cuts the loop thus forming a tuft (Section 1.5). Each needle-looper-knife combination operates through the same cycle simultaneously when plain tufted fabric is being produced. The needle, looper, and knife motions must be correctly phased at all operating speeds to ensure continuous production of fabric of acceptable quality. Figure 3.01 shows the envelopes of these motions and the phase relationships required for satisfactory tufting.

In the hydraulically actuated tufting machine constructed in the Department, the needles, loopers and knives are driven by actuators which are powered by rotary valves. Thus, the machine uses rotary valves driving actuators which describe prescribed displacement/time profiles to carry out the tufting process. The actuators therefore
FIG 3.01

POSSIBLE NEEDLE AND LOOPER DISPLACEMENT-TIME ENVENLEPS

LOOPER DISPLACEMENT

NEEDLE DISPLACEMENT

\[ \frac{d^2 X_a}{dt^2} + \frac{K dX_a}{M dt} + \frac{R X_a}{M} = \frac{PA}{M} \]

\[ \frac{d^2 X_a}{dt^2} + \frac{K dX_a}{M dt} + \frac{R X_a}{M} = \frac{PA}{M} \]

Acceleraton

Constant velocity

Deceleration (Cushioning)

Acceleraton

Constant velocity

Deceleration (Cushioning)

Dwell forward

Dwell down

Dwell back

REQUIRED LOOPER AMPLITUDE

REQUIRED NEEDLE AMPLITUDE

FIG 3.01 POSSIBLE NEEDLE AND LOOPER DISPLACEMENT-TIME ENVENLEPS
have to move particular masses certain distances in specified time
intervals. For satisfactory system performance, there are certain
points that must be taken into account when specifying the problem
and implementing the procedures for the design of an actuator-rotary
valve combination for application to a specific manipulative cycle.
While it is clear that some parameters like the dwell times, amplitude
of motion, frequency of operation in terms of number of pulses per
second will have to be specified, it is also certain that some para­
meters like actuator masses, diameter of the driving rotary valve etc
will have to be selected to achieve the required responses.

In this section, factors that must be taken into account in the
specification of the problem, and the design and operation of the
system are pointed out. Also derived are equations of motion of actua­
tors under different operating conditions. These equations can be
used to calculate the time increments of the cycle for actuator acce­
leration, constant velocity and deceleration as indicated in Figure
3.01. Also discussed, is the relationship between the diameter of the
rotary valve used to drive actuators, the speed at which the valve is
driven and the resulting durations of the pulses. From this relationship,
it is possible to select the diameter of the bobbin and the speed at
which it is driven so that the actuators it drives move with the speci­
fied motions to meet specified dynamic requirements. At the end of
this section, a procedure for designing hydraulic circuits involving
rotary valves and reciprocating mechanisms is proposed. Expressions
for estimating leakage flows, actuator flow and power requirements,
apparent effective mechanism mass in the circuit, viscous drag coeffi­
cients and the viscous torque on a rotating shaft, are derived in
Appendix Two.
3.2 Factors to be Considered in the Specification of the Problem, and the Design and Operation of an Hydraulic Valve-Actuator System

From Figure 3.01 which shows the envelopes of motion and phase relationships that exist when tufting, the importance of a suitable system specification is clear. Because the situation is characterized by the need to move particular masses certain distances in specified time intervals, the specification will include:

i) The masses \( M \) being moved. In addition to the mass of the actuator and everything attached to it, this will include the apparent mass of fluid being moved around in the circuit. The involvement of this mass of fluid makes it difficult to determine an accurate value of \( M \) but an approximation can be found from the proposed circuit configuration. It is essential that the value be determined because it has a significant effect on system performance as indicated in Figure 3.06 which shows that on average, the actuator took 15 milliseconds to travel 50 mm whereas neglecting the apparent mass of the fluid in the circuit, the calculated value would have been about 5 milliseconds. In Appendix Two, a method is given for the calculation of this apparent mass which must be added to that of the actuator and its attachments.

ii) The required actuator displacements or amplitudes, \( X_a \). The specified values will be those necessary for the required displacement of each driven element to carry out its part of the manipulative process as required for say tufting, knitting and sewing.
iii) Actuator external work or resistance $R_x$. In addition to any spring loading that the actuator might be working against, motion of the actuator will be resisted by both mechanical and viscous friction between parts in relative motion. The viscous drag force can be calculated from $F = \frac{A_w \mu}{C_r} \frac{dX_a}{dt}$, Appendix Two, and the mechanical friction force can be estimated from the fact that it will be proportional to the product of a friction coefficient and the normal force. These expressions will give approximate values, as a single value for reciprocating movements is impossible to specify.

iv) The sum of actuator acceleration, constant velocity, deceleration and dwell times $t$. An actuator port must remain connected to a valve pressure/exhaust section long enough for the actuator to describe the prescribed motion for the required action. Therefore this time will determine the machine cycle rate in terms of say number of stitches per second. It will also have an effect on the diameter of the valve driving the actuator, Section 3.6. As indicated in Figure 3.01 $t$ consists of four parts; acceleration, constant velocity, cushioning, and dwell. For given values of $X_a$, $M$, $R_x$, $P$, $A$ and $K_v$ ($K_v =$ effective circuit damping coefficient, $A =$ effective actuator working area and $P =$ pressure driving the actuator) equation

$$\frac{d^2 X_a}{dt^2} + \frac{K_v}{M} \frac{dX_a}{dt} + \frac{R_x}{M} X_a = \frac{PA}{M}$$

can be used to calculate actuator acceleration + constant velocity
time or deceleration time. Although these times can be adjusted by varying the values of some of the parameters e.g. P, A and M, the dwell times on the other hand cannot because they will depend on what the system is used for. So in a sense, these will set the upper speed limit at which the system can be operated to successfully perform the desired task. When drawing up a specification, it is important to remember that for fixed parameter values, running the system at higher speeds first diminishes the dwell times and at still higher speeds, these may be reduced to zero, leading to attenuation of the actuator amplitude of travel and system malfunction. On the other hand running at lower speeds increases the dwell times.

Another time interval that might have to be specified is $t_c$ (Figure 3.01), the period during which there is no movement of either the needle or looper. If both needle and looper are to be driven by the same valve, then $t_c$ must be taken into account in the design and manufacture of the valve. This will be a simple matter since it will involve only the relative positioning of the pressure/exhaust sections to be machined on the bobbin. If the needle and looper are to be driven by separate valves, $t_c$ will be set simply by adjusting the relative starting positions of the pulses generated by the valves (Section 5.7.1). But as noted above, running the system at higher speeds diminishes the dwell times and therefore $t_c$.

v) The force $F$ (Force = effective pressure * Effective area), required to move all the masses against all the resistances so that the actuators move with
the specified motions to produce the required action. This force will depend on the effective pressure $P$ and the working area of the actuator $A$. After the whole system has been designed and constructed, all other variables except the pressure will have fixed values. It is also important to note that the flow of fluid between valves and actuators will be resisted by conduits and bends resulting in further pressure drops (Figure 3.07).

So while a nominal value of pressure must be specified for the calculation of the other variables such as the mechanism response time, final tuning by pressure adjustment will be necessary after the system has been constructed and tested.

vi) The hydraulic power pack supplying the system and the rotary valve drive motor. The maximum flow rate available and the system operating pressure will depend on the power rating of the pack selected to drive the system. Since there will be leakage and viscous drag between parts in relative motion, it is necessary to consider the leakage characteristics, flow, and power requirements of the actuator and rotary valve. The motor selected to drive the rotary valve must be capable of overcoming the inertia of the valve, friction between the bobbin and housing and the viscous drag of the fluid. Like the actuator resistance $R_x$, some of these will be functions of the wetted surface area $A_w$, radial clearance $C_r$ and fluid coefficient of viscosity $u$ ($F = \frac{A_w u}{C_r} \frac{dX}{dt}$). They will therefore depend on the attained accuracy of machining and fluid operating temperature. All the same a rough estimate will be necessary for the preliminary calculations. After the
system has been constructed it may be possible to obtain more realistic values from experimental results and therefore tune the system for optimum operating conditions.

These are some of the factors that must be considered when specifying, designing and operating a functionally efficient valve/actuator system. One aim of these investigations was to try and set a procedure that can be followed in carrying out these processes. Expressions which can be used to determine some of the relevant parameters are given in the following sections and in Appendix Two.

3.3 Equations of Motion of the Actuator

While developing the algorithm for producing geometric developments of balanced bobbins of rotary valves used to control actuators operating in phase related sequences, it was noted that the actual individual actuator pulses, \( GP(I,J) \) representing the sums of acceleration, constant velocity, deceleration and dwell times, would have to be input. For given dimensions, these will set the upper limit at which the rotary valve can be driven without the actuator amplitudes of travel attenuating. To obtain these, the dynamic behaviour of the hydraulically powered mechanism would have to be known at the outset of the design. The required times can then be obtained by solution of the differential equations of motion.

The general force balance equation for an actuator can be written as:

\[
\frac{d^2 X_a}{dt^2} + \frac{K_v}{M} \frac{dx_a}{dt} + R_a \cdot x_a = PA \quad \text{or} \quad \frac{d^2 X_a}{dt^2} + \frac{K_v}{M} \frac{dx_a}{dt} + \frac{R_a}{M} \cdot x_a = \frac{PA}{M}
\]
which is the standard form of a nonhomogeneous second order differential equation. The characteristic equation will have roots 

$$ h_{1,2} = -\frac{K_v}{2M} \pm \sqrt{\left(\frac{K_v}{2M}\right)^2 - \frac{R_x}{M}} $$

which, depending on the relative values of $K_v, M$ and $R_x$, may be:

i) Real and distinct \( \left(\frac{K_v}{2M}\right)^2 > \frac{R_x}{M} \) (overdamped system)

ii) Real and repeated \( \left(\frac{K_v}{2M}\right)^2 = \frac{R_x}{M} \) (critically damped system)

iii) Complex conjugates \( \left(\frac{K_v}{2M}\right)^2 < \frac{R_x}{M} \) (underdamped system)

and in all cases the Particular Solution due to the constant term \( \frac{P_A}{M} \) will be \( X_{ap}(t) = \frac{P_A}{R_x} \).

This is the general case of a spring-loaded hydraulically actuated mechanism. For different actuator configurations and operating conditions, appropriate interpretations must be given to the coefficients.

3.3.1 Interpretation of the product $P_A$ in the case of a spring-loaded actuator moving a constant load e.g. a single-acting piston

The term $P_A$ is interpreted as the algebraic sum of the initial compression force of the spring, the constant load force (may be including that due to gravity) and the hydraulic pressure force.

3.3.2 Roots of the characteristic equation and interpretation of the product $P_A$ in the case of a constant load without a spring

The above differential equation reduces to
\[
\frac{d^2 X_a}{dt^2} + \frac{K_v}{M} \frac{dX_a}{dt} = \frac{PA}{M}
\]

and the characteristic equation will have roots \( h_{1,2} = 0, -\frac{K_v}{M} \) which are real and distinct. The Particular Solution due to the constant term \( \frac{PA}{M} \) will be \( X_{ap}(t) = \frac{PA}{K} t \). As in the case of the spring-loaded piston above, \( PA \) is interpreted as the algebraic sum of the constant load force and the hydraulic pressure force.

3.4 The Total Solution and Procedure for Determining Values of Some Parameters From Observed Performance

The Total Solution for each case can be written and on successive differentiation with respect to time, expressions for the velocity \( \dot{X}_a(t) \) and acceleration \( \ddot{X}_a(t) \) are obtained. These are given in Table T3.1 where in determining the various constants, it has been assumed that the system is initially at rest, that is \( \dot{X}_a(0) = \ddot{X}_a(0) = 0 \).

The object of carrying out these investigations was to obtain expressions from which acceleration, constant velocity and deceleration times can be calculated. The predicted values of these times will depend on the accuracy with which the parameters \( A, P, R_x, M \) and \( K_v = f(A_w, \mu, C_r) \) used in the expressions are determined. \( A \) and \( P \) can be determined reasonably accurately by instrumentation and measurement. Therefore the problem is reduced to that of fixing \( R_x, M \) and \( K_v \). For the critically damped case and the case of constant loading without a spring, \( R_x, M \) and \( K_v \) can be determined from observed performance using the expressions
<table>
<thead>
<tr>
<th>Constant Load with Spring (overdamped)</th>
<th>( x_a(t) )</th>
<th>( \dot{x}_a(t) )</th>
<th>( \ddot{x}_a(t) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( h_1 = \frac{-K_v}{2M} + \sqrt{\left(\frac{K_v}{2M}\right)^2 - \frac{R_x}{M}} )</td>
<td>( x_a(t) = \frac{PA}{R_x} \left[ 1 + \frac{h_2 e^{h_1 t}}{h_1 - h_2} - \frac{h_1 e^{h_2 t}}{h_1 - h_2} \right] )</td>
<td>( \dot{x}_a(t) = \frac{PA h_2}{R_x(h_1 - h_2)} \left[ e^{h_1 t} - e^{h_2 t} \right] )</td>
<td>( \ddot{x}_a(t) = \frac{PA h_2}{R_x(h_1 - h_2)} \left[ h_1 e^{h_1 t} - h_2 e^{h_2 t} \right] )</td>
</tr>
<tr>
<td>( h_2 = \frac{-K_v}{2M} - \sqrt{\left(\frac{K_v}{2M}\right)^2 - \frac{R_x}{M}} )</td>
<td>( x_a(t) = \frac{PA}{R_x} \left[ 1 + (h_1 - 1) e^{h^2 t} \right] )</td>
<td>( \dot{x}_a(t) = \frac{PA h^2 t e^{h^2 t}}{R_x} )</td>
<td>( \ddot{x}_a(t) = \frac{PA h^2}{R_x} \left[ h^2 t + 1 \right] e^{h^2 t} )</td>
</tr>
<tr>
<td>Constant Load without Spring (critically damped)</td>
<td>( h = \frac{-K_v}{2M} )</td>
<td>( x_a(t) = \frac{PA}{R_x} \left[ 1 + \frac{e^{h_1 t}}{h_1} \right] )</td>
<td>( \dot{x}_a(t) = \frac{PA}{R_x} \left( \frac{g^2 + h^2}{g} \right) e^{h_1 t} \sin gt )</td>
</tr>
<tr>
<td>( h = \frac{-K_v}{2M} )</td>
<td>( g = \sqrt{\frac{R_x}{M} - \left(\frac{K_v}{2M}\right)^2} )</td>
<td>( x_a(t) = \frac{PA}{R_x} \left[ 1 + \frac{e^{h_1 t} / g \sin gt - \cos gt}{h_1} \right] )</td>
<td>( \dot{x}_a(t) = \frac{PA}{R_x} \left( \frac{g^2 + h^2}{g} \right) e^{h_1 t} \sin gt )</td>
</tr>
</tbody>
</table>

**Table T 3.1** Equations for the displacement \( x_a(t) \), velocity \( \dot{x}_a(t) \), and acceleration \( \ddot{x}_a(t) \) of an hydraulic mechanism.
derived below. These expressions indicate that the pressure to
drive the actuator, the size of the actuator to drive a given load
and the hydraulic circuit configuration designed for the purpose,
can all to some degree be manipulated to yield a required response.

3.4.1 Determination of $R_x, M$ and $K_v$ for the critically damped case

The velocity of the actuator $\dot{X}_a(t)$ is determined at various
instants using dynamic measuring and recording equipment. From the
expression for the velocity, the ratio of velocities $(VR)$ at two
successive instants, $t_n, t_{n+1}$ is given by:

$$\frac{\dot{X}_a(t_n)}{\dot{X}_a(t_{n+1})} = \frac{PA h^2 t_n e^{ht_n}}{PA h^2 t_{n+1} e^{ht_{n+1}}} = \frac{t_n e^{ht_n}}{t_{n+1} e^{ht_{n+1}}}$$

Taking logarithms, $\ln(VR) = \ln(t_{n+1} e^{ht_{n+1}}) = \ln(t_n e^{ht_n})$

Having determined $h$, since $P$ and $A$ are known, $R_x$ can be found from
any value of $\dot{X}_a(t)$. 
and since the system is critically damped, \( \left( \frac{K_v}{2M} \right)^2 = \frac{R_x}{M} \)

\[ M = \frac{R_x}{h^2} \text{ and } K_v = 2\sqrt{R_x M} = \frac{2R_x}{h} . \]

Therefore from known values of \( A, P \) and \( \dot{x}_a(t) \) at various instants, the other circuit parameters can be determined.

### 3.4.2 Determination of \( R_xM \) and \( K_v \) for the case of constant loading without a spring

Similarly the acceleration of the actuator \( \ddot{x}_a(t) \) is determined at various instants. From the expression for acceleration, the ratio of accelerations \( (A R) \) at two successive instants \( t_n, t_{n+1} \) is given by

\[
(A R) = \frac{\ddot{x}_a(t_n)}{\ddot{x}_a(t_{n+1})} = \frac{-\frac{PA}{K_v} h t_n^2 e^{ht_n}}{-\frac{PA}{K_v} h t_{n+1}^2 e^{ht_{n+1}}} = \frac{h t_n^2}{h t_{n+1}^2} = \frac{t_n}{t_{n+1}}
\]

\[
(A R)_{t_n} = e^{ht_n}
\]

Taking logarithms, \( \ln (AR) + h t_{n+1} = h t_n \)

\[
\therefore \quad h = \frac{\ln (AR)}{t_n - t_{n+1}}
\]

Again since \( A \) and \( P \) are known, \( K_v \) can be determined from any value of \( \dot{x}_a(t) \) or \( \ddot{x}_a(t) \).
\[ K_v = \frac{PA}{\dot{X}_a(t)} [1 - e^{ht}] \text{ or } K_v = \frac{PA}{X_a(t)} e^{ht} \]

and since \( h = -\frac{K_v}{M} \), \( M \) is then given by

\[ M = -\frac{K_v}{h} \]

Therefore it is again possible to determine circuit parameters from observed performance.

By providing an indication of the magnitudes of some parameters in a specific hydraulic system, and the variables on which these parameters depend, this will help in the specification of operating conditions for other tasks and in the manipulation of the variables to obtain a desired response. As indicated above, determination of these parameters can be done for the two cases from the displacement-time traces obtained using dynamic measuring and recording equipment. Velocity and acceleration values at various instants are then determined from the traces and the various parameters calculated. The procedure and results can be presented as in Table T3.2. However, because of the logarithmic nature of the response, this technique is not directly applicable to the overdamped and underdamped cases.

3.5 Effect of Spring Loading the Actuator and Definition of the Effective Circuit Mass \( M \) and Damping Coefficient \( K_v \) when Using the Derived Expressions to Calculate the Time Increments of the Cycle for Actuator Acceleration, Constant Velocity and Deceleration

The manner in which the derived expressions are used to predict the system response will depend on the system operating conditions.
From the displacement/time trace of a uv recorder, derive average values of \( \dot{X}_a(t) \) over the interval.

\[ \dot{X}_a(t) = \frac{X_a(t_{n+1}) - X_a(t_n)}{t_{n+1} - t_n} \]

From \( \dot{X}_a(t) \) derive average values of \( \ddot{X}_a(t) \) over the interval.

\[ \ddot{X}_a(t) = \frac{X_a(t_{n+1}) - X_a(t_n)}{t_{n+1} - t_n} \]

From the derived values of \( \ddot{X}_a(t) \). From \( \dot{X}_a(t) \) and the derived values of \( \ddot{X}_a(t) \).

From working pressure \( P \), effective area \( A \) and the derived values of \( h \) and \( X_a(t) \) at instant \( t \). From the derived values of \( X_a(t) \).

<table>
<thead>
<tr>
<th>( t )</th>
<th>( X_a(t) )</th>
<th>( \dot{X}_a(t) )</th>
<th>( \ddot{X}_a(t) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( t_1 )</td>
<td>( X_a(t_1) )</td>
<td>( \dot{X}<em>a(t</em>{1,2}) )</td>
<td>( \ddot{X}<em>a(t</em>{1,2}) )</td>
</tr>
<tr>
<td>( t_2 )</td>
<td>( X_a(t_2) )</td>
<td>( \dot{X}<em>a(t</em>{2,3}) )</td>
<td>( \ddot{X}<em>a(t</em>{2,3}) )</td>
</tr>
<tr>
<td>( t_3 )</td>
<td>( X_a(t_3) )</td>
<td>( \dot{X}<em>a(t</em>{3,4}) )</td>
<td>( \ddot{X}<em>a(t</em>{3,4}) )</td>
</tr>
<tr>
<td>( t_4 )</td>
<td>( X_a(t_4) )</td>
<td>( \dot{X}<em>a(t</em>{4,5}) )</td>
<td>( \ddot{X}<em>a(t</em>{4,5}) )</td>
</tr>
<tr>
<td>( \vdots )</td>
<td>( \vdots )</td>
<td>( \vdots )</td>
<td>( \vdots )</td>
</tr>
<tr>
<td>( t_n )</td>
<td>( X_a(t_n) )</td>
<td>( \dot{X}<em>a(t</em>{n-1,n}) )</td>
<td>( \ddot{X}<em>a(t</em>{n-1,n}) )</td>
</tr>
</tbody>
</table>

In theory constant values would be obtained for these four quantities. This will not always be so because of experimental errors. So average values would be taken.

**TABLE T3.2**

DETERMINATION OF SOME PARAMETERS FROM OBSERVED PERFORMANCE
A spring load and the fluid in the conduits affect the accuracy of this prediction:

i) For a spring loaded actuator, the solid length of the spring imposes a limit on the maximum displacement obtainable which, for miniature actuators will be very small. Also because of the varying force of the spring, it will be difficult to get an accurate solution to the differential equations of motion.

ii) Because of water hammer effects, it is recommended to limit fluid flow velocities in the conduits to a maximum of 4.5 m/sec. The 'instant valve closure pressure rise' above the steady state level will then be about 52 bar which is generally considered to be a safe design value and is sometimes taken as a criterion for conduit selection. From theoretical considerations, the displacement of an actuator with a constant load, without a spring, will increase to infinity and the velocity tend to the maximum value of $\frac{PA}{K_v}$ as $t \to \infty$. Obviously these values are far in excess of the 4.5 m/sec and in practice the actuator displacement will be much less than infinity. So to keep fluid velocities in the conduits to a maximum of 4.5 m/sec while maintaining high actuator cycling rates (the needle actuator of the prototype tufting machine, having a stroke of 50 mm, can operate at 3000 stitches per minute. This is equivalent to a steady speed of 5 m/sec) the size of the conduits will have to be increased. This has the effect of increasing the effective circuit mass $M$ and reducing the effective circuit damping coefficient $K_v$, (Appendix Two). This increases the system response time and can lead to attenuation of the actuator amplitude of travel.
Therefore to calculate system response times using the derived expressions, M and K_v must be defined to include the effects of fluid motion in the conduits and the effects of circuit restrictions. With properly defined and accurately determined values of A, P, R_x, M and K_v the program MASTER time given below can be used to calculate mechanism acceleration + constant velocity times or deceleration times. The flow chart is given in Figure 3.02. The acceleration, constant velocity, deceleration, together with the dwell times specified for the particular application are required in the program that processes this information to give a map of the pressure and exhaust sections (Section 2.4). These times are also required for the determination of the diameter of the rotary valve and the speed at which it is rotated as noted in the following section.

3.6 Derivation of the Relationship Between the Diameter of the Rotary Valve (D), Valve Rotational Speed (N), Total Duration of Pulses per Cycle (T), Valve Land Width (l) and Actuator Port Diameter (d)

The rotary valve used to supply simple reciprocating mechanisms consists of a series of alternate pressure and exhaust regions machined round a bobbin to produce fluid sources and sinks. The fluid sources are separated from the sinks by narrow lands. The width of these lands must be accurately determined, because they affect the overall size of the valve, crossport leakage from the sources to the sinks, and the power required to turn the bobbin in its housing.
**DOCUMENT FORTRAN**

**MASTER TIME**

C A = EFFECTIVE AREA OF ACTUATOR
C C = CLEARANCE BETWEEN ACTUATOR AND HOUSING
C R = SPRING CONSTANT
C A W = WETTED AREA OF ACTUATOR
C A K = LINEAR VISCOUS DAMPING COEFFICIENT
C A M = ACTUATOR MASS
C F V = ABSOLUTE VISCOITY OF THE FLUID
C X A = ACTUATOR DISPLACEMENT
C X AV = ACTUATOR VELOCITY
C X AA = ACTUATOR ACCELERATION

AK=AW*FV/C

IF(R.EQ.0.0)GO TO 9
IF((AK/(2.0*AM)**2)-(R/AM)**3,5,7
3 WRITE(2,15)
WRITE(2,17)
H=-AK/ (2.0*AM)
G=SQRT(R/AM-{AK/(2.0*AM)**2})
DO 4
I=1,30
T=I
XAT=(P*A/R)*(1.0+(H*T)-EXP(H*T))
XAV=(P*A/R)*(1.0-EXP(H*T))
XAA=-XAV/R
WRITE(2,18)XAT,XAV,XAA
4 CONTINUE
GO TO 11
5 WRITE(2,13)
WRITE(2,17)
H1=-AK/(2.0*AM)
DO 6 I=1,30
T=I
XAT=(P*A/R)*(1.0+((H1*EXP(H1*T)))/(H1-H2))-(H1*EXP(H1*T))
XAV=((P*A-H1**2)/((H1-H2)**2)-((H1**2-H2**2)/(H1-H2))
XAA=-(P*A-H1**2)/(R*(H1-H2)**2)*((H1**2-H2**2)/(H1-H2))
WRITE(2,18)XAT,XAV,XAA
6 CONTINUE
GO TO 11
7 WRITE(2,12)
WRITE(2,17)
H1=-(AK/(2.0*AM))*SQRT(((AK/(2.0*AM)**2)-(R/AM))
H2=-(AK/(2.0*AM))*SQRT(((AK/(2.0*AM)**2)+(R/AM))
DO 8 I=1,30
T=I
XAT=(P*A/R)*(1.0+((H2*EXP(H2*T)))/(H1-H2))-(H2*EXP(H2*T))
XAV=((P*A-H2**2)/((H1-H2)**2)-((H1**2-H2**2)/(H1-H2))
XAA=-(P*A-H1**2)/(R*(H1-H2)**2)*((H1**2-H2**2)/(H1-H2))
WRITE(2,18)XAT,XAV,XAA
8 CONTINUE
GO TO 11
9 WRITE(2,12)

****

**SOLUTION**

**CONSTANT LOAD WITHOUT SPRING**

<table>
<thead>
<tr>
<th>XA</th>
<th>XAV</th>
<th>XAA</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.198E 04</td>
<td>0.299E 04</td>
<td>0.199E 04</td>
</tr>
<tr>
<td>0.545E 04</td>
<td>0.437E 04</td>
<td>0.428E 03</td>
</tr>
<tr>
<td>0.102E 05</td>
<td>0.501E 04</td>
<td>0.389E 03</td>
</tr>
<tr>
<td>0.154E 05</td>
<td>0.531E 04</td>
<td>0.428E 02</td>
</tr>
<tr>
<td>0.204E 05</td>
<td>0.552E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.262E 05</td>
<td>0.552E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.318E 05</td>
<td>0.552E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.375E 05</td>
<td>0.552E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.429E 05</td>
<td>0.557E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.486E 05</td>
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<td>0.557E 04</td>
</tr>
<tr>
<td>0.548E 05</td>
<td>0.557E 04</td>
<td>0.557E 04</td>
</tr>
<tr>
<td>0.596E 05</td>
<td>0.557E 04</td>
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<tr>
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<td>0.557E 04</td>
</tr>
<tr>
<td>0.708E 05</td>
<td>0.557E 04</td>
<td>0.557E 04</td>
</tr>
</tbody>
</table>

**CONSTANT LOAD WITH SPRING - OVERDAMPED**

**CONSTANT LOAD WITH SPRING - CRITICALLY DAMPED**

**CONSTANT LOAD WITH SPRING - UNDERDAMPED**

**FINISH**

**DATA**

4.4E-5 1.54E-3 1.41E-3 5.0E-5 5.5E6 5.66E-5 5.66E-2

**END**
FIG 3.02

FLOW CHART (3.1) FOR THE DETERMINATION OF ACTUATOR DISPLACEMENT $X_a(t)$, VELOCITY $X_a(t)$ AND ACCELERATION $X_a(t)$ FROM GIVEN VALUES OF A, P, $R_y(AM)$ and $K_y = f(AW, L, C_L) = (AK)$
i) To reduce cross port leakage the land should be as wide as possible because leakage depends on the pressure gradient $\left(\frac{\delta P}{\delta x}\right)$ across the land. But the overall diameter of the bobbin and therefore power consumption increases with land width.

ii) To reduce power consumption, the contact area between the lands and the valve housing must be reduced, which requires a reduction in land width resulting in increased leakage.

Related to the width of the lands, is the size of the actuator ports. For complete fluid control, the size of the valve increases with increasing port size. Cameron recommends that the land width should be made approximately equal to the actuator port diameter. This is valid for small valves but for large valves, the larger resultant contact area may increase the valve power consumption to an unacceptable level.

The speed at which the valve is driven also affects the diameter of the valve, because an actuator port must remain connected to a source or sink long enough for the actuator to go through its set displacement/time profile. At higher speeds the actuator has less time to complete its traverse and its amplitude of travel is therefore more likely to attenuate.

The diameter of the valve depends also on the number and total duration of pulses required during each rotation of the valve. It increases with the number of pulses and their durations.
A functional relationship between these variables was derived and can be used to calculate the nominal diameter of the valve.

If it is assumed that:

i) The sum of actuator acceleration time, constant velocity time, deceleration time and dwell time is equal to $t_n$, that is $GP(I,n) = t_n$ (Section 2.4).

ii) The actuator describes the required time-displacement profile only when the actuator ports are fully open.

iii) The valve speed is $N$ radians per second and $n$ pulses are required during each rotation of the valve whose diameter is $D$.

Then referring to the figure above, with all angles measured in radians,

![Diagram](image-url)
\[ \alpha_n = \frac{2\ell}{D}, \quad \beta_n = \frac{2d}{D}, \quad \therefore \phi_n = \frac{2}{D} (\ell + d) \]

For all the \( n \) pulses, the total angle taken up by the lands and ports will be

\[ \phi = \sum_{n=1}^{n} \phi_n = \frac{2n}{D} (\ell + d) \]

If the speed of the valve is \( N \) radians per second, then in \( t_n \) seconds, it turns through an angle \( \theta_n = Nt_n \) radians.

\[ \therefore \text{For all } n \text{ pulses } \theta = \sum_{n=1}^{n} \theta_n = \sum_{n=1}^{n} Nt_n \]

\[ \theta = N(t_1 + t_2 + \ldots + t_n) = NT \]

\[ \therefore \text{For the whole valve } 2\pi = \frac{2n}{D} (\ell + d) + NT \]

Solving, the diameter of the valve is obtained as

\[ D = \frac{2n (\ell + d)}{2\pi - NT} \]

and if the actuator port diameter \( d \) is equal to the land width \( \ell \) separating the pressure and exhaust regions as suggested by Cameron, then

\[ D = \frac{4nd}{2\pi - NT} \]
This equation reveals important facts related to problem specification, design and operation of a rotary valve system as indicated below.

### 3.6.1 Relationship between total duration of pulses per cycle (T) and valve rotational speed (N)

For real actuator diameters \( D > 0 \), the denominator \( (2\pi - NT) \) in the expression \( D = \frac{2\pi (x + d)}{2\pi - NT} \) must be positive. This is so because \( (2\pi - NT) \) represents the total angle taken up by the lands and ports. If it were negative or zero, it would mean that land widths were negative and therefore there would be no separation between the pressure and exhaust regions. Therefore \( T \) must be less than \( \frac{2\pi}{N} \). At constant supply pressure, the actuator/valve operating point will always be below the curve \( T = \frac{2\pi}{N} \). This means that a specified value of \( T \) determines the maximum valve speed \( N \). Running at higher speeds first reduces the dwell times and at still higher speeds, actuator movement attenuates. At lower speeds the dwell times, and therefore \( T \), are increased. Therefore \( T \) and \( N \) must be specified such that the operating point will always be below the curve \( T = \frac{2\pi}{N} \). Figure 3.04 which is a graph showing the relationship between \( N \) and \( T \) indicates that it is possible to design a valve that will give a \( T \) value of 104.7 milliseconds running at speeds less than 60 radians per second. At higher speeds, \( T \) will correspondingly decrease. For example at 150 radians per second, \( T \) will be reduced to less than 42 milliseconds.
RELATIONSHIP BETWEEN TOTAL DURATION OF PULSES PER CYCLE \( (T) \) AND VALVE ROTATIONAL SPEED \( (N) \)

\[ T = \frac{2\pi}{N} \]

FIG 3.04
3.6.2 Influence of valve land width ($\varepsilon$) and actuator port diameter ($d$) on valve design and performance

If $N$ and $T$ are specified such that the actuator/valve operating point will always be below the curve $T = \frac{2\pi}{N}$, equation $D = \frac{2n(\varepsilon + d)}{2\pi - NT}$ indicates the relationship that must exist between the diameter of the valve $D$ and $NT$. There will be very little freedom in the selection of suitable values of $\varepsilon$ and $d$ because:

i) $d$ must be large enough to meet the actuator flow requirements. Smaller values of $D$ cause flow starvation of the actuator by increasing restriction to flow. This results in large pressure drops across the orifice, so that less thrust is available to drive the actuator. Since $T$ is a function of the pressure, the actuator might not have sufficient time to move through the travel required to perform the prescribed task.

ii) The land width $\varepsilon$ must be selected appropriately. Large values increase the bobbin-housing contact area and therefore the power consumption of the valve. Small values result in increased leakage from the pressure to the exhaust regions. This again increases the power consumption of the valve because enough flow must be supplied to drive the actuator and to compensate for leakage losses.

iii) The relationship between the product $NT$ and $D$ can be presented graphically as in Figure 3.05. This figure indicates that having fixed $NT$, $D$ must be selected such that the operating point is above the curve $\frac{D}{n(\varepsilon + d)} = \frac{2}{2\pi - NT}$. If $NT$ is specified to be 4.0 radians for example, $D$ must be selected such that $\frac{D}{n(\varepsilon + d)}$
RELATIONSHIP BETWEEN VALVE DIAMETER (D), VALVE ROTATIONAL SPEED (N) AND TOTAL PULSE DURATION (T)

\[
\frac{D}{n(l+d)} = \frac{2}{2\pi - NT}
\]
is at least equal to 0.88 because if it is less, say 0.5, the maximum value of NT will be less than 4.0 radians. For 0.5 radians it is 2.28. Therefore to avoid actuator travel attenuation, valve speed will have to be reduced. The system will then be operating at a lower frequency than that specified. On the other hand if it is greater, 2.5 say, the system can be operated at higher frequencies without actuator travel attenuation or, if the speed N is to be kept constant, the actuator will have longer dwell periods.

3.6.3 An alternative method of specifying and measuring the total duration of pulses per cycle (T)

T represents the total time during which the actuated mechanism moves and dwells to perform the desired task. It does not include the time taken by the valve to open and close the actuator ports. While N and T can be specified such that \(2\pi - Nt > 0\), once the valve and actuators are manufactured, it will be difficult to accurately measure values of T at different speeds N in order to find out whether the system is operating as specified. An alternative is to specify the periodic time \(T_p\) for one revolution of the valve. This can be accurately measured and from this value, T can be deduced. \(T_p\) is the time \(\frac{2\pi (\phi + d)}{ND}\) required to turn through the port and land widths, (angle \(\phi\)), + time \(T\) required to turn through the pulse generating sections, (angle \(\theta\)).

\[T = T_p - \frac{2\pi (\phi + d)}{ND}\text{ and since } N = \frac{2\pi}{T_p}\]
\[
T = T_p \left[ 1 - \frac{2n(\ell + d)}{2\pi D} \right]
\]

For a manufactured valve, \( n, \ell, d \) and \( D \) will be constant. Therefore \( T \) can be deduced from the above expression by measuring the periodic time \( T_p \) and multiplying this by the constant \( \left[ 1 - \frac{2n(\ell + d)}{2\pi D} \right] \). In a similar way to the above consideration if a value of \( T_p \) is specified, running the system at a higher speed (lower value of \( T_p \)) will result in less dwell time and eventually, actuator travel attenuation. Running at lower speeds, that is to say higher values of \( T_p \), results in more dwell time. This is illustrated in Figure 3.06. Here a valve/actuator assembly was run at various speeds and the periodic time \( T_p \) deduced from cyclic recordings. This figure indicates that as a result of increasing the speed of the valve, \( T_p \) is reduced. The dwell times and therefore \( T \) are reduced, making it more likely to attenuate at higher speeds.

In conclusion, equation \( D = \frac{2n(\ell + d)}{2\pi - 2nT} \), where \( T \) is the total time in a cycle during which both pressure and exhaust pulses are required, sets the maximum cycling rate at which the system can be operated without actuator travel attenuation or reduction in dwell times. The acceleration, constant velocity and cushioning part of \( T \) will be calculated from the equation of motion

\[
\frac{d^2 x_a}{dt^2} + \frac{K_v}{M} \frac{dx_a}{dt} + \frac{R_x x_a}{M} = \frac{PA}{M}
\]

while the dwell part of \( T \) will be specified and will depend on the displacement/time relationship required for a specific application.
3.7 Proposed Procedure for Designing Hydraulic Circuits Involving Rotary Valves and Reciprocating Mechanisms

Figure 3.07 shows a schematic hydraulic circuit in which a reciprocating cushioned actuator is controlled by a rotary valve. Under the action of the applied pressure, the mechanism will reciprocate as the valve is rotated. The resultant displacement/time profile will then depend on the imposed boundary conditions and the losses occurring within the system. At the specification and design stage, it is important to have an idea of how these factors will affect system performance. Expressions for estimating leakage flows, actuator flow and power requirements, apparent effective mechanism mass in the circuit, viscous drag coefficients etc are given in Appendix Two and can be used to calculate approximate values of the various quantities from the proposed circuit configuration and load characteristics.

The design procedure can be broken down into a number of related operations:

i) Inputting the load-displacement specification,

ii) Selecting a suitable driving pressure at the actuator, pipe size and circuit configuration to give minimum power loss due to flow resistance,

iii) Determining suitable actuator dimensions to meet specified operating conditions. The load-displacement specification will enable determination of the relationship between the pressure at the operating piston and its effective working area.
ACTUATOR/ROTARY VALVE COMBINATION — SCHEMATIC HYDRAULIC CIRCUIT AND LOSSES

Hydraulic pump

Reservoir

Rotary supply valve

Leakage flow

Effective circuit mass \( M \)

\[ M = m + \left[ A_1 + A_2 + A_3 + \ldots \right] \]

Equation of motion

\[ \frac{d^2}{dt^2} = \frac{PA}{M} \]

Friction

\[ F = N \cdot \frac{v}{V} \]

Resistances due to abrupt changes in pipe cross section and due to the geometry of pipe entrances and exits

Resistance coefficients of pipe bends (J.J. Taborak, "Fundamentals of Fire Flow" machine design magazine, April 16, 1959)

Resistance coefficients due to the geometry of pipe entrances and exits with large reservoirs.

Effective circuit damping coefficient \( K_w \)

Must include viscous friction, mechanical friction and effect of fluid movement in conduit connecting valve and mechanism.

FIG 3.07
iv) From actuator dimensions and required dynamic performance, determining the valve dimensions, the power required to drive it, the expected leakage through clearances and the power losses due to frictional effects in the flow circuit.

v) Selecting the appropriate hydraulic pump, its drive and associated circuitry.

Figure 3.08 shows in detail the proposed procedure in the form of a flow chart. It must be emphasized that while some parameters will be specified and others determined from the expressions given in Appendix Two, final system tuning may be necessary to achieve the required dynamic performance.
ACTUATOR/ROTARY VALVE COMBINATION — HYDRAULIC CIRCUIT DESIGN PROCEDURE

START

INPUT LOAD — DISPLACEMENT SPECIFICATION
Number of actuators to be controlled, their positions, and the positions of the supply and exhaust lines to the rotary valve.
For each actuator:
- Load mass,
- Load spring characteristics,
- Actuator cycle rate,
- Maximum actuator displacement,
- Maximum actuator (Load ⇒ Fluid) velocity,
- Duration of dwell periods.

Select appropriate driving pressure $P$

Select appropriate pipe size

Determine suitable circuit configuration to give minimum power loss due to flow resistance.

Calculate effective circuit mass $M$ and effective circuit damping coefficient $R_H$

Determine appropriate actuator dimensions

Calculate:
1. Actuator (Load) acceleration + uniform travel time:
   \[ \frac{d^2 x_a + k_s x_a}{m} + \frac{dx_a}{dt} + \frac{R_h x_a}{M} = \frac{P A}{m} \]
   Leakage flow $Q_L = \frac{(P_x^2 - P)^2}{2C}$ + $V Q$

Because of strength and space limitations, required performance may not be available with the selected pressure. Therefore a different driving pressure must be set.

Select rotary valve land width $l$ and actuator port diameter $d$ to give:
1. Minimum contact area between valve and its housing.
2. Minimum leakage across valve lands — will be $\propto$ pressure gradient if laminar.
3. Enough flow to the actuator to meet specified performance.

Performance satisfactory

Compare total durations of pulses with specification:
- Acceleration times + constant velocity times
- Deceleration times + dwell times

Select rotary valve dimensions $D = \frac{2\pi(l+d)}{2\pi(l+d)}$

Calculate:
- Circuit flow requirements — Actuator displacement flows + Leaks due to valve and actuator clearances.
- Total system pressure — Actuator driving pressure $P$ + Losses due to pipe friction, bends, fittings and changes in flow section and direction.

Hydraulic power supplied to circuit $= R Q$
from actuator dimensions and working speed, total volume of accumulator is equal to the volume of fluid stored + volume of compressed gas (accumulator could be of a different type e.g. spring or weight) necessary to supply the fluid to the actuator at its working pressure.

SELECT HYDRAULIC PUMP, DRIVE, RESERVOIR AND ACCUMULATOR

STOP

FIG 308
SECTION FOUR
THE USE OF LIMITED ROTARY STROKE VALVES TO CONTROL
TUFTING MACHINE NEEDLE POSITIONS FOR
PATTERNED FABRIC PRODUCTION

4.1 Introductory Summary

The continuing trend towards automation of processes often requires manipulative machinery to have greater flexibility of operation and more precise control of phase and position of mechanisms than manually supervised machines. Features which are important in a system being considered for an automatic process such as textile fabric production include the cost, reliability and ability of the system to be easily re-programmed to modify the design and improve the quality of the product. For this reason attention must be focussed on components that respond fast and with good repeatability to electrical commands. The fact that hydraulic systems can now be designed to operate at speeds far in excess of existing mechanical mechanisms in certain categories of fabric producing machinery has been clearly demonstrated in the review of work on these techniques set out earlier. This is because of the superior cycling rates due to higher possible accelerations and velocities. It is also certain that electronic devices like microprocessors can now be used to achieve accurate and virtually infinitely variable control of many industrial processes.

The review of earlier work given in Section One indicated that as a continuation of the development of miniature hydraulic actuation techniques, there was need for developing suitable hydraulic/electronic interfacing methods. If this was successfully done, it would then be
possible to make full use of the following major advantages offered by both technologies:

1) the flexibility, compactness and increased speed of operation offered by hydraulic devices;

ii) the ease of programming, reliability, speed and low power consumption offered by electronic components.

It was therefore desirable at the start of the project to develop techniques which employ inexpensive hydraulic components that interface well with electronic equipment. As an example of the great variety of applications to which hydraulically actuated high speed reciprocating mechanisms can be put, and also as an illustration of the flexibility and adaptability of hydraulic system components, and as a further development of the concept of miniature hydraulic actuation and control, it was decided to carry out theoretical and experimental investigations into the design of a system that can be programmed to produce tufted fabric having multi-colour designs in a plane. Figures 4.01 to 4.06 illustrate examples of designs that were regarded as feasible to produce on an hydraulically actuated candlewick tufting machine.

Production of any of the designs in Figures 4.01 to 4.06 requires control of the reciprocating movement of individual needles, or groups of needles in addition to control of the sideways shift of the fabric relative to the needles. There has to be an actuator to give the needle the required reciprocating movement in and out of the fabric, and a system for positioning the needles with the required colours of yarn at the appropriate places to maintain continuity of tufting with
AN INDICATION OF THE RANGE OF MULTI-COLOUR DESIGNS THOUGHT POSSIBLE TO PRODUCE ON A HYDRAULICALLY ACTUATED TUFTING MACHINE

2 rows of individual needles carrying coloured yarn for double-coloured designs

Control of movement of individual needles can result in production of tufted fabric with names, diagrams, maps, e.t.c.

FIG 4.01

Movement caused by colour-selecting signal

Line of symmetry

FIG 4.02
AN INDICATION OF THE RANGE OF MULTI-COLOUR DESIGNS THOUGHT POSSIBLE TO PRODUCE ON A HYDRAULICALLY ACTUATED TUFTING MACHINE.

2 rows of groups of needles carrying coloured yarn for double-coloured designs

control of the movement of groups of needles can result in production of multi-coloured chess-board type patterns

FIG 4.03

2 rows of groups of needles carrying coloured yarn for multi-coloured designs

sideways shifting of fabric carrier-bed results in production of chess-board type patterns

can be produced (without carrier-bed sideways shift) by multi-coloured yarn needles on same column as in (c) but variety limited by number of needles on column, required stroke lengths, and complexity of control system for the required movements

FIG 4.04
AN INDICATION OF THE RANGE OF MULTI-COLOUR DESIGNS THOUGHT POSSIBLE TO PRODUCE ON A HYDRAULICALLY ACTUATED TUFTING MACHINE.

FIG 4.05

FIG 4.06
the required stitch length.

4.1.1 Production of patterned fabric on an hydraulically actuated tufting machine

In order to produce patterned fabric on an hydraulically actuated tufting machine, it is necessary to program the tuft-miss sequence of needle reciprocation. This enables tufted zones of predetermined shapes to be produced. However, this method of producing patterned tufted fabric still has the limitation of one yarn colour per needle, i.e. each line of tufting along the web of fabric is of one colour only. It would therefore be a major technical advance in tufting machine design if a facility to change yarn colour along a line of tufting could be incorporated in the machine. The tufted fabric designs would then have a combination of loop-miss patterning, individual needle colour change and variable pile height to create unique pattern effects which cannot be obtained from any existing machine. Thus a research and development program including production of an hydraulically actuated needle transfer mechanism was carried out.

Under normal operating conditions, each needle is threaded with yarn of one colour. This means that in order to produce multi-colour designs on a single line of tufting, more than one needle must be provided for each line so that less time is taken up in re-threading the needle when another colour is required. Also, the needles that are not working must be left clear of the fabric and needle change-over can take place only when all needles are out of the fabric. From these considerations, the needle position control problem naturally splits into two:
i) Needle reciprocation control;
ii) Mechanism for effecting needle transfer for yarn colour changes.

In addition to various braking techniques that are currently being used with rotary and linear reciprocating mechanisms, control of fluid flow to or from the needle actuator can be used to control needle position.

In this section, brake and clutch techniques that are currently being used with rotary and linear reciprocating mechanisms are reviewed. The object of these investigations was to find out whether any of these techniques can be adopted and used for patterned tufted fabric production on an hydraulically actuated tufting machine. The possible use of such devices was ruled out because of their size, high cost and the slow speed of response. The idea of using solenoid-operated two-position limited rotary stroke valves for on/off flow switching, to control the GO-NO GO sequences of high speed reciprocating actuators was therefore developed. Advantages offered by this technique compared with other alternatives are set out. To aid valve design and solenoid selection, an expression relating the relevant dimensions is presented. Diagrams showing how the limited rotary stroke valve is used to control needle reciprocation and effect yarn colour change are also presented.

4.2 Clutch and Brake Devices Currently Being Used with Rotary and Linear Reciprocating Mechanisms

4.2.1 Electromagnetic clutches and brakes

These are best suited to low torque, high speed applications. The most common designs used in industry are electrically actuated
mechanical friction units which are capable of quickly stopping a machine and restarting it fast in a start-stop cycle for instance. A magnetic field, usually created by a coil, is the source of the forces which cause coupling and uncoupling. Most units engage when electrically energised and disengage when de-energised. Other types are engaged with springs and released electrically. Yet another type uses mating crowned teeth instead of friction discs.

One important aspect of electrical control is the rate of current rise, which determines clutch/brake actuation time. Speeding up current rise reduces engagement time. One simple method of accelerating current rise is the fast-engagement circuit, where a shunted resistor is connected in series with the coil. When the coil is first energised, the shunt switch is closed, sending a high surge of current to the coil. Soon after, the switch is opened automatically by a timing device. The initial, momentary overvoltage applied to the field coil causes magnetic flux, and hence torque to build up faster. An automatically timed solid state circuit can be used instead of the switch and timer and several other circuits are available for the same purpose.

Figure 4.07 is a diagrammatic representation of the various types of rotary configuration units. A summary of the functions normally performed is given in Table T4.1 with separate columns indicating whether the components are used primarily in a zero slip, an intermittent slip or a continuous slip mode for each function. Table T4.2 summarises the characteristics of the various types.
With resistor shunted, high current energizes coil for fast engagement; normally closed contactor opens later to limit holding current. Solid-state fast-engagement circuits are also available.

Electromagnetic clutches and brakes are often used to control web tension:
(a) On-Off method senses the position of a loop of material and applies a clutch or brake as required.
(b) Continuous control method senses roll diameter and adjusts brake and clutch torque to hold web tension constant.

TYPES OF ELECTROMAGNETIC CLUTCHES AND BRAKES

1. POSITIVE ENGAGEMENT DEVICE
2. FRICTION CLUTCH/BRAKE — Single disc
3. " — Multiple disc
4. SHOE AND BAND BRAKE
5. EDDY CURRENT DEVICE
6. MAGNETIC PARTICLE DEVICE
7. HYSTERESIS DEVICE
## Table T 4.1

<table>
<thead>
<tr>
<th>Major Function Performed</th>
<th>Zero Slip</th>
<th>Intermittent Slip</th>
<th>Continuous Slip</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Clutches</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coupling and uncoupling</td>
<td>X</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Transmitting running torque</td>
<td>X</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Controlled acceleration</td>
<td></td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Speed changing</td>
<td></td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Reversing</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Indexing</td>
<td>X</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jogging</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Positioning</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Torque limiting</td>
<td>X</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tensioning</td>
<td>X</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Adjustable speed</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Brakes</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Emergency stopping</td>
<td></td>
<td></td>
<td>X</td>
</tr>
<tr>
<td>Controlled deceleration</td>
<td></td>
<td></td>
<td>X</td>
</tr>
<tr>
<td>Holding or locking</td>
<td>X</td>
<td></td>
<td>X</td>
</tr>
<tr>
<td>Tensioning</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Friction disc, Magnetic particle, Hysteresis clutches and brakes.**

No relative motion between input and output members (static application).
1. Ability to transmit torque without slipping.
2. Ability to disengage quickly and reliably.

**Friction disc, Magnetic particle clutches and brakes.**

Acceleration or deceleration of rotary loads on a cyclic basis.
1. Ability to transmit torque for acceleration and deceleration.
2. Ability to dissipate the heat generated.

**Friction disc, Magnetic particle, Hysteresis, Eddy current clutches and brakes.**

Components subjected to constant wear and heat generation.

**Intermittent Slip**

**Continuous Slip**
<table>
<thead>
<tr>
<th>TYPE</th>
<th>OPERATING PRINCIPLE</th>
<th>ADVANTAGES</th>
<th>LIMITATIONS</th>
<th>INDUSTRIAL APPLICATIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>POSITIVE ENGAGEMENT CLUTCHES</td>
<td>A toothed armature is drawn or forced axially into engagement with a magnetic body that has mating teeth.</td>
<td>Large torque capacity for its size. Can operate dry or in an oil atmosphere. Accurate indexing possible. No drag torque when disengaged.</td>
<td>Essentially an on-off device. Maximum permissible slip-speed at engagement about 300 rev/min.</td>
<td>Packaging machines, Printing-presses, Rolling-mill screw-downs, Conveyors, N/C machine tools, Payoff and windup reels.</td>
</tr>
<tr>
<td>SINGLE DISC</td>
<td>A metallic disc is drawn or forced axially into engagement with a ring of friction material.</td>
<td>Versatile, fast-acting device. is low in cost. Torque is easily controlled by adjusting coil current. Available in wide range of sizes with numerous mounting arrangements.</td>
<td>Friction faces eventually wear to a point requiring adjustment or replacement. Units must be derated to operate in continuous slip mode.</td>
<td>Machine tools, Business machines, Textile machinery, Packaging equipment, Materials handling equipment, Recording instruments.</td>
</tr>
<tr>
<td>FRICITION CLUTCHES AND BRAKES</td>
<td>A stack of alternate driving and driven discs is compressed to effect coupling. Torque is relatively constant at slip speeds above 300 rev/min. It rises to a maximum at zero slip speed as for single disc units.</td>
<td>High torque to diameter ratio. Can operate in dry or in an oil atmosphere. Long life.</td>
<td>Hold-in may be erratic where coil current is less than about 40% of nominal rating. Not generally used for continuous slip operation.</td>
<td>Speed changing mechanisms and transmissions for machine tools such as multiple spindle chuckers, automatic screw machines and boring machines.</td>
</tr>
<tr>
<td>MULTIPLE DISC</td>
<td>Friction shoes are moved radially inward or outward to engage a drum; a friction band is tightened around the drum. Units of this kind are generally spring-set and solenoid-released.</td>
<td>Simple, rugged units are highly reliable. Available in large torque capacities. Fail-safe operation.</td>
<td>Torque is usually not easily electrically adjustable.</td>
<td>Hoists, Cranes, Winches, Elevators, Metal-working machines, Printing-presses and processing lines.</td>
</tr>
</tbody>
</table>

**TABLE T 4.2**
<table>
<thead>
<tr>
<th>TYPE</th>
<th>OPERATING PRINCIPLE</th>
<th>ADVANTAGES</th>
<th>LIMITATIONS</th>
<th>INDUSTRIAL APPLICATIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAGNETIC PARTICLE Clutches and Brakes</td>
<td>A magnetic field causes magnetic particles to form 'chains' that mechanically link an inner and outer rotating member. Particles tend to congeal and therefore transmit torque. Strength of linkage is a function of coil current.</td>
<td>Fast response and smooth operation. Torque independent of slip speed. High torque-to-signal linearity over a wide control range. No wear adjustments required. No stick-slip or cogging. Produce torque under forward, reversing, or stationary conditions. Capable of 100% lockup with no slip.</td>
<td>More expensive than friction units.</td>
<td>Lathes, Milling machines, Boring machines, Case loaders, Printing presses, Crushers, Conveyors, Tensioning and Positioning devices</td>
</tr>
<tr>
<td>HYSTERESIS BRAKES AND CLUTCHES</td>
<td>A drag cup follows the rotation of a multipole rotor because of the torque developed by hysteresis losses in the magnetic material of the drag cup.</td>
<td>Smooth operation. Torque independent of slip speed. High torque-to-signal linearity over a wide control range. Long life. Environmentally stable.</td>
<td>High cost. Low power gain. Limited to low torque applications (less than 14Nm).</td>
<td>Instrument, Servomechanisms, Tape drives, Wire and strip tensioners</td>
</tr>
<tr>
<td>EDDY CURRENT Clutches and Brakes</td>
<td>Eddy currents induced in the inner surface of an input drum create a magnetic field that couples the drum and a multipole output rotor. Drum must rotate faster than rotor to produce torque. Transmits torque by magnetic drag resulting from eddy currents induced in a driven member by a rotating magnetic field.</td>
<td>Smooth, shock-free operation. Torque proportional to slip-speed. Indefinite life.</td>
<td>High cost. No torque at zero slip. Moderately slow response. May be temperature sensitive.</td>
<td>Used widely in adjustable-speed-drive systems for web processing equipment, Extruders, Conveyors, Hydraulic-test-stands, Printing presses and Packaging machines</td>
</tr>
</tbody>
</table>

continued
<table>
<thead>
<tr>
<th>TYPE</th>
<th>OPERATING PRINCIPLE</th>
<th>ADVANTAGES</th>
<th>LIMITATIONS</th>
<th>INDUSTRIAL APPLICATIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>LOW VOLTAGE ELECTROSTATIC DIELECTRIC CLUTCHES</td>
<td>Electrostatic forces hold together a metal member and a closely adjacent dielectric member.</td>
<td>Fast acting if suitable metal-dielectric combination is chosen. Can be driven by low A.C. or D.C. voltages (5-100 V) depending on dielectric type, thickness and required permissible torque. (Piezoelectric materials need lower voltages than other dielectrics)</td>
<td>Elastic, Thermal and other mechanical properties of the dielectric must be good to meet conditions for high power transmission</td>
<td>On-off applications, for accelerating or decelerating the driven shaft, as slipping clutches, and for step-by-step functions.</td>
</tr>
<tr>
<td>AIR OPERATED DEVICES</td>
<td>Powerful radial compression force of inflated flexible tube actuates friction shoe assembly.</td>
<td>Only one moving flexible part. Total disengagement on air release. No lubrication, wear, heat buildup, or adjustment. Transmission shock and axial pressure absorption.</td>
<td>Best for low speed and high torque requirements.</td>
<td>Marine propulsion, Continuous strip mills, Forging presses, Transfer presses, Continuous pipe weld mills, Flying shears Slitting lines, Strip de-coilers and reelers, Tube draw benches, Guillotines.</td>
</tr>
<tr>
<td>FERROFLUID DEVICES</td>
<td>Effective magnetic pressure generated and results in fluid motion and other unusual mechanical fluid properties and characteristics in the presence of an applied magnetic field. Fluid retains liquid characteristics.</td>
<td>Fluid can be used in gravity-free conditions. Fluid retains liquid characteristics in presence of applied field.</td>
<td>Very high cost. Induction of fluid motion for brake operation is difficult.</td>
<td>Avionics, Flight control, Energy conversion, Novel pumps, Viscous dampers, Specific gravity meters, Gyroscope supports employing levitation bearings, Fluid seals, Triaxial accelerometers.</td>
</tr>
</tbody>
</table>
4.2.2 Air-operated clutches and brakes

These harness the powerful radial compression force of an inflated flexible tube. For large power applications, the ideal heavy duty design of clutch is based on a well-proven system of compressed air inflation of a pneumatic tube to actuate the friction brake shoe assembly. These power transmission assemblies have the advantage of only one moving part; the mechanism, a flexible actuating tube, which simplifies operation and maintenance. Pressure is applied evenly over the friction surface and the tube absorbs shock, and a degree of angular or parallel misalignment. The release of air ensures total disengagement of the friction shoes, so there is no drag to cause heat and wear. The assemblies require neither lubrication nor adjustment. The resilience of the tube means cushioning, a smooth take up of brake/clutch engagement and the ability to absorb transmission shock and axial pressures. In general, the best applications are low speed and high torque requirements.

4.2.3 Low voltage electrostatic dielectric clutches

Operation of electrostatic devices depends upon the electrostatic forces which, due to the Johnson-Rehbek effect, tend to hold together a metal member and a closely adjacent semiconducting member of resistivity between $10^6$ and $10^7 \text{ohm}\cdot\text{cm}$ when a voltage is applied between them. If the semiconducting member is replaced by a dielectric member and particularly by a piezo-electric member, electrostatic forces may be generated at low applied voltage and exploited for the construction of low voltage clutches or similar electrostastic devices.
An interesting application of these devices is in the step-by-step operation that can result if stationary and driving cylinders of the piezoelectric type have a flexible metal band of low inertia around them. In this system, the band is driven by the driving cylinder and slides over the stationary one. When a voltage is applied between the stationary cylinder and the band, the band stops and if the driving cylinder is connected to a motor by means of a clutch that can transmit only up to a given torque, when the voltage is applied, the driving cylinder also stops. In this manner the metallic band can be driven step-by-step by applying pulses to the system.

Electrical time constants for a suitable dielectric-metal combination can be rendered very small, less than a microsecond, and the slip period can be reduced to a few tenths of a microsecond if the inertia of the driven member is small and the friction properties of the materials being used are appropriate. Some characteristics of electrostatic clutches are given in Figure 4.08.

4.2.4 An electro-mechanical latch as a mechanical stop

This is another method that can be used to stop needle reciprocation. It is the method that was adopted for use in the programmed control of loop-miss sequences in the first hydraulically actuated tufting machine. The latch is pushed in by a solenoid-operated pushrod, against a moving bar stop at some time during the preceding cycle of the needle reciprocation, ready to latch-in as soon as the needle bar stop clears the top of the latch, Section 1.5.
Schematic arrangement for an electrostatic clutch.

Maximum transmissible torque as a function of the applied ac electrical field of frequency 16kzh.

Transmitted torque as a function of relative speed of slip for constant ac applied electrical fields of frequency 16kzh.

(a): lead zirconate titanate ceramic-brass pair;
(b): quartz-chromium pair;
(c): hard anodized aluminum graphite of resistivity 900-10 ohm cm.

Supplied torque as a function of the steps per second for a device comprising a metal band (berillium-brass alloy) slipping on two quartz cylinders.

Maximum transmissible torque as a function of the number of on-off operations.

Duration of the slip time as a function of the prime mover speed with \( \omega(t)=0 \) and for fixed values of the applied ac fields of frequency 16kzh and different loads. \( I_e \) inertia of the driven member of the clutch without load.

Characteristics of low-voltage electrostatic clutches (Oliveir. A. IEEE IND IE17(2) 76 1970)
4.2.5 The possibility of using electromechanical latches, electromagnetic clutches or brakes for needle position control

To provide an alternative to the electromechanical latch that was used on the first hydraulically actuated tufting machine, it was decided to investigate alternative methods of needle position control. The latch mechanism presents two major operational problems:

i) Due to the metal-to-metal impact, the latch produces noise levels as high as 90 dB when the machine is in the patterning mode. The impact stresses are likely to reduce the life expectancy of the patterning mechanism.

ii) The latch does not work when the machine is running at a speed higher than 1600 cycles a minute. This is because the actuator stroke depends on the pressure at the actuator and the speed of rotation of the valve supplying the actuator. Above this speed, at constant supply pressure, the stroke decreases as the valve speed increases, thus making it impossible to operate the latch.

Electrical and electromagnetic devices are widely used for many industrial applications as discussed above. The possible use of such techniques for the production of patterned tufted fabric was ruled out because:

i) For high speed operation, say 20 Hz upwards, the forces generated and the distances moved are very small.

ii) The devices are expensive and bulky. It would therefore be difficult to package them in a small space if control of individual miniature actuators was to be realised.
iii) Although it is possible to control needle reciprocation using a mechanical latch or brake, it would be difficult to incorporate the needle transfer facility in the design of the machine in order to effect yarn colour changes. It was therefore decided to investigate hydraulic means of controlling needle positions.

4.3 Control of Fluid Flow for Patterned Fabric Production on an Hydraulic Tufting Machine

4.3.1 The use of patterning blocks to control fluid flow

To increase the versatility of the hydraulic knitting action using one actuator per needle on the circular weft knitting machine, Garside suggested that a method of alternating between the three types of knitting action could be introduced and in his second investigations (Garside, Ref 2) he looked at the size of the pattern repeat with a view to increasing the pattern capacity of the machine. The pattern capacity of the machine is limited by the size of the memory store for selecting the individual needles or the number of needles between consecutive feeders. It can be increased by making use of a larger memory for example paper tape, punched card or magnetic tape, external to the knitting machine. This in turn involves a technique for transmitting the relevant information from the logic elements to the needles.

For limited patterning capability, full control over the knitting cycle can be obtained by selecting the needle position, that is, whether the needle operates in the knit, tuck or miss mode. Garside designed a
system for this. Referring to Figure 4.09 for example, to obtain the tuck motion, the actuator amplitude has to be constrained by preventing the pressure pulse from reaching port 2 of the actuator. The miss stroke is similarly obtained by preventing the pressure pulses from reaching ports 1 and 2. To select needles into a particular configuration of knit, tuck, and miss combinations for relatively simple programmes, i.e. stitch combinations requiring up to eight selections before repetition, the rotary valve can be used as a selecting device thus enabling fixed patterning blocks to be used. For example in a knitting machine where it is required to make four selections of knit, tuck or miss for each individual needle in sequence before repeating, the following technique can be used:

The rotary valve is designed so that it will supply oil to the actuator from four independent rotor positions for every revolution of the rotary valve. This involves machining the desired slots into a 90° segment of the bobbin, thus the cycling rate of the actuator would be four cycles per revolution, with the pulses from the rotary valve being supplied from a different hydraulic circuit in each instance. Pictorial diagrams of such a valve, designed by Garside are shown in Figure 4.09. The four separate supply paths, namely a, b, c and d for actuators ports 1 and 2 are taken via a pattern block which consists of two steel plates with a plastic shim clamped between them. If a hole is drilled, the actuator receives the pulse of oil and if the hole is omitted, the supply line is blocked. This hole punching technique enables each individual needle to be programmed for four selections before repetition. Once the pressure signal has passed through the pattern block, the four separate lines can be taken via a
A diagram to show the hydraulic circuit for a single actuator and a rotary valve together with a sectional view of the slots required in the bobbin to produce the time displacement profile required for knitting.

A diagram to show a sectional view of a rotary valve suitable for programming repetitive needle combinations.
manifold before being connected to the actuator. A circuit diagram for such a valve-actuator combination is shown in Figure 4.10. A non-return valve has to be introduced to enable the residual oil at port 2 to be exhausted when the actuator, operating in the tuck mode is returning from the tuck position. This is because supply line 2 is blocked for patterning purposes and the oil trapped between the piston rod and port 2 cannot be exhausted. This technique could usefully be employed when producing mono-colour, texture fabric, where the tuck stitch is used to produce a regular surface pattern. Such a system for the selection of needles in particular configurations of knit, tuck and miss combinations apart from making it possible to produce fabric with fixed patterns, does not fully exploit the simplicity, economy, flexibility and the numerous advantages offered by the application of hydraulically actuated mechanisms.

This technique was not adopted for use in the production of patterned tufting because:

i) To produce continuous yardage with varied patterns, it is necessary to have control over individual needles at every instant so that individual needles can be selected into particular configurations according to a program. This is not possible when using patterning blocks. It is necessary to change the patterning block if yardage having different patterns is to be produced. This puts a limit to the variety of patterns that can be produced on a particular machine because each pattern block gives a particular fabric pattern and having more pattern blocks increases the cost of the system.
A CIRCUIT DIAGRAM TO SHOW THE HYdraulIC CONNECTIONS BETWEEN A ROTARY VALVE, WITH A PATTERNING FACILITY AND AN ACTUATOR.

FIG 4.10
ii) The patterning blocks would be very bulky and expensive to produce.

iii) The speed of operation of the machine would be restricted by the need to move or change the block in order to control reciprocation of individual needles.

It was therefore necessary to look for alternative methods of controlling positions of individual miniature hydraulic actuators so that varied patterns could be economically produced without having to alter the mechanical set up of the machine.

4.3.2 The possibility of using an electroviscous process for the sequential GO-NO GO programming of actuators

Ferrofluids are colloidal dispersions of single-sub-domain magnetic particles (diameter less than 150 Ångstrom units) that retain their liquid characteristics in the presence of an applied magnetic field. As a result, they are unique liquid media in which it is possible to induce substantial magnetic forces, resulting in liquid motion and other unusual mechanical fluid properties because of the generation of an effective magnetic pressure. In Table T4.2, a summary is given of the principle of operation, advantages, limitations and industrial applications of ferrofluids. Phenomena which have been observed and which have resulted in a number of unique applications as a result of the interaction of ferrofluids with magnetic fields include:

i) Ability to suspend the fluid in space by the application of a magnetic field, which indicates their possible use as magnetic fluid seals;
ii) Stable levitation of an object by application of a magnetic field, e.g. gyroscope supports employing levitation bearings, triaxial accelerometers, incremental velocity meters etc.

iii) Variable apparent specific gravity that is controlled by an applied magnetic field, e.g. the variation of specific gravity, by thermal or magnetic means without any moving mechanical parts (e.g. novel pumps and other energy conversion devices, and viscous dampers for gravity satellites).

iv) Fluids' ability to flow and conduct magnetic flux.

v) Spontaneous formation of stable liquid spikes in the presence of perpendicular magnetic fields.

Each one of these applications places unique requirements on the combination of all properties of the fluid being used. These properties can be controlled by varying the composition, size and concentration of the magnetic particles. By proper choice of the stabilizing agent, these magnetic properties can be conferred to a wide range of liquids such as water, hydrocarbons, fluorocarbons, glycerol and silicones. Electroviscous technology has now advanced to such an extent that it is possible to formulate fluids which respond at 1 KHz. Results of work done in this field at Sheffield University indicate that the characteristics and physical properties of these fluids could be beneficially exploited to program needle loop-miss combinations of hydraulic tufting machines. The use of these fluids with suitably designed control devices for actuator switching appears to be a very suitable method for sequential programming of actuators but at the moment, con-
siderable development work on both fluids and locking devices will be necessary before the technique can be successfully applied. Worthwhile work could therefore be done in this field with a view of developing a method of interfacing hydraulic and electronic equipment.

4.3.3 The possibility of using an electrically controlled on/off flow control valve for the sequential GO-NO GO programming of actuators

At the outset of the investigations presented in this thesis, the applicable work on electroviscous fluids was of classified status and relevant technical data was not available, whilst response limitations, lack of controllability, bulk and cost ruled out the use of conventional electro-mechanical clutch devices. Thus work was directed to determining methods of using a small electric signal to switch an on/off flow control valve in an hydraulic supply circuit. This would enable a low power level signal from an electronic programming system to be used to select required needle positions. Using flow control, incorporation of the needle transfer facility to effect yarn colour changes in individual lines of tufting would be feasible. The needle reciprocation control problem was investigated first.

4.4 Needle Reciprocation Control by Hydraulic Means

4.4.1 Use of a linear solenoid-operated two-position spool valve to control needle reciprocation

To produce patterned tufting, the double acting piston, Figure 1.07, supplied by the rotary valve, has to be locked in the up position with the needles clear of the backing fabric in order to miss loops or to change needles in a line of tufting when producing multi-colour designs.
An initial idea for locking up the needle was to use the electro-hydraulic system pictorially presented in Figure 4.11 which also shows typical servovalve configurations commonly used for fluid flow control purposes. The hydraulic circuit shown employs a two-position solenoid-operated valve to allow free flow in one position and no flow in the other. Together with the non-return valve which admits fluid only into the cylinder, this would permit free up and down movement of the actuator under running conditions and would lock the actuator in the up position for needle transfer, or when stitches are to be missed, when the solenoid operated valve changes to the other position.

In an attempt to select a suitable configuration out of those indicated in Figure 4.11, the system given in Figure 4.12 was designed and manufactured. In one position with the piston withdrawn, there would be free flow between the rotary valve and the actuator, thus permitting free up and down movement. In the other position, flow would be possible only to the cylinder through the spring loaded non-return valve, making it possible to lock the actuator in the up position with the needles clear of the fabric as indicated in Figure 4.13. Tests showed that this system worked satisfactorily at supply pressures less than 6 bar. Higher pressures caused locking of the valve in one position making it impracticable to operate with a small solenoid. If this idea was to be adopted, it would be necessary to have two systems; a high pressure system to drive the needle actuator and a low pressure system to control needle reciprocation. It was judged better to operate the needle actuator and control needle reciprocation using the same pressure supply. This would eliminate the need for additional valves and control circuits.
TYPICAL HYDRAULIC SERVOVALVE CONFIGURATIONS

Two-jet flapper valve

Four-land-four-way spool valve

Two-land-three-way spool valve

Jet pipe valve

Two-land-four-way spool valve

Three-land-four-way spool valve

NEEDLE RECIPROCATION CONTROL BY HYDRAULIC MEANS

FIG 4.11
Two-position spring-loaded solenoid-operated

Needle reciprocation control — Linear solenoid system
NEEDLE RECIPROCA TION CONTROL — LINEAR SOLENOID SYSTEM

FIG 4.13
4.4.2 Use of a limited rotary stroke valve to control needle reciprocation

Conventional spool type servovalves of the type depicted in Figure 4.11 when used for on/off switching or reversal of fluid flow in selected portions of an hydraulic circuit call for a high standard of design and precision manufacture if the transient damping forces, steady state spring forces and friction are to be kept to a minimum. Very close and matching tolerances must be kept and the need to reduce the valve forces may require shaping of the valves lands, chambers and ports thus increasing the unit costs.

Use of the sculptured rotary valve to supply high speed reciprocating mechanisms in previous investigations, had indicated that with inexpensive equipment it is much easier to use rotary rather than linear motion to switch fluid flow at high speeds. Instead of the two-land valve and linear solenoid of Figure 4.12 an attempt was made to use a rotary solenoid to actuate a limited rotary stroke valve controlling fluid flow. The linear system operated simply by connecting the supply and exhaust lines to either side of a piston. This function can be performed by a sculptured rotary valve as indicated in Figure 4.16. The Ledex rotary solenoid used, employs a ball detent mechanism resulting in almost frictionless conversion of linear to rotary movement. Originally developed for use with bomb-release mechanisms, the solenoid is a robust and efficient electromechanical device that has been adopted for many applications in the precision control of industrial and commercial equipment. Designed for use with direct current, they are available in different rotary stroke sizes and torque ratings.
However, for satisfactory performance of the system, it is very important to match the angular stroke of the selected solenoid with the size of the valve and fluid passage diameters, and the torque capability with the operating torque requirements. Malfunction can result if the fluid passages are not fully opened or closed as required, especially for high speed applications. Therefore the relationship between the relevant variables must be known so that an appropriate valve can be designed.

4.4.3 Relationship between fluid passage bore diameter \(d\), rotary stroke valve diameter \(D\), rotary valve land width \(z\) and rotary solenoid stroke angle \(\theta\)

Referring to Figure 4.14, line \(XX_1\) must move to position \(XX_2\) for complete fluid control. The angle through which it turns, \(\theta\), is
equal to the angular stroke of the selected solenoid. With all angles measured in radians:

\[ \theta = \frac{D\beta}{2} \text{ giving } \beta = \frac{2\pi}{D} \]

\[ d = \frac{D\alpha}{2} \text{ giving } \alpha = \frac{2d}{D} \]

\[ \theta = \alpha + \beta \]

\[ \therefore \quad \theta = \frac{2(\lambda + d)}{D} \]

This expression, which is similar to the expression \[ D = \frac{2n(\lambda + d)}{2\pi - NT} \]
relating the diameter of a rotary valve, land width and fluid passage diameter to the speed of rotation, Section 3.6, reveals some of the conflicting requirements of miniaturization which require a compromise solution;

i) \( \theta \) is set by the solenoid selected. For fixed \( \lambda \) and \( d \), the bigger \( \theta \) is, the smaller \( D \) will be. However the solenoid operating speed is dependent upon the size, load, duty cycle, torque available and hence input power. The solenoid operating speed will decrease with rotary stroke.

ii) For fixed \( D \) and \( \lambda \), \( \theta \) decreases as \( d \) decreases. But \( d \) must be sufficiently large not to cause undue restriction to flow.

iii) \( \lambda \) must be large enough to minimize cross-port leakage flow. For fixed \( d \), the larger \( \lambda \) is, the larger \( D \) will have to be.
4.4.4 Advantages offered by the hydraulic technique over the electromechanical latch in programming needle reciprocation sequences

With the above considerations noted, and with the aim of using a rotary stroke valve operating at the same pressure as the high speed reciprocating actuator to be controlled, the system indicated in Figure 4.16 was designed and manufactured.

Instead of the two-land valve and linear solenoid of Figure 4.12, this system employed a rotary stroke valve and rotary solenoid for the purpose of switching fluid to either side of the piston which had an integral spring loaded non-return valve. Fluid switching was in response to the actuator-position selecting signal. The piston was used in a similar way to that shown in Figure 4.12. In terms of low noise levels and availability of needle patterning at higher operating speeds, tests showed that the limited rotary stroke valve system had better performance than that of the electro-mechanical latch mechanism. The improved characteristics can be explained by the following considerations:

![Diagram showing Time to actuate rotary hydraulic system vs Time to actuate mechanical system](FIG 4.15)
Two-position solenoid-operated (rotary)

Return

Supply

Ball

Spring

Non-return valve

FIG 4.16

NEEDLE RECIPROcation CONTROL — ROTARY SOLENOID SYSTEM
i) If the up and down movement of the actuator is represented as in Figure 4.15, it is true that the mechanical latch system can be designed such that the latch is pushed inwards in advance, ready to drop under a stop as soon as the actuator reaches the upper limit of its movement. However, the amplitude of the actuator motion must be a certain minimum before the latch can operate. This imposes an upper limit on the speed at which the actuator can be run to avoid attenuation and thus maintain consistent latching for repetitive patterning. At a speed higher than this, the latch cannot be pushed in, even in advance. This is not true of the rotary hydraulic system as it can be actuated at any time in the cycle even when the actuator is not moving through its full stroke. This enables repeatable loop-miss sequence control at higher cycling rates, thus giving increased production rates of patterned fabric. Provided the looper-knife and needle positions have been properly set to account for the possibility of actuator amplitude attenuation at the designed operating speed, an attenuation as much as 3 mm will not affect the performance of the system.

ii) The electrical circuit controlling the hydraulic rotary system can be designed for "FAIL SAFE" operation. This is done by arranging that with power off, the angular position of the rotary stroke valve corresponds to a non-actuating condition of the needle drive piston which will then remain in the up position with the needles clear of the fabric while the rest of the system is running.

iii) Use of the hydraulic technique eliminates the high impact forces characteristic of the mechanical latch system when needle descent
is stopped for patterning. This results in a reduction of noise levels and longer life expectancy.

4.4.5 Elimination of separate supply and return lines by directly using the rotary valve to control needle reciprocation without a pilot system

The above considerations and the encouraging performance of the limited rotary stroke valve system indicated that further design and manufacturing simplification could be realised. Elimination of the pilot system would result in a faster speed of system response, thus enabling patterning at higher cycling rates. Provision of one pressure supply instead of two separate ones to drive the needle actuator and control needle reciprocation would result in a reduction of manufacturing costs.

Figure 4.17 shows that using the actuator-position selecting signal, the limited rotary stroke valve connects exhaust to either side of the double acting piston that has an integral non-return valve. The resulting position of this piston then determines whether the needle actuator reciprocates or stays up. This requires the provision of separate supply and return lines for the limited rotary stroke valve. Also, because of the pilot system, response times are further increased. It was therefore decided to look for a method of eliminating the need for separate supply and return lines and to investigate how response times could be reduced by directly using the valve for actuator position control.
NEEDLE RECIPROcation CONTROL—ROTARY SOLENOID SYSTEM

FIG 4.17
4.4.6 Valve construction and principle of operation

The system shown in Figure 4.18 was therefore designed and manufactured. This employed a bush with appropriately cut slots for fluid passage. The bush was press-fitted into a bore across the fluid passage from the main rotary valve to the actuator. A spool with small holes drilled through was fitted to turn in the bore of the bush. This spool design was different from the sculptured version of the previous design. Turning of the spool to align the holes with the slots in the bush allowed free fluid flow to and from the actuator. With the holes blocked, the actuator locked in the up position due to the non-return valve.

4.4.7 Problems of manufacture and operation

During manufacture and test a number of problems leading to malfunction were encountered:

i) The bush had to be straight and parallel to the bore across the line from the rotary valve powering the actuator. There was no certainty that this would be so without elaborate methods of manufacture and metrology.

ii) Similarly the blind bore in the bush had to be straight and parallel and the end had to be as perpendicular to the axis as possible. Again there was no certainty that this would be achieved by simple manufacture and inspection.

iii) The thin material left after sculpturing the bush and cutting the slots led to distortion of the bush on press fitting into the bore across the line to the actuator. This resulted in the
NEEDLE RECIPROCATION CONTROL—ROTARY SOLENOID SYSTEM

To and from power rotary valve

Two position solenoid operated (rotary)

FIG 4.18
bush having a barrel shape which led to an increase in leakage and therefore movement of the actuator in the locked up position. The amplitude of this movement was as much as 5.0 mm.

iv) Because of leakage, pressure built up at the end of the spool and this resulted in the spool vibrating in tune with the pressure pulses from the power valve. This seriously increased the torque requirements of the solenoid required to turn the spool in its bore and had adverse effects on the response times of the system, the accuracy with which needle reciprocation could be controlled and the power consumption of the solenoid.

Clearly this system required redesigning if the idea was to be employed successfully. The system shown in Figure 4.19 was therefore designed and manufactured to eliminate or minimize the problems referred to above. This employed the same principle as that of Figure 4.18 but the construction of the two-position solenoid-operated rotary valve is similar to that of Figure 4.17. The bobbin lies across the line from the main rotary valve to the bottom of the reciprocating actuator. When the rotary solenoid is actuated, the spring loaded non-return valve can admit fluid to the bottom of the actuator but once the actuator is in its up position, it cannot descend because the two position valve seals off the exit by blocking the small holes to the main rotary valve. In the other position when the solenoid is not operated, there can be free flow through the small holes.

The idea of using a series of small holes instead of one big one resulted from a consideration of the expression $\theta = \frac{2(\delta + d)}{D}$ relating the required stroke angle $\theta$, the diameter of the fluid passages $d$,
land width \( l \), and the diameter \( D \) of the bobbin of the stroke valve. The diameter \( D \) and rotary stroke angle \( \theta \) can be made small if fluid passage diameter \( d \) is kept small. So to maintain the same flow area, a series of small holes were used. Their total cross-sectional area must be sized to ensure that flow starvation of the actuator does not occur. For the same reason, a series of small holes had been used in the system of Figure 4.18.

4.4.8 Accuracy of needle reciprocation control when using the rotary stroke valve to trap a volume of fluid in the cylinder below the piston

It was noted that under operating conditions, the speed of system response would be set by the electrical solenoid being used and that because of leakage, it would be impossible to stop the actuator moving when in the locked-up position. The amplitude of this movement can be kept small by reducing leakages to the minimum possible. This technique was developed after considering the simplicity of the design and manufacture of the components, the fact that no separate supply or return line was required, and the possibility of being able to program the loop-miss sequence at higher needle actuator cycling rates.

Preliminary tests with the available actuator indicated that the performance would be acceptable especially for single colour patterned tufting where the small amplitude of actuator movement in the locked up position would have no undesirable effect. To allow movement of the needle carrying components so as to effect yarn colour changes, a more accurate method of locking the actuator would be necessary. The imperfect operation of the non-return valve and leakage are the cause of this movement of the actuator in the locked up position. The amplitude
of this movement depends on the system pressure, speed of operation, non-return valve spring characteristics and manufacturing accuracies achieved. If the amplitude of this movement is greater than the maximum that allows free movement of the needle transfer components to effect yarn colour changes in individual lines of tufting, it will be necessary to look at other alternatives. For instance in the system adopted, it is the flow from the bottom of the cylinder housing the needle actuator, back to tank through the limited rotary stroke valve, and through the main rotary valve supplying the needle actuator, that is blocked. It is possible to employ the same principle to stop flow from the main rotary valve to the top of the piston and thus hold it up for patterning purposes, Section 1.5. This might have certain advantages in that the top of the actuator would then be maintained at a pressure nominally equal to exhaust and therefore the actuator might be more steady in its top null position. Originally, it was believed that the piston could move down if in addition to gravity, the top of the piston was pressurized when the bottom was connected to exhaust. This is why in the system presented, flow out of the bottom of the cylinder is controlled. Durrant, Ref 7, in a parallel experimental investigation, used the developed rotary valve principle to control flow to the top of the cylinder and obtained very good results indicating that friction forces and the resistance to flow from the bottom of the cylinder are enough to overcome those due to gravity and any out of balance downward pressure forces. A sketch of the hydraulic circuit is given in Figure 4.20 and a diagram showing the set up of this system is shown in Figure 4.21.
In view of the results obtained from that investigation, it may be necessary to change the design of the system if the amplitude of actuator movement is greater than acceptable to allow needle transfer for multicolour designs. A photograph of the valves used in both systems to control needle reciprocation is given in Figure 4.25.

4.5 Mechanism for Effecting Needle Transfer for Yarn Colour Changes

In order to produce any of the designs indicated in Figures 4.01 to 4.06, control of the movement of the individual needles, or groups of needles is required, in addition to the facility to control the sideways shift of the fabric relative to the needles. There has to be an actuator to give the needle the desired reciprocating movement in and out of the fabric and a system of positioning the needles with the required colours of yarn at the appropriate places, to maintain conti-
CONTROLLING NEEDLE RECIPROCATION BY STOPPING FLOW TO THE TOP OF THE PISTON USING A LIMITED ROTARY STROKE VALVE

Limited rotary stroke valve

Patterning block

Needle actuator

Spring-loaded non return valve

Rotary valve

FIG 4.21
nuity of tufting at the required uniform stitch length. Also, correct alignment of the needles, with the looper-knife combination is essential.

For normal tufting, each needle is threaded with yarn of one colour. This means that in order to produce multicolour designs on a single line of tufts, more than one needle must be provided for each line. This minimizes the time taken up in rethreading the needle when another colour is required.

It was first proposed to employ a hydromechanical system for positioning the needles. With this system, schematically shown in the upper part of Figure 4.22, control of the movement of each individual needle, or group of needles, would be achieved by driving each by a separate actuator supplied by the main rotary valve, through a multiposition selector spool valve. Multicolour designs would then be produced by employing more than one needle or one group of needles, one for each required colour, in each line of tufting. This required a system for directing fluid flow to the appropriate actuator to drive the needle having the desired coloured yarn and a system for positioning the needles and the selector spool valve appropriately as required by the signal selecting the colours. This signal could be derived from paper tape, optical detector, punched card or magnetic tape system.

Although it had earlier been decided to limit the number of colours on a line to only two, this system was not designed. To produce patterned tufting, this system required movement of the needle actuator blocks and the two-position selector spool valves so as to always place the working needles in the same tufting position while leaving those not working, up, out of the backing fabric.
FIG 4.22

Proposals of Needle Position Control Methods

Actuator blocks of double acting cylinders driving the needles

Movement would be caused by signal selecting required colour

2 Position selector spool valve for production of double-coloured designs

Supply from rotary valve

Actuator block of double acting cylinders driving the needles

Supply from rotary valve

Actuator-needle coupling-decoupling system

Movement would be caused by signal selecting required colour

FIG 4.22
This would be necessary in order to maintain uniform stitch length. It was therefore decided to investigate alternative methods that would involve a smaller number of moving components and reduce the cost of manufacture.

The second proposal was that instead of each needle, or group of needles, being driven by a separate actuator, a system could be designed whereby one actuator would drive more than one needle, or group, on each line of tufting. With this system, as shown in the lower part of Figure 4.22, needles are moved to the appropriate place by the colour-selecting signal. Compared to the previous system, this results in only one actuator per line of tufting, eliminates the selector spool valve, simplifies the circuitry, reduces the moving masses and therefore the required stroking forces, as well as the cost of manufacture. A fast acting system using a quick acting coupling-decoupling mechanism for effecting needle transfer, was therefore required. The transfer action MUST be timed to occur when the needles are in the upward position, clear of the fabric.

Without a pilot system, it is not possible to use a rotary stroke valve to provide control of the motion necessary for the transfer of the tufting action from one set of needles to another in order to produce multicolour patterned tufting. This is because the needle carrying components have to be moved to the appropriate places in order to effect needle transfer for yarn colour changes. However, operation of the system of Figure 4.17 had indicated that a rotary stroke valve can be satisfactorily used to exhaust fluid from either side of a piston according to a colour-selecting signal. The resulting
movement of the piston can then be used to effect needle transfer. The system shown in Figure 4.23, with the changeover elements attached to the piston, was therefore designed to provide a facility for effecting needle transfer in order to produce patterned multicolour designs on an hydraulic tufting machine. Figure 4.24 shows a unit of the rotary valve controlled tufting machine that was designed to produce patterned fabric. Needle reciprocation and transfer are controlled by limited rotary stroke valves shown in Figure 4.25.

4.6 The Use of Limited Rotary Stroke Valves to Transfer the Tufting Action from One Set of Needles to Another in Order to Produce Multicolour Patterned Tufting

Figure 4.24 shows a unit that was designed to produce multicolour patterned tufting. The main rotary valve supplies a double acting piston that employs labyrinth seals and auxiliary sleeves to provide hydraulic cushioning at the end of each piston stroke. Bars, N1 and N2 in Figure 4.23, carrying needles threaded with yarn of the required colours, are alternately coupled to the double acting piston and the reciprocating movement of the piston drives the working needles in and out of the backing fabric. Motion of the needle bar driving components is along rectangular grooves lubricated by the leaked fluid from the double acting pistons. The system works as follows:

1. To continuously tuft with a set of needles, e.g. set 2 in Figure 4.24, first, the rotary stroke valve controlling needle reciprocation is actuated to lock the double acting needle actuator in the up position, with all needles out of the backing fabric. This is the position shown in Figure 4.24. The needle
NEEDLE TRANSFER MECHANISM TO EFFECT COLOUR CHANGE

FIG 4.23
SECTIONAL VIEW SHOWING THE NEEDLE ELEMENTS OF AN HYDRAULIC ACTUATORS FOR A TUFTING MACHINE CONTROLLED BY ROTARY VALVES

TITLE ROTARY VALVE CONTROLLED HYDRAULIC ACTUATORS FOR A TUFTING MACHINE

LOUGHBOROUGH UNIVERSITY OF TECHNOLOGY

TOLERANCES

WHOLE NUMBERS ± 1-00

DECIMALS ± 0-10

All dimensions in millimetres unless otherwise stated.

This drawing is the property of Loughborough University of Technology. It is strictly confidential and must not be copied, loaned or traced without their written consent.
CONTROLS NEEDLE RECIPROCATION BY TRAPPING A VOLUME OF FLUID BELOW THE PISTON

CONTROLS NEEDLE RECIPROCATION BY STOPPING FLOW TO THE TOP OF THE PISTON (DURRANT, REF. 7)

USED TO TRANSFER TUFTING ACTION FROM ONE SET OF NEEDLES TO ANOTHER

FIG 4.25 PHOTOGRAPH SHOWING TWO-POSITION SOLENOID OPERATED ROTARY STROKE VALVES USED TO CONTROL NEEDLE RECIPROCATION AND EFFECT YARN COLOUR CHANGE
reciprocation control rotary stroke valve cuts off flow from the bottom of the needle actuator back to tank. Since flow is possible from the main rotary valve through the spring loaded non-return valve, to the bottom of the needle actuator, the actuator will stay in the up position with the needles clear of the backing fabric.

The rotary stroke valve used to effect yarn colour change is then actuated. This exhausts the left side and pressurizes the right side of the double acting piston to which the needle bar coupling-decoupling components are attached. Therefore as the piston moves from right to left, these components also move from right to left to bring set 2 into the tufting position as indicated in Figure 4.26a.

The horizontal movement of the needle-bar coupling-decoupling components Z1 and Z2 will allow pin A which is attached to the needle actuator to leave the groove in Z1 thus decoupling needle set 1 from the needle actuator, and enter the groove in Z2, coupling the needle actuator and the needle bar carrying set 2. The set 1 needle-bar-carrying components will be left resting on pin B with the needles out of the fabric.

The needle reciprocation control rotary stroke valve is then actuated to allow free needle reciprocation and set 2 will continuously tuft as the main rotary valve is turned.

2. To produce mono-colour patterned tufting, the double acting needle actuator is locked in the up position as above. The rotary stroke valve used to effect yarn colour change is then actuated
FIG. 4.26  THE USE OF LIMITED ROTARY STROKE VALVES TO CONTROL TUFTING MACHINE NEEDLE POSITIONS FOR PATTERNED FABRIC PRODUCTION
to bring the required set of needles into the tufting position. The needle-bar coupling-decoupling action will be the same as above, movement of the double acting piston to which the needle-bar coupling-decoupling components are attached being either to the left or to the right, and components carrying needles which have yarn that is not required, being left resting either on pin B or pin C as required. Programmed on/off switching of the rotary stroke valve used to control needle reciprocation will then result in loop-miss sequences which will produce tufted fabric having the required pattern in individual lines of tufting as the main rotary valve is turned.

3. To produce multicolour patterned tufting, it is necessary to transfer the tufting action from one set of needles to another. This is done by first locking the double acting needle actuator in the up position as above. The rotary stroke valve used to effect yarn-colour change is then actuated to bring the required set of needles into the tufting position. The needle-bar coupling-decoupling action will again be the same as above and will result in the transfer of tufting action from one set of needles to another, while leaving the non-working needles clear of the backing fabric.

As the main rotary valve is turned, programmed on/off switching of both the rotary stroke valve that controls needle reciprocation and that which effects yarn colour change will result in loop-miss sequences which will produce multicolour tufted fabric with the required patterns in individual lines of tufting. Figure 4.26a
shows needle set 2 working with components Z1 resting on pin B, whilst 4.26b shows needle set 1 working with components Z2 resting on pin C.
SECTION FIVE
SUMMARY OF MAIN FINDINGS AND CONCLUSIONS

5.1 Introduction

The objectives of the studies presented in this thesis were to formulate procedures for designing reciprocating mechanisms using hydraulic actuators controlled by rotary valves and to investigate practical aspects of programming so as to provide complete control over individual or groups of actuators. The studies were a continuation of investigations into the possible application of miniature hydraulic actuation and control techniques to high speed reciprocating mechanisms in manipulative machinery.

A review of earlier work on rotary valves and miniature high speed reciprocating actuators indicated that there was a need for advancing by computer aided techniques the design procedures for rotary valve-hydraulic actuator combinations to meet specific dynamic requirements. Another area requiring investigation was into methods of sequential programming of hydraulic actuator positions and the development of techniques for interfacing the programming system with the hydraulically actuated mechanism.

Two programs have been developed. One can be used to calculate the displacement, velocity and acceleration of an actuator at any instant under various operating conditions. The other processes input data specifying the diameter of the bobbin of the rotary valve, axial positions of supply and exhaust lines to the bobbin, the dynamic
cycle characteristics in terms of acceleration, constant velocity, cushioned deceleration and dwell times of individual actuators and the positional phase relationship between them. The output copy from the program is a geometric development of the balanced bobbin.

The use of limited rotary stroke valves to control fluid flow in response to an electrical signal has been put forward as a technique for interfacing the programming system with the hydraulically actuated mechanism.

In order to illustrate the significance of the variables involved in the design of miniature actuators for a specific machinery application and to obtain a practical performance evaluation of such mechanisms, the developed techniques have been used in the design of a unique hydraulic tufting mechanism which incorporates both loop-miss programming of individual actuators and a programmable hydraulically actuated mechanism to effect needle transfer and hence yarn colour change in individual lines of tufting.

5.2 A Comparison of the Operating Features of Hydraulically Actuated Candlewick Tufting Machines and Conventional Solid Member Mechanism Types

An innovative step in the candlewick tufting process has been achieved by completely changing from the long accepted methods of using solid member mechanisms to control needles and manipulate yarn. The hydraulically actuated tufting mechanisms designed and constructed in the Department use individual actuators to move each needle and looper-knife combination independently, in pure straight line
motion. The motions are in a correctly controlled time relationship, to form and cut loops when generating tufted structures. This innovation is the result of research work aimed at advancing the technology and engineering of:

i) Miniature hydraulic actuators with geometries that enable actuation rates of up to 30 per second, at amplitudes of up to 50 mm and provide inherent tuned cushioning to minimize impact stresses and prolong actuator life.

ii) Rotary valves with the correct geometries to control a number of such actuators, through prescribed displacement-time profiles, in a predetermined sequence, with particular phase relationships between them.

Performance advantages of these machines over those employing solid member mechanisms have been demonstrated and films have been made showing the working of the prototype machines designed and built in the Department. Extensive experimental work by Isham, Ref. 6, and Durrant, Ref 7, has also been done on the prototype hydraulic tufting machine. Advantages demonstrated in the film that shows the working of the tufting machine, can be evaluated in terms of:

5.2.1 Pile height variation

This technique of needle control and yarn manipulation enables large amplitudes of needle travel, at high cycling rates to be attained, e.g. 75 mm at 50 penetrations per second compared with 25 mm at 16 per second for a conventional mechanically actuated tufting
mechanism. This fundamental advantage allows completely new machine design ideas to be incorporated, and has led to wide and easy variation of pile height. This is done by simply moving the backing fabric working surface, up or down, in relation to a fixed amplitude of needle travel. The needle-looper-knife positional relationship is set only once and is kept fixed. On the machine built, pile height can be varied in steps of 3.175 mm from 9.525 mm to 25.4 mm, the change from one setting to another being effected in less than 20 seconds.

In contrast, the only competitive mechanical machine, having limited pile height variation (12.7 mm to 25.4 mm), requires replacement of links and cams in the mechanism and separate adjustments of loopers and knives. Retiming of the machine is also necessary. This process can take as long as three hours of a setter's time.

5.2.2 Stitch length variation

The independent hydraulic actuation of the needle and looper-knife assembly completely decouples the backing fabric feed mechanism and allows the fabric feed mechanism to have a much larger amplitude of throw than the mechanical machine. This gives a wider range of stitch lengths. Stitch length can be varied from 118 to 394 per metre on the machine built. Mechanical machines have a range from 197 to 315 per metre over a smaller range of pile heights.

5.2.3 Looper-knife setting

A simple hydraulic piston is used to move both the looper and the knife. The looper moves along a horizontal straight line path
at the correct loop height while movement of the knife produces a correctly timed scissors cutting action. This ensures that only one initial fixed looper-knife setting is necessary for all combinations of pile height, stitch length and running speed.

When operating conditions are changed on the mechanical machine, looper-knife adjustments are necessary to ensure correctly timed cutting action. This is due to the fixed positional relationships imposed on the looper and knife bars by the mechanism.

5.2.4 Programming needle loop-miss combinations

An hydraulic actuator has a maximum thrust limit set by the pressure setting of a relief valve. It can therefore be easily and safely stalled by stopping fluid flow or by interposing a mechanical latch in its path. Needle control into any sequence of loops and misses can therefore be easily achieved by using solenoid operated valves and/or latches, correctly timed to hold the needle actuator at the top of its stroke. This causes a missed penetration and at the same time enables generation of a pattern sequence of tufts and misses on the candlewick fabric. The needle latching method also enables the machine to be stopped with the needle in the UP position allowing an operator to run out at the end of the fabric, with the needle up. This gives automatic cut-off of the retained loops.

The needle programming facility provides virtually unlimited scope for candlewick patterning in preset loop-miss sequences using electronic logic control of a solenoid, which may be applied to a single needle, or to a group of needles. An entirely new range of
candlewick product designs is therefore possible, hitherto not feasible to produce at economic cost. The hydraulic machine developed has an electronic needle program selector which allows any combination of loops and misses to be selected, in programs up to 16N needle motions long, where N is any integer from 1 to 9. This is done simply by setting a sequence of switches into a loop or miss position, the resetting of the program being automatic. This technique of needle control for loop-miss programming provides patterning for any combination of pile height and stitch length.

To avoid a crunch up condition that can result when a machine is mechanically stalled, the mechanical machine is manually wound on to cut off retained loops at the end of a run. The technique of latching up of the needle at the end of a run and the automatic cut off of loops reduces cycle time thereby increasing productivity.

5.2.5 Over tufting

Special timing or setting up of an hydraulic machine to over-tuft designs on to previously tufted base fabrics is not necessary. Because of the dynamic characteristics of the hydraulic needle actuator, this overtufting operation can be efficiently done. This is due to the high constant velocity of fabric penetration and withdrawal, its fixed timing relationships with the looper and knife for all operating conditions, and the separately actuated fabric feed mechanism.

It is necessary in mechanically operated machines, to specifically adjust the mechanisms to perform the overtufting operation.
5.3 Limitations Imposed by the Solenoid Operated Mechanical Latch Mechanism used to Prevent Needle Actuator Descent in Programming

Figure 1.08 shows the operation of the latch mechanism on the built hydraulic machine. Energising the solenoids actuates the pushrod forcing the latch against the needle bar stop. As the actuator approaches the top of the stroke, the latch drops under the needle bar stop thus preventing the downward stroke of the needle. De-energising the solenoids causes the pushrod to retract under the force of the return spring. The latch is then disengaged by the action of its return spring. A sequence of loop-miss combinations is produced by setting the required pattern on an electronic controller.

5.3.1 Speed of operation

The prototype machine is capable of plain tufting at speeds of up to 30 loops per second. Beyond this, the cam control of the backing fabric feed mechanism becomes ineffective resulting in undesirable stitch length variation. Patterning with constant stitch length is available at 20 loops per second beyond which speed the latch fails to latch in or out in one cycle. This results in a 2 loop/2 miss set sequence producing a 1 loop/3 miss pattern. Similarly a 4 loop/4 miss set sequence produces a 3 loop/5 miss pattern. This malfunction is due to the fact that the amplitude of actuator stroke depends on the rotary valve and actuator dimensions, actuator cycling rate and system pressure, \[ D = \frac{2n (\frac{z + d}{r} N - N)}{2\pi} \]. At high cycling rates, the actuator has less time to clear the top of the latch. Even a variable timing system designed to trigger the solenoids so
that the latch can be pushed in well in advance, cannot work when the top of the latch is not cleared by the needle bar stop (Section 4.4.4). This means that the amplitude of the actuator has to be a certain minimum before patterning can be achieved. This sets an upper limit to the cycling rate of the actuator.

5.3.2 Noise and life of the latch and needle bar stop

Due to the metal-to-metal impact between the latch and the needle bar stop, noise levels increase when the machine is in the patterning mode. Levels in the region of 90 dB have been recorded. Noise levels increase with cycling rate and with system operating pressure. Also the impact forces reduce the life of the unit.

5.4 The Use of Limited Rotary Stroke Valves to Control Needle Reciprocation and Effect Needle Transfer

Another hydraulic tufting mechanism designed by the author is being developed in the Department. It has been designed to include features that reduce some of the performance limitations imposed by the mechanical latch mechanism. The looper-knife unit and the needle actuating unit employ one valve each. These valves have been designed to drive 12 loopers and knives or 3 double acting pistons, each piston driving 4 needles at a pitch of 9.525 mm. The design of the units is such that the machine can be used on its own or the units put on the back of a conventional yardage machine to produce patterned fabric. Figure 4.24 shows a sectional view of the needle actuating unit and a diagram showing the working of the looper-knife mechanism is given in Figure A3.06
Manufacture of some components is still continuing and further development of this tufting mechanism will be necessary. Preliminary running tests on the tufting and transfer mechanisms have been carried out to confirm the design principles and functional operation. Photographs of the manufactured components are given in Appendix Three.

5.4.1 Control of needle reciprocation

An hydraulic technique has been developed to control needle reciprocation making it possible to program needle loop-miss combinations. The technique employs a limited rotary stroke valve to trap a volume of fluid below the piston thus preventing its descent. The system has been observed to perform well at high cycling rates. At low cycling rates, leakage from the bottom of the cylinder causes the piston to move with a small amplitude. For overtufting or plain patterned tufting this is not a problem. However, if needle transfer is to be effected in order to produce multicolour tufted fabric, the amplitude of this movement will need to be limited to 2 mm if satisfactory needle transfer is to be effected. The alternative is to employ the same technique for blocking flow to the top of the piston. This has been tried and the results are very promising. Therefore if after testing, the piston movement from the zero displacement position prevents satisfactory needle transfer, it is recommended that the rotary stroke valve be used as shown in the circuit below. This will have the effect of eliminating any differential downward pressure at zero stroke and the upward resistive force will then be sufficient
to hold the needle actuator in the up position. Any leakage will be made up during the next half cycle of the rotary valve rotation.

Hydraulic circuit used to control needle reciprocation by blocking flow to the top of the actuator

FIG 5.01

Patterning will then be available at speeds beyond those at which the mechanical latch can be used. Due to the elimination of the cause of impact forces and hence the source of noise, longer life and quieter operation will be obtained.

5.4.2 Needle transfer to effect yarn colour change

Limited rotary stroke valves have been designed to control the needle coupling-decoupling mechanisms. This will make it possible to produce tufted fabric with more than one colour on each line of tufting. Satisfactory results have been obtained and initial tests indicate that performance of the system will be limited only by the
requirement to hold the needle actuator at the top of its stroke during the needle change-over period. Used on its own, or in conjunction with a yardage machine, this is bound to increase the range and production rate of multicolour products.

Equation \( \theta = \frac{2(e + d)}{D} \) should be used to determine the size of rotary stroke valve, whilst selection of the matching rotary solenoid will be based on this also.

5.5 Design of Rotary Valves Used to Drive Reciprocating Mechanisms

The pressure distribution around the bobbin of the valve affects the power requirements of the valve-actuator system. Garside developed a program that analyses the pressure distribution. The analysis was then used to design compensating pads to balance the bobbin. As a result, he obtained 50% reduction in the power required to turn the bobbin in its housing. Cameron improved on this when he put forward the technique of sculpturing the bobbin surface to leave fluid sources and sinks separated by narrow lands. This has the effect of reducing the metal-to-metal contact area and hence the friction forces. The ideal solution is when the valve behaves like an unloaded shaft turning in a concentric housing.

A computer program has therefore been developed to produce geometric developments showing the bobbin pressure-exhaust sections. The power required to turn the bobbin in its housing will be kept to a minimum by reducing the metal-to-metal contact area and by making sure that pressures at diametrically opposite points are equal.
The resultant pressure system acting on the bobbin will then produce no unbalanced moments.

The development of the bobbin can then be drawn to any scale and wrapped round a metal bar. The contours will guide a machinist in sculpturing the bobbin and the valve produced will automatically be balanced. In addition to being able to manufacture valves of the same design in different sizes, actuators describing the same motion with phase differences can be connected in a circumferential configuration as in Garside's circular weft knitting machine. Figure 5.02 shows the valves designed by Garside and Cameron alongside the new valve. This illustration indicates the simplicity of design and manufacture of the new valve.

The equation $D = \frac{2n (\varepsilon + d)}{2\pi - N\bar{T}}$ has been put forward, to relate the valve diameter $D$, number of pulses per valve rotation $n$, valve land width $\varepsilon$, actuator port diameter $d$, cycling rate $N$ and the sum of incremental times $T$ for actuator acceleration, constant velocity, deceleration and dwell. This expression and the related variables must therefore be taken into account in the specification, design and operation of any system. For a given valve with $D$, $n$, $\varepsilon$ and $d$ fixed, running the system at higher speeds first reduces the actuator dwell periods and at still higher speeds, the actuator displacement attenuates. With the exception of the dwell times which have to be specific for a particular application, the other increments of time can be derived from the equation $\ddot{X}_a + \frac{K_a}{M} \dot{X}_a + \frac{R_a}{M} X_a = \frac{PA}{M}$. This indicates that a high degree of control over the response can be exercised by adjusting the system operating pressure and appropriate sizing of the actuator.
FIG 5.02 DIAGRAM SHOWING THE DEVELOPMENT OF
ROTARY VALVE DESIGNS

THE NEW INHERENTLY BALANCED VALVE

INHERENTLY BALANCED VALVE - CAMERON, REF. 5

THE 'UNIVERSAL' INHERENTLY BALANCED VALVE - CAMERON, REF. 5

THE ORIGINAL VALVE DESIGNED BY GARSIDE - REF. 2
5.6 Hydraulic Cushioning and Sealing Methods

The selection of sealing and cushioning methods used was based on simplicity of design, manufacture and the need to eliminate a heat source. The disadvantage with the use of labyrinth seals is that the leaked fluid requires containing and draining back to tank. On the machine being developed, the leaked fluid is usefully employed to lubricate ancillary mechanisms while allowing free needle reciprocation and free movement of the needle transfer elements. The problem has been to keep the working area free of oil. A rubber gasket was therefore incorporated in the tufting mechanism design to contain the oil at atmospheric pressure within the needle head chamber.

Because of space limitations, it was difficult to use the conventional methods of an adjustable needle valve and non-return valve to provide piston cushioning. The single acting pistons of the looper mechanism employ auxiliary pistons and the double acting pistons driving the needles employ auxiliary sleeves instead, Figure A2.04. Towards the end of a stroke, the volume of oil trapped is allowed to escape to tank through small clearances. The amount of cushioning achieved depends on the amount of fluid trapped, the system pressure, reciprocation speed of the piston, size of clearances and the viscosity of the fluid. There are bound to be small variations of the actual clearance geometries due to manufacturing imperfections whilst the other variables could change significantly during operation. The amount of cushioning will therefore change with operating conditions. There is need therefore for further studies to optimise methods of cushioning high speed reciprocating mechanisms.
5.7 *Suggestions for Further Work*

The work presented has further developed the techniques of design, manufacture and operational control of miniature hydraulic actuators to a stage where machines incorporating these devices as reciprocating mechanisms controlled by rotary valves can be designed to meet specified continuous cycling requirements. However, in engineering research, the acquisition of more analytical and experimental information generally leads to a better understanding of the technology and a generation of new ideas. Three such ideas are put forward as worthy of further investigation to give improved controllability of hydraulically actuated mechanisms.

5.7.1 *Using two rotary valves to control dwell times, provide damping or vary the stroke amplitude of a reciprocating mechanism*

The movements of reciprocating masses have to be adequately damped at the end of each stroke so as to limit the impact forces to an acceptable minimum. The specification might require compound movements, for instance, the knit, tuck or miss stroke of a knitting machine requires amplitude control of the reciprocating mechanism. The actuator could be required to stay in various positions for varying periods of time.

These demands can be satisfied by a simple tuning operation if the mechanism is reciprocated by two rotary valves. Figure 5.03 illustrates this using two identical valves, one connected to each port of the actuator. If the phase difference between the valve pulses is $\frac{\pi}{2}$, the duration of the pressure difference across the actuator will be a maximum and therefore the actuator will go through its full displacement/time cycle - that is, acceleration, constant velocity, deceleration and maximum dwell time. If the phase difference is increased to say $\frac{3\pi}{4}$, duration of the pressure pulse will be less than the maximum.
FIG 5.03  ACTUATOR AMPLITUDE CONTROL USING TWO ROTARY VALVES
hold the load in position, and therefore a pressure difference across the actuator must be maintained. But if the system configuration is such that the actuator will not move under the action of gravity and/or external load forces, then two conditions could occur:

i) Referring to Figure 3.01, if \( t \) is equal to actuator acceleration time + constant velocity time, then at the end of this period, the velocity of the actuator and load will be reduced by gravity, friction and viscous drag caused by shearing of the fluid in the clearances. This is because before the actuator completes its full stroke, the pressure driving the actuator is reduced to zero. Therefore depending on the system operating conditions, the actuator and load may go through the remaining part of full stroke before coming to rest, and staying in the full stroke position, before the reversing pulse is initiated. Friction, gravity and the viscous effects will provide some cushioning and the duration of the dwell period could be varied.

ii) The actuator and load could go through only part of the remaining fraction of full stroke, come to rest and remain in that position before the reversing pulse is initiated. Depending on the phase difference between the pulses from the two valves, the actuator could still be moving when the reversing pulse is initiated, the resulting amplitude of the motion being only a fraction of full stroke.
It is therefore possible to vary the amplitude of the motion of an actuator from maximum, right down to zero, when the pulse phase difference is π, simply by adjusting the phase difference between the pulses generated by the valves. If the two valves are not identical, the length of time the actuator stays in one position relative to the other can be varied again by varying the phase difference between the pulses. This facility could find application in machinery used in the packaging, automatic component handling and assembly, and machine tool control industries.

In all these instances, phase adjustment would be a simple setting up operation to adjust the starting position of the valve pulses relative to one another as indicated in Figure 5.03.

5.7.2 Low voltage electromagnetic devices as the interface between hydraulic and electronic components

Low voltage electromagnetic devices are now widely used in the hosiery industry and are steadily replacing the traditional Jacquard and Pattern wheel which have been in use for a very long time. They are very fast acting and because of their small size and low power consumption, they are suitable for use with microprocessor controlled systems. A virtually unlimited number of patterns can therefore be produced, simply by changing the source programme.

Their major disadvantages are the small force they can exert and the small amplitude of movement they are capable of producing. If it is possible to design the mechanical elements they are required to actuate so that the forces required and distances moved are within
the ranges obtainable from these devices, this will increase the variety of possible patterns and the system speed of operation.

Researchers at UMIST have for instance successfully replaced the Jacquard system in weaving with an electromagnetic system which employs permanent magnets to hold mechanical elements which are moved by cams. The magnetic effect of an applied electronic pulse cancels that of the permanent magnet, thereby releasing the mechanical element. This provides a means of rapid warp position selection. Electromagnetic devices have also been successfully used to actuate miniature pneumatic valves for weft selection, by workers in the Department of Mechanical Engineering, Loughborough University. In both these cases the electromagnet is not used to cause mechanical movement, but merely to hold light elements in position according to the pattern required. So if hydraulic control elements can be made of very small mass and requiring very low operating forces, these miniature electromagnetic devices could be of considerable value in patterned fabric production.

Materials having suitable magnetic properties have been developed to meet widely varying conditions of industrial operation. Thus investigations could be carried out to select materials that will be light enough and at the same time have the mechanical properties necessary to withstand the wear and tear in cyclic control of hydraulic actuators.

Figure 5.04 is a schematic set-up of such a programming system proposed as being worth investigating. It consists of two sets of
LOW VOLTAGE ELECTROMAGNETIC DEVICES AS THE INTERFACE BETWEEN HYDRAULIC AND ELECTRONIC COMPONENTS

Electrical leads to information processing system

Permanent magnetic bit

Very light return spring

Very light mechanical element(s) (magnetic)

Actuator(s) 1

Actuator(s) 2

LOAD

ROTARY VALVE

FIG 5.04
reciprocating actuators both driven by the same rotary valve. Actuator 1, by acting on the light mechanical element shown, would be used to move the load, whilst actuator 2 would move the magnetic element to stop actuator 1 from moving the load. The magnetic element would be held in position by the permanent magnet and the magnetic effect of an applied electronic pulse from the information processing system, would cancel that of the permanent magnet thus allowing the light spring to return the element to the position where it can be moved by actuator 2.

The actuators would therefore be run continuously, to provide the necessary mechanical movement, the electromagnetic device being used only to hold as required by the pattern. With such a system, another method of individual needle control additional to those already developed for the hydraulic tufting machine, would be available. Such a method would probably offer the advantage of greater ease of packing a large number of programming devices into a small space due to the small size of the electromagnetic devices.

5.7.3 The use of Electroviscous Fluids to control programmed actuator GO-NO GO sequences

Electroviscous fluids have been developed for various applications and it is now possible to formulate fluids that will respond at 1 kHz. Therefore in addition to removing speed and space limitations imposed by cams, linkages, clutches, etc that are used in conventional mechanical machines, using the electroviscous technique to control the hydraulic actuation of reciprocating mechanisms could result in higher production rates in the textile fabric manufacturing, or similar manipulative process industries, because
of this high speed response of the electroviscous process. Worthwhile work could therefore be done on developing control devices to use these fluids for the high speed switching of actuator GO-NO GO sequences and would result in another method of interfacing electronic and hydraulic equipment.
APPENDIX ONE

COMPUTER AIDED DESIGN OF ROTARY VALVES - AN ILLUSTRATION OF HOW THE DEVELOPED PROGRAM CAN BE USED TO OBTAIN A GEOMETRIC DEVELOPMENT OF A BALANCED BOBBIN

Presented in Section Two is the development of a computer program which processes input data specifying the diameter of the bobbin of the rotary valve, axial positions of supply and exhaust lines to the bobbin, the dynamic cycle characteristics of individual actuators and the positional phase relationships between them. The output copy from the program is a geometric development of the balanced bobbin, showing the required pressure and exhaust regions to be machined in order to meet the specified dynamic requirements. Taking as a specific example a rotary valve controlling ten individual reciprocating actuators operating in phase related dynamic sequences, this Appendix illustrates how the program can be used. Using the calculated coordinates of the corner points separating the pulse generating sections, a map is produced. For illustration purposes, basic elements of the curves are shown with arrow heads indicating the direction of movement, Section 2.4.2, and hatching showing the nature, i.e. pressure or exhaust, of the different regions. The flow charts are given in Figures 2.10 - 2.13 and a copy of the output is shown in Figure 2.14.
\[ Y(I,J+8) = Y(I,J) + 0.5 \times \Delta Y \]
\[ X(I,J+1) = X(I,J) + 0.5 \times \Delta X \]
\[ Y(I,J+10) = Y(I,J) + 0.5 \times \Delta Y \]
\[ X(I,J+11) = X(I,J) + 0.5 \times \Delta X \]
\[ \text{IF}(X(I,J+1)) = \text{EQ} \text{SUM} \text{E1} \text{SUM} \text{I}1, \text{EQ} \text{SUM} \text{B}1, \text{GO} \text{TO} 76 \]

\[ 63 \text{NPLOT} = 8 \times (N_1 + 1) + 1 \]
\[ \text{SUMA}(1) = \text{TOTAL}(0.0, (L+1), A, I) \]
\[ \text{SUMB}(1) = \text{TOTAL}(0.0, (L+2), B, I) \]
\[ \text{IF}(X(I,J+1) = \text{EQ} \text{SUM} \text{A1}, \text{EQ} \text{SUM} \text{B1}, \text{GO} \text{TO} 87 \]

\[ \text{GO} \text{TO} 78 \]

\[ 64 \text{NPLOT} = 8 \times (N_1 + 1) + 1 \]
\[ \text{SUMA}(1) = \text{TOTAL}(0.0, (L+2), A, I) \]
\[ \text{SUMB}(1) = \text{TOTAL}(0.0, (L+2), B, I) \]
\[ \text{IF}(X(I,J+1) = \text{EQ} \text{SUM} \text{A1}, \text{EQ} \text{SUM} \text{B1}, \text{GO} \text{TO} 76 \]

\[ 70 \text{NPLOT} = 2 \]
\[ (X(I,J+12)) = X(I,J) + B(I,L+1) \]
\[ \text{SUMA}(1) = \text{TOTAL}(0.0, (L+1), A, I) \]
\[ \text{SUMB}(1) = \text{TOTAL}(0.0, (L+2), B, I) \]
\[ \text{IF}(X(I,J+12) = \text{EQ} \text{SUM} \text{A1}, \text{EQ} \text{SUM} \text{B1}, \text{GO} \text{TO} 76 \]

\[ 71 \text{NPLOT} = 2 \]
\[ (X(I,J+12)) = X(I,J) + B(I,L+1) \]
\[ \text{SUMA}(1) = \text{TOTAL}(0.0, (L+1), A, I) \]
\[ \text{SUMB}(1) = \text{TOTAL}(0.0, (L+2), B, I) \]
\[ \text{IF}(X(I,J+12) = \text{EQ} \text{SUM} \text{A1}, \text{EQ} \text{SUM} \text{B1}, \text{GO} \text{TO} 76 \]
SUBROUTINE Z3102(I)
COMMON/ABC1/A(I,100),SUMX,NSEP(250),NPLT(10),S1UMB(10),S1UMA(10)
COMMON/ABC2/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC3/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC4/P
COMMON/ABC5/SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC6/N3GP(10),N4GP(10),ISA,ISB
COMMON/ABC7/A(10,100)
COMMON/ABC8/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC9/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC10/P
GO TO 16
14 ISA=I+1
ISB=I+2
GO TO 16
15 ISA=0
ISB=1
16 CONTINUE
RETURN
END

SUBROUTINE Z3112(I)
COMMON/ABC1/A(I,100),SUMX,NSEP(250),NPLT(10),S1UMB(10),S1UMA(10)
COMMON/ABC2/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC3/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC4/P
COMMON/ABC5/SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC6/N3GP(10),N4GP(10),ISA,ISB
COMMON/ABC7/A(10,100)
COMMON/ABC8/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC9/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC10/P
GO TO 16
14 ISA=I+1
ISB=I+2
GO TO 16
15 ISA=I+1
ISB=I+2
16 CONTINUE
RETURN
END

SUBROUTINE Z3122(I)
COMMON/ABC1/A(I,100),SUMX,NSEP(250),NPLT(10),S1UMB(10),S1UMA(10)
COMMON/ABC2/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC3/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC4/P
COMMON/ABC5/SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC6/N3GP(10),N4GP(10),ISA,ISB
COMMON/ABC7/A(10,100)
COMMON/ABC8/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC9/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC10/P
GO TO 16
14 ISA=I+1
ISB=I+2
GO TO 16
15 ISA=I+1
ISB=I+2
16 CONTINUE
RETURN
END

SUBROUTINE Z3132(I)
COMMON/ABC1/A(I,100),SUMX,NSEP(250),NPLT(10),S1UMB(10),S1UMA(10)
COMMON/ABC2/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC3/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC4/P
COMMON/ABC5/SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC6/N3GP(10),N4GP(10),ISA,ISB
COMMON/ABC7/A(10,100)
COMMON/ABC8/B(I,100),X(I,100),Y(I,100),N3GP(I),N4GP(I),ISA,ISB
COMMON/ABC9/ISTA(I),ISTB(I),SUMA(I),SUMB(I),ISP,IX,YY,N1,N2,K
COMMON/ABC10/P
GO TO 16
14 ISA=I+1
ISB=I+2
GO TO 16
15 ISA=I+1
ISB=I+2
16 CONTINUE
RETURN
END
SUBROUTINE ZZZ103(I,M)
COMMON/ABC1/A(10,100),SUMX,NSEP(250),NPLT(10),STMB(10),S1UMA(10)
COMMON/ABC2/B(10,100),X(10,200),Y(10,200),N3GP(10),N4GP(10),ISA,DT
COMMON/ABC3/ISTA(10),ISTB(10),SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC4/P
X(I,M+1)=XX
X(I,M+2)=X(I,M+1)+A(I,(N1+1))
X(I,M+3)=X(I,M+2)+A(I,(N1+1))
X(I,M+4)=X(I,M+3)
Y(I,M+1)=YY
Y(I,M+2)=Y(I,M+1)+0.5*OY
Y(I,M+3)=Y(I,M+2)
Y(I,M+4)=Y(I,M+3)-0.5*OY
ISA=1
ISB=1
K=2
XX=X(I,M+4)
YY=Y(I,M+4)
NPLT(I)=NPLT(I)+4
CONTINUE
RETURN
END

SUBROUTINE ZZZ103(I,M)
COMMON/ABC1/A(10,100),SUMX,NSEP(250),NPLT(10),STMB(10),S1UMA(10)
COMMON/ABC2/B(10,100),X(10,200),Y(10,200),N3GP(10),N4GP(10),ISA,DT
COMMON/ABC3/ISTA(10),ISTB(10),SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC4/P
X(I,M+1)=XX
X(I,M+2)=X(I,M+1)+A(I,(N1+1))
X(I,M+3)=X(I,M+2)+A(I,(N1+1))
X(I,M+4)=X(I,M+3)
Y(I,M+1)=YY
Y(I,M+2)=Y(I,M+1)-0.5*OY
Y(I,M+3)=Y(I,M+2)
Y(I,M+4)=Y(I,M+3)+0.5*OY
ISA=1
ISB=1
K=2
XX=X(I,M+4)
YY=Y(I,M+4)
NPLT(I)=NPLT(I)+4
CONTINUE
RETURN
END

SUBROUTINE ZZZ103(I,M)
COMMON/ABC1/A(10,100),SUMX,NSEP(250),NPLT(10),STMB(10),S1UMA(10)
COMMON/ABC2/B(10,100),X(10,200),Y(10,200),N3GP(10),N4GP(10),ISA,DT
COMMON/ABC3/ISTA(10),ISTB(10),SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC4/P
X(I,M+1)=XX
X(I,M+2)=X(I,M+1)+A(I,(N1+1))
X(I,M+3)=X(I,M+2)+A(I,(N1+1))
X(I,M+4)=X(I,M+3)
Y(I,M+1)=YY
Y(I,M+2)=Y(I,M+1)+0.5*OY
Y(I,M+3)=Y(I,M+2)
Y(I,M+4)=Y(I,M+3)-0.5*OY
ISA=1
ISB=1
K=2
XX=X(I,M+4)
YY=Y(I,M+4)
NPLT(I)=NPLT(I)+4
CONTINUE
RETURN
END

SUBROUTINE ZZZ103(I,M)
COMMON/ABC1/A(10,100),SUMX,NSEP(250),NPLT(10),STMB(10),S1UMA(10)
COMMON/ABC2/B(10,100),X(10,200),Y(10,200),N3GP(10),N4GP(10),ISA,DT
COMMON/ABC3/ISTA(10),ISTB(10),SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC4/P
X(I,M+1)=XX
X(I,M+2)=X(I,M+1)+A(I,(N1+1))
X(I,M+3)=X(I,M+2)+A(I,(N1+1))
X(I,M+4)=X(I,M+3)
Y(I,M+1)=YY
Y(I,M+2)=Y(I,M+1)+0.5*OY
Y(I,M+3)=Y(I,M+2)
Y(I,M+4)=Y(I,M+3)-0.5*OY
ISA=1
ISB=1
K=2
XX=X(I,M+4)
YY=Y(I,M+4)
NPLT(I)=NPLT(I)+4
CONTINUE
RETURN
END

SUBROUTINE ZZZ103(I,M)
COMMON/ABC1/A(10,100),SUMX,NSEP(250),NPLT(10),STMB(10),S1UMA(10)
COMMON/ABC2/B(10,100),X(10,200),Y(10,200),N3GP(10),N4GP(10),ISA,DT
COMMON/ABC3/ISTA(10),ISTB(10),SUMA(10),SUMB(10),ISB,XX,YY,N1,N2,K
COMMON/ABC4/P
X(I,M+1)=XX
X(I,M+2)=X(I,M+1)+A(I,(N1+1))
X(I,M+3)=X(I,M+2)+A(I,(N1+1))
X(I,M+4)=X(I,M+3)
Y(I,M+1)=YY
Y(I,M+2)=Y(I,M+1)+0.5*OY
Y(I,M+3)=Y(I,M+2)
Y(I,M+4)=Y(I,M+3)-0.5*OY
ISA=1
ISB=1
K=2
XX=X(I,M+4)
YY=Y(I,M+4)
NPLT(I)=NPLT(I)+4
CONTINUE
RETURN
END
APPENDIX TWO

DESIGN PROCEDURES FOR ACTUATOR/ROTARY VALVE COMBINATIONS - ESTIMATION OF HYDRAULIC SYSTEM PARAMETERS IN CIRCUITS INVOLVING RECIPROCATING MECHANISMS CONTROLLED BY ROTARY VALVES

In Section 3.7 a procedure was proposed for designing hydraulic circuits involving reciprocating mechanisms and rotary valves. To implement this procedure some parameters will be specified and others determined from given data. In this Appendix expressions that can be used to obtain approximate values of some of them are derived. As pointed out in Section 3.2 it will be necessary to have an idea of their magnitudes before the system is properly designed and constructed. The actual values can only be determined from experimental tests performed on the system and also, it will be necessary to tune the system for optimum performance.

A2.1 Estimation of Leakage and Actuator Flow Requirements

In order to obtain expressions that can be used in the sizing of the actuator and the rotary valve; in the selection of the system drive mechanism and in predicting system performance and power requirements, an analysis of the motion of the actuator, leakage characteristics and flow requirements when operating under various load conditions is necessary.

A2.1.1 Leakage between a steadily moving piston and the concentric cylindrical housing

With the notation of Figure A2.01, the total force acting on an elemental annular shell of fluid of thickness $dr$ is:
LEAKAGE BETWEEN A STEADILY MOVING PISTON AND THE CONCENTRIC CYLINDRICAL HOUSING

\[ Q = \pi R R \left( \frac{\partial P^*}{\partial x} \right) \frac{C_f}{6\mu} + V_G \]
\[ F = \{P^* - (P^* + \delta P)\} \delta r 2\pi (r + \frac{\delta r}{2}) + 2\pi r \tau \delta x - 2\pi \delta x (r + \delta r)(\tau + \delta \tau) \]

and for infinitesimal increases in \(r\), \(r = r + \frac{\delta r}{2} = r + \delta r\)

this becomes:

\[ F = -2\pi r \delta r \delta P^* - 2\pi r \delta x \delta \tau \]

For steady-state conditions, there is no acceleration and so this total force is zero, giving

\[ \delta r \delta P^* = -\delta \tau \delta x \]

Dividing by \(\delta r \delta x\) and taking the limit as \(\delta r \to 0\), gives

\[ \frac{\delta P^*}{\delta x} = -\frac{\delta \tau}{\delta r} \]

For laminar leakage flow, the stress is \(\tau = \mu \frac{\partial u}{\partial r}\)

Therefore

\[ \frac{\delta P^*}{\delta x} = -\frac{\partial}{\partial r} (\mu \frac{\partial u}{\partial r}) \]

As \(P^*\) no where varies in the radial direction, \(\frac{\delta P^*}{\delta x}\) is independent of \(r\) and the equation can therefore be integrated with respect to \(r\) to give

\[ \frac{\delta P^*}{\delta x} = -\mu \frac{\partial u}{\partial r} + A \quad (A\ being\ the\ constant\ of\ integration) \]
Integrating further with respect to \( r \) gives

\[
\frac{\delta P^*}{\delta x} \frac{r^2}{2} = -\mu u + A_r + B \quad (B \text{ being a constant of integration})
\]

This solved gives \( u \) as:

\[
u = \frac{1}{\mu} \left( B + A_r - \frac{\delta P^*}{\delta x} \frac{r^2}{2} \right)
\]

If the shell under consideration is far from the edges, \( A \) and \( B \) will be constants independent of both \( r \) and \( x \) and can therefore be determined from the boundary conditions.

For no slip at the boundaries, \( u = 0 \) when \( r = R_2 \). With this condition, \( B \) is obtained from the equation for \( u \) as:

\[
B = \frac{\delta P^*}{\delta x} \frac{R_2^2}{2} - AR_2
\]

Also for no slip at the boundaries, \( u = V \) when \( r = R_1 \). This condition, with the value of \( B \) from above gives the value of \( A \) from the equation for \( u \) as

\[
A = \frac{\delta P^*}{\delta x} \frac{(R_1 + R_2)}{2} - \frac{\mu V}{R_2 - R_1}
\]

Inserting these values of \( A \) and \( B \) in the equation for \( u \) obtained above gives the value of \( u \) at any radius \( r \) as:

\[
u = \frac{1}{\mu} \left( \frac{\delta P^*}{\delta x} \frac{R_2^2}{2} - R_2 \frac{\delta P^*}{\delta x} \frac{(R_2 + R_1)}{2} - \frac{\mu V}{R_2 - R_1} \right) + \frac{\delta P^*}{\delta x} \frac{(R_2 + R_1)}{2} - \frac{\mu V}{R_2 - R_1} r - \frac{\delta P^*}{\delta x} \frac{r^2}{2}
\]
which on simplification gives \( u \) as:

\[
    u = \frac{1}{\mu} \left\{ \frac{1}{2} \frac{\delta P^*}{\delta x} \left( - R_1 R_2 + (R_1 + R_2) r - r^2 \right) + \frac{\mu V}{R_2 - R_1} (R_2 - r) \right\}
\]

The discharge through the annular shell under consideration will be \( 2\pi ur \delta r \) and therefore the total leakage flow rate \( Q \) will be given by

\[
    2\pi \int_{R_1}^{R_2} ur \, \delta r
\]

With the value of \( u \) from above,

\[
    Q = \frac{2\pi}{\mu} \int_{R_1}^{R_2} \left( \frac{1}{2} \frac{\delta P^*}{\delta x} \left( - R_1 R_2 + (R_1 + R_2) r - r^2 \right) + \frac{\mu V}{R_2 - R_1} (R_2 - r) \right) r \, dr
\]

On integration, this simplifies to:

\[
    Q = \frac{\pi}{12\mu} \frac{\delta P^*}{\delta x} (R_2 + R_1)(R_2 - R_1)^3 + \frac{\pi V}{3} (R_2 - R_1)(R_2 + 2R_1)
\]

In terms of the annular clearance \( C_r = (R_2 - R_1) \), and if \( R = (R_1 = R_2) \) is the nominal radius, this equation can be written as:

\[
    Q = \pi R \left[ \left( \frac{\delta P^*}{\delta x} \right) \frac{C_r^3}{6\mu} + VC_r \right]
\]
A2.1.2 Estimation of actuator flow requirements

The total volume flow rate of fluid to be supplied to the actuator depends on the speed at which the actuator is working, its dimensions, and the dimensions of its housing. Because of leakage, the required volume flow rate will also depend on the pressure driving the actuator and the viscosity of the fluid. For a double acting actuator as the one shown in Figure A2.02 the total volume of fluid \( Q \) is made up of the displacement volume \( Q_d \) causing actuator movement and leakage flows \( Q_1 \) and \( Q_2 \) through the clearance spaces between the piston and its housing. Equation \( Q = \pi R \left[ \frac{\delta P}{\delta x} \cdot \frac{C_r^3}{6 \mu} + V C_r \right] \) can be used to estimate the leakage flow rate through the various clearances, while the volume flow rate due to the displacement of the actuator will be given by the product of the area of the annulus \( \pi \left( D_2^2 - D_1^2 \right) \) and the velocity of the piston \( V \). For instance if it is assumed that the actuator is working at 2000 cycles a minute while supplied at a constant pressure of 55 bar, the actuator stroke being 50 mm, then the average speed can be taken as 3.33m per second. If the significant dimensions are those given in Figure A2.02 then the total volume flow rate of fluid required \( Q \) would be calculated as shown in the following table.
ESTIMATION OF ACTUATOR FLOW REQUIREMENTS

\[ Q = Q_a + Q_1 + Q_2 \]

FIG A2.02
TABLE TA2.1:

Estimation of Actuator Flow Requirements Using the Derived Equations

\[ V = 3.33 \text{ m sec}^{-1} \quad P = 55 \text{ bar} \quad \mu = 1.38 \times 10^{-2} \text{ N sec m}^{-2} \]
\[ = 5.62 \times 10^{5} \text{ kg m}^{-2} \quad = 1.38 \times 10^{-3} \text{ kg sec}^{-1}\text{m}^{-1} \]

| \( Q_1 \) | \( 2.5 \times 10^{-3} \) | \( 4.68 \times 10^{7} \) | \( 0.025 \times 10^{-3} \) | \( 0.25 \times 10^{-7} \) |
| \( Q_2 \) | \( 4.5 \times 10^{-3} \) | \( 1.87 \times 10^{7} \) | \( 0.02 \times 10^{-3} \) | \( 11.92 \times 10^{-7} \) |
| \( Q_a \) | \( 5 \times 10^{-3} \) | \( 9 \times 10^{-3} \) | \( \frac{\pi}{4} (D_2^2 - D_1^2) \) | \( V \text{ m}^3 \text{ sec}^{-1} \) |

\[ Q = Q_a + Q_1 + Q_2 = 1.477 \times 10^{-4} \text{ m}^3 \text{ sec}^{-1} \]
\[ = 0.148 \text{ litres per sec} \]
\[ = 8.88 \text{ litres per min} \]

A2.2 Viscous Forces

Different mechanism types will be represented by different differential equations but, in all cases, there will be a viscous drag damping force acting on the actuator as a result of the relative motion between the actuator and its housing. This will be in addition to any other load, gravitational, frictional and driving pressure forces. It is therefore necessary to develop an expression for this force. Assuming that there is no slip at the boundaries and that the clearances between parts in relative motion are so small that a linear velocity distribution can be assumed (that is \( \frac{\partial u}{\partial y} = \frac{u}{C_r} = \text{constant} \),
then an estimate of this force can be obtained.

By definition, the stress in a viscous fluid is given by \( \tau = \mu \left( \frac{\partial u}{\partial y} \right) \). By taking the value of this function at the moving boundary, the stress on the moving surface can be calculated as the product of the stress and the area over which this stress acts.

Therefore viscous drag force = viscous stress \(*\) wetted surface area

\[
= \mu \frac{\partial u}{\partial y} * A_w
\]

\[
= \frac{A_w \mu}{C_r} \frac{dx_a}{dt} = \text{constant} \times \frac{dx_a}{dt}
\]

A2.3 Viscous Torque on a Cylinder Rotating in a Concentric Housing

\[
\text{FIG A2·03}
\]

\[
\Omega = N \text{ rad/sec}
\]
Referring to the Figure A2.03 overleaf the net torque on the element of fluid shown is

\[(\tau + \delta \tau)(r + \delta r)^2 \delta \theta - \tau r^2 \delta \theta\]

Under steady state conditions, the element does not undergo angular acceleration and so the net torque on it is zero. Therefore neglecting higher orders of small quantities:

\[2 \tau r \delta r + r^2 \delta \tau = 0\]

Dividing by \(r^2\) and rearranging gives

\[\frac{\delta \tau}{\tau} = -2 \frac{\delta r}{r}\]

Integration gives \(\ln \tau = 2 \ln r + \ln A = \ln \frac{A}{r^2}\) which yields \(\tau\) as \(\tau = \frac{A}{r^2}\) where \(A\) is a constant of integration.

The velocity gradient will be \(\frac{\partial u}{\partial r} = \frac{\partial}{\partial r} (r\omega) = \omega + r \frac{\partial \omega}{\partial r}\) and therefore the rate of shear representing the relative motion between the particles will be \(r \frac{\partial \omega}{\partial r}\). (If \(\omega\) is independent of \(r\), \(\frac{\partial \omega}{\partial r} = 0\) and \(\frac{\partial u}{\partial r} = \omega\) and therefore there will be no relative motion between the particles of the fluid). With this simplification:

\[\tau = \mu r \frac{\partial \omega}{\partial r}\] or \[\mu \frac{\partial \omega}{\partial r} = \frac{\tau}{r} = \frac{A}{r^3}\]

Integrating, this yields: \(\mu \omega = -\frac{A}{2r^2} + B\) where \(B\) is a constant of integration.
For no slip conditions at the boundaries, \( \omega = 0 \) when \( r = R_2 \) and \( \omega = \Omega \) when \( r = R_1 \). Therefore from the above

\[
B = \frac{A}{2R_2^2} \quad \text{and} \quad A = \frac{2R_2^2 R_2^2 \mu \Omega}{R_1^2 - R_2^2}
\]

Torque on the concentric housing of radius \( R_2 = \text{stress*area*radius} \). If \( L \) represents the length of the cylinder in contact with the fluid, including end effects which act in a very similar way to an additional length of cylinder in contact with the fluid, then the torque \( T_Q \) is given by:

\[
T_Q = (\mu \frac{3\omega}{3r})_{r=R_2} \times 2\pi R_2 L R_2^2
\]

\[
T_Q = 2\pi \mu R_2^3 L \left( \frac{3\omega}{3r} \right)_{r=R_2} = 2\pi \mu R_2^3 L \left( \frac{A}{\mu R_2^3} \right) = 2\pi LA
\]

\[
T_Q = \frac{4\pi LR_1^2 R_2^2 \mu \Omega}{R_1^2 - R_2^2}
\]

or simply

\[
T_Q = K_q \mu \Omega
\]

where \( K_q \) is a constant for the system.

**Note:**

In using these equations the radial displacement \( R_d \) at the inner surface of the thick-walled cylinder due to the internal pressure increase
\( \Delta P \) must be taken into account. This is given by

\[
R_d = \frac{D \Delta P}{2E \left( \frac{D_0^2 + D^2}{D_0^2 - D^2} + \gamma \right)} = \frac{D \Delta P}{2E \left( \frac{(1+\gamma)D_0^2 + (1-\gamma)D^2}{2T_h (D_0 + D)} \right)}
\]

where \( D \) is the inner diameter, \( D_0 \) the outer diameter and \( T_h \) the wall thickness. \( 2T_h = D_0 - D \).

A2.4 The Combined Linearized Viscous Damping Coefficient \( K_v \)

When using the equation \( \frac{d^2X_a}{dt^2} + \frac{K_v}{M} \frac{dX_a}{dt} + \frac{R_v}{M} X_a = \frac{P A}{M} \), the viscous damping coefficient \( K_v \) and the mass \( M \) must be appropriately defined. Values calculated with the use of the above equations will indicate faster system responses. From experimental results, it was noted that their values must include the effects of fluid flow in the conduits. This is why Cameron combined the viscous and momentum effects when he defined a LINEARISED VISCOUS DAMPING COEFFICIENT for the whole circuit (Cameron (5), Section 1.5) and proposed a method for its determination.

A2.5 Effective Circuit Mechanism Mass \( M \)

With the accumulator located as close to the rotary valve as possible, the valve pressure/exhaust sections are in effect rotating constant pressure fluid sources and sinks which supply or exhaust the two ends of a reciprocating actuator. As the actuator and load move, fluid in the circuit section between the valve and actuator moves to satisfy continuity. Therefore in addition to the actuator and load, this mass must be considered when analysing the motion of the mechanism.
Considering continuity, its acceleration in the conduits $a_f$ is related to that of the actuator $\ddot{X}_a$ by $a_f = \frac{A}{A_c} \ddot{X}_a$ where $A$ is the effective area of the actuator and $A_c$ the cross-section area of the conduit connecting the valve and the actuator. The total inertia force resisting motion is then given by

$$F = M \ddot{X}_a = (m + A \left[ \frac{m_1}{A_1^2} + \frac{m_2}{A_2^2} + \frac{m_3}{A_3^2} + \frac{m_4}{A_4^2} + \frac{m_5}{A_5^2} + \ldots \right]) \ddot{X}_a$$

where $M = \text{effective circuit mechanism mass}$

$$= m + A \left[ \frac{m_1}{A_1^2} + \frac{m_2}{A_2^2} + \frac{m_3}{A_3^2} + \frac{m_4}{A_4^2} + \frac{m_5}{A_5^2} + \ldots \right]$$

$m = \text{mass of actuator and load}$

$m_1, m_2, m_3, m_4, m_5, \ldots = \text{mass of fluid in circuit section 1, 2, 3 respectively,}$

$A_1, A_2, A_3, A_4, A_5, \ldots = \text{cross-section flow area 1, 2, 3 respectively.}$

The term $A \left[ \frac{m_1}{A_1^2} + \frac{m_2}{A_2^2} + \frac{m_3}{A_3^2} + \ldots \right]$ due to the fluid in the conduit represents an additional mass and could have a significant effect on system dynamic performance. It is therefore necessary to reduce its value by locating the accumulator, valve and actuator as close together as possible.

A2.6 Hydraulic Cushioning

Effective sealing of miniature high speed reciprocating shafts is difficult. The conventional use of 'O' rings limits the speed of operation, introduces friction and heat sources and the seals leak
after a short period of running. Also towards the end of each stroke, the actuator and load have to be brought to rest gradually so as to limit impact forces and noise to a minimum. Controlled deceleration in this case will increase the life expectancy of the actuator. This problem was investigated by Cameron who came up with the idea of using 'Labyrinth' or clearance seals which provide no physical contact between adjacent sliding members but rely on a small clearance between members in relative motion to limit the resultant leakage to an acceptable level. The leaked fluid can also be used to lubricate ancillary mechanisms. This simplifies manufacture, eliminates friction and heat sources and improves actuator response times. This technique, which was adopted for use on actuators used in this work, employs auxiliary pistons on single acting actuators and auxiliary sleeves on double acting actuators. Cushioning is obtained when a trapped volume of fluid is allowed to exhaust through the clearances, Figure A2.04. Cushioning time will therefore depend on the total volume of fluid trapped, the pressure and viscosity of the fluid driving the piston, and the size of clearances through which the trapped volume is allowed to escape.

Equation \( Q = R \pi [(\frac{dP}{dx}) \frac{C^3}{6\mu} + VC_p] \) cannot be applied to the trapped volume to determine the deceleration time because the velocity \( V \) varies with piston position and the pressure gradient \( \frac{dP}{dx} \) does not remain constant. In general it will therefore be better to set the deceleration time when the pistons are lapped in to suit the housing. The initial tuning operation is best done by:
SINGLE ACTING ACTUATOR WITH HYDRAULIC CUSHIONING

DOUBLE ACTING CYLINDER WITH HYDRAULIC CUSHIONING

FIG A2·04
i) manufacturing the piston and housing so that the clearances are smaller than required, and will therefore result in severe cushioning of the piston;

ii) the clearances are then progressively increased in steps of say, 0.01 mm by grinding the cushioning lands. At each stage, the system is re-assembled and the cushioning checked at the expected system working pressure until the observed performance is satisfactory.

Since the amount of cushioning will vary with the fluid temperature, final tuning will then be done by adjusting the system operating pressure to give optimum cushioning at the operating temperature.

A2.7 Pipe Size and Circuit Configuration

Pipe size will depend on the selected driving pressure and load characteristics. The circuit configuration will determine the amount of power loss due to friction, bends and fittings. It is desirable to keep the actuators as near the rotary valve as possible so as to reduce the effective mechanism mass M in the circuit and the effective damping coefficient Kv.

In conventional hydraulic circuits, the flow is determined from load-velocity requirements, and pipe size is selected so that the pressure drop is moderate. Pipe selection criteria of 4.57 m/sec maximum fluid flow velocity in the conduits and $2.3 \times 10^3$ kg/m$^2$ pressure drop per metre of conduit length are common. The empirical equation giving the pressure drop for fully developed turbulent flow
is $\Delta P = f \frac{1}{D} \frac{\rho u^2}{2}$ where the friction factor $f$ depends on Reynolds number and pipe roughness. The additional pressure drop due to the transition length is about $0.09 \frac{\rho u^2}{2}$ and is negligible in most computations. Pressure drops due to entrance and exit losses are also usually negligible. For smooth pipes and Reynolds numbers less than 100,000, Blasius (Merrit (8), Section 2.5), found the friction factor to be $f = \frac{0.3164}{Re^{0.25}}$ where $Re$ is the Reynolds number and this covers the cases in hydraulic control where pipes are smooth and fluid velocities kept below 4.57 m/sec to avoid large pressure surges due to sudden valve closures. Minor losses caused by bends, fittings and sudden changes in flow section and those due to centrifugal forces and secondary flow patterns are empirically described by $H_L = k \frac{u^2}{2g} = \frac{k}{2g} \left( \frac{Q}{A} \right)^2$ or $\Delta P = \frac{k}{2} \left( \frac{Q}{A} \right)^2$. Figure 3.07 gives numerical values of the constant $k$ for various flow situations. Since pressure losses in the various sections are additive, the total pressure loss can be expressed in terms of the velocity of the actuator as $\Delta P = K \dot{X}$, $K$ will then be obtained from the selected pipe size, circuit configuration and expected flow velocity.

In hydraulic circuits involving miniature high speed reciprocating actuators controlled by rotary valves, the accumulator, rotary valve and actuator will be located as close together as possible in order to reduce the effective circuit mechanism mass $M$ and damping coefficient $K_v$. The valve will then behave like a source or sink and the pressure drop and fluid flow velocity limits mentioned above will not be significant. However the effect of the fluid in the section between the valve and the actuator, sections 1, 2 and 3 in Figure 3.07, will have to be taken into account because it affects the system response as pointed out in Section 3.5.
A2.8 Other Mechanism Losses

There will be resistance to motion because of the drag force imparted as adjacent surfaces move with a relative velocity, for instance pistons moving in their housings or a slide moving in its bearing. With dry surfaces, the force will depend on the nature of the surfaces and normal reaction \( N_F = nN \). If the surfaces are lubricated, the force will be a measure of the internal friction of the fluid or its resistance to shear. In this case, it will be proportional to the area in contact \( A_w \), the relative velocity \( \dot{x} \), the absolute viscosity \( \mu \), and will be inversely proportional to the film thickness \( C_r \). \( (F = \mu \frac{A_w \dot{x}}{C_r}) \). It can be expressed in terms of the effective area of the actuator \( A \) and the pressure required to overcome it \( \Delta P \). Therefore the pressure loss will be given by \( \Delta P = \frac{A_w \dot{x}}{AC_r} = \text{constant} \times \text{relative velocity} \).
APPENDIX THREE

PHOTOS OF MANUFACTURED COMPONENTS OF THE EXPERIMENTAL PROTOTYPE HYDRAULICALLY ACTUATED TUFTING MACHINES

In order to obtain a practical performance evaluation of actuator and valve designs, the developed techniques have been used in the design of unique hydraulic tufting machines which incorporate loop-miss programming of individual needle actuators. A current development involves the use of limited rotary stroke valves,

i) To control needle reciprocation thus making it possible to program loop-miss sequences, and

ii) For hydraulically actuating a programmable mechanism that effects needle transfer and hence yarn colour change in individual lines of tufting.

Manufacture of some components is still going on and development testing of the components is yet to be done. Given in this Appendix are photographs of some components of the latch controlled experimental prototype tufting machine that has been exhaustively tested, and the manufactured components of the rotary stroke valve controlled tufting machine being developed.
FIG COMPONENTS OF THE LOOPER-KNIFE MECHANISM A3.01 OF THE DEVELOPED EXPERIMENTAL TUFTING MACHINE
FIG ASSEMBLY OF THE LOOPER-KNIFE COMPONENTS A3.02 OF THE DEVELOPED EXPERIMENTAL TUFTING MACHINE
FIG COMPONENTS OF THE LATCH CONTROLLED NEEDLE ACTUATING MECHANISM OF THE DEVELOPED EXPERIMENTAL TUFTING MACHINE
FIG 10

SUB-ASSEMBLY OF THE LATCH CONTROLLED NEEDLE ACTUATING MECHANISM OF THE DEVELOPED EXPERIMENTAL TUFTING MACHINE
FIG. 3.05 GENERAL VIEW OF THE DEVELOPED A3.05 EXPERIMENTAL CANDLEWICK TUFTING MACHINE
Auxiliary piston

Main pistons

Actuator

Auxiliary piston

Knife

Looper

FIG A3.06

DIAGRAM SHOWING THE WORKING OF THE LOOPER-KNIFE MECHANISM OF THE TUFTING MACHINE BEING DEVELOPED

(Employed two opposed single acting pistons with hydraulic cushioning)
FIG COMPONENTS OF THE LOOPER-KNIFE MECHANISM A3.07 OF THE CANDLEWICK TUFTING MACHINE BEING DEVELOPED
FIG SUB-ASSEMBLY OF THE LOOPER-KNIFE
A3.08 COMPONENTS OF THE CANDLEWICK TUFTING MACHINE BEING DEVELOPED
FIG A3.09
ASSEMBLED UNIT OF THE LOOPER-KNIFE
COMPONENTS OF THE CANDLEWICK TUFTING
MACHINE BEING DEVELOPED
COMPONENTS FOR EFFECTING TRANSFER OF TUFTING ACTION

NEEDLE BAR COUPLING-DECOUPLING COMPONENTS

ROTARY SOLENOID

DOUBLE ACTING PISTON

NEEDLE BARS

ROTARY STROKE VALVE

FIG COMPONENTS USED TO TRANSFER THE TUFTING ACTION FROM ONE SET OF NEEDLES TO ANOTHER IN THE CANDLEWICK TUFTING MACHINE BEING DEVELOPED
FIG 13.11 FRONT VIEW OF AN ASSEMBLED UNIT OF THE A3.11 NEEDLE ACTUATING COMPONENTS FOR THE CANDLEWICK TUFTING MACHINE BEING DEVELOPED
Fig. Side view of an assembled unit of the A3.12 needle actuating components for the Candlewick Tufting Machine being developed.
REFERENCES

1. GARSIDE, J D:
"The Development of a Miniature Hydraulic Actuator for Application to a Circular Weft Knitting Machine".

2. GARSIDE, J D:
"An Investigation into Miniature Hydraulic Actuation Techniques for Needle Control on Industrial Knitting and Sewing Machines".

3. PRIESTLEY, T P and GARSIDE, J D:
"A Hydraulic Actuation Technique to Eliminate the Cam Mechanism in Circular Weft Knitting".

4. PRIESTLEY, T P and CAMERON, D S:
"Sewing by Means of an Hydraulically Actuated Mechanism".

5. CAMERON, D S:
"An Investigation into Miniature Hydraulic Actuation and Control Techniques for Use on High Speed Reciprocating Mechanisms".

6. ISHAM, K:
"Development Testing of the Needle Actuator for the Hydraulically Actuated Tufting Machine".
Final year project submitted in partial fulfilment of the requirements for the award of the Degree of Bachelor of Technology, Loughborough University, Department of Mechanical Engineering, May 1981.
7. DURRANT, R G:
"Development of Programming Techniques for Hydraulically Actuated Tufting Needles".
Final year project report submitted in partial fulfilment of the requirements for the award of the Degree of Bachelor of Technology, Loughborough University, Department of Mechanical Engineering, May 1982.

8. MERRITT, H E:
"Hydraulic Control Systems".
John Wiley and Sons Inc. 1967.